Developments in Waste Heat Systems for Motor Tankers

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In order to cover the application of waste heat recovery in motor ships it is necessary to consider other auxiliary plant on board, the economics of changes in the plant and the estimating of auxiliary loadings. Thereafter a decision may be made on the extent of waste heat recovery which it is desirable to include. There may be several alternative solutions and the one chosen will be influenced by operating experience, economics and staffing considerations.

This paper reviews suitable equipment for use in heat recovery circuits and to some extent follows the process of development which has been followed in one company. Details are given of service performance of prototype plant and the reasons which led to the particular circuit components being chosen.

INTRODUCTION

The engineer's aim is efficiency, provided by the simplest possible plant at the lowest true cost, bearing in mind that these requirements often conflict and that any one may take over priority due to changing external circumstances. This is shown in the development of motor ship auxiliary machinery, where the increased hotel services and propulsion auxiliary loads in the modern high powered ships, which are necessary to maintain competitive trading ability, now call for careful examination of auxiliary plant economics.

Total expenditure on auxiliary plant is phased into initial cost and maintenance and operating (i.e. fuel) costs which are spread over the life of the ship. It is apparent that fuel costs can be reduced if heat from a waste heat source can be utilized to avoid burning fuel at sea. If waste heat recovery plant is installed the initial cost of the auxiliary plant will increase and it must be decided if the added investment is justified. While there are other methods of reducing the true cost, an alternative, which will be dealt with at length in this paper, is that of providing all auxiliary power requirements at sea from a steam turbine-driven generator with the steam being produced from a waste heat unit in the main engine exhaust ducting. This is one correct technical and economic answer for ships which spend about two-thirds of their time at sea at designed service power.

The use of steam-driven plant in a motor ship is not unusual since motor tankers normally have the auxiliary system of a steamship superimposed upon their Diesel plant to supply their large cargo pumping and heating loads. It is considered that the quality of marine engineers now available is such that no fear need be felt in putting steam auxiliaries in any type of motor ship, provided too much complication is resisted and the equipment provided is of good quality.

EXTENT OF WASTE HEAT RECOVERY

Before deciding to instal waste heat recovery plant for both power and heating services it is necessary that the following assumptions and controlling factors should be considered:

a) A capital amortization rate of 15 per cent is reasonable and both maintenance and fuel cost savings are available to offset initial expenditure. Fuel costs of £9 10s./ton for Diesel oil and £5/ton for boiler fuel have been assumed.

b) If it is necessary to run a Diesel generator in parallel with the turbo set whilst at sea, or frequently to use supplementary oil firing of the boilers, except when using steam for deck machinery or cargo purposes, then a near maximum heat recovery system is not justified.

c) At designed main engine service power, the exhaust gas heat must be adequate to enable the normal ship's electrical load to be provided by steam supply to a turbo-generator and steam must also be available for ship's heating and hotel services. If the ship is to be operated at a lower power during initial service to meet a charter speed, due allowance should be made.

d) The electrical load must be carefully assessed and allowance made for peak loads due to air conditioning, galley, etc.

e) The normal heating service load at sea must be similarly assessed and the optimum steam pressure decided. A generous allowance on quantity must be made for radiation and other parasitic losses.

f) If the ship is low powered, it must spend the major part of its time at sea to enable the additional investment to be recovered in a reasonable time, i.e. tankers and bulk carriers are more suitable than short haul cargo ships.

THE GRADE OF HEAT

When a turbocharged Diesel engine is operated at full load the heat balance is approximately:

i) 38 per cent to useful work on the propeller shaft;

ii) 35 per cent to heat in the exhaust gases after the turbocharger;

iii) 11 per cent to heat in jacket water system;

iv) 16 per cent to other coolant/lubrication services and mechanical losses.

The ratios vary with engine type and loading. Practicable heat recovery is limited to heat remaining in the exhaust gases and the jacket water cooling system. Because the jacket water temperature is normally restricted to about 150 deg. F. (66

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deg. C.) the grade of heat is low and heat recovery is restricted to services requiring heat at a lower temperature level.

ELECTRIC GENERATING PLANT

A substantial annual cost is incurred if Diesel generators are used to provide power at sea. Fig. 1 shows this cost for typical ship sizes and engine powers. Initial costs of such machines are much less than for turbo-generators of high efficiency type. Typical costs for such units and for single-stage turbo-generators are shown in Fig. 2.

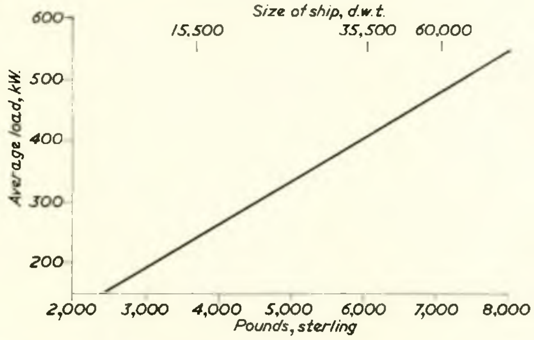


FIG. 1—*Annual fuel cost for power generation at sea (based on 250 days p.a.—Fuel cost £9-10s.-0d. per ton)*

In view of the apparent high extra cost of using a multi-stage self-contained turbo set instead of a Diesel-driven set, the relative maintenance costs become a very significant factor. Disregarding the costs of breakdowns some random figures taken five years ago showed the comparative annual maintenance costs per ship to be:

£1,460 for two 600 r.p.m., 150 kW Diesel generator sets.

£480 for two 600 kW turbo-generator sets (in steamships).

On a capital and amortization differential of 15 per cent, this would justify an additional initial expenditure of about £7,000 for installing turbo sets instead of Diesels for generating duty. Some up-to-date costs taken on the same basis as previously show the current random figures for ships about four years old to be:

£1,560 p.a. for two 600 r.p.m. 250 kW Diesel generator sets.

£520 p.a. for two 750 kW turbo-generator sets (in steamships).

There is a need for both types of machines in the engine room of a motor tanker. The turbine set is well suited for continuous operation and can usefully employ steam from waste heat plant at sea. The Diesel-driven unit is quick starting in emergency and more economical if fuel is being burnt to provide power.

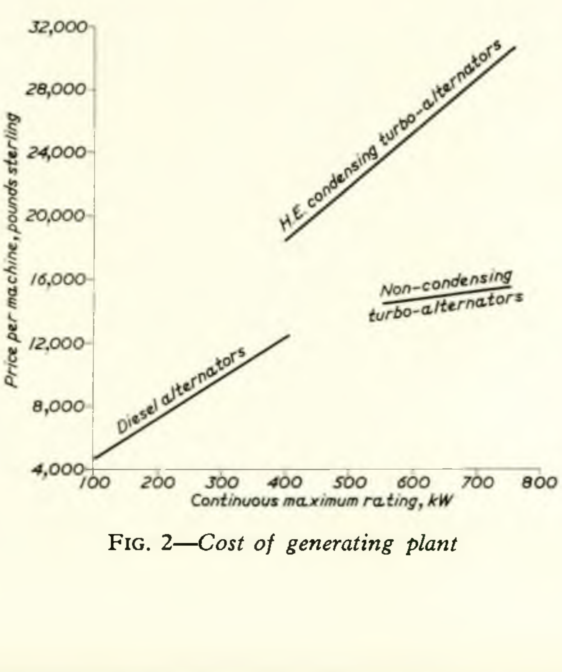
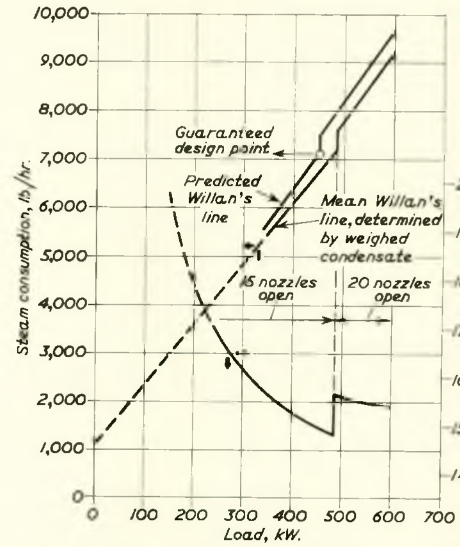


FIG. 2—*Cost of generating plant*

Turbo-generators for Waste Heat Systems

Several manufacturers have produced ranges of steam turbines specially designed for use in motor ships and incorporating features developed from their long experience in building similar machines for steamships. These can operate satisfactorily on saturated steam but, as the expansion of wet steam in a turbine is not ideal and may lead to blade erosion in the last rows, it is desirable to arrange for steam conditions

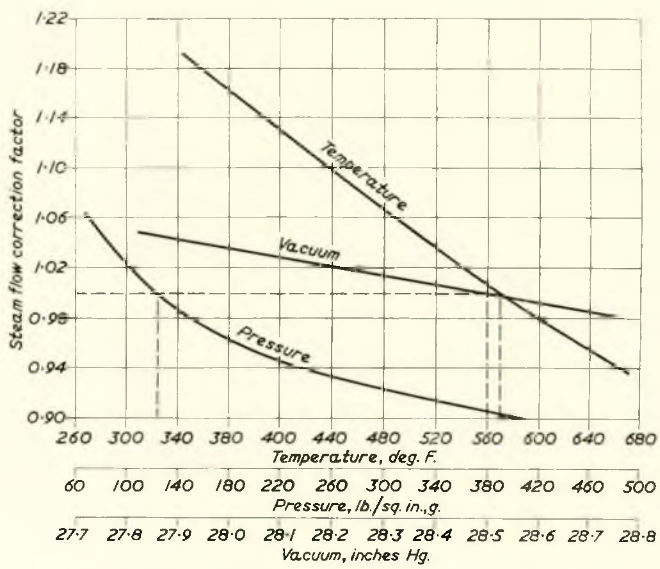


450 kW/600kW. 6,000 r.p.m. Multi-stage turbine

Steam conditions: (at sea)

Inlet pressure	125lb./sq. in. gauge
Inlet temperature	570 deg. F.
Exhaust vacuum	28·5in. Hg.

FIG. 3—*Steam rate of turbo-generator set*



Turbine designed for—

Inlet pressure	125lb./sq. in. gauge
Inlet temperature	570 deg. F.
Exhaust vacuum	28·5in. Hg.

FIG. 4—*Correction factors for turbo-generators steam rate 450 kW./600 kW. 6,000 r.p.m. multi-stage turbine*

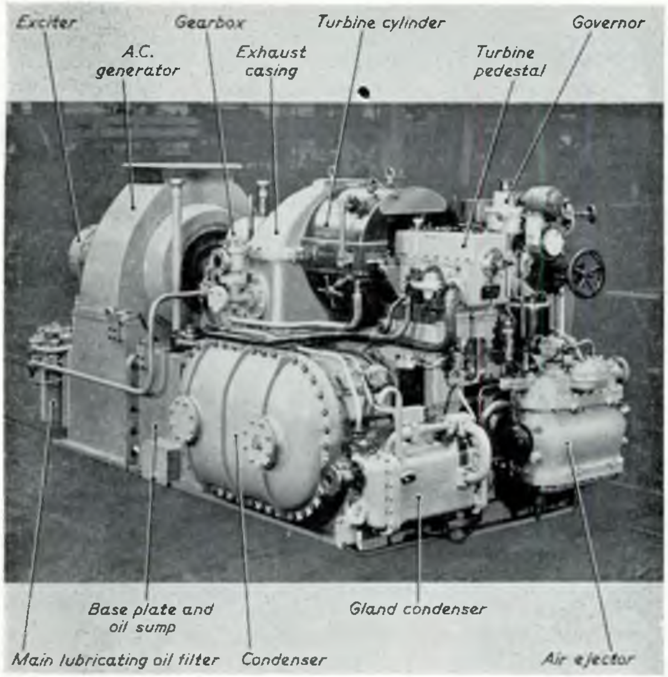


FIG. 5—*Principal components of turbo-generating set*

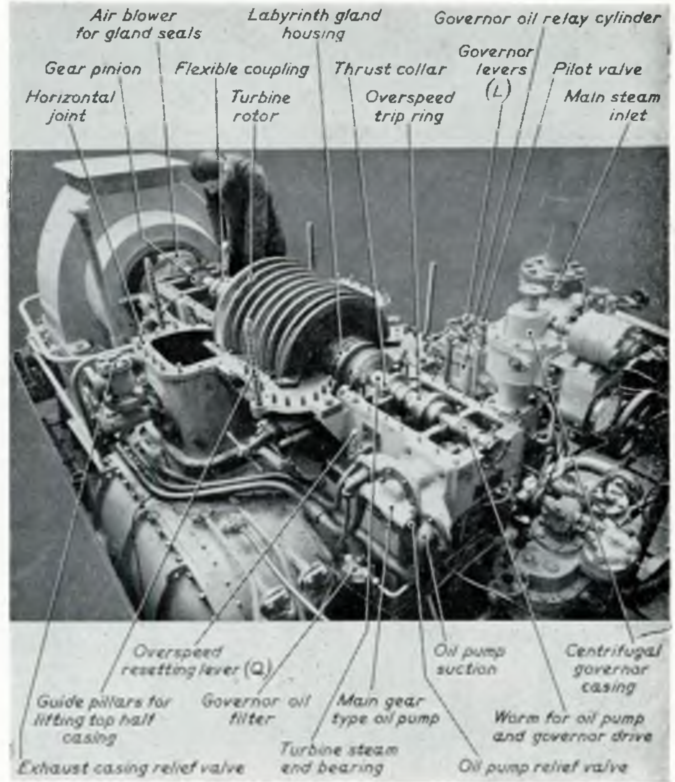


FIG. 6—*Turbine details* (*top half of cover removed*)

which will limit the exhaust moisture content to not more than 12 per cent. It will also improve the efficiency to provide some superheat whenever possible, at least when at sea.

Fig. 3 shows the steam rate obtained from the multi-stage set installed in *British Venture* and Fig. 4 shows a correction curve which can be used for varying conditions. The machine was designed for optimum efficiency at the estimated maximum load of 450 kW, when the ship was operating at sea under tropical conditions, and for the maximum load of 600 kW that would be imposed when manœuvring under the same conditions.

Particular attention is required to the provision of clear and explicit instruction books for this type of machine that may be operated by personnel unfamiliar with turbines. In ships now in service these are supplemented by instruction plates for reference when watchkeeping and typical examples of such reference data are shown in Figs. 5 and 6.

As an alternative, or in addition to, the multi-stage turbines a simpler design with a single wheel turbine may be employed if a much higher steam consumption is acceptable.

EVAPORATING PLANT

A class of 15,500-d.w.t. product carrying tankers was ordered by the company in 1956 and each ship had a separate waste heat boiler and several steam engine-driven auxiliary units. The quantity of exhaust steam substantially exceeded the requirements for the feed heater and it was decided that the best use for this surplus could be made in an evaporator of better design than the submerged coil type then in use.

Flexible Element Evaporators

In steamships the efficient but expensive low pressure distilling plants were firmly established but economics did not justify their use in motor ships. The more desirable attributes of these plants were the flexible heating elements with their ability to shed scale, the weir control of water level and the provision of brine pumps to maintain steady conditions.

TABLE I

m.v. *British Cygnet* — Engine voyage data
On passage from: Isle of Grain — To: Port Said

(Readings once per day)

Day ending noon						
	Day:	25	26	27	28	29
	Month:	10	10	10	10	10
	Year:	1963	1963	1963	1963	1963
Evaporator — hours run			23·5	23	23	6
Evaporator — shell pressure, lb./sq. in.		0	0	0	0	0
Evaporator — coil pressure, lb./sq. in.		10	10	10	10	10
Evaporator — on live or exhaust steam		exh.	exh.	exh.	exh.	exh.
Evaporator — water made, tons		5	10	8	11	3
Evaporator — density thirty seconds		1·5	1·5	1·5	1·5	1·5
Evaporator — water level		½	½	½	½	½
Evaporator — hours between cold water shocking			23	23	23	6
Evaporator — condition of element when sighted				slight scale		

Remarks:—Hours run between descaling 296.
No manual scaling required. Scale brushed off.

A design was evolved which incorporated these attractive features of the steamship units, but was simpler and less costly. The units were arranged to provide an output of 10 tons/24 hr. on exhaust steam and 20 tons/day on live steam and some typical service results, abstracted at random, are shown in Table I.

Flash Evaporators

Large crude oil carriers are expected normally to satisfy all water requirements by distillation and when *British Venture* (36,000 d.w.t.) was designed it was decided to fit an evaporator of the flash type in view of the thermal advantage possible with the multi-stage construction. This is a 25 tons/day unit, to provide 10 tons of make-up feed and 15 tons of potable water, and Fig. 7 shows the flow diagram.

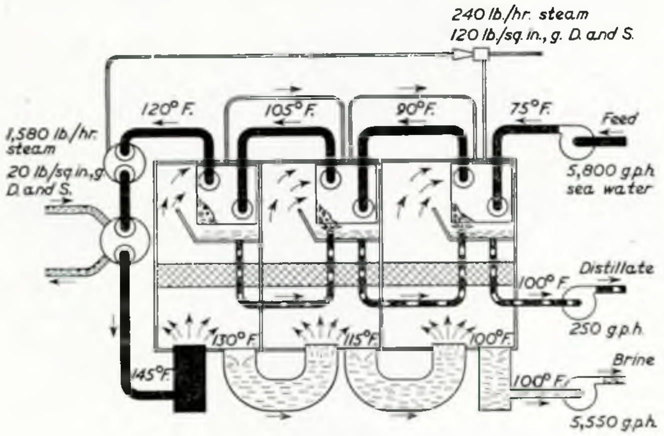


FIG. 7—*Three-stage "flash" evaporator*

Jacket Water Distillers

An ample quantity of waste heat is available in jacket water systems, but the heat is of such a low grade that distillation is almost the only duty in which it can be applied on shipboard. The jacket water distillers are therefore usually of single-stage type. Although widely used elsewhere none are yet in service in the company, but some are on order.

Fig. 8 shows a packaged distiller in which the heat exchangers are of plate type and where an alternative steam heating connexion has been provided in case it should be necessary to produce good quality boiler feed water in emergency under port conditions. Other types of plant, e.g. a unit where the sea water is "flashed" after leaving the engine heat exchangers, are available but space considerations preclude detailed comments on them in this paper.

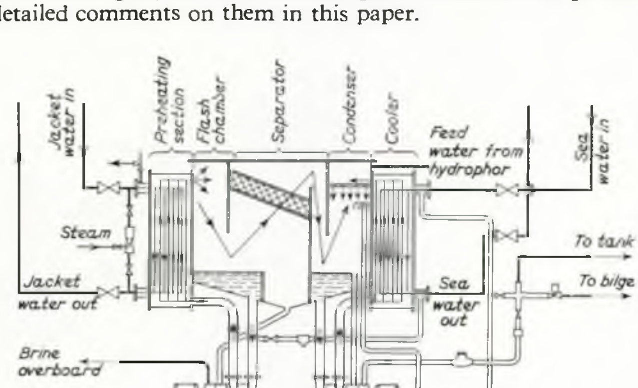


FIG. 8—*Jacket water distillation plant*

GAS/WATER HEAT EXCHANGERS

These may be constructed either as smoke tube boilers, in which the exhaust gases pass through tubes surrounded by the water being evaporated or as economizer-type units where the water is inside the tube and the gases outside. Both types are in service in the company's ships. Alternative gas bypass arrangements and silencers are not provided for these units, thus reducing installation costs and complexity in the exhaust system. These heat exchangers may be operated in a dry condition in emergency and act as spark arresters and silencers at all times.

The cost of these heat exchangers for any given steam production must obviously vary with the mean temperature difference across the heating surfaces and the type of heating surface. Typical costs, therefore, vary from say, 20s.-30s./lb. of steam produced.

Waste Heat Boilers

These offer the advantages of simple construction and low cost since the shell is cylindrical and the tubes extend between the end tube plates, with gas inlet and exhaust ducts shaped to direct the gas flow through the tube bank. Tubes are expanded and sometimes seal-welded into the tube plates. Tube pitching is usually very close which makes manual cleaning, on the water side of tubes, virtually impossible. The effectiveness of the plain tube type is low but a specially shaped type of tube is available which is claimed to increase turbulance of the gases and increase heat transfer.

Of 12 waste heat boilers in service for periods up to six years it has been found necessary to completely re-tube one, owing to repeated tube end leakage, and to make good tube leaks in several others. Owing to the large water content of such boilers it is considered essential to preheat them before the main engine is first started.

Waste Heat Economizers

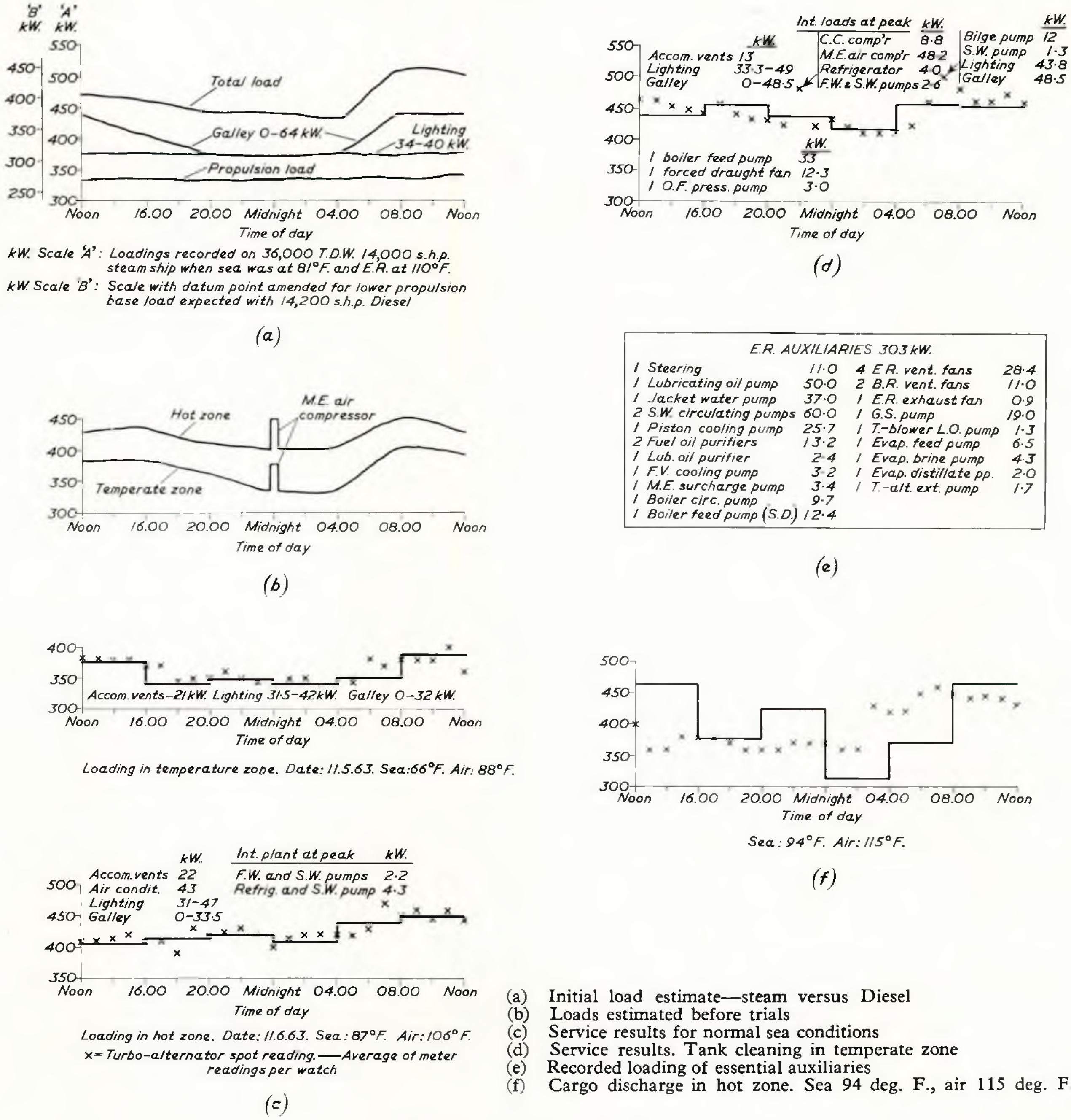
Engineers responsible for the operation of steamships know that the economizers fitted to the boilers need almost no maintenance or repairs, and are extremely reliable. It is thus reasonable to assume that similar units of all-welded design should be used for steam generation, from waste heat in motor ships, on the grounds that reliability should be the criterion rather than first cost.

These economizer type units are usually provided with an efficient extended surface of one of the several types commercially available. The extended surface of the elements can be of all-steel construction throughout if precautions are taken against metal temperature in normal operation being below the acid dew-point of the gases. If it is considered that elements in the lower temperature end of the gas stream may operate at lower than dew-point, then steel tube cast iron clad elements should be used. The elements may be arranged to form either a staggered gas path or a straight-through gas path. The staggered path has the advantage of increased heat transfer due to greater gas turbulence. The straight-through path has the advantage of freedom from restriction with reduced pressure drop and uniform cross-sectional area; this allows easy sighting of surfaces to ensure they remain clean. Water washing arrangements or soot blowers were provided in the first units to enter service, but experience has shown permanent fittings for cleaning duty are not initially necessary; provision is now made in the casing dimensions for later fitting if this is required.

One disadvantage of such units is that the large flat casing sides are liable to vibration due to the exhaust gas pulsation and structure transmitted vibration. This occurred in units of two separate makes and additional stiffening was found necessary in the early service life. In fairness to the designers it must be stated that similar vibrations have been encountered with end casings of waste heat boilers.

METHODS OF ESTIMATING ELECTRICAL LOADING

Whatever the type of auxiliary plant fitted, it is important to make a careful estimate of the auxiliary loading. Some



(a) Initial load estimate—steam versus Diesel
(b) Loads estimated before trials
(c) Service results for normal sea conditions
(d) Service results. Tank cleaning in temperate zone
(e) Recorded loading of essential auxiliaries
(f) Cargo discharge in hot zone. Sea 94 deg. F., air 115 deg. F

FIG. 9—*Variations in electrical loadings for 36,000-d.w.t. tanker with 14,200 s.h.p. Diesel engine*

margin must be allowed and, with memories of how electrical loads in ships have increased in the past as additional equipment was fitted in service, there is a human tendency to make the margin a safe one. If the estimate is too low the service results will be unpleasant; if it is too high then unnecessary capital expenditure will be incurred for larger plant than necessary. Methods of estimating the loading vary between builders but usually involve a calculation involving connected load and a diversity factor; the end result gives a probable load for one sea condition only.

A preferable alternative method is shown in Fig. 9(a). Since similar sized turbine ships of nearly the same power were already at sea, from log book data it was possible to draw graphs to show the electrical load under various service conditions. The base load for the Diesel propulsion auxiliaries was then calculated and the graph scale corrected. Fig. 9 also shows some service loadings which may be of interest.

HEATING SERVICES

If waste heat recovery plant is used to produce steam for power generation in addition to steam for heating purposes, a balancing action may take place, since the relative demands vary with climatic conditions. This reduces the effect of error in estimating steam requirements for hotel services, an estimate which is usually obtained by tabulating requirements and applying a usage factor.

Alternatively, the total requirements for all heating services plus radiation losses may be obtained from records of ships in service and corrected to form a blanket estimate for projected ships. Service results from a number of ships operating

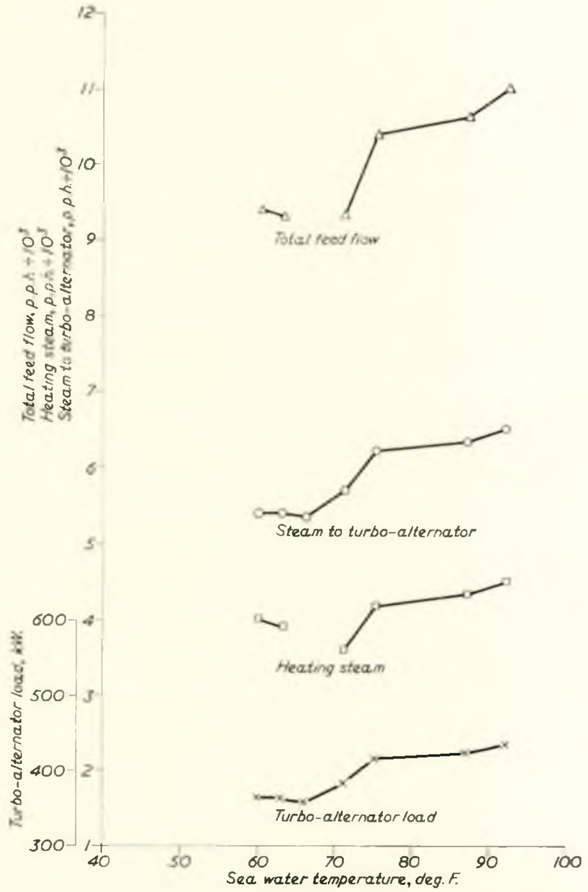


FIG. 10(a)—*36,000 d.w.t. tanker—14,200 (service) b.h.p. Steam and electrical loadings at sea—Daily average readings*

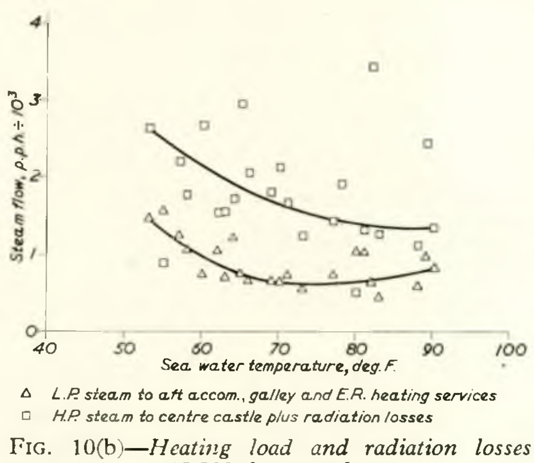


FIG. 10(b)—*Heating load and radiation losses 15,500 d.w.t. tanker*

at about 7,500 s.h.p. indicate that this blanket figure is seldom less than 2,000lb./hr. of steam. Records obtained from heat recovery plants in two ships are given in Fig. 10(a) and (b) and illustrate the range of heating steam requirements.

Exhaust Gas Conditions

Exhaust gas weight and temperature vary considerably between engines of different designs and even successive engines built to the same design may have variations of the order of 5 per cent. Fig. 11 shows detailed test bed results of some engines now in service and since outline test bed data for new designs are usually given in the technical press there is no point in covering too wide a range in this paper. It is suggested that by modifying the various scales shown on the figure, reasonable curves may be obtained for other engine designs than those named. In order to design waste heat recovery plant it is necessary to predetermine the precise operating conditions at the design point. The effect of these on the gas temperatures and weights obtained from test bed results are shown to some extent on the figure and amplified by the following comments.

1) *Engine Rating*: There are very few ships which operate at the builders' designed maximum ratings. The normal service rating at which the ship will operate must be determined.
2) *Engine R.P.M.*: If r.p.m. drop off in service due to hull and propeller surface deterioration—as happens to a marked extent in tankers with their bluff hull forms—gas weight will be reduced and temperature will increase if the original power is maintained.
3) *Tropical Conditions*: Test bed trials are usually carried out at low ambient temperatures which give a good fuel rate. If a constant main engine power is maintained under all conditions the weight of gases will be reduced and the temperature increased when operating under tropical conditions. The effect of temperature is usually seen during sea trials where a 20-30 deg. F. (11-17 deg. C.) increase over the shop test readings is evident. If the shop test has been carried out on Diesel fuel there will also be a rise of 10-20 deg. F. (6-11 deg. C.) when the engine is operated on heavy fuel.

This temperature effect is being exploited to aid heat recovery in two ships which are to be fitted with Sulzer 6RD76 engines and where automatic temperature control at the turbocharger air cooler outlets is being provided to increase the gas temperature under cold weather conditions. The waste heat recovery plant in these two ships is designed for the gas conditions expected in service in the tropics.

EXHAUST GAS ANALYSIS

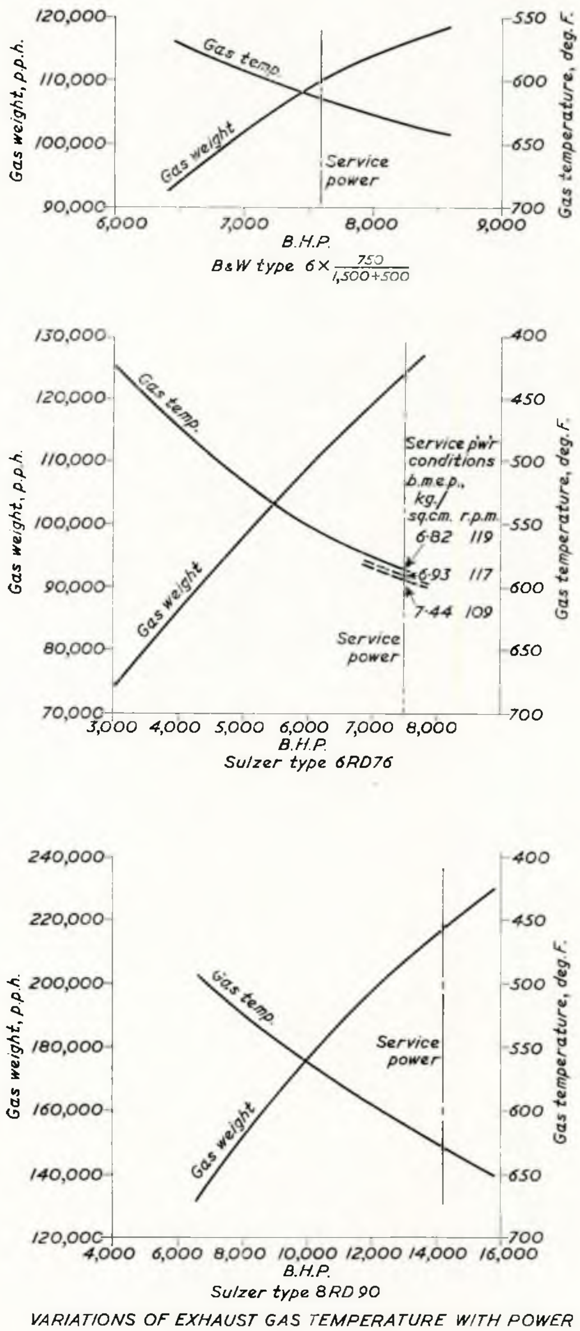
Although the results of some detailed work on small engines is available there has been little published on exhaust gas analysis of large two-stroke turbocharged engines. The effect of this was seen when designers for specialized companies were estimating for Diesel engine waste heat recovery plants and the specific heats used by them varied from 0·24 to 0·26 depending on their optimism. In order to provide data which could be used with confidence for large engines, the company requested the Research Branch of the parent organization to carry out full scale tests with the following main objectives:

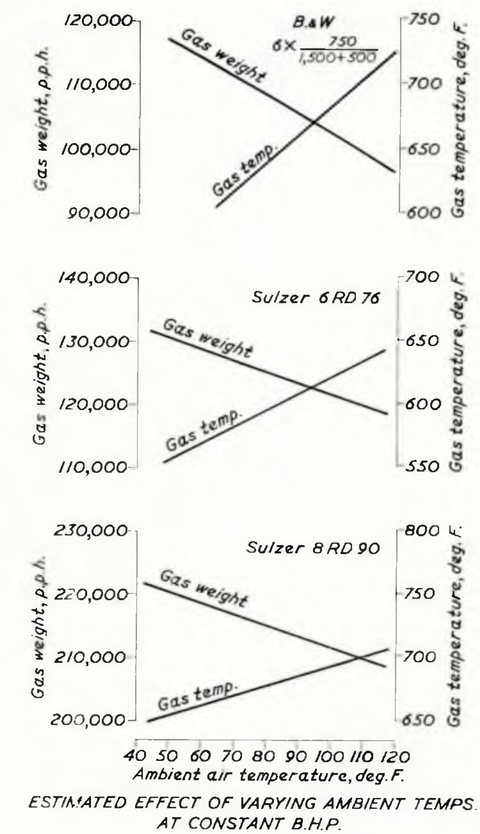
a) To determine the corrosive potential of the exhaust gases and the minimum temperatures which should be maintained in the low temperature zones of waste heat units to prevent excessive corrosion.
b) To make a full analysis of the exhaust gases so that the effective specific heat could be determined for use in calculations of waste heat recovery.

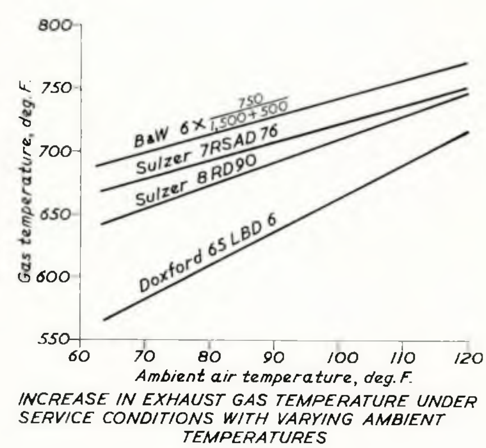
The engine was a six-cylinder 750 mm. bore opposed piston engine of Kincaid/Harland and Wolff/Burmeister and Wain type which was ultimately installed in *British Kestrel*

TABLE II.—INSPECTION DATA OF FUEL

Specific gravity, 60 deg. F.		0·956
Gross calorific value	B.t.u./lb.	18,365
Total sulphur content	per cent wt.	3·35
Kinematic viscosity, at 100 deg. F.	cs.	215·4
,, ,, at 140 deg. F.	cs.	65·08
,, ,, at 210 deg. F.	cs.	16·84
Redwood No. 1 viscosity, 100 deg. F.	sec.	879
Ash content, at 550 deg. C.	per cent wt.	0·030
Water content	per cent vol.	trace
Water + sediment	per cent vol.	trace
Carbon hydrogen ratio		7·60







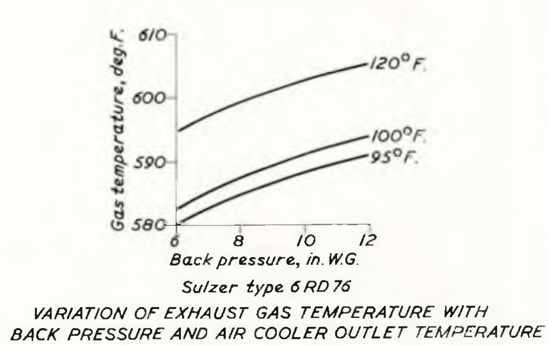


FIG. 11—*Exhaust gas conditions*

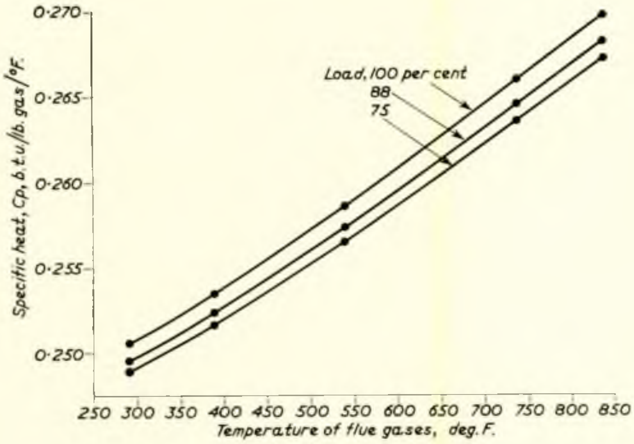


FIG. 12—*Specific heat at various gas temperatures, for three engine loads*

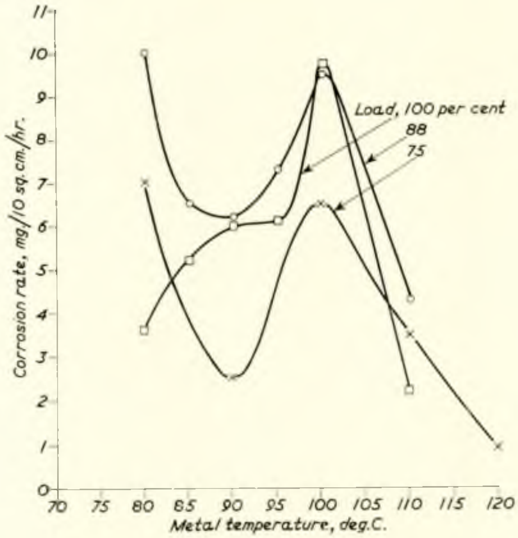
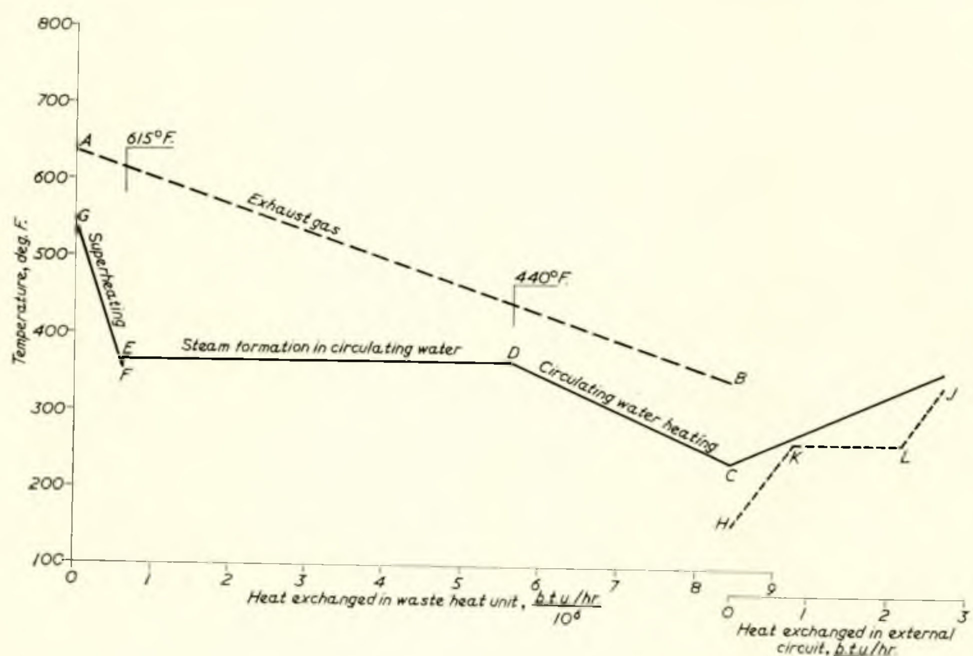


FIG. 13—*Rate of corrosion with varying metal temperature at three engine loads*

but which was specially run for this purpose some time after the normal test bed trials were completed. Details of the fuel used, the corrosion rate and the specific heat are given in Table II and Figs. 12 and 13. The detailed report covered the following features:

c) *Acid Dew-point and Corrosion Rates*: The observed acid dew-point of between 228 and 234 deg. F. (109 and 112 deg. C.) was about the level expected from the type of fuel judging from boiler experience, but rates of acid build-up were fairly high, implying a high rate of conversion of SO_2 to SO_3. The peak iron losses from the probe were considered to be equivalent to a penetration rate in practice of between 0·005 and 0·012 thousandths of an inch per thousand hours, but could be prevented by operating all metal surfaces above, say, 248 deg. F. (120 deg. C.).

d) *Gas Analysis and Specific Heat Calculations*: The method of calculation of specific heat should give a result accurate to within ± 0·002 B.t.u./lb. gas/deg. F. The effect of changes of ambient air temperature on the engine air/fuel ratio and hence on the specific



A	Exhaust gas from main engine	F	Steam from boiler
B	Exhaust gas to funnel	G	Steam at superheater outlet
C	Circulating water at economizer inlet	H	Feed water from pump
D	Circulating water at saturation temperature	J	Feed inlet to boiler
E	Water/steam emulsion to boiler	KL	Evaporation in steam generator

FIG. 14—*Heat transfer in waste heat system for a 6RD76 Sulzer engine operating at 7,500 b.h.p. in tropical conditions. Gas wt. lb./hr. 119,016*

heat of the exhaust gases was calculated to be negligible.

In view of these findings the author considers that unprotected steel heat exchange surface may be safely used in low temperature gas zones, provided that precautions are taken to avoid low metal temperatures occurring. These precautions ensure that water below the gas dew-point temperature cannot enter the heat exchanger and that where electrically-driven scavenge air fans are provided for manœuvring purposes these are operated for a few minutes after the engine has stopped.

HEAT TRANSFER CONSIDERATIONS

Fig. 14 shows the heat transfer which will take place in the waste heat recovery system of a new class of 19,000-d.w.t. tankers. Further reference to this figure will be made in a later section of this paper but in the meantime it is only being used to illustrate the basic heat transfer considerations which apply, whatever the type of waste heat unit used. The comments directly refer to a forced circulation unit of economizer type and the changes of phrase to make them apply to a waste heat boiler of smoke tube type are deliberately omitted to avoid complexity of wording.

The heat transfer conditions in a waste heat unit are rather different to those in an oil fired boiler where the air/fuel ratio is adjusted to give good combustion with minimum excess air and high furnace temperature. The steam thus produced in a fired boiler is related directly to the amount of fuel burnt and, in an efficient watertube boiler, the flue gases can readily be reduced to 300 deg. F. (149 deg. C.) by fitting a feed heating economizer. In comparison the Diesel engine exhaust gas weight and temperature and the heat available in the gas (at a steady engine power) at entry to the waste heat unit is constant and is independent of the steam demand.

To effect heat transfer between the exhaust gas and the water/steam streams in the heat exchanger, the gas temperature at any point must obviously be the higher of the two and there must be some minimum temperature difference, at which the heat transfer rate becomes so low that the cost of providing further heat exchange surface is not justified. The rate of heat transfer at any point must also depend upon:

i) The temperature difference.
ii) The extent of fouling on gas and water sides.
iii) The resistance of the tube metal.
iv) Gas velocity and distribution over the heating surfaces.

Referring to Fig. 14, point D, generally referred to as a "pinch point" has the minimum temperature difference mentioned above. Heat transferred from the gas to the left of this point will form steam bubbles in the circulating water; the steam is later separated from the water/steam emulsion in the boiler drum. Heat remaining in the gas to the right of point D can only be utilized in raising the circulating water, or feed water, to the saturation temperature corresponding to the pressure.

Operating Steam Pressure for Waste Heat Recovery Systems

The optimum steam pressure at which waste heat recovery systems should operate is a subject of controversy. Some authorities contend that a pressure of about 70lb./sq. in. gauge gives the minimum overall cost for the heat exchanger and the turbo-alternator set. It is considered that the optimum is mainly related to the cost of heat exchange surface and the low pressure premise is only true if the cheaper smoke tube type waste heat boiler is employed.

The author prefers a pressure of about 125lb./sq. in. gauge at the turbine, for tanker service, despite the higher cost, for the following reasons:

1) The corresponding boiler pressure of 140lb./sq. in. gauge allows any surplus steam to be diverted to other steam-using equipment on board, for example, deck machinery for handling ropes in and out of storage, forehold fuel transfer pumps and forward bunker heating.
2) A reserve of steam capacity is available to meet steam surges. If the plant is normally operated at its minimum pressure, the engineer will be tempted to light boiler fires at the first sign of any pressure fall. This will adversely affect fuel economy.

At lower pressures the steam rate of the turbo-generator will increase but, if a single pressure steam system is used, a greater weight of steam can be produced from the waste heat unit. For example, reverting to Fig. 14, if steam at 25lb./sq. in. gauge only was being produced, then line ED on the figure would be redrawn at a lower position corresponding to the temperature of 267 deg. F. (131 deg. C.) and point D would, relatively, move to the right of its former position. A greater weight of steam would thus be produced. Since the amount of heat required to raise water to saturation temperature is much less than the latent heat, it will be seen that if near-maximum heat recovery is required it is necessary to adopt either a very low steam pressure or a two-pressure steam system.

Within narrower limits, provided that the turbine throttle valve is not fully open initially, a similar action to the foregoing will occur in service if the steam demand increases, i.e. the system pressure will fall until a balance is again obtained. Under such conditions the volume of water contained in the working boiler acts as a heat bank for the system and will reduce the speed at which pressure changes occur with changes in steam demand, i.e., the boiler acts as a steam accumulator when the system is self-sustaining. Balancing effects under reduced steam demand have the effect of increasing the steam pressure unless controls are provided.

In considering this boiler heat bank concept, the heat added to the boiler contents is represented on the figure by the distance E-D and the bulk of the heat abstracted from the boiler is by steam withdrawal. Further heat is abstracted from the boiler water content in raising the incoming feed water to saturation temperature and since the boiler heat input (E-D) available is fixed for any steady pressure, the effect of low feed temperature is either to reduce the steam output or reduce the steam pressure. In terms of steam output, the effect of supplying cold feed water can reduce the maximum boiler evaporation by an amount approaching 30 per cent.

A further balancing action is superimposed due to the rise in temperature of the exhaust gases which occurs in hot weather. Although this is accompanied by a reduction in weight of the gases, it enables a greater weight of steam to be produced. This offsets the increase in turbo-alternator steam rate per kW hr. which follows the reduced vacuum in the condenser of the machine.

HEAT RECOVERY CIRCUITS—EXHAUST GAS

There are many possible circuits, each with its advantages and disadvantages, and the particular circuit adopted for an installation should be chosen to provide the correct economic solution. Various circuits are shown in Figs. 15 and 16.

Natural Circulation Boilers

The natural circulation smoke tube type boiler shown in Fig. 15(a) provides the simplest, the most widely used and usually the cheapest possible installation. It can be located at a high level in the engine room or in the funnel and is widely used in cargo ships where only a limited amount of heat recovery is required to satisfy heating services at sea. This popular design has some minor disadvantages, which are also applicable to units in some of the other circuits later described. For example the weight of the boiler and contained water is large for the output and preheating arrangements are necessary to avoid thermal stressing when the boiler is brought into circuit. Output control may be effected by providing gas bypass arrangements and a silencer in parallel with the boiler or by using a variable water level with the obvious consequent disadvantages. Introducing cold feed water directly into the boiler drum reduces steam output and may lead to scale formation on the heating surfaces; in view of the low mean temperature difference across the surfaces, such scale may cause a marked reduction of output.

There are a number of variations on this simple circuit which can be used to overcome certain of the disadvantages but which involve cost increases. For example, external ducting can be arranged to give a downflow gas path and the feed water introduced at the bottom of the boiler drum: it is claimed that the contraflow thus effected enables the final gas temperature to be reduced to near the steam temperature. Alternatively a thimble tube construction can be used which changes the basic design to a watertube type. This reduces the demand for overhauling space and improves the circulation in the (short) tubes in which all the surface is used. Output control can also be effected by arranging a movable plug type damper in the central gas passage and positioning the damper to constrain the gas to pass over the tube area required to give fractional outputs.

If only small quantities of steam are needed for port operation a composite boiler may be used. While the exhaust gas and oil fired heating surfaces can be contained in a single boiler shell, the gas passages must be kept entirely separate to comply with classification requirements.

Simple Forced Circulation Systems

The adoption of a simple forced circulation system, overcomes many of the disadvantages of the circuit in Fig. 15(a). A typical circuit is shown in Fig. 15(b). By taking the circulating pump suction from near the bottom of the main boiler drum, the boiler is maintained in a condition suitable for immediate lighting off if necessary. The pump is usually arranged to circulate water at about ten times the steam production rate of the waste heat boiler, so improving heat transfer therein. The steam/water emulsion leaves the waste heat boiler through either one of two valves arranged at different levels. This gives a coarse control of output, by varying the effective heat transfer surface in the waste heat boiler, and a fine control

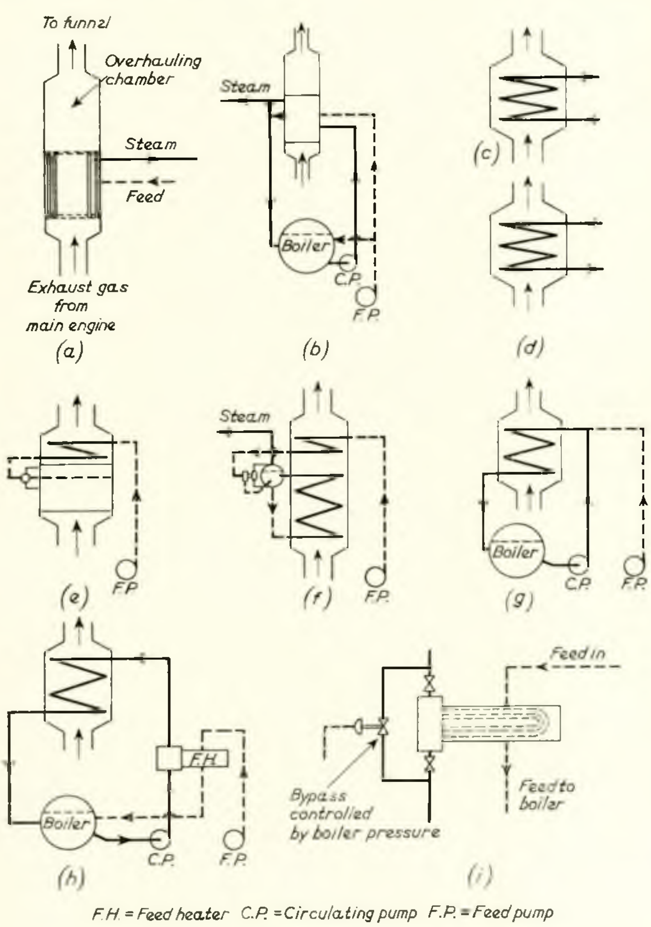


FIG. 15—*Exhaust gas heat recovery circuits*

can be obtained by operating the circulating pump bypass (not shown). The steam/water emulsion is discharged into the water space of the main boiler drum and the steam bubbles separate out in exactly the same manner as if the boiler was being fired. If it is necessary to isolate the main boiler at any time, the waste heat boiler is arranged to be operated in the same way as the unit described in the circuit in Fig. 15(a) with the output controlled by hand control of the check valve to vary the water level. The feed water is normally delivered to the water drum of the main boiler. This is advantageous in that any scale-forming impurities present in the feed may be neutralized by the chemical water treatment now usually applied to all boilers. Since the main boiler is normally acting as a steam reservoir only at sea, the impurities which separate out may be removed through the boiler blow-down without settling on the heating surfaces.

In operating Scotch boilers and smoke tube type waste heat boilers in such circuits, it has been found that a pressure difference of about 20lb./sq. in. normally exists between the two. When the Scotch boiler is oil fired to increase steam production the difference is decreased. This is considered to be mainly due to:

i) radiation losses from the Scotch boiler decreasing the temperature and therefore the pressure of the contents;
ii) the heat losses from the boiler contents when raising the feed water to saturation temperature.

Due allowance for this should be made when estimating the mean temperature difference in the waste heat boiler and the steam pressure expected in the main boiler drum. A similar forced circulation system is also employed with economizer type waste heat boilers and the equivalent connexions are shown in Fig. 15(c). Parallel flow of gas and water is usually arranged and this reduces the possibility of vapour locking occurring if the feed flow only is passed through the economizer unit. If the circulation rate is a multiple of the feed flow, the advantages of using contraflow as shown in Fig. 15(d) are evident from the preceding comments in this paragraph.

Feed Heating Circuits

If maximum steam production is required from any exhaust gas heat recovery system, the feed water should be raised to near saturation temperature before entry to the steam generating section. The reasons for this have previously been outlined but, in view of the importance of this basic truth, they will be re-stated in a different form.

At constant main engine power conditions, similar to those in a ship at sea, the heat content of the exhaust gases is also constant and only part of the heat can be used to produce steam at a reasonable pressure. The temperature drop of the exhaust gases over this steam-producing range is the difference between the inlet to the waste heat unit and the outlet from the steam producing section, i.e. the section in which heat transfer from the gases is sufficient to form steam bubbles in the water content of the unit. Even if an infinite area of heat exchange surface was provided the exhaust gases could not be reduced to the same temperature as the steam because of the resistance to heat flow of any stagnant film of gas on the heat exchange surface, the resistance of the tube metal and the resistance of any scale on the water side of the tubes.

The outlet temperature of the exhaust gases from the steam producing section must therefore be equal to the steam temperature therein plus the temperature difference necessary to effect heat transfer. With a boiler as shown in Fig. 15(a), the feed water entering the boiler mixes with the contents and abstracts heat therefrom until the temperature levels are the same. The colder the feed water is on entry, the lower the mean temperature level, boiler pressure and steam output become.

If the feed water heating section is separated from the steam producing section, a gas/water temperature difference will obviously be available to effect sensible heat transfer. If the feed water is heated to saturation temperature before entering the boiler section only latent heat will be required and the steam output will be at maximum.

Several ways of effecting this feed heating are shown in Fig. 15, and it will be seen that all employ gas/water contraflow in the feed heating section.

Feed Heating Economizers

The circuits in Fig. 15(e) and Fig. 15(f) are similar and superficially attractive, but in the opinion of the author, incorporate undesirable features. It will be seen that the economizer section is arranged exactly the same as in an oil fired watertube boiler plant, with the feed pump discharging through the economizer to the feed regulator or feed check valve, to the boiler drum. This is satisfactory in steamships where the heat release in the boiler furnace is proportional to steam demand and feed flow: any interruption of feed flow has to be rapidly cleared to safeguard the boiler. Conditions in a motor ship are different in that the gas heat quantity is constant at any steady power and that the steam demand will fluctuate with the time of day and the climatic conditions: any change in feed flow is reflected in the economizer feed outlet temperature. If this temperature is kept low for normal operation then more heat is abstracted from the boiler contents at entry and steam production suffers: if it is kept high then danger of vapour locking will be present when gas/water contraflow is provided and downflow of water is arranged.

Unstable conditions may therefore occur in an economizer circuit as shown in Fig. 15(e) and Fig. 15(f) if the feed regulator valve is closed due to manual action or automatic control in the event of a sudden steam surge causing an apparent increase of water level in the boiler. Vapour will form in the economizer elements once the temperature of the water therein rises to that corresponding to saturation at feed pump discharge pressure. Subsequent opening of the feed regulator will reduce the pressure in the economizer section and part of the water content may flash into steam. The increased rate of feed flow necessary to restore the boiler water level may cause thermal shock due to rapid cooling of the economizer elements.

In the event of such fluctuating conditions occurring, perhaps due to the not unknown event of a sluggish feed regulator, the sequential passage of steam, hot water and cold water form a potential basis for water hammer action taking place. It is known that a number of fairly new ships is at sea with such circuits operating without mishap and it must be repeated that the preceding comments express an opinion only.

If the feed water in this type of circuit is not preheated the economizer element metal temperature near the water inlet point may fall below the gas dew-point. In addition to the possible attack on the metal surface of sulphurous compounds, the probability of these providing the starting point for a build-up of deposits leading to fouling of the gas passages must not be overlooked.

Feed Heating by Admixture

The circuit in Fig. 15(g) employs a method of feed heating by admixture of feed and circulating water. In comparison with the circuits in Fig. 15(e) and Fig. 15(f), the circuit in Fig. 15(g) has the advantages that the feed water is normally preheated before entering the gas water heat exchanger and complete stoppage of the flow through the unit is unlikely to occur. It retains some of the disadvantages of the other circuits in that any solid impurities in the feed water may deposit in the small bore watertubes so impairing heat transfer and necessitating more frequent cleaning of the unit. Air, CO_2 or other gases entrained with the feed water may attack the tube surfaces, probably near the inlet end of the tube elements when the gases are liberated from the water following heating. The quality of feed water in a motor ship is, of course, not expected to be as high as in a steamship where precautions are taken to de-aerate, to keep the CO_2 content of make-up feed low, and to maintain a suitable *p*H level.

This type of circuit also has the following general disadvantages:

a) The mixing arrangements require careful attention. If centrifugal pumps are used, as nearly always, the feed pump and circulating pump characteristics require careful matching over a wide feed flow range to avoid the potential danger of successive slugs of hot and cold water following each other through the combined pipeline. If the maximum boiler working pressure is low, the matching of pump characteristics is much easier than if a harbour service pressure of say 300lb./sq. in. gauge is used and a lower pressure is used when operating the waste heat system. This is because the circulating pump gives a differential pressure only, whereas the feed pump must overcome maximum boiler pressure plus circuit resistance. Despite this difficulty the use of centrifugal pumps is most desirable. A reciprocating circulating pump is not well suited to the high and varying temperature of water coming from a boiler drum. A reciprocating feed pump could give intermittent flow with a loss of heat recovery potential during non-flow periods.

b) Unless complicated flow controllers are provided the circulating/feed ratios are fixed at the design point; if feed flow is more than design the inlet temperature to the economizer may be too low to safeguard the heat exchange surface on the gas side from deposits and attack. Because of this the first few rows of economizer elements, at least, should be cast iron clad. The total flow through the economizer will vary with steam output.

c) Boiler feed level control has to be indirect.

d) The amount of heat transfer between the circulating and feed water on admixture is limited to that required to raise the feed to the mean mixture temperature, which in turn controls the amount of cooling effected to the circulating water.

Indirect Feed Heating

The circuit in Fig. 15(h) shows the indirect feed heating system preferred by the author. It is obviously more costly than the other circuits described because of the inclusion of the water/water feed heater. The apparent anomaly of taking heat out of the circulating water in the feed heater and replacing the heat in the gas/water heat exchanger is justified in that thereby the circuit has none of the disadvantages of the other circuits described.

The indirect heat exchange ensures that only water from the boiler drum is passed through the gas/water heat exchanger. This water has been freed from solid and gaseous impurities during its sojourn in the drum; the chemical treatment of the boiler water in the drum is usually adequate to ensure that scale formation in the heat exchanger is eliminated. Any deposition or gas release from the boiler feed water which may occur during heating to near saturation temperature takes place in the feed heater. This heater may easily be isolated from the circuit for cleaning without interference with the bulk of the heat recovery from the exhaust gases.

Considerable flexibility is available with the circuit in Fig. 15(h), for example:

1) Because the feed heater area is fixed the extent of heat exchange is limited and the circulating water inlet temperature to the economizer cannot be reduced to a low level if feed flow exceeds design. This safeguards the economizer elements which can be of mild steel throughout.

2) The flow through the economizer is normally fixed at the optimum and is independent of feed flow. Some variation can be effected, however, by initially making the circulation pump larger than necessary and operating the bypass (not shown) to increase the flow if necessary. This can be useful if feed flow exceeds design—as happens when, say, deck machinery is being used at sea for handling ropes—or if an additional water/water heat exchanger has to be installed in the circuit for some special purpose at a later date.

3) Since the equivalent of the feed heating process and the steam generating process is a continuous action—as it also is in Fig. 15(g)—the whole surface of the gas/water heat exchanger is used to its full extent. This improves the thermal storage capacity of the system at light loads which is limited only by the gas entering temperature and the boiler safety valve setting, i.e., the circuit has a rising pressure characteristic at falling loads.

4) Control of heat extraction from the circulating system, and hence from the gases can easily be effected; the arrangement used is shown in Fig. 15(i). If the boiler pressure approaches the safety valve setting, the bypass valve in the circulating water line is automatically opened. This increases the temperature of water at the economizer inlet and decreases the amount of heat abstracted from the exhaust gases because of the reduced mean temperature difference between them. At the same time the feed heating potential is reduced because of the reduction in circulating water flow through the feed heater. The colder feed water entering the boiler abstracts heat from the boiler contents and further reduces the pressure. Alternatively a similar bypass may be fitted in the feed line to give the same control features.

Dual Pressure Steam Systems

The use of a two-pressure steam system is obviously only justified when a single-pressure system will not give adequate heat recovery to enable normal ship's power and heating requirements to be satisfied without recourse to burning additional fuel. The cost of a two-pressure system will be higher because of the added equipment on the steam generating side and the greater surface area necessary in heat exchangers using the low pressure steam.

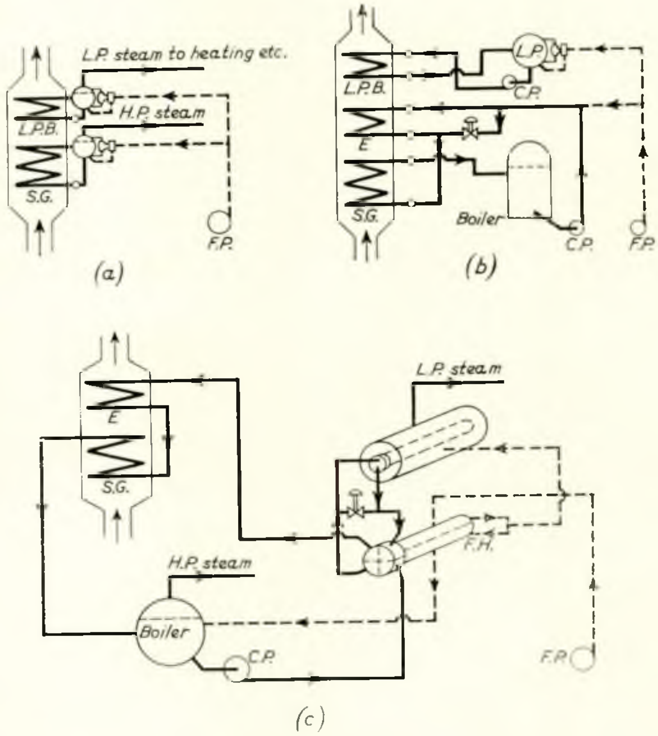
Since several grades of heat are required in a ship at sea the steam supply problem may be divided into two parts, the first being to provide high pressure, high temperature steam to operate a turbo-generator which will carry all the electrical load. The second and much simpler part of the problem is to provide low pressure steam for heating services. There are several circuits which will provide near the practicable maximum of heat recovery from exhaust gases. The essential feature is usually to obtain near maximum output from the high pressure steam system, as the residual heat in the gases after this duty has been performed can easily be exploited.

Comments have already been made on the optimum steam pressure for power generation, but slightly different considerations apply in choosing the operating steam pressure for heating services. The essential feature is to protect the gas/water heat exchanger from possible damage at the low temperature end of the gas stream. There is little experience of low temperature operation in motor ships but much experience has been gained in steamships. In the direct fired boilers used in steamships, the optimum temperatures which are often commercially accepted for economizers, or other heat recovery units in the lower temperature end of the gas stream, are an outlet (tail end) temperature of 320 deg. F. (160 deg. C.) for the gas and a water inlet (entering) temperature of 240 deg. F. (116 deg. C.) giving a gas/water temperature differential of 80 deg. F. (44 deg. C.) These temperatures, on the one hand, avoid an unduly large heat exchange area due to low temperature differences and, on the other hand, avoid the metal temperature in the heat exchanger being so low that the gases may be locally reduced below the dew-point, due to flow stratification or pocket formation. In view of this steamship experience it is reasonable to expect that temperatures of this order should be acceptable for waste heat recovery systems in motor ships and should be borne in mind when considering the low steam pressure system. The lower the pressure, the lower will be the cost of the gas/water heat exchanger due to the mean temperature difference across the heating surface. Since the quantity of steam required for heating services is low, in relation to that required for power generation, the use of low pressure will not unduly affect the cost of pipelines. Much of the hotel services equipment is normally arranged for operation with low pressure steam and the cost is scarcely affected by steam pressure change. It is considered by the author that the minimum pressure in the low pressure steam system should be 25lb./sq. in. gauge, as this allows all engine room and hotel heating services—except for final heating of fuel to the main engine or boilers—to be supplied and to maintain a reasonable temperature difference in the heat exchangers.

Some typical double pressure circuits are shown in Fig. 16, and many of the advantages and disadvantages of components have already been dealt with in detail. The following comments are therefore mainly applicable to the additional features incorporated to provide a dual pressure system, with a working pressure of 25lb./sq. in. gauge, in the low pressure steam system.

Separate L.P. Boiler

Fig. 16(a) shows a separate low pressure boiler of water-tube type with an external steam drum and the comments are also generally applicable to a natural circulation boiler of smoke tube type. The advantages and disadvantages are generally as described for single pressure circuits (Fig. 15(a), (b) and c)). With L.P. steam drum pressure of 25lb./sq. in. gauge 267 deg. F. (131 deg. C.)—the metal temperature in the outlet gas zone is at a safe level while steam is being generated. At low engine powers, for example when manœuvring, if steam is not being generated, the pressure in the steam drum will fall and the metal temperature may enter the gas acid dew-point range. Arrangements will normally be made to supplement the L.P. steam range from the H.P. range, but are not shown in the diagram. If this is done then the pressure fall will only be due to radiation losses but, nevertheless, it is desirable to isolate such units under transient power conditions. The L.P. heat exchange elements in the gas stream should pre-



L.P.B. = Low pressure boiler E = Economizer S.G. = Steam generator
F.H. = Feed heater C.P. = Circulating pump F.P. = Feed pump

FIG. 16—*Dual pressure steam systems*

ferably be of steel tube, cast iron clad type, to delay the effect of acid attack. While heat may be transferred from H.P. to L.P. systems in such circuits the reverse action is impossible. The heating surface in each section must therefore be designed for the maximum steam demand expected.

Forced Circulation L.P. Boiler

The circuit in Fig. 16(b) shows a separate L.P. boiler drum with a forced circulation system using economizer type elements in the gas/water heat exchanger. Gas/water contraflow is arranged to enable the gas temperature, for equal L.P. steam pressure, to be lower than that possible for the circuit in Fig. 16(a) where the steam pressure (and hence temperature) in the heating elements must be slightly higher than in the drum to allow the steam to flow.

In the circuit shown in Fig. 16(b) the water inlet temperature to the L.P. economizer/steam generator elements is often considered as being the same as that of the water in the steam drum. This is only true if no stratification of circulated hot water and of cold feed water occurs in the steam drum. Particular care should be given to mixing arrangements to avoid, on the one hand, wet steam being provided with possible pitting attack on the drum at normal water level, which will follow if the feed water is sprayed into the steam space and, on the other hand, excessive cooling at the circulating water take-off point. The main advantage of this forced circulation L.P. unit is that it allows the position of the steam collecting drum to be varied to suit the machinery arrangement.

The H.P. system shown in Fig. 16(b) is one of the several possible variants which can be devised to give small advantages by lowering water inlet temperatures and so improving mean temperature difference between gas and water. One disadvantage of using such separate heat exchange sections in the gas stream is that a header is necessary at each entry and exit point, to distribute the flow in the multi-pass elements of economizer type units.

Indirect L.P. Steam Generation

Fig. 16(c) shows the double pressure steam system used by the author's company for some ships operating at a service power of about 7,500 s.h.p. As stated for circuit in Fig. 15(h) it has the apparent anomaly of taking heat out of the circulating water and replacing it in the gas/water heat exchanger. This is again justified in that thereby the circuit has none of the disadvantages of others described and can give the maximum practicable heat recovery simply by inserting additional units in the circulating system without introducing complexity into the more expensive gas/water heat exchanger unit.

The heating services and power requirements on shipboard vary with climatic conditions. In cold weather the heating (L.P.) steam requirements are above average and power demands (H.P. steam) are below average. In hot weather the conditions are reversed owing to ambient temperature increase and the use of air conditioning plant and greater demand for coolant services. In Fig. 16(c) the forced circulation economizer unit can be considered as a single feed heating and steam raising entity in which the division between the functions is widely variable. Some interchangeability of total surface area between the equivalent of L.P. and H.P. sections exists. The effect of this can best be visualized by analogy with Fig. 14. If the horizontal distance between A and B represents a continuous feed heating and steam generating economizer element, then D would represent the point on the element where the first steam bubbles formed in the circulating water. Due to the interchangeability of the surface area, point D would move to the left under cold weather steam demand conditions and to the right under hot weather steam demand conditions.

Two feed water heaters, contained in a single shell, are shown in Fig. 16(c). These are provided to meet the needs of both primary and secondary steam systems and contribute both to maximum heat recovery and maximum output or steam pressure in the system. The low temperature first stage heater reduces the circulating water temperature to extend heat extraction from the exhaust gases and the second stage raises the boiler feed water temperature to near saturation point to provide maximum output/pressure from the H.P. steam system.

In this instance the L.P. steam generator is interposed in the system between the two feed heaters because the grade of heat required to produce L.P. steam is intermediate between those required for the feed heating duty. The heat exchange in this external circuit is shown at H, K, L, J in Fig. 14. The position of the units in circuit can, in the design stage, be varied to suit requirements. One advantage of the circuit shown is that, by isolating the second stage feed heater, the L.P. steam pressure or output can be increased due to the higher initial temperature of the circulating water. Both H.P. and L.P. steam systems have a rising characteristic at falling loads. Control of maximum pressures is effected by means of an automatic bypass valve as described for the circuit shown in Fig. 15(h) and by arranging a similar bypass to control the L.P. steam generator pressure. In practice these valves are arranged so that in the event of a high main boiler pressure the one on the L.P. steam generator opens first, so leading to H.P. steam being passed through a reducing valve (not shown) to supplement the reduced output of the L.P. steam generator which will ensue: the second valve operates only if the main boiler pressure continues to rise and this second valve is arranged to bypass the feed water direct to the boilers. The overall effect of this is to give control over a wide range.

COMPOUND HEAT RECOVERY CIRCUITS

The power and heating requirements on shipboard are normally such that they can both be supplied by steam generated from exhaust gases. Since the heat from the coolant systems is of such a low grade it can seldom be usefully absorbed elsewhere than in distillation plants. For example, since the jacket water outlet temperature from the main engine is normally about 150 deg. F. (66 deg. C.) the maximum temperature of a secondary circuit in indirect heat exchange would be limited to about 140 deg. F. (60 deg. C.). Such a secondary circuit could be used for accommodation heating but would be more complex than the usual steam heating arrangement. The temperature would, of course, be too low for cargo heating services in a tanker since a further indirect heat exchange is involved with corresponding temperature degredation. The direct use of jacket water in external heat dissipation circuits is undesirable because of the larger water volume required in circuit and the possibility of water contamination.

Proposals have been put forward to use the main engine coolant heat source to preheat water before entry to a gas/water heat exchanger and thereafter use the high temperature, high pressure water in external heat dissipation circuits. Such installations would be of value in small powered ships or for special installations where the heating load is high, but the author does not know of such plant being used in large ships. A Russian proposal for a compound heat recovery circuit to produce steam only is shown in Fig. 17(a); this circuit would involve cylinder wall lubrication problems due to high coolant temperature and the problem of providing water treatment compatible with both engine and boiler protection.

Compound heat recovery circuits are widely used in land installations where, in addition to power generation, large space heating and process heating requirements are usual. This has caused greater attention to be paid to maximum heat recovery circuits and overall thermal efficiencies of about 80 per cent have been claimed. Despite the basic simplicity of obtaining maximum heat recovery once a large external low temperature heat dissipation requirement is superimposed on the power generation plant, many patented circuits have been evolved. The most complicated circuit proposed had a Diesel engine, gas turbine, oil fired boiler and steam turbine all connected one to the other to obtain maximum heat recovery: fortunately for operating engineers this system has not yet been used. A relatively simple circuit providing three systems at different temperature levels is shown in Fig. 17(b).

A further recent development in land work has been the

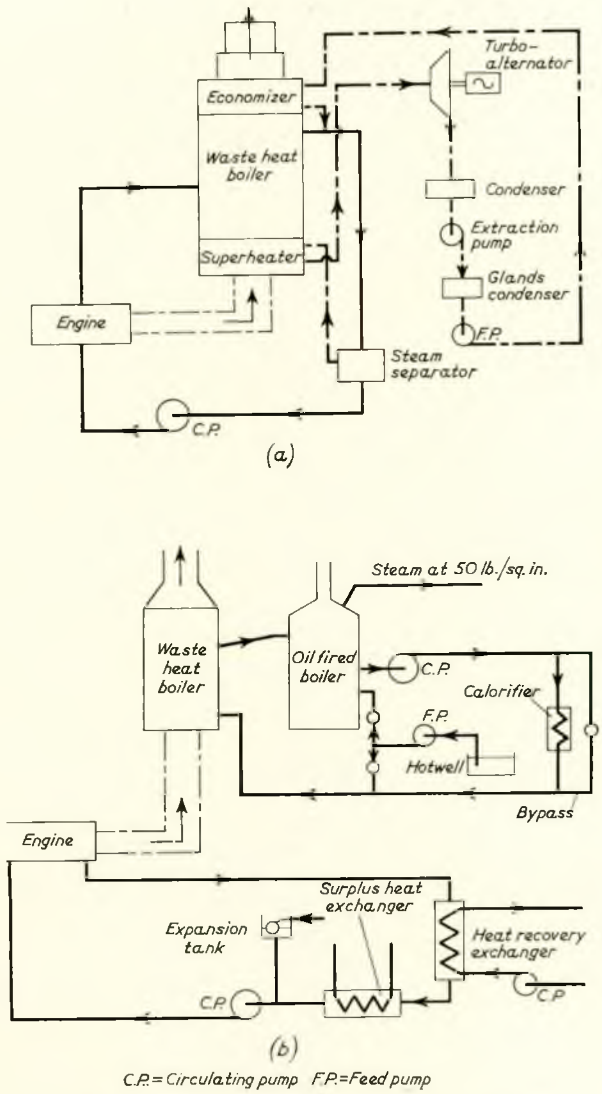


FIG. 17—*Compound heat recovery circuits*

use of vapour phase, or latent heat, cooling. In this application the jacket water system is at a pressure of about 15lb./sq. in. gauge at the engine with a jacket outlet temperature of 250 deg. F. (121 deg. C.). An elevated steam separator is necessary and water circulation may be either thermo-syphonic or forced circulation. The steam which is liberated in the steam separator may be used in external space heating circuits, in distillation plant or for power generation and the final cooling is effected by condensing the steam. The condensate has, of course, to be ultimately returned to the system. The steam produced may be supplemented by the output from a waste heat boiler to supply a steam turbine-driven fan supplying air blast to a condenser. Such installations are expensive and mainly used in isolated locations where fuel supply costs are very high. They are not suitable for shipboard application where heat radiation from the jackets presents a problem in enclosed engine rooms and where an inexhaustible supply of coolant is available from the sea.

FUTURE DEVELOPMENTS OF WASTE HEAT PLANT

While developments must be governed by overall economic considerations, one attractive line which is being pursued is to evolve means of using the gas/water heat exchanger for steam generation in port by passing hot gases from a combustor over the heat exchanger surface. A further possible development is to improve the reliability of main engines of the usual pressure charge type by supplying scavenge air from a steam turbine-driven blower, the steam being derived from the waste heat system.

CONCLUSIONS

Nearly all motor ships require some form of waste heat recovery plant but the extent and type of plant will vary with the ship and its trading pattern. It is suggested that a graded and careful examination of the potential economics of heat recovery plant should be made for all ships which require more than 6,000-7,000 s.h.p. for more than 200 days/year. At lesser powers it is probable that the answer found will be that only heating services and water distillation requirements at sea should be met from heat recovery services. As powers progressively become greater the case for also providing power from heat recovery plant will become so strong that economics will forbid it being ignored. It is probable that it will soon be unusual for a large powered motor ship to be built without at least one turbo-generator being provided to utilize the steam produced from waste heat plant to meet all normal power requirements at sea.

APPENDIX A

A TYPICAL DEVELOPMENT

The result of investigations into the factors discussed will indicate auxiliary plants that will differ with the emphasis that is placed on certain points. The number of possible alternatives is large. Figs. 18 and 19, which reproduce the Guidance Diagrams issued for the principal circuits in a class of 7,500 b.h.p. service power tankers which are being built for the company, are therefore given to show one solution. The principal units in circuit are:

Turbo-alternators

Two are fitted, one being a multi-stage machine which will normally be in operation at all times, and the other a single wheel turbine driving the auxiliary set which will be in use during manœuvring periods and when cargo discharge at over 50 per cent of the maximum rate is being carried out.

Evaporator

In the earlier ships the evaporator is of low steam pressure, flexible element type. In later ships a jacket water plant is installed.

Waste Heat Unit

This is of the economizer type with extended surface. In ships fitted with a heat recovery circuit generally as in Fig. 16(c) the economizer elements are of all-steel type. In other ships where another design is used, cast iron protection is provided in the low pressure section of the unit. Only one boiler circulating pump is provided and this is of the "canned" type.

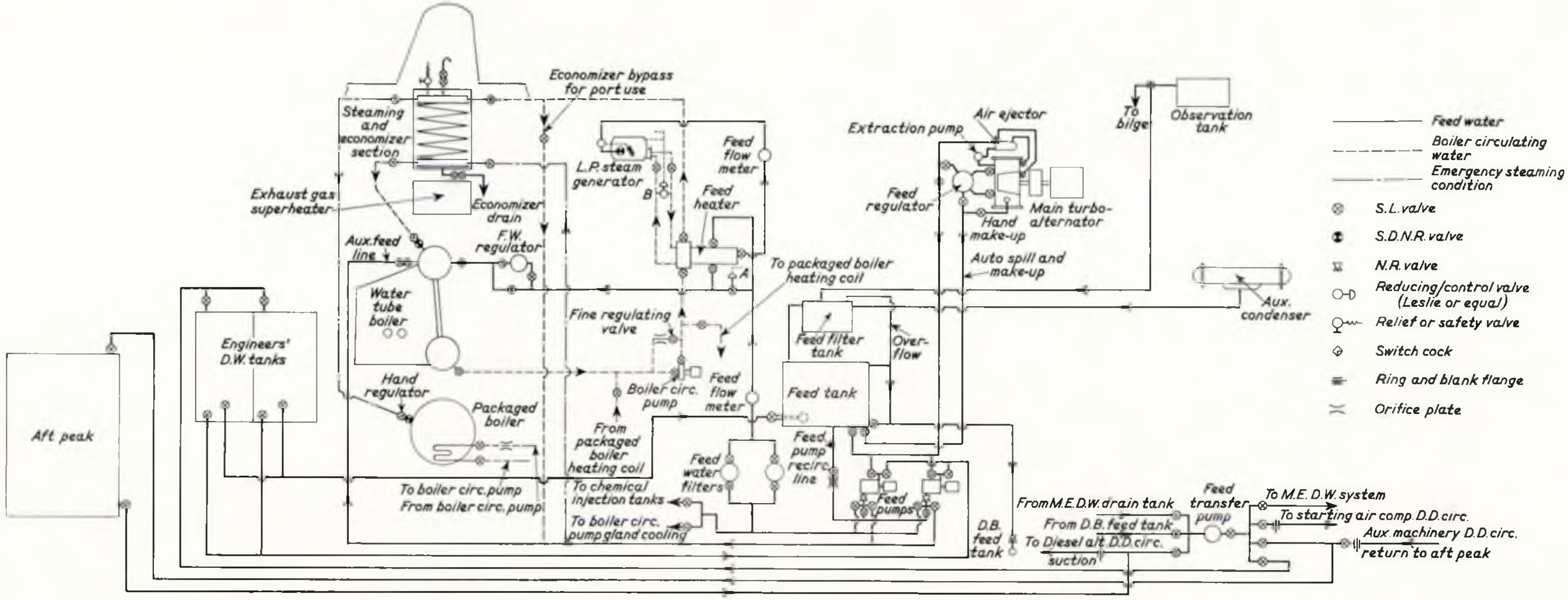


FIG. 18—*Diagrammatic arrangement of boiler feed water system—19,000-d.w.t. 7,500 b.h.p. motor ship*

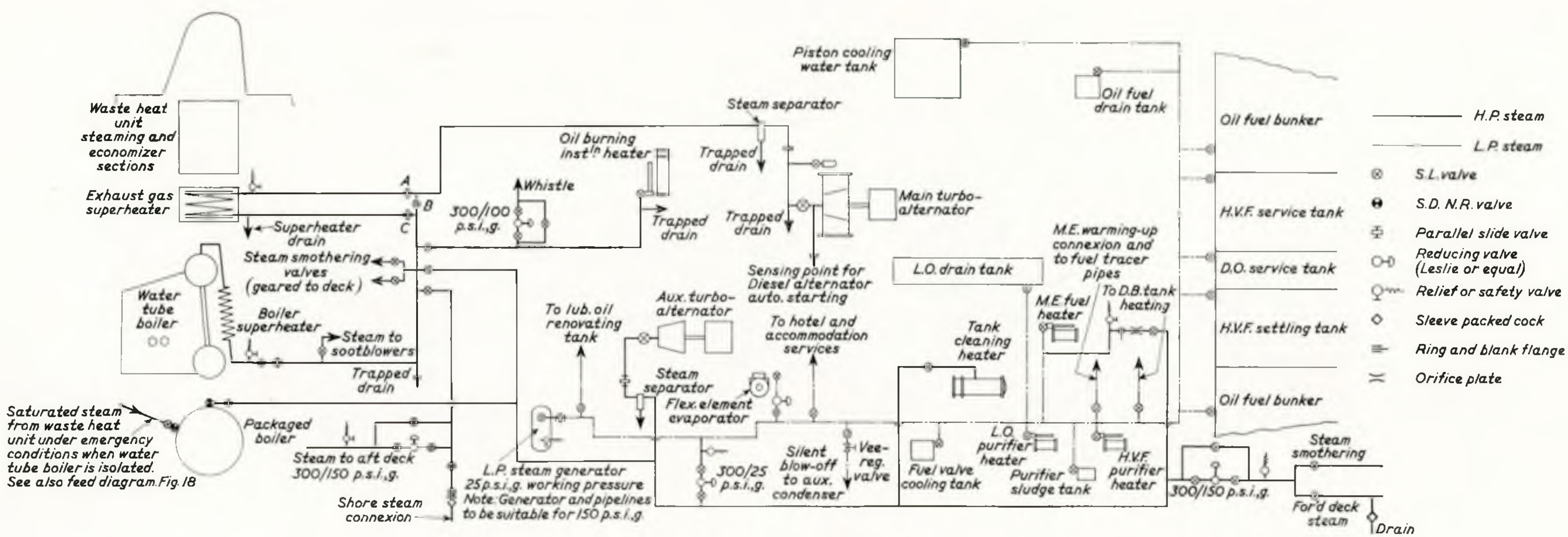


FIG. 19—*Diagrammatic arrangement of auxiliary steam principal connexions—19,000-d.w.t. 7,500 b.h.p. motor ship*

Appendix B

SERVICE RESULTS

M.V. British Kestrel (*15,500 d.w.t., 7,600 service s.h.p.*)

This ship was fitted with a prototype exhaust gas heat recovery plant arranged as shown in Fig. 16(c). One 400 kW C.M.R. turbo-alternator, arranged for optimum efficiency at 320 kW was fitted and the waste heat plant design conditions were:

H.P. steam at superheater outlet—5,900lb./hr. at 130lb./sq. in. gauge, 580 deg. F. (304 deg. C.).

L.P. steam at steam generator—1,850lb./hr. at 25lb./sq. in. gauge, D.S.

Feed water temperature to boiler = 346 deg. F. (174 deg. C.). To steam generator = 263 deg. F. (128 deg. C.).

Main engine exhaust gas—120,000lb./hr. at 650 deg. F. (343 deg. C.), reduced to 320 deg. F. (160 deg. C.).

The connected heating load for the hotel services was 2,697lb./hr., distribution being:

	Amidships H.P. steam	Aft L.P. steam
Air Conditioning, lb./hr.	434	577
Calorifiers and radiators, lb./hr.	639	683
Galley and pantrys, lb./h.	35	339

Engine room heating services connected to L.P. steam system.

Figs. 10(b), 20, 21, 22, 23, show service loadings. One feed flow meter was fitted in the main feed line and a second unit in a branch line led to the L.P. water/steam generator. The steam flow to the turbo-alternator set has been estimated from shop trial results. The total heating steam load has been deduced from the main feed flow meter reading less the estimated steam to the turbo-set. Thus the top curve in Fig. 10(b) shows an estimated amount and the bottom curve, the meter reading in the L.P. steam generator line. Fig. 20 includes a curve for total heating steam plus radiation and other parasitic losses. The amidships steam plus radiation and other losses may be deduced from the other information shown in Figs. 21 and 22. Excepting for Fig. 23, the readings were taken when the flexible element steam heated evaporator was out of circuit and when the waste heat system was self-sustaining, i.e. no fuel was being burnt for auxiliary services. A large scatter of readings is shown in Fig. 20, but it was considered reasonable to show this scatter as indicative of the range of outputs under changing temperature conditions. The curves shown

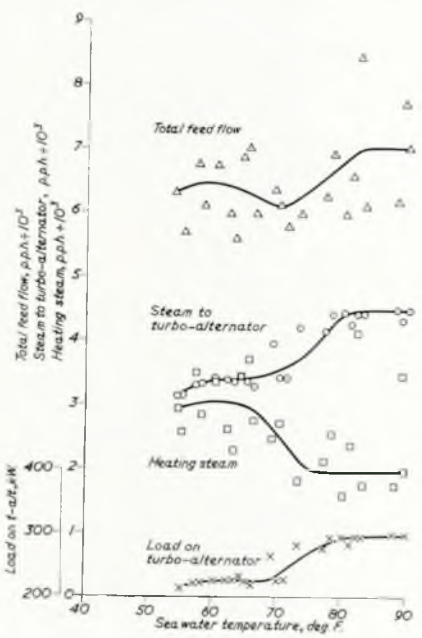


FIG. 20—*15,500 d.w.t. tanker—7,600 (service) s.h.p. Steam and electrical loadings at sea—Daily average readings*

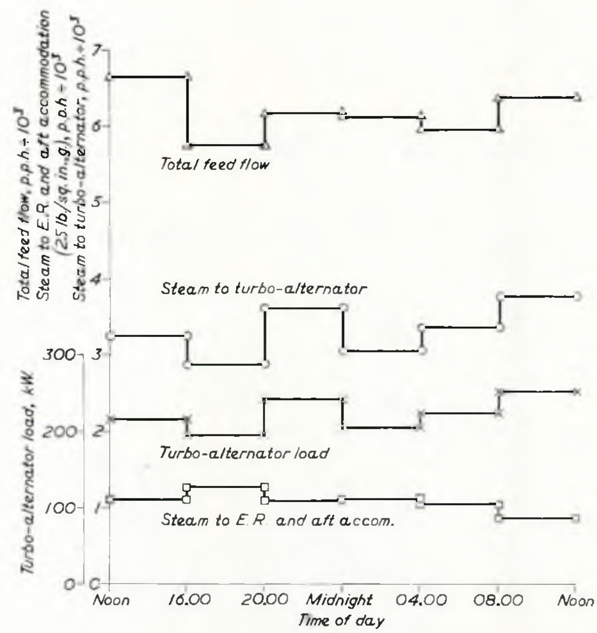


FIG. 21—*15,500-d.w.t. tanker—7,600 (service) s.h.p. steam and electrical loadings at sea. Loading in temperate zone*

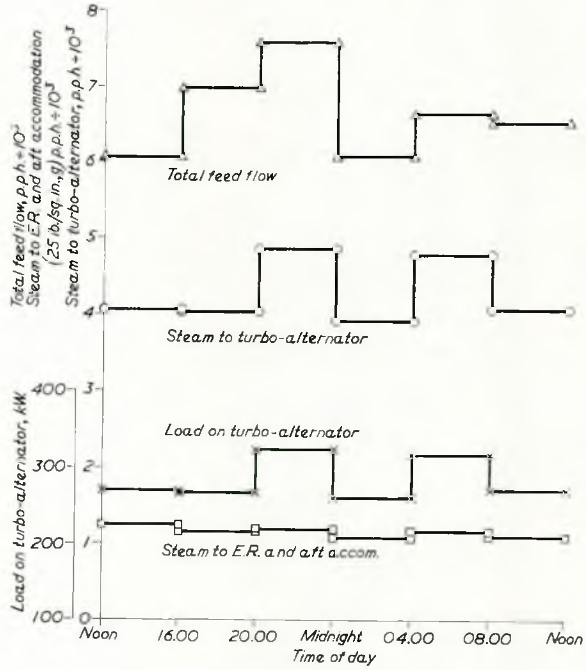


FIG. 22—*15,500-d.w.t. tanker—7,600 (service) s.h.p. steam and electrical loadings at sea. Loading in hot zone*

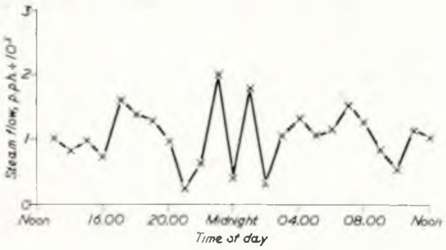


FIG. 23—*15,500-d.w.t. tanker. L.P. steam flow with evaporator in use. Sea temperature 71 deg. F.*

have been superimposed to show the general trend of readings.

The significant feature on all these figures is the large amount that must be apportioned to radiation and other losses. Allowance must be made for this "missing quantity" when estimating loading. Failure to do so may have been responsible for unsuccessful earlier attempts to provide self-sustaining waste heat systems.

Throughout the first year of service of the ship the turbo-alternator was stopped for about one day only. The average sea load for world wide trading, including winter months on U.K./Scandinavian services, was 245 kW.

Eleven similar installations are on order for a new class of 19,000-d.w.t. ships.

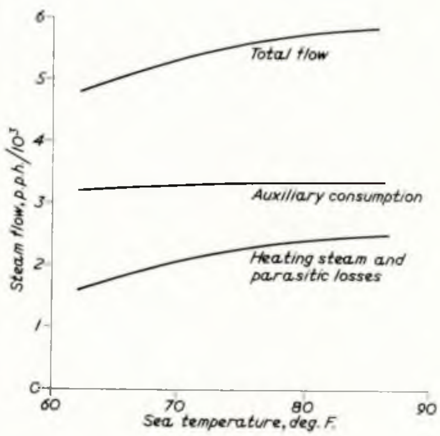


FIG. 24—*15,500-d.w.t. tanker with conventional auxiliaries. Steam distribution*

15,500-d.w.t., 7,600 s.h.p. Ship with Conventional Auxiliaries

Fig. 24 shows readings taken on another ship of the 15,500-d.w.t. class. The total flow was taken from flowmeter readings and the auxiliary steam consumption deduced from loading of a steam engine-driven generator. Attention is again drawn to the large heating steam and parasitic losses. This ship has an extensive engine room steam pipe range and a waste heat system as shown in Fig. 15(b).

M.V. British Venture *(36,000 d.w.t., 14,200 s.h.p.)*

The waste heat recovery system in this ship is as shown in Fig. 15(h). One 600 kW turbo-alternator set (see Figs. 3, 4, 5, 6), arranged for optimum efficiency at 450 kW load, is fitted and a "flash" evaporator (see Fig. 7) is in circuit at all times at sea. The waste heat plant design conditions were:

Steam at superheater outlet	—	7,300lb./hr. at 130lb./sq. in. gauge, 580 deg. F. (304 deg. C.)
	=	125lb./sq. in. gauge, 570 deg. F. (299 deg. C.) to T/A.
Steam to heating services and evaporator	=	4,200lb./hr. at 142lb./sq. in. gauge, D.S.
Feed water temperature to boiler	=	342 deg. F. (172 deg. C.)
Main engine exhaust gas (design)	=	220,400lb./hr. at 653 deg. F. (347 deg. C.) reduced to 392 deg. F. (200 deg. C.)
Main engine exhaust gas (shop trial)	=	217,200lb./hr., 630 deg. F. (332 deg. C.)

The ship is built with all-aft accommodation and the connected heating load for the hotel services is about 3,000lb./hr., plus a total of 1,740lb./hr. of steam allowed for the evaporator at full output. An early estimate of steam requirements, at the conditions given, gave:

Turbo-alternator at 390 kW (cold weather), lb./hr.	6,180
Turbo-alternator at 450 kW (hot weather), lb./hr.	7,130
Evaporator air ejector, lb./hr.	220
Evaporator heating (corrected for pressure), lb./hr.	1,490
Fuel and L.O. heating, main engine, lb./hr.	540
Calorifiers and galley, lb./hr.	550
Accommodation heating, lb./hr.	1,300
Bunker heating, lb./hr.	750
Radiation loss at 5 per cent of total, lb./hr.	550

Figs. 9, 10(a), 25 and 26 show readings taken during service. The total feed flow readings are taken from a meter, the turbo-alternator steam flow estimated from shop test results and the heating steam, including that for the evaporator, deduced from the difference between the meter reading and T/A estimate.

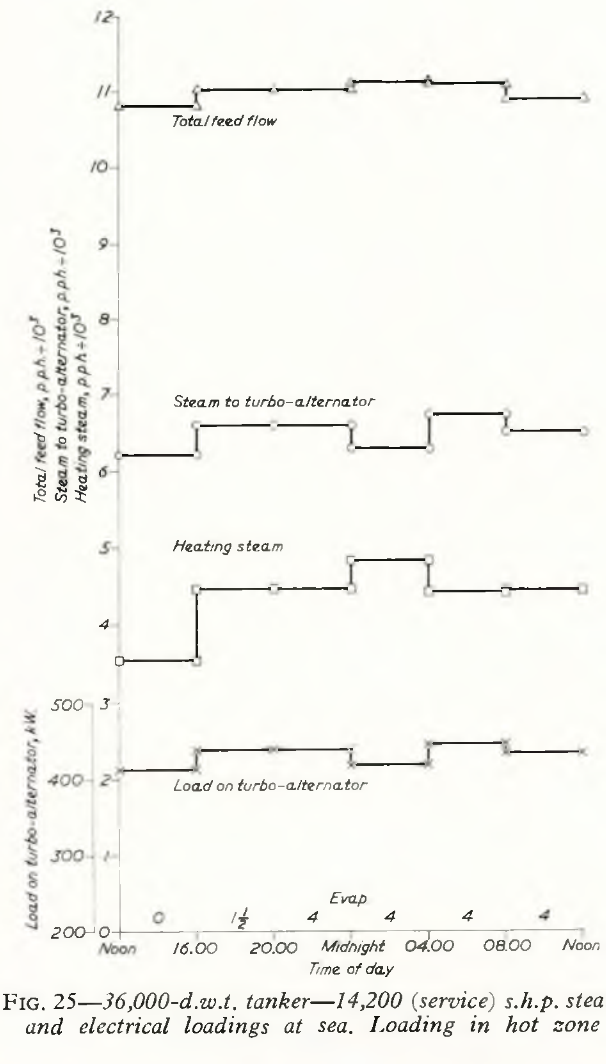


FIG. 25—*36,000-d.w.t. tanker—14,200 (service) s.h.p. steam and electrical loadings at sea. Loading in hot zone*

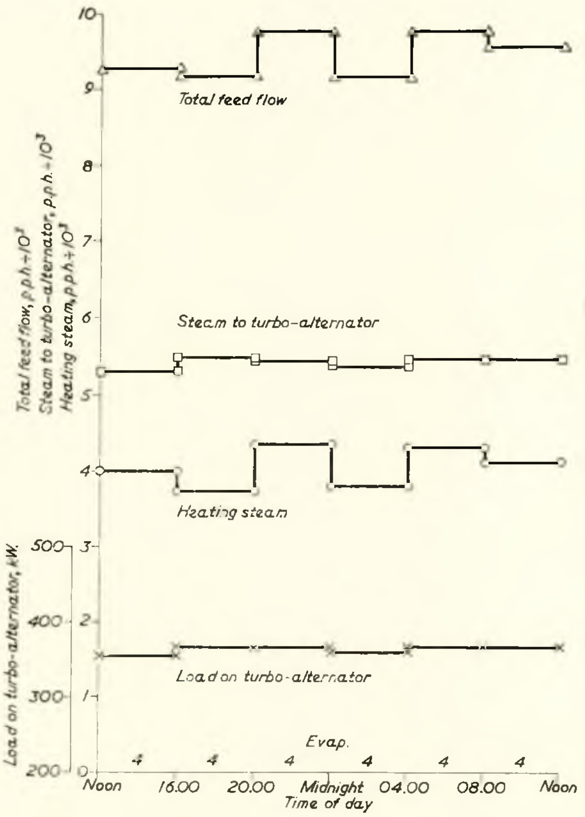


FIG. 26—*36,000-d.w.t. tanker—14,200 (service) s.h.p. steam and electrical loadings at sea. Loading in temperate zone*

Average sea load over the first nine months' service was 408 kW. The chief engineer then advised that the only time the turbo-alternator had to be stopped was to remake two joints in the steam supply line. He stated the average boiler make-up feed was about 8½ tons/day, the main engine water losses about 3 tons/day, and the evaporator output range from 15 to 24 tons/day.

ACKNOWLEDGEMENTS

The author wishes to thank the management of BP Tanker Co. Ltd. for permission to publish the service results given in this paper. Thanks are also due to members of the Engine Design Section of the company for assistance given in preparation of the figures shown in the paper, and to the chief engineers of the ships for collecting the data so used. It is also appropriate to thank the three firms that supplied material for presentation of Figs. 3, 4, 5, 6, 7 and 8.

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Discussion

Mr. E. G. Hutchings, B.Sc. (Member) said that his first reaction on being asked to take part in the discussion of Mr. Norris' paper had been one of confusion, as he and the author had been good friends for a long time, but he had never really agreed with Mr. Norris on many of his ideas on the subject. As he had read the paper his spirits had fallen as step by step the author had carefully shot down most of his own (Mr. Hutchings') hobby horses and had built up an extremely strong case for his point of view.

After the initial shock to his own ego had passed, he was therefore delighted to be given the honour and privilege not only of taking part, but in actually opening the discussion on what was, in his opinion, an extremely valuable contribution to marine engineering.

The author was to be congratulated not only on his knowledge and study of the subject, but also on the clear and concise way in which he had covered a fantastic amount of ground in a comparatively short paper.

A lot was heard about " package machinery " in respect of steam ships. He thought this an ill-chosen phrase, but he believed that most people understood what it meant, which was the investigation of each single piece of equipment with its fellows in order to obtain the best possible compromise between initial cost, efficiency and maintenance for the particular service. In fact, Mr. Norris, in his paper, had carried this approach most effectively into the realms of waste heat plant for motor ships. In many respects this was a more difficult problem than that of main propulsion machinery in a steamship, although the results might not be as spectacular to the uninformed.

The author had not only outlined the train of thought which had led up to certain particular installations, but he had included in his paper a fund of information which would prove invaluable to others who wished to give proper consideration to the problem for their own particular application.

Before passing the floor to others who would no doubt comment on the many and varied points in the paper, he said there were a few specific remarks he would like to make. Referring to the graph which Mr. Norris had shown but which was not in the paper, he said that if he understood it correctly it indicated the average cost of a pound of steam. If he had done his arithmetic correctly, the cost for 13,200lb./hr. was £1 per pound; the cost for 14,200lb./hr. was £15,400, an increase of 8 per cent. This might not sound very much, but if one worked it out each extra pound of steam generated would cost 115 per cent more than the average cost of the first 13,000, which was a lot of money to pay.

The author had frequently referred to " waste heat economizers " and usually meant by that phrase a watertube heat exchanger. Many such heat exchangers could readily be constructed as self-contained boilers by the simple, but perhaps expensive, expedient of adding a steam drum in the appropriate places. Indeed, such a plant was shown in Figs. 15(f) and 16(a), and referred to elsewhere in the paper.

The author's remarks on acid dew-point and corrosion rates were interesting, as Fig. 13 showed that the peak corrosion rate occurred at about 210-215 deg. F. (99-102 deg. C.), while the acid dew-point was stated as being only about 15-20 deg. F. (8-11 deg. C.) higher, at 228-230 deg. F. (109-110 deg. C.). Mr. Hutchings was interested to know whether the corrosion rates shown were for cast iron or mild steel. In any case, this was not in line with oil fired steam boiler experience, where peak corrosion rates occurred some 40 deg. F. (22 deg. C.) below the acid dew-point with mild steel and some 80 deg. F. (44 deg. C.) below the dew-point for cast iron.

The author's remarks on page 406 with regard to the difference in pressure between the waste heat boiler and the Scotch boiler, were difficult to understand and if, as the author had stated, the Scotch boiler operated some 20lb./sq. in. below the pressure of the waste heat boiler which was discharging into it, he suggested that this was for some other reasons than those stated, although he could not think what they would be.

There was one particular problem in the design of water tube waste heat boilers which only merited one paragraph on page 400, and that was casing vibration. This was a very difficult problem since if the designer used circular casing the heating surface and the maintenance thereof became complicated. On the other hand, if flat casings were adopted a more attractive plant resulted, but casing vibration was a much more serious possibility. To design flat casings so that there was no risk whatsoever of vibration was extremely expensive and, consequently, any supplier who tried this would price himself out of the market. However, if a more reasonable design were offered and vibration did occur, who was to pay for it, as the cure might cost somebody (be it the owner, the shipbuilder or the supplier) a considerable amount of money, sometimes as much as the heat exchange unit. He said he would be interested in the author's views on this problem. A similar problem occurred on page 407 with regard to the effect of sluggish feed creating conditions ideal for water hammer. In several waste heat boilers the water level was varied in order to control the output, and this theoretically might result in fatigue and leakage. In other words, several of the waste heat units which were on the market today had built into them features which were theoretically undesirable, but suppliers seemed to get away with it. He said he would appreciate Mr. Norris' remarks on this.

In conclusion he said that the paper had proved most valuable to him personally and had cleared away a lot of false impressions he had had. He was sure that many others would learn a great deal from it. The Institute owed a strong debt of gratitude to Mr. Norris for the very searching examination he had made of the subject and the clear manner in which he had presented his findings.

Mr. P. Alsen said that as he was in agreement with Mr. Norris on most of what he had said in the paper, he preferred, instead of discussing the paper in detail, to give some views on waste heat systems used by his company.

Talking about steam engines and turbines, he said that round about 1950 they used to have a small waste heat boiler producing steam for heating purposes, especially to heat the heavy fuel, which the late J. Lamb had shown could be used for main engines. The tankers at that time also had a small steam engine-driven generator to cover harbour requirements. At sea, two Diesel generators, each big enough to take the load, were used. If it happened that one of the Diesels was damaged,

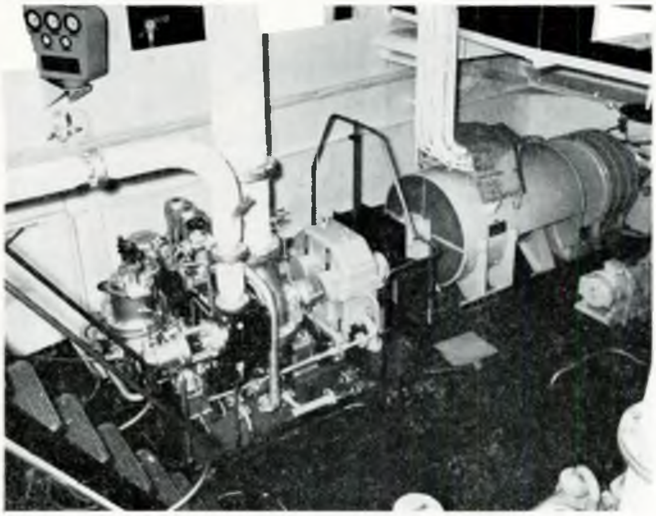


FIG. 27

a great deal of responsibility fell on the other one. Therefore, the steam engine generator grew in size. Furthermore, with surplus steam from the exhaust boiler, it was natural to keep the steam generator running also at sea.

The next step was to substitute the steam engine with a steam turbine. This arrangement with two Diesel generators and one steam turbine generator had been their standard arrangement for many years. With one Diesel and the steam turbine running in parallel, a comparison between the noisy, fuel-consuming and laborious Diesel and the smooth running turbine fed with free steam was not always in favour of the Diesel. Thus, they had entered the field of what Mr. Norris called the near maximum waste heat utilization. In 1956 they had delivered their first ship in which power for all auxiliaries was generated in steam turbines, using steam from a waste heat boiler. Also, in other respects, the arrangement in that ship was a little uncommon (Fig. 27). The tanker had two cargo pump rooms.In the aft pump room were two centrifugal pumps installed. Forward in the engine room there were two sets of machines, each comprising a multi-stage steam turbine, a generator and an hydraulic coupling driving through the bulkhead to the cargo pumps.

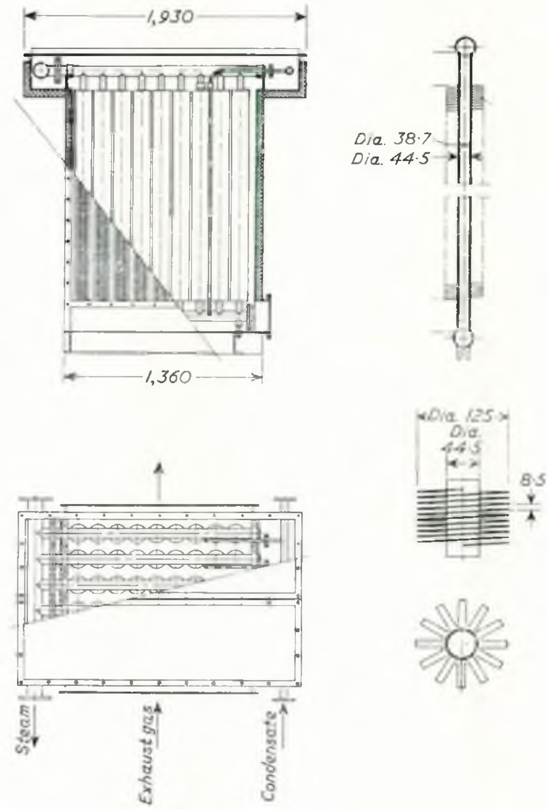


FIG. 29

On the boiler side, the following steps were taken: in the early days they had used La Mont exhaust gas boilers. For obvious reasons there was soon a demand for a boiler with a more generous space to harbour scale and sludge. Thus they had got the vertical cylindrical smoke tube type. In his opinion this boiler was suitable for the purpose. However, as Mr. Norris mentioned, troubles did occur. There were tube-end leakages and corrosion in tubes near the bottom where they were buried in deposits. In order to give this boiler another chance it had been placed horizontally (Fig. 28). With the tube ends seal-

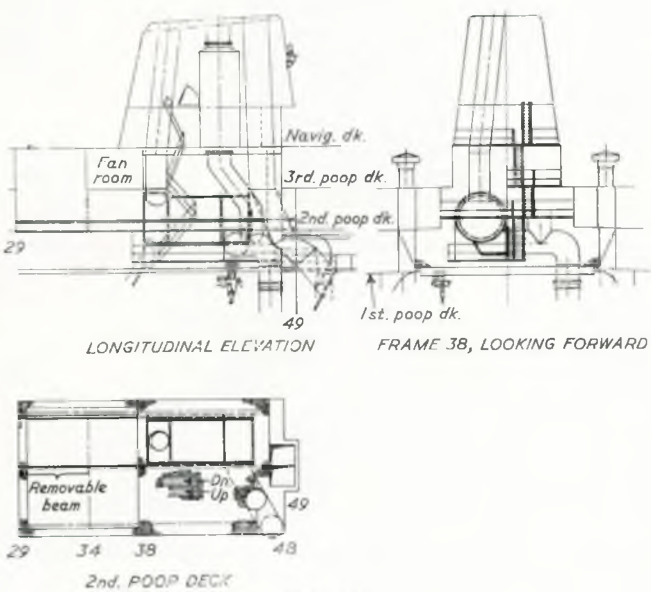


FIG. 28

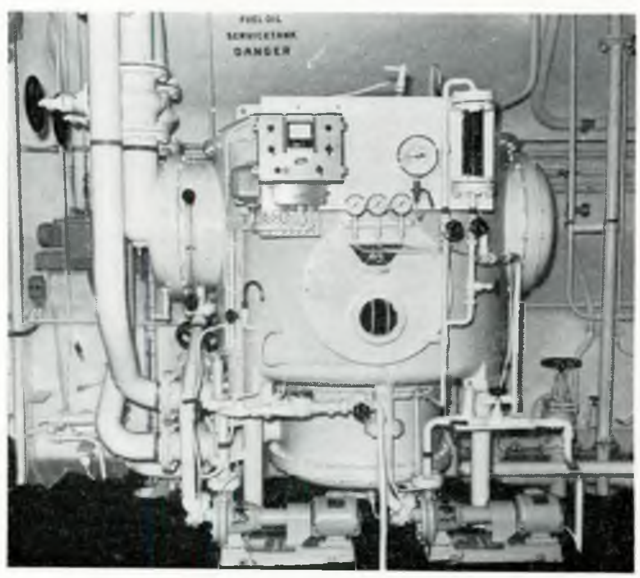


FIG. 30

welded and no possibility of deep layers to form on the hot bottom tube plate, they felt confident that they would obtain good service results.

However, the smoke tube boiler was rather heavy and difficult to repair or replace. They had therefore designed a new type of exhaust gas boiler (Fig. 29). This had vertical tube banks of steel tubes with extended surfaces.

Mr. Norris was apparently looking forward to getting some fresh water generators heated by jacket water. He was sure he would not be disappointed. His firm had installed some equipment, originally Danish, in 55 ships (Fig. 30). Except for a few teething troubles, the performance had been very good. So far they had avoided the type of distiller where sea water was flashed without preheating by jacket water. To throttle the cooling salt water, in order to raise its temperature, was to call for trouble.

A high exhaust gas temperature was desirable for the waste heat system, but might have undesirable effects on the main engine itself. With the turbocharged engines they had found a continuous reduction in gas temperature, due to improved turbocharger performance and better tuning of the components. In ships where the electrical load was based on waste heat systems it might be necessary to stop this downward trend. The easiest way to do this was probably to reduce the cooling of air leaving the turbochargers.

In his paper Mr. Norris had referred to one particular problem which, some time previously, had also troubled Mr. Alsen's company, that was the rather high pressure difference between the exhaust gas boiler and the Scotch boiler (Fig. 31).

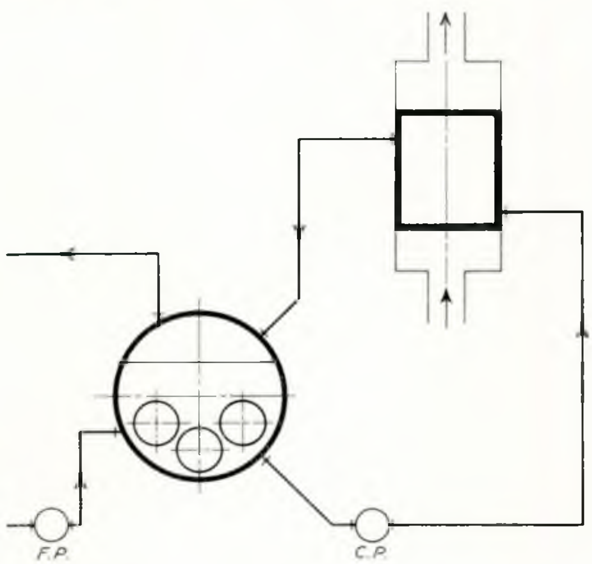


FIG. 31

Mr. Norris considered that heat losses due to radiation and feed water heating in the Scotch boiler should cause the pressure difference. He wondered if this was the right explanation. Was it not rather the flow of steam and water which, under special circumstances, needed a higher pressure differential than normally calculated? Anyhow, a bigger pipe had solved his company's problem.

The waste heat system with steam turbine generators was no doubt an economic solution for large motor ships when both maintenance and fuel costs were considered, which naturally was the case with ships run by, for instance, the oil companies. Independent owners with ships sailing under charter sometimes got no benefit if the auxiliaries did not consume any fuel. For these ships it was not so easy to show the overall advantage with a full size waste heat system. He had the feeling, however, that charterers now were more inclined to give a little preference to motor ships with waste heat systems which, apparently, was enough to make such a system attractive. It might be a coincidence, but his company now had ten ships on order, for which exhaust boilers and multi-stage steam turbines were specified or anticipated. They therefore very much welcomed Mr. Norris' paper, which would be of great value for future progress in the field.

MR. J. H. MILTON (Member of Council) pointed out that the paper provided a fund of extremely useful information on waste heat recovery, particularly in connexion with tankers. Generally speaking, waste heat recovery installations could, he thought, be divided into four categories:

1) Tankers and bulk carriers (not self-discharging) in which, with long distances between terminal ports and a large number of days at sea per annum, waste heat steam in addition to supplying heating and domestic services, ran a turbo-generator set carrying all the sea electrical load.
2) Bulk carriers with self-unloading gear which, in spite of the fact that they probably had three Diesel generators for cargo purposes, still had long hauls between ports, during which savings could be effected on Diesel fuel and maintenance by fitting a waste heat system turbo-generator set.
3) Fast high powered cargo liners which were not such an attractive proposition for waste heat recovery, due to their smaller number of days at sea per annum.
4) The 10,000-15,000-ton dry cargo vessels of, say, 6,000-8,000 b.h.p., where 3,000-5,000lb./hr. of steam was required, at sea and in port, for heating and domestic purposes.

His reasons for mentioning the foregoing was that, apart from the schemes diagrammatically illustrated in the paper, there were many others and each required its own design of heat exchanger and boiler equipment. In other words, there was no standard approach—each installation had to be separately tailored.

He said he would like to ask the author, bearing in mind that:

a) different builders of the same engine quoted different exhaust gas weights and temperatures;
b) the sea electrical load was never definitely known;
c) the output of the waste heat unit was a maker's figure dependent on the heat transfer rate of different types of extended surface, etc.;
d) domestic steam requirements and radiation losses were at best an estimate;
e) ambient temperatures and fuel quality affected the weight and temperature of exhaust gases, etc.;

how close to the anticipated result did his company get? It would appear quite possible, in view of the number of uncertainties, to finish up with a waste heat turbo-generator installation which at normal operating main engine power was just " not man enough "—which would be a disaster.

On page 400, when discussing exhaust gas economizers or heat exchangers, the author pointed out the disadvantages, from the vibration aspect, of large flat casing sides. As there were units available of cylindrical form with pancake type elements, it would be interesting to hear the author's views on the relative merits of this and the rectangular type.

Tanker companies with motor tonnage generally specified waste heat recovery with a turbo-generator set for sea use, but the barriers against fitting such installations, even in bulk carriers without winches, were very formidable. Financially, it could usually be argued that the extra initial cost was recoverable in about four years, but generally owners' superintendents did not want steam plant in their motor ships, fearing operational difficulties, and were prepared to forego the saving on auxiliary engine fuel and maintenance on this account.

He said he would be grateful if the author would comment on the commercial implication that the extra initial cost of such plant could be recovered in four years, and also whether he had experienced any difficulties in the operation of such plant by Diesel engineers.

In conclusion, he said that it would appear from the section on exhaust gas conditions, on page 402, that by decreasing the cooling of the air discharge from the turboblowers, additional usable heat could be obtained from the exhaust gases. The limitations of this practice were obviously related to the cooling of the cylinder liners in way of the ports in a Sulzer engine, and exhaust valve seats in a Burmeister and Wain engine. He asked the author to indicate which of these types was more applicable to this practice.

MR. J. R. FRANK, B.Sc. (Member) thought that the author had provided much interesting information on a subject which was becoming of increasing importance to the economic operation of motor ships.

It was of interest to consider the operating characteristics of these waste heat recovery systems and, as the author had pointed out, it was particularly important to design the system for the lowest service power anticipated. Illustration of this could be provided by the typical electricity generating capacity determination for a 600 kW C.M.R. turbo-alternator set, for which the main engine rating was specified as 82·5 per cent of the maximum (Fig. 32). Although a five per cent margin had

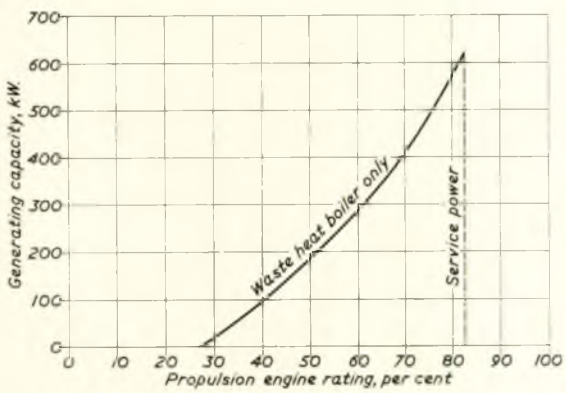


FIG. 32—*Normal operating characteristics for waste heat recovery system*

been provided, by an increase in boiler surface over that required to sustain the 600 kW, the maintenance of a sea load of 540 kW would only be sustained provided the engine rating was held above 78 per cent.

Considering manœuvring aspects, an order for half speed would require only an eighth power, and the electrical generation would be nil. Thus, for a crash stop or similar large reductions in power, the system must be so designed that the essential services might be sustained.

A further point of interest was in the control of the system to maintain a substantially constant steam pressure. The prototype plant, described by the author, utilized a bypass valve to divert water at evaporation temperature to the economizer elements, so reducing the temperature difference to the gas and thereby the heat transferred. This, coupled with the injection of relatively cold feed water, would result in a measure of control. Perhaps the author could advise what evaporation turn-down could be achieved with the main engine at full service power. This was particularly important as, without the gas bypass feature on the waste heat boiler and the undesirability of restricting the waste heat boiler circulation, it was desirable to provide for full flexibility in operation such that the turbo-generator could be shut down if required. In such a case the steam to ship's services would account for only about 20 per cent of the total evaporation.

The remarks on dew-point corrosion and the prediction of a temperature of 248 deg. F. (120 deg. C.), or 30lb./sq. in. gauge as a desirable minimum were of great interest. If such a low temperature could be used in service, then the surface installed in the L.P. boiler of such a system as shown in Fig. 16(b) would be considerably less than if steam were generated at, say, 70lb./sq. in. gauge or 320 deg. F. (160 deg. C.). However, with steam at 30lb./sq. in. gauge for general heating services, the steam for residual fuel oil heating might have to be taken from the H.P. circuit.

Considering the cost of the boiler installation, it was necessary to consider the total cost of the whole plant and the repayment period. In this respect, his own company had found that smaller than normal pinch temperatures were sometimes necessary to give an economic result, particularly for the simple single pressure systems. For such systems the composite boiler offered an attractive proposition as no circulation pumps were required. However, the use of separate oil fired and waste heat boilers enabled the oil fired boiler and feed regulation equipment to be sited on the generator flat.

MR. N. J. JAMES (Associate Member) did not wish to disagree in any way with the views expressed in the excellent wide coverage of waste heat recovery systems, but he thought that the author (whom he knew appreciated well enough all the auxiliary makers' problems) would tolerate some comments from the turbo-generator designer's aspect.

The question of margins was, of course, perennial. As stated by the author, the actual rating of a turbo-generator depended firstly on the total connected load which, although arithmetically simple to arrive at, might contain many hidden margins. The performance and peak load factors applied to the total connected load to give the turbo-generator rating must be a matter of experience with similar ships.

The shipbuilder's experience, one assumed, would be limited to sea trials data which, although extremely useful, could not be truly representative since every item of equipment was in mint condition.

Full and reliable performance data from auxiliary makers in the early stages of a projected ship design would help the shipbuilder to improve the accuracy of the total connected load.

Operating experience data logged over long periods was the only logical way to arrive at the required factors and it was gratifying to see this approach being adopted, as illustrated in Fig. 9. Automatic data logging should materially contribute towards more precise ratings.

Until this desirable state of affairs existed the turbo-generator designer was likely to keep a little up his sleeve to avoid what the author had described as unpleasant service results.

A further fear factor in the designer's mind was the inevitable fall-off in heat exchanger performance after long periods of service. It was one thing to produce x kilowatts from y pounds of steam under near laboratory conditions on the test bed with a team of specialists, but the true criterion was maintaining the "at sea" load throughout the working life of the ship.

Despite reasonable margins, early experiences of waste heat recovery systems bordered on the unpleasant and there was a natural tendency to play even safer.

Thanks to the author's company, and other progressive companies, and the increasing number of ships in service with waste heat recovery systems, a stage was being approached where, given full co-operation between auxiliary makers, ship and engine builders and operators, design margins might be relaxed and more accurate running costs for a ship could be calculated, which was of obvious concern to builders and owners alike.

With regard to steam conditions, superheat was, of course, always welcomed by the designer. Many schemes, however, less sophisticated than those described, used dry saturated or wet steam. A turbo-generator designed specifically for this purpose had been developed in parallel with the set described by the author, and showed no sign of blade wear after some five years' service.

Provision for bleeding steam from the turbo-generator to supply the ship's heating service was made on a number of sets that had been supplied. He said he would appreciate the author's views on this system.

Where a moderate degree of superheat was employed, the geometry of the heat-entropy chart allowed an inlet pressure to be chosen which gave little or no superheat in the bled steam.

The author's chosen optimum of 125-140lb./sq. in. gauge was ideal for existing frame sizes. High pressures meant expensive boilers, and very low pressures with large mass flows gave larger and more expensive condensers.

In addition, with smaller powers of around 300 kW the inlet piping (or ducting) size began to approach the turbine cylinder diameter and necessitated larger, more cumbersome valve gear and attendant physical design problems which tended to increase the first cost.

In the highly competitive marine world, the highest possible degree of standardization was clearly desirable in order to reduce costs, and the U.K. designers had concentrated on the self-contained packaged type turbo-generator with integrated condenser oil sump, air ejector, gland condenser, gears and generator, to give the minimum installation worries to the builders.

Another approach which appeared to be, at any rate superficially, somewhat cheaper, was for the builder to purchase these items separately in what an engineer friend of his had recently described as a " do it yourself kit ", and integrate the several components into the ship.

Experience would indicate the better approach, indeed there might be room for both.

Mr. A. M. Scrimgeour (Associate Member) said that he wished to acknowledge Mr. Norris' personal help and advice when his (Mr. Scrimgeour's) company had decided to install its first waste heat recovery plant in a 47,000-ton motor tanker. The system was fundamentally that described in Fig. 15(h) and that it was now proving a real success in service, was the most worthwhile of tributes he could pay to the author.

He remarked that he was fortunate in that, as well as attending the trials of his company's ship late in 1963, he had remained on board for the first ten days of the maiden voyage, during which the main engine was operated at different powers, varying loads being put on the turbo-alternator, and the general flexibility and ease of operation of the system were proved under normal seagoing service conditions without extra supervision of any kind.

The majority of the comments he would like to make, he said, were to amplify and confirm some of Mr. Norris' points. In his case, Mr. Norris had been preaching to the converted, and there were only a few minor items on which he would like to take issue with the author.

Although the paper was entitled " Waste Heat Systems in Motor Tankers ", his company felt that this principle was equally attractive for large bulk ore carriers, particularly as many of these ships owned by his company were frequently exceeding an average of 80 per cent of their time at sea. These vessels were already fitted with some steam plant, admittedly of a pretty basic type, to provide steam for heavy fuel oil heating, but it was an economic fact that as auxiliary boiler size increased, so the relative costs in £/lb. evaporation decreased, and he thought it became well worth while increasing the boiler capacity to ensure continuous operation of the turbo-generator. The claim that burning heavy fuel under the boiler was less economical than employing a Diesel generator, when all factors were considered, was only marginally true. Even the largest vessels now loaded in a matter of hours and rapid discharge too made continuous turbo-alternator running the simplest routine for the ships' engineers. The almost continuous operation of turbo-alternators quoted in Appendix B fully supported this contention.

The author's reference to quality of engineers and the inference that a high standard was available, was slightly at variance with the examples of instruction book illustrations (Figs. 5 and 6) where it had been thought necessary to identify such items as the gearbox, turbine, horizontal joint, and even the generator itself.

Referring to the electrical load which must be catered for, Fig. 9(b) showed very clearly what would happen if the main engine or compressor were started on the peak at 08·00 hours. His company favoured a system employing a small garage type air compressor automatically topping-up the air start bottles as required at sea. This saved the steep rise in load when a main compressor was started and, as a secondary benefit, provided an emergency electrical generator was fitted, by wiring the small compressor directly to it and providing a connexion to the generator air start bottles, this, he believed, might be classified as the emergency compressor, and a financially significant saving would be made by not having to fit the usual small hand start Diesel engine-driven emergency compressor.

He had been very interested to see that in both instances where annual maintenance costs of Diesel generators were compared with turbo-generators, 600 r.p.m. machines had been cited. He wondered if the differential would be even greater had figures for high speed machines been quoted. He would go so far as to say that without Diesel generator engine maintenance to cater for, a possible reduction in manning could be considered. Overall, he thought that the reduction in generator engine maintenance was an extremely important factor and he would have liked to have seen it given more prominence in the paper.

Also worthy of greater emphasis was the " heat bank " concept which, in terms of benefit to the watchkeepers, whose attitude to such an installation was of paramount importance, represented the order of several minutes of continued dependable operation of the turbo-alternator following an emergency stop of the main engine at sea. This was a very comforting aspect for the operators and allowed sufficient time to put a burner under the boiler and retain the turbo-generator or to start, parallel, and load a Diesel generator without interruption of electric supply.

To digress slightly, he said that should a sudden shutdown of the turbo-alternator take place in his company's ship, due, say, to low vacuum or low lubricating oil pressure, a preselected Diesel generator was automatically started and, on reaching correct running speed, loaded automatically and sequentially in order of importance, starting with main engine auxiliaries. Subsequently, on restoration of the turbo-alternator to its correct running speed, the latter in turn re-accepted the load automatically and sequentially.

The recommendation that allowance should be made for lower charter speeds was interesting. During the maiden voyage of his company's vessel they had established a minimum satisfactory self-sustaining condition at 75 per cent power. This raised the very vexed question of what was " normal " power. Mr. Norris' reference to engine rating was non-commital; even he skated around this nice point by stating " The normal service rating at which the ship will operate must be determined ". It was high time that there was some disinterested " which " type frank assessment of optimum day-in day-out, overall, truly economic ratings of the principal main engine types.

Speaking on a minor point, he said that he thought that the omission of soot blowers might be " penny wise ". As long as some of the deposit, which was inevitable, was removed, surely this very minor expense was warranted.

Finally, backing up the author's warning to resist complication, he said that simplicity was the essence of good design. He was glad that it was only suggested as a possible development to have steam turbine-driven main engine blowers. Frank reference was made by Mr. Norris to casing troubles in one vessel of his company. He was quite sure that a proportion of this pulsation was ironed out by the turboblowers. Anyway (and he trusted main engine manufacturers would accept this in good spirit) an exhaust gas turbocharger should be a thoroughly reliable straightforward piece of equipment, provided it was not fed with morsels of piston rings.

Commander V. M. Lake, R.N. (Member), speaking as one of the auxiliary manufacturers involved in the production of the system, said that his firm had been set many interesting problems with regard to the equipment under discussion, most of which had been covered in detail by other speakers. However, he emphasized the fact that the whole of the paper had been based on detailed consideration of auxiliaries and their combina-

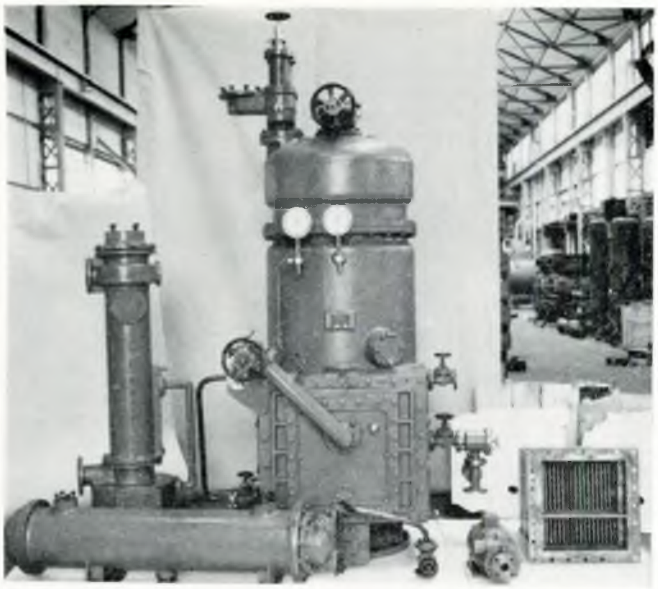


FIG. 33

tion. Mr. Norris had indeed done a very great service in covering this ground and had again emphasized the question of margins. A combination of auxiliaries could get completely out of hand if margins were hidden. His company had had to change their approach to their heat exchanger design quite considerably, in order to eliminate these margins. He suspected that some were probably not completely eliminated and this was a pity.

He thought that the BP Company should be congratulated on the number of results which they passed back to sub-contractors. It was a great pity that the measuring equipment in this particular system was not really up to the small temperature differences that were involved. These were essential for the combination to make the system work.

Figs. 33 and 34 pointed out how things had changed. Fig. 33 showed the simple evaporators referred to by Mr. Norris, with the flexible heating elements. This might be considered rather old-fashioned when compared with the plant shown by Mr. Alsen from Sweden, but it did represent, in Mr. Norris's opinion, the optimum for space, simplicity and price.

Fig. 34 showed a packaged unit which combined the steam generator and the two-stage feed heater underneath. This was now a long thin heat exchanger, which was really two heat exchangers in one shell. The controls were quite clearly shown. It was of interest that certain shipbuilders preferred not to have this packaged unit built up as one complete kit, but to combine the steam generator and the heat exchanger by their own efforts. It would be interesting to have Mr. Norris' opinion on this approach.

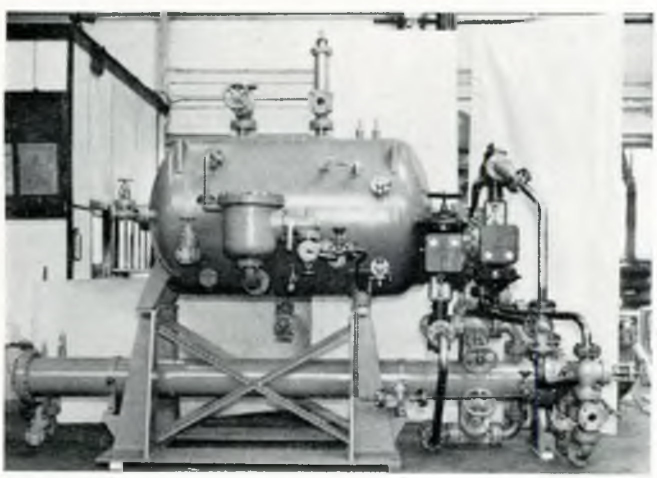


FIG. 34

MR. A. WILSON said that on page 397 Mr. Norris had quoted a figure of 15 per cent as being reasonable in regard to amortization rate. He had heard it said in regard to a number of large organizations, such as BP Tanker Company, that if one were putting a scheme up to these organizations, one was required to show a return within three years. He wondered how the author reconciled the three year period with the 15 per cent amortization rate quoted in the paper.

On page 399 Mr. Norris had mentioned the figure of 12 per cent for moisture in steam. He said he would be most grateful if the author could give information on whether there had been any problems with erosion on final stages of turbine rotor under those conditions.

On page 400 mention was made of expanded tubes and seal-welded tubes. He thought that seal-welding was not only a desirable thing, but most probably something which should be standardized in the future for all boiler construction. However, he thought the term "seal welding" was wrong. Either a weld was a strength weld or it was not anything that mattered at all.

Also on page 400 Mr. Norris had referred, quite rightly, to the question of heat exchange relative to the velocity of gases passing through the heat exchanger, but he gave no indication of acceptable Reynolds numbers, although discussing the merits of turbulence. He said he would like the author's comments on this.

On page 405 it was proposed that temperatures on the water side could be reduced to around 300 deg. F. (149 deg. C.). Work of BP Research, Shell Research and others, had indicated that there was, in fact, quite a large loop in the dew-point curve which was most probably due to the fact that, because of a mixture between the acid and the water, an elevated water dew-point of something above 212 deg. F. (100 deg. C.) was obtained, and a depressed acid dew-point of something below 300 deg. F. (149 deg. C.). He asked if Mr. Norris could expand on this and say whether, in fact, he had found any indication in ships which his company operated, that there might be two points of corrosion, say of 220 deg. F. (104 deg. C.) for water or lower dew-point, and possibly 290 deg. F. (143 deg. C.) for the acid or higher dew-point, and whether this would affect proposal of 300 deg. F. (149 deg. C.) Also, in view of the operation of boilers at other than full power, he wondered whether perhaps a slightly higher temperature—possibly 350 deg. F. (177 deg. C.)—should be accepted.

With regard to latent heat cooling, he said that he was pleased that the author had made reference in the appendix to Brett-Littlechild, who had possibly done more than anybody else in this country in this field. He wondered if the author could give further indications on how successful this work had been in the marine field.

Correspondence

CAPTAIN W. A. STEWART, C.B.E., R.N. (Member) wrote that his interest in the paper, which he had found most informative, lay in the application of waste heat in Diesel locomotives for carriage warming, by feed heating for the existing boilers. It would appear from this most detailed and informative paper that, owing to the low load factor of about 50 per cent, which was common on British Railways, the amount of heat that could be recovered would be very limited and that this application might not be justified financially.

However, there was one significant point which was new to him in the paper, namely, in Fig. 11 it showed that there was a notable drop in temperature of the exhaust gas with increasing load, and this was presumably typical of all turbo-charged two-cycle engines, of which he had had no experience. It was the opposite to that with which one had been brought up to expect with other types of engines, such as turboblown four-cycle engines, which were used.

It would be interesting to know just how the 35 per cent of total heat in the exhaust gases at full power, quoted on page 397, varied with part load in these particular engines. It seemed possible that it could remain fairly constant over quite a wide range of power. Could the author confirm this?

MR. W. FRANCIS (Member) wrote that Mr. Norris had presented a paper giving most useful information which might not be readily available to the marine engine builder and that the information would be closely studied and used by installation designers and others.

He was particularly interested by the opinions on future developments and the concluding remarks.

Mention was made of using the engine exhaust gas heat exchanger in port for steam generation, using a combustor. Was this a development to make " inerting " of tanks during loading and discharging economic?

There was some questioning of the reliability of exhaust gas turbocharging in the suggestion that steam-driven scavenge blowers should be fitted. This solution would appear to offset some of the economic benefits of a waste heat system by introducing lower efficiencies; further, a rather complicated automatic control system might be required for the steam-driven blower to follow engine load.

The author set a lower limit of 6,000 b.h.p. for considering a turbo-generator driven from waste heat. A limit of 8,000 b.h.p. was suggested as more reasonable. This power should provide sufficient steam without supplementary firing for about 240 kW loading.

It was most important, with the lower powered vessel, that excessive power margins should not be added at the design stage, so that in service the engine operated at reduced powers. He knew of one case where an owner expected full generator output under sea trial conditions and when the engine was operating in service, considerably de-rated.

It must not be overlooked that the main engine was primarily a propulsion unit and the utilization of waste heat was a secondary consideration. Fig. 11 illustrated clearly that the most efficient engine had the lowest exhaust temperature. As a means of increasing exhaust temperature, the paper showed the effects of raising exhaust back pressure and air cooler temperature. Had the author considered maintaining the final exhaust temperature constant at all loads for maximum heat recovery?

There was reference to the use of a Diesel generator in conjunction with the turbo-generator. The speed governing characteristics of the two prime movers were widely different and it was found in practice that certain skill was required to parallel the two units on to load.

The point was made that, as powers increased, the economics made it essential for waste heat to provide all normal power requirements. At some horsepower the exhaust heat quantity might become an embarrassment. Did Mr. Norris foresee some of the heat being made available at the shaft for propulsion?

A compound heat recovery system using both jacket water and exhaust heat for steam generation seemed attractive. Practically, complete heat recovery was possible from jacket water and even low powered installations were able to produce adequate fresh water. At higher powers, surplus heat was available. At the beginning of the paper, 35 per cent of the fuel heat was apportioned to the exhaust gas, but only about one-third of this was recoverable, which was the same order as the jacket water heat. Perhaps the author had some opinions on a single package unit for production of power *and* fresh water.

Fig. 12, showing the specific heat of the gas mixture over a range of loads and temperatures, was most useful and he wondered if, when the paper was published in the TRANSACTIONS, Mr. Norris could supply some of the background information and calculations used to obtain these curves from the dry gas analysis.

MR. J. E. CHURCH (Member of Council) wrote that Mr. Norris had contributed a very comprehensive paper containing a wealth of information. There was, however, very little that was new in it and many superintendent engineers had investigated and tried all that he described and had not been so convinced. The writer had in fact been associated with similar installations which made use of steam, raised by Diesel waste heat to the exclusion of any other fuel expenditure at sea, on and off, for the past 30 years, but was still unable to agree entirely with the author's view that it would soon be unusual for a large powered motor ship to be built without at least one turbo-generator, using only steam from waste heat, for all normal auxiliary power requirements at sea.

There were many reasons, amongst them the following:

1) Success depends on high exhaust temperatures. It is comparatively easy to produce these, but all marine Diesel engine operators know that, other things being equal, trouble-free operation, reliability and low maintenance costs depend upon a high proportion of excess air for combustion purposes to obtain the lowest exhaust temperatures possible. To maintain steam entirely from exhaust gas invariably means designing or adjusting for a degree of air starvation and then regulating engine speed, even forcing the engines more than would otherwise be the case. Piston ring and cylinder liner wear, scavenge fires and worse are, in the writer's experience, the price one has to pay in the long run for maximum waste heat recovery from exhaust gas. Many have found it not worth while, because ultimately economy is not merely a question of the lowest possible fuel rate. All that matters is the lowest cost in £.s.d. per annum to operate the ship.
2) In the lifetime of all ships now contemplated, the manning problem, particularly in the engine room, will become such that successful operation will only be possible in conjunction with maximum simplicity and a minimum number of pieces of equipment to be watched and maintained in the engine room. Most of that represented by the diagrams shown in the paper (Fig. 19) together with everything else *not absolutely essential* for the propulsion and safety of the ship will sooner or later have to go. In fact it is inevitable that before long semi-skilled engine drivers will take the place of certificated engineers at sea, and ship and machinery design should already be thinking on these lines.
3) Most owners seem agreed that reduced manning in all departments is now desirable and inevitable. It is doubtful if a high powered motor ship, with the full treatment outlined in the paper, could be sent to sea with only one engineer and one rating on each watch. Without it the writer would have no hesitation.

The paper did not describe exactly how the whole system was controlled. On the one hand it was shown how the electrical load and therefore the steam demand fluctuated; on the other hand, the main engines presumably were set to run at the required power for the ship speed demanded. It followed that either there must be too much steam—in which case the surplus might be dumped to condenser, or some exhaust gas bypassed, or evaporation reduced by lowering the water level in the boiler—or the engine speed adjusted to balance the firing rate to the steam required. Alternatively, there might often not be enough in which case either a Diesel generator would have to be paralleled, or oil firing used to maintain steam. None of these procedures were thought to be particularly attractive, but it could be that Mr. Norris had evolved a better system of control. In the writer's experience it was difficult in practice to avoid making the main engine match the steam turbo-generator, which to say the least was the " tail wagging the dog " and not very good engineering.

The comparable costs given, of maintenance for steam turbine and Diesel-driven generators, were not altogether convincing. £520 p.a. for two turbo-generators in a steamship was not likely to be achieved in a motor ship and the writer could produce figures of Diesel generator maintenance, up to 300 kW. in size, amounting to nil except for spare gear supplied and fitted by the ship's engineers at a cost of £400 p.a. for *three* machines.

All in all, therefore, it seemed that the balance would often be about even and the promised " something for nothing " was as elusive as ever.

MR. J. ELLISON (Member), in a written contribution, pointed out that the idea of using steam produced from the heat of Diesel engine exhaust gases for generating power was not by any means new, as his company had fitted 200 kW reciprocating steam-generating sets in large refrigerated cargo liners some thirty years ago, and a considerable saving in Diesel fuel was effected thereby. It was of course conceded that the modern large-bore supercharged Diesel engine, made available a very much larger amount of recoverable heat in the exhaust; also it was important to run at or near the normal designed power in order to obtain maximum supercharging efficiency, thus it was more likely that a steady supply of heat was available for producing steam to drive a turbo-generator or alternator.

He did not agree with the statement under paragraph (b) on page 397 "If it is necessary to run a Diesel generator in parallel with the turbo-set whilst at sea . . . then a near maximum heat recovery system is not justified ". This might be true for a tanker with which the author was particularly concerned, but it was certainly not true for a refrigerated cargo vessel where the electrical load in one voyage direction was very much higher than in the other. Under these circumstances incompatibility of the two methods of generating power might sometimes be found, when say an output of 500 kW was required, and 375 kW could be obtained from the turbo-alternator. It was highly desirable to run both sets at or near maximum load, especially in the case of the turbine set, and the Diesel set would probably find itself producing the remaining 150 kW with consequent deleterious effects to valves and fuel injectors, through running on less than half its designed load.

On page 399, the author dealt with "Flexible Element Evaporators " and mentioned their ability to shed scale. This might be so in theory but it did not quite work out in practice, and some form of de-scaling liquid was required, especially for elements incorporating open-ended tubes through which the brine circulated. The author did not mention if scale prevention treatment was applied to the evaporator feed.

It was interesting to note, on page 402, that the increase in ambient temperature under tropical conditions was being exploited to take full benefit of the heat in the exhaust gases and that when the vessel entered temperate conditions the intercooling was automatically reduced to maintain the heat in the exhaust. On the other hand this could only be effected at the expense of increased fuel consumption. This might be considered worth while in a tanker which was on a regular Persian Gulf run, but would not be warranted on, say, a U.K.-Australasian run, where only 25 per cent of the voyage was spent in tropical waters.

The exhaust gas temperatures for Sulzer engines shown on page 403 were evidently at the cylinder outlets as the temperatures at the turbine inlets were usually at least 100-150 deg. F (56-83 deg. C.) higher, say in the region of 800-900 deg. F. (426-482 deg. C.).

No mention appeared to be made of controlling the steam output of the steam generator, and it would be interesting to have the author's opinion as to whether it was better to allow the full capacity to be generated under all conditions and to lead the surplus to the turbo-alternator or auxiliary condenser via a dump valve, or to isolate sections of the generating tubes by hand-operated or motorized valves.

MR. C. W. PARRIS (Member) and MR. H. WAGNER (Member), in a joint written contribution, felt that this important and valuable paper was particularly appropriate at the present time; it would be used as a reference by the many interests concerned in current developments and, for that reason, the present contributors concluded at the meeting that it would be wrong to attempt any compression of their views.

It was pleasing to note the appreciation that marine economizers were extremely reliable pieces of apparatus and of a construction eminently suited to waste heat recovery from Diesel exhaust gases.

Where feed temperatures were 240 deg. F. (104 deg. C.), cast iron protected tubes were selected for economizers of oil fired boilers. It was expected that SO_2 formation from Diesel gases would demand the same material and it was considered advisable to instal cast iron protection with internal fluid temperatures up to 300 deg. F. (149 deg. C.). The vibration damping qualities of cast iron were highly desirable for use with pulsating gas flow and it was expected that tubes in staggered formation would better contribute to silencing and spark-arresting duties.

The cast iron construction was naturally expensive and, where operating temperatures might permit, tubes with steel gills provided the most competitive solution for economizer type surface. For convenience these were often basically of in-line formation, but, none-the-less, a nominal back pressure of four inches w.g. under service power conditions was generally considered adequate to provide an acceptable silencing effect.

Economizer type tubes permitted the maximum exploitation of waste heat recovery from the relatively low temperature gases. The extended heating surfaces were efficient and compact and the construction might be readily manipulated into economizer, low pressure and high pressure generator, and superheater sections according to requirements and all within the same casings. The paper well illustrated the requirement for such composite heat exchange from the waste gas source to meet fully the electrical and domestic loads at sea.

They would agree that the so called optimum pressure of 70lb./sq. in. could well apply in principle to shell type boilers where a high steaming rate was only permitted by utilization of a reduced saturation temperature. With economizer type surface the author's selection of dual pressures was considered quite appropriate to match general requirements from the grade of heat available.

On the question of flow systems within economizer type steam generators, there were pertinent considerations where, with upward gas flow, a reduced mean temperature difference could result from a contra-flow arrangement. Downward water flow as in Fig. 15(d) required high velocities to propel steam bubbles towards the lower outlet and high resistances were entailed. Ignoring the initial, but negligible, amount of sensible heat, steam with the highest saturation temperature was generated local to the coolest gases. For high pressure systems the approximate increase in saturation temperature in deg. F. was numerically equal to one half the system resistance in lb./sq. in. and was more for low pressure systems, and the effect could be quite marked. With a more natural and upward water/steam flow, a lower resistance was permitted and in any event the highest saturation temperature was associated with the hottest gases and contra-flow benefits could result from parallel flow arrangement. Economizer sections associated with changing temperatures below those of saturation were of necessity contra-flow for maximum heat recovery.

Still referring to Fig. 15 they would agree that circuits (e) and (f) were undesirable, more because the feed temperatures aboard motor vessels were usually too low. Where a de-aerator was fitted, however, and a temperature satisfactory to the heat exchange surfaces might be maintained, then of course the system could be successful. Steam formation was permitted, as indeed would occur in any flow system under certain conditions, and it was desirable to arrange the feed regulators before the economizer.

Feed heating by admixture should be accompanied by high velocities to maintain solid particles and liberated gases freely in motion through the tube bank and it was considered preferable for the economizer to discharge direct into the drum. The recirculated feed heat quantity should be controlled thermostatically to provide the required feed temperature and. with feed regulators properly positioned, problems from intermittent operation were not anticipated. Boiler water levels were maintained in the usual manner and certainly centrifugal pumps were recommended. On single-pressure steam generation, the maximum permissible economizer design duty was that to raise feed water as received up to saturation temperature. This could not be improved on by any system and cooling of circulating water was purely incidental.

Indirect feed heating was expected to be more costly and in a competitive market such as this, the manufacturers of waste heat equipment had to consider how best to satisfy requirements at lowest cost. Apart from the necessity for supplementary heat exchangers, the primary unit within the gas stream usually entailed some extra expense due to reduced temperature differences. With boilers in this class it was highly desirable to arrange a good circulation rate, which, with a series flow system, was severely limited. Where low pressure steam was additionally generated indirectly in cooling circulating water to 240 deg. F. (104 deg. C.), the rate might be increased to a figure more in keeping with forced circulation practice. Generally, the higher the circulation rate achieved with indirect heating, the higher the proportionate cost as compared with direct heat exchange; this taking into account cast iron protected surfaces for economizer duties.

Low pressure steam might be generated direct from the gases with less capital expenditure in the primary unit. The cost might be further reduced where, for less stringent duties and considering the possibilities of corrosion, a selected pressure of 50lb./sq. in. with a saturation temperature around 300 deg. F. (149 deg. C.) would permit the less costly steel tube design.

Finally the paper well illustrated that the owner alone really knew his changing requirements and that design, incorporating the desired controls, whereby the amount of steam generated was maintained constantly in accordance with ship's requirements, should be finally settled by close co-operation with the supplier of the waste heat recovery unit.

Mr. A. G. Howe, O.B.E. (Member) wrote that he was a little surprised to see that Mr. Norris had stated that, with the quality of engineer available, there should be no trouble in running both Diesels and turbo-electric sets; he had been under the impression of late that engineering skills at sea were not what they used to be.

For smaller tankers which did not spend most of their life at sea—coastal tankers, etc.—Mr. Norris seemed to admit that there was a good case for purely Diesel generating sets. Although his company could burn marine Diesel oil in their engines, they cou d not, of course, take advantage of the cheaper, heavier grade of fuel.

Although it was clear that Mr. Norris accepted the higher capital cost of turbo-electric sets, this was undoubtedly due to the fact that no fuel was burned at sea and maintenance costs were of a very low order, as highlighted in the paper.

The following details under the heading "Grade of Heat", so far as his company's engines were concerned, might be of interest.

Grade of Heat (Full Load):

	YLC Range (900 r.p.m. 2,270-1,135 b.h.p)	12 YHX (1,200 r.p.m., 730 b.h.p.)	6 RPH (1,000 r.p.m., 200 b.h.p.)	6 RPHX (1,200 r.p.m., 288 b.h.p.)
Percentage heat to work	36·5	36·0	33·5	33·0
Percentage heat to exhaust	26·0	31·5	30·0	33·0
Percentage heat to jacket water	25·0	24·0	28·0	24·5
Percentage heat to other services	12·5	8·5	8·5	9·5

It was rare that engines operated on full load. At three-quarter load, on YLC engines, the heat to water and exhaust would have each dropped 0·25 per cent—increasing the percentage heat to work to 37·0 per cent.

Did the costs of maintenance for the turbo-alternators include the waste heat recovery plant (or a proportion of the cost since this equipment could be used for other purposes)?

In conclusion, it would appear that both types of generating plant had their use, the Diesel-driven sets up to 400 kW. and the turbo-alternators above this output (see Fig. 2). It would be useful if Mr. Norris could elaborate on smaller vessels than those mentioned in the paper.

Author's Reply

The author thanked those who took part in the discussion, either verbally or by written contribution, for their complimentary remarks and for the stimulating questions posed. In replying he proposed to deal firstly with the question of acid dew-point referred to by several contributors and thereafter to reply separately to each contributor, endeavouring to avoid repetition where others had raised related points. In comparison with an oil fired boiler plant, the location of the acid dew-point was influenced by the large excess air requirement of the Diesel engine. The measured carbon dioxide content of the gases at the three engine loadings given was 5·0, 5·9 and 6·7 per cent (volume). Some difficulty was experienced in obtaining data at the low temperature end because of difficulty in cooling the (air-cooled) corrosion probes. Nevertheless the loop in dew-point curves, expected in boiler investigations, was also evident in Fig. 13. The acid dew-point was measured by the British Coal Utilization Research Association electrical dew-point technique and corrosion probes were of B.C.U.R.A. type. The method used to establish the specific heat from the gas analysis was to calculate the water vapour content from the measured carbon dioxide content of the gases and the carbon/hydrogen ratio of the fuel; the nitrogen content was then determined by difference, thus giving the mean analysis for the major components of the gases.

If low gas outlet temperatures were likely from the waste heat unit, then the precautions, mentioned in the paper, should be taken. Safeguards were built into the circuits shown in Figs. 15(h) and 16(c), since the "tail end" metal temperature automatically increased when the steam output required was less than design, or when oil firing of the boiler was necessary if the main engine was operated at reduced power.

The graph, referred to by Mr. Hutchings, was intended to show the effect on cost of making provision for excessive margins on steam output from a waste heat unit of economizer type, installed in the exhaust ducting of an 18,000 (service) s.h.p. Diesel engine. At the designed output of 13,200lb./hr. of steam, the supply cost of the unit had been assumed to be £13,200, i.e., £1/lb. of steam. The addition of a 1,000lb./hr. margin would increase the cost by eight per cent per pound of steam; a decrease of 1,000lb./hr. would reduce the cost by seven per cent per pound of steam.

The pressure difference, between a waste heat boiler and a Scotch boiler linked by a forced circulation system, had been observed in several vessels. While it might be due to other reasons than those given, radiation loss was considered to be the most significant factor—the effect was well known and demonstrated whenever a boiler was shut down with all stop valves closed. With waste heat plant the rate of circulation and the boiler water capacity were usually such that the water component of the water/steam emulsion, returned from the waste heat unit, had an appreciable residence time in the boiler drum, before being recirculated. Radiation losses must occur at this time. Readings taken over a long period indicated that the circulating water withdrawn from a Scotch boiler was approximately 20 deg. F. (11 deg. C.) below the saturation temperature corresponding to the pressure, when the boiler was not being oil fired.

The author agreed that it would be expensive to design the flat casings of waste heat recovery units to avoid all risk of vibration. Any flat panel vibration on shipboard was difficult to combat in view of structure-borne vibration induced by secondary engine unbalance, propeller blade impulses and other sources. It was considered reasonable to design initially to withstand the exhaust gas impulses and to provide external stiffening in easily accessible positions on the casing. Thereafter, if vibration was found, the panel sides could easily be modified to change the natural frequency and damp out the panel vibration. Undue emphasis should not be given to this feature, however, since, in the author's experience, there had only been one difficult case which was worsened by engine and propeller effects. Care was necessary in dealing with engines of an unfamiliar type, particularly where seven or eight-cylinder engines were fitted. The most recent units to come into service with six-cylinder engines had no panel vibration at all as far as could be seen. Earlier experience with six-cylinder engines of another design was also very good.

It was also agreed that several of the waste heat units on the market had built-in features which were theoretically undesirable. The author would not agree with Mr. Hutchings that the suppliers seemed to get away with it, since their liberty only existed until service experience disclosed the shortcomings of the design. In the example given by Mr. Hutchings, the maker supplied an automatic level control which adjusted the water level, i.e. the "drowned" section of the boiler tube, to suit the steam demand. Such controls needed careful consideration if steam demand was likely to fluctuate. For example, if steam demand was low, the water level was low; if a sudden demand for steam arose, the steam pressure would fall, because of the small water quantity having low thermal capacity, and the controls then allowed cold feed water to enter the boiler drum, thus further depressing the steam pressure. Such rapid changes in water level and boiler temperature or the adoption of circuits liable to water hammer hazard, as described on page 407 of the paper, were features which were both theoretically and practically undesirable.

In the author's opinion, good initial design should never be sacrificed to commercial expediency since, although defects might take years to develop, the ultimate consequences might be disastrous.

Mr. Alsen had provided a most interesting description of the evolution of his company's heat recovery plant; an evolution based on sound engineering thinking and logical development to which he had made an important contribution. The new design of exhaust gas boiler shown as Fig. 29 appeared to be of a desirably simple construction which should keep the manufacturing cost down. The extended surface shown appeared to be of spiral-wound type, but of new design. A related integral fin type of heat exchange surface was known to have been used in a few instances in ships built in the United Kingdom, but details of long term performance had not been disclosed. Whatever type of waste heat boiler, i.e. steam generator, was used, it was usually necessary to make some provision for feed heating (except where the main engine was operated at, say, more than 15,000 b.h.p.) if all power and heating services were provided from this source. The design shown could, obviously, easily be arranged to incorporate a feed heating section of forced circulation type.

The reasons for the pressure difference of boilers in forced circulation systems were, of course, subject to argument and any reduction in frictional losses in the piping system must improve the position.

With reference to Mr. Alsen's comments on charter rates, it was within the author's personal knowledge that, in the past, ships were presented for charter with claims made that they would not require fuel for auxiliaries. Analysis of such claims disclosed that considerable optimism had been shown in the calculations and the quality of the equipment put forward was such that the initial performance was unlikely to be sustained. It had now been demonstrated that where sound design had been applied and good quality equipment used, the claims could be substantiated. In such cases and in a competitive charter market, the owners must benefit, either by having their ships chartered against the competition from less efficient ships, in situations where the charterer paid for the fuel, or by the savings in fuel cost if the owner paid for the fuel.

Mr. Milton had listed the variables which made accurate estimation difficult and it must be accepted that the decision on output had often to be made on inadequate and incomplete information. Nevertheless, reasonably accurate estimates could be made if care was taken and the estimator was reasonably experienced in marine engineering. For example, details were given, on page 413 of the paper, of the estimated requirements for the plant in *British Venture*. It would be seen, from the left hand column, that the design covered for a total steam output of 11,500lb./hr. under cold weather conditions against an estimate of 11,580lb./hr. Fig. 26 showed a typical measured feed flow of 10,000lb./hr. under temperate zone conditions and Fig. 25 showed a flow of 11,100 under hot zone conditions. The apparent over-estimate was probably due to: i) turbo-generator actual steam rate on shop test being better than design (−450lb./hr., say, from Fig. 3); ii) bunker heating being less than that allowed due to the use of 1,500 sec. fuel (compared with 3.500 sec. design) which might be transferred with reduced preheating; iii) reduction of evaporator output in cold weather when potable water requirements usually decreased. The effect of this in service was to provide a steam pressure of about 180lb./sq. in. gauge instead of the 142lb./sq. in. gauge of design. The electrical loading estimate was accurate as could be seen from comparison of curves (b) and (c) in Fig. 9.

The author did not have access to any service records of cylindrical-cased pancake type element heat exchangers which would enable him to comment on them, but some criticisms were made on pages 69 and 92 of the De Lavals Angturbin proceedings listed in the bibliography. Such units obviously had the disadvantage of any "once through" type of boiler in that any impurities in the feed water were deposited on the heating surface when the evaporation was completed.

Recovery time for the extra initial cost of waste heat plant needed to be separately calculated for each installation. The four-year period quoted by Mr. Milton could easily be met for ships operating at about 10,000 b.h.p. or above.

The control of air temperature to give a slight increase in exhaust gas temperature, as referred to on page 402, would only be applied under cold weather conditions. The application would not affect the engine component cooling since the cylinder cooling temperature was kept constant and the effect would only be to restore the scavenge air temperature to that which would apply under temperate zone operation of the engine.

In reply to Mr. Frank, the author said that, although a crash stop of the main engine at sea or a rapid power reduction without warning were very rare occurrences, precautions were taken to guard against interruption of essential services. One safeguard was shown in Fig. 19, where a steam pressure sensing point was shown near the turbo-alternator steam inlet. In the event of an emergency stop, the steam pressure would begin to fall and the rate of fall would depend upon the thermal capacity of the gas/water heat exchanger and of the boiler. This provided a time margin of several minutes in which manual lighting-off of the boiler could be carried out, or a Diesel generator set could be started. If the steam pressure fell below design, either the watertube boiler or the package boiler firing sequences would be automatically started and the Diesel generator set would also start automatically. Ships already in service did not have the automatic arrangements for boiler firing and the reason for adding this refinement was to make the whole sequence automatic and obviate the need for manual intervention in emergency.

The evaporation turn-down ratio possible could be illustrated by reference to Fig. 14. Under maximum control conditions, the steam pressure would rise and the sequence of events described for Fig. 16(c), on page 409 of the paper, would have taken place. The location of the valves was shown in Figs. 18 and 19. Reverting to Fig. 14, the circulating water would now enter the waste heat unit at the saturation temperature of 417 deg. F. (214 deg. C.) corresponding to the maximum boiler pressure, instead of at 240 deg. F. (116 deg. C.) as at the design conditions. If line ED was now redrawn at a temperature of 417 deg. F. (214 deg. C.) and the gas water temperature difference remained the same as indicated at D on the figure for the design condition (in practice it reduced slightly), the total heat extracted from the gases would be about 3,500,000 B.t.u.

In comparison with the 8,500,000 B.t.u. designed extraction, this gave a turn-down ratio of 7 : 17. Since the turbo-generator set was intended to be operated at all times, at sea, the turn-down ratio available was greater than required and the 20 per cent evaporation postulated by Mr. Frank was not related to the plant operational requirements. Nevertheless, reference to Fig. 19 would disclose a silent blow-off connexion arranged to the auxiliary condenser from the L.P. steam line and Fig. 18 showed emergency steaming connexions which allowed the waste heat unit to be operated as a "once through" unit with feed flow regulated to steam demand. It was most unlikely that these latter devices would ever be required for output control only, but their use would enable an infinite turn-down ratio to be applied.

While the author agreed that economics might sometimes indicate that a lower than normal pinch temperature should be used, he was reluctant to accept a "tail end" temperature difference of much less than 80 deg. F. (44 deg. C.) between gas and water. This was because of the effect that a thin layer of carbon on the gas side, or of scale on the water side, would have on heat transfer at the small temperature differences involved.

Replying to Mr. James the author said that he appreciated the reasons for the turbine designer retaining a margin for contingencies, but suggested that it was unreasonable that he should have to include in the margin an amount to offset the undue optimism or design errors of others. Some one person or company must be responsible for co-ordinating the overall design and component outputs of the waste heat recovery system, including the generating plant. Designers of the calibre of Mr. James could well do this, but the author considered that the owner's superintendent was best suited for the duty, since he had access to operating records which indicated the extent that the many variables might influence output requirements.

At the modest steam pressures used in motor ships, it was accepted that a turbo-generator could operate satisfactorily with saturated steam. As Mr. James was aware, superheat should be welcomed by the operators, in addition to the designer, since it reduced the possibility of water carry-over damaging the turbine. The major advantage followed from operation under tropical conditions where the steam rate of a machine operating on saturated steam would rapidly increase as vacuum fell under the influence of sea temperature and increasing load. If superheated steam was provided, the increased exhaust gas temperature increased the superheat and would keep the steam rate of the machine almost constant. The effect was indicated by Fig. 4 and by the relevant curves in Fig. 11.

The provision of bled steam from the turbo-generator to ships' heating services would reduce the cost of the gas/

water heat exchanger in certain cases, since the total steam requirements from a single-pressure steam system were thereby reduced due to the extraction factor at the turbine bleed point. As mentioned on page 408 of the paper, the provision of heating steam was a simple problem; provision of adequate steam for power generation was more difficult. Any artifice which increased the demand for H.P. steam—as did a bled steam connexion—must impose a more arduous requirement on the most critical part of the system. In addition, the pressure available at the bleed point must be low, in view of the modest initial pressures usually employed, and would vary with the electrical load. It would seem that this arrangement should only be used where the H.P. steam potential of the system substantially exceeded the steam required for power generation.

It was agreed that large bulk carriers could use waste heat recovery plant with advantage and obtain substantial reductions in operating costs. As Mr. Scrimgeour pointed out, the additional cost for fuel, incurred by running the turbo-generator set in port, was only small, since although the fuel required per kW/hr. for the turbo-generator was about twice that of the Diesel, high viscosity fuel was only about half the cost of Diesel grade fuel. A further marginal advantage followed from reducing or eliminating the need to ship separate grades of bunkers.

The maintenance costs of 600 r.p.m. Diesel generators, quoted in the paper, were taken from company records. As the higher speed machines were just entering service, similar direct evidence was not yet available, but estimates and advice from others indicated that the costs would be similar. In view of the reliability of the turbo-sets and the requirement that they should be operated continuously, the Diesel generator sets in all motor ships on order for the company would operate at 1,200 r.p.m. The cost, weight and size of the Diesel sets would thus be reduced in comparison with the medium speed machines and, since they would only be operated when manœuvring or when the ship was in narrow waters, the maintenance costs should be small.

As mentioned on page 400 of the paper, soot blowers were not being fitted to the waste heat units in ships now on order. The decision was made on evidence, from two ships, that they made little difference to performance; since that time some contrary evidence had been obtained from another ship and it might be necessary to reconsider the decision. In the latter case the main engine was of a type that required a comparatively large amount of cylinder oil per b.h.p. and this might well be the controlling factor. Portable boiler water washing equipment was, in any case, being provided for use on the oil fired boilers of ships now on order and this equipment could, of course, be used on the waste heat units if required.

The author accepted that the legends on the illustrations in the turbo-generator instruction book might appear to be too numerous. The detail did give an advantage, in that components could be clearly identified in correspondence by using the correct part name, and avoided ambiguity and uncertainty which followed if a part was described by a name which might vary from one part of the country to another, because of local usage. This particular instruction book was an exceptionally good one and could well be used as a model by U.K. manufacturers of other components who, in general, did not produce satisfactory manuals.

The detailed work carried out by Commander Lake's company on water/water heat exchangers, designed for low temperature differences and the progressive elimination of margins, had been successfully applied to components described in the paper. Temperature measuring equipment was now being improved and it should be possible to obtain more accurate readings in the future, which would help in considering new designs of heat exchangers. The packaged steam generator/feed heater shown in Fig. 34 was a neat design which could be expected to reduce shipbuilders' installation costs. The fact that certain builders preferred to purchase the components separately did not disprove this, but probably indicated that they had labour available which it was desirable to employ on this type of work. Nevertheless, it was considered by the author that the package principle for all types of plant was one that would need to be increasingly applied in future, in order to reduce machinery installation costs.

In his reply to Mr. Wilson, the author said that he did not subscribe to the view that initial extra costs of waste heat recovery plants should necessarily be recovered within three years, even though this could be achieved in ships operating at more than 14,000 b.h.p. without recourse to tax allowances to prove the case. Although quite outside both the subject of the paper and the author's duties, it should be pointed out that the present complex U.K. tax structure allowed a 40 per cent investment allowance and a depreciation allowance of up to 15 per cent of the investment to be offset against profits after the end of the first year of operation. If such allowances were viewed in conjunction with savings in fuel cost and reduced maintenance cost, the justification of the increased investment for good quality plant became even stronger.

With steam pressures in the modest range of 100-140lb./sq. in. gauge usually applicable to waste heat recovery plant at sea, there was no danger of turbine blade erosion with saturated steam, provided that the steam off-take point from the boiler was positioned in accordance with normal marine practice. Under such conditions the steam could be assumed to be about 95 per cent dry. If the boiler pressure was arranged to be in the 350-400lb./sq. in. gauge range, when oil firing was being used in port and saturated steam was supplied to the turbine, then the last stages of blading could be in the potential damage zone. The author had no knowledge of damage being sustained in machines fitted in motor ships, where the pressure was usually less than 300lb./sq. in., but there had been, of course, many instances of erosion on the last stages of main turbine blading in steamships which had been operated at reduced powers.

Acceptable Reynolds numbers for heat exchanger design were outside the scope of the paper but, as Mr. Wilson was well aware, the designers of such equipment should rely upon calculations based on experimental data in evolving proprietary designs.

Latent heat cooling did not seem to have been applied in the marine field for the reasons given on page 410 of the paper. The principle was well known and a circuit, such as that shown in Fig. 17(a) and utilizing a low steam pressure, could obviously provide a substantial amount of power. Many technical problems would be involved, however, and the use of a low steam pressure would make the steam producing and using components large and costly.

The author would agree with Captain Stewart that only a limited amount of heat recovery could be economically justified in Diesel locomotives, in view of their 50 per cent load factor. While the author had never examined the issue in detail, it would seem likely that the heat recovery could best be directed towards carriage warming, by means of a small heat exchanger in the exhaust ducting used to preheat feed water, where boilers of the "once through" types were used, or in a forced circulation loop where boilers with a submerged heating surface were used.

The exhaust gas temperature change with increasing load was shown in Fig. 11; in each case the temperatures were on the right-hand scale and showed the increase as the load approached maximum. This would perhaps have been made more clear if arrows had been attached to the curves to indicate which scale should be read.

The heat in the exhaust gases from a large marine engine at part load probably remained at approximately 35 per cent of the total heat, as could be expected from the rather flat fuel consumption curves from such engines. The variation of fuel consumption with load was shown in the following results from an 18,000 h.p. engine recently tested:

	50 per cent	75 per cent	100 per cent
Engine r.p.m.	96·0	110·0	121·0
B.h.p.	9,000	13,630	18,130
Specific fuel rate	0·366	0·350	0·348

Gas temperature after turbine, deg. F.	495	560	600
Total air flow, lb./hr.	192,100	250,800	312,300
Total gas flow, lb./hr.	195,390	255,570	318,600

It would be seen that the small percentage variation of heat to exhaust for smaller four-cycle engines was given in the comments made by Mr. Howe.

Replying to Mr. Francis, the author said that the attraction of using a combustor for port use in association with the exhaust gas heat exchanger might be peculiar to tankers but was two-fold. It removed the steam generating surface from the radiant heat zone usual in a boiler (and where tube failures were most likely to occur) and hence should improve safety and reliability factors. It should also reduce the overall installation cost since the large heat exchanger surface for use at sea would also be used in port. Since the combustor gases might be as high as 1,100 deg. F. (593 deg. C.) at entry to the heat exchanger, in comparison with, say, 650 deg. F. (343 deg. C.) of the exhaust gases at sea, the steam production under port conditions would be greater than the sea output. This dual duty would thus reduce the size and cost of the boiler plant fitted for port use. It was attractive for tankers where the large steam output was dependent on the requirements for cargo pumping, cargo heating, tank cleaning and steam-operated deck machinery. Gas from a combustor could be used for inerting tanks during cargo discharge periods, but, in view of the usual excess air requirements of combustors, some form of gas recirculation would be necessary to enable a gas CO_2 content of at least 12 per cent to be obtained to ensure that the remaining oxygen content was sufficiently low for inert gas applications. The source for inert gas for large motor tankers now building was from the boiler uptakes—flue gases from the main boiler would be used under cargo discharge or ballasting conditions and from the package boiler at sea when small quantities were required for topping up. Since the gas was thus from a source which was free, no change of supply cost would be involved in substituting a combustor for the boiler. The economy offered by inert gas installations was, of course, a long term reduction in repair costs following reduction of corrosion in the cargo tanks, and the attraction of inert gas lay in the enhanced safety feature.

With reference to the exhaust gas-driven turbochargers, the author considered these to be the most vulnerable part of the modern highly rated engines. While present requirements for larger powers in smaller machinery spaces enforced the use of turbocharged engines, it was considered that their reliability was less than that of their normally-aspirated predecessors. This was particularly true in tankers, where opportunities for piston ring renewal were less frequent than in other ships and where broken piston rings, passing exhaust gas guard grids, had frequently caused blower damage. Such damage might become less frequent in future, however, as cylinder oils were now available which reduced piston ring wear to a much lower rate. Substitution of a steam turbine-driven blower would undoubtedly increase the overall reliability of the plant and improve operation at part engine loads. The initial cost of the plant would be high, however, and controls would be necessary to avoid blower surging at part engine loads.

Waste heat recovery plants used up to the present time, had been provided for auxiliary power and heating services only. Since the hotel and ship's services portions of these loads varied independently of main engine power, the final exhaust temperature must vary accordingly at outputs below the nominal design quantities. Only at larger outputs, where supplementary oil firing was necessary, would the main engine exhaust temperature at the funnel base be constant. This latter condition was unlikely to apply with the large Diesel engines of 16,000 s.h.p. and above which were now in service and where a reduction of gas temperature from, say, 650 deg. F. (343 deg. C.) to 400 deg. F. (204 deg. C.) enabled all normal power/heating loads to be met. The gases could safely be reduced by a further 100 deg. F. (56 deg. C.) but, although it would be possible to evolve means of transmuting the corresponding amount of heat into power at the propeller shaft, it was unlikely that this could be justified on an economic basis. The author's preference would be to use this surplus heat to improve the scavenge process by driving a steam turbine-driven blower, as previously discussed. A compound heat recovery system, using both jacket water and exhaust heat with heat exchangers in series, could also be evolved and, provided that a very low steam pressure was acceptable, this could provide a substantial amount of power to divert to scavenge air supply or propulsion duty.

The paralleling of turbo and Diesel-driven generators, as mentioned by Mr. Francis, presented a problem, but the provision of check synchronizing devices was now an accepted safety feature to prevent machines being coupled out of phase.

With isolated exceptions, the similar but earlier installations, which were cited by Mr. Church, did not require to support auxiliary loadings of the orders now imposed. Many of the earlier main engines had attached pumps, hence the remaining auxiliary load was only that required for lighting and power. Also, it was only in the last few years that full air conditioning of accommodation spaces and the provision of electric galley ranges had become accepted features, with a consequent increase of auxiliary loadings. The plant shown as Figs. 15(h) and 16(c) in the paper was designed to accept these greater present day loadings.

It was agreed that to adjust an engine for a degree of air starvation in order to increase waste heat recovery potential would be most unwise. The author did not, in fact, recommend this, nor would any other responsible engineer. The troubles mentioned by Mr. Church were those which were usually due to thermal overloading of a turbocharged engine. The overloading might, of course, be caused by forcing the engine to maintain a charter speed under conditions where hull fouling had led to a deterioration in r.p.m. in comparison to the original condition, by operating with turbocharger air filters dirty or with fouled blading, or by excessive back pressure on the turbochargers due to choked air coolers, or deposit in cylinder liner ports, or leaking and fouled tubes in a waste heat boiler.

The author also agreed that simplicity was desirable and considered that the provision of an unfired waste heat unit, operating at sea in conjunction with a reliable turbine-driven generator, gave such simplicity insofar as the power generating plant was concerned. If the systems were properly designed, the plant needed little attention and hence was suitable for any reduced manning scale which was likely to be applicable in future. In considering the diagram shown in the paper as Fig. 19, it must be borne in mind that it was for a tanker and, in such ships, an auxiliary steam installation of some type had to be superimposed on the Diesel plant in order to meet the large steam demands for heating cargo, for tank cleaning and for deck machinery. There was little additional complication in providing the steam supply to turbo-generator sets. In a Diesel ship of the dry cargo type, the steam installation would be greatly simplified.

Control of the circuits shown in Figs. 15(h) and 16(c) of the paper was described on page 408, item (4), and in the last paragraph under "Indirect L.P. Steam Generation" on page 409, respectively. The valves used were of standard air-operated type, widely used for other shipboard applications. Correct initial designing, to make the waste heat system self-supporting at normal sea load, meant that, at electrical or steam loadings below design, the maximum steam pressure was simply controlled by automatic operation of these valves; the other methods suggested by Mr. Church were not necessary. At steam pressures below the maxima permitted by the valves, the steam pressure would vary inversely with loadings, but the pressure variation was not passed to the turbo-set, since the steam was throttled in the control valve responsive to electrical loading. If total steam requirements exceeded design, for example when deck machinery or cargo handling equipment was in use at sea, then supplementary oil firing of the boiler was necessary.

Replying to Mr. Ellison the author said that the use of reciprocating steam generating sets to absorb surplus steam from waste heat boilers had been widely accepted for some years. The disadvantage of such machines was that the steam rate was too high to enable them to carry the full sea load of a modern ship, when supplied with steam solely from a waste heat source. The alternative of using turbines operating on superheated steam and with high vacuum effectively reduced the steam rate to one half. The refrigerated cargo ship could form an exception to the general rule mentioned, on economics not justifying parallel operation of turbo and Diesel-driven generating sets at sea. The governing factor for near maximum waste heat recovery on such ships was probably the number of days spent at sea per annum. If economics justified use of a turbo-generator operating on steam from a waste heat source to meet the (lighter) loadings imposed on outward passage, then the greater loadings on the homeward passage could be met either by operating a Diesel set in parallel or providing additional steam for the turbo-set by supplementary oil firing of the boiler. Since the cost per kW of the turbo-generator decreased as size increased (Fig. 2) it was probable that the most economical generating installation would be obtained by increasing the size of the turbo-set to meet the greatest sea loading, and reducing the number of standby Diesels sets by one.

No scale prevention treatment was used with the flexible element evaporators mentioned in the paper. The units used had been of similar design, but of two separate makes, and the make that gave most satisfactory service had a well-positioned "shocking" water pipe arranged for rapid quenching of the element; the thermal shocking allowed the element to shed scale effectively.

With reference to the arrangements for air attemperation, this had not been carried out as yet. When and if applied, it was not expected to increase the fuel consumption and would certainly not involve any measurable amount. It would be borne in mind that the air flow was not reduced, but, because of the slight change in temperature of the air trapped in the cylinder at the beginning of the compression stroke, the weight would be slightly reduced. Therefore the main engine specific fuel consumption should correspond to that which would apply under the ambient temperature which allowed the waste heat system to carry the normal loading. The temperatures quoted on page 403 were all at the turbocharger exhaust, i.e. the waste heat unit inlet.

The method of control of the steam generator was referred to under the heading of "Indirect Feed Heating" on page 407, and, although very simple—in essence controlling the temperature of water entering into heat exchange with the gases and so reducing output—gave output control over a wide range, as discussed in the reply to Mr. Frank. The author did not favour systems where the full capacity was generated under all conditions, since any steam dumping arrangements involved a condenser having to be available to receive the dumped steam. Therefore, if the dump-line was arranged to be led to the turbo-alternator condenser, the heat dissipation might not be available if the machine was stopped and, if alternatively arranged to an auxiliary condenser, this unit must be circulated at all times. In addition there was always a danger in allowing high pressure steam to enter a condenser where impingement on tubes might occur. The other alternative of isolating generating tubes was also undesirable, insofar as when any section of tubes was isolated, the water therein was evaporated to form steam. This would have the effect of initiating deposits on the generating surface. These deposits did not occur if the tube was continuously swept by water while steam formation was occurring.

In reply to the joint contribution by Mr. Parris and Mr. Wagner, the author said that the internal fluid temperature of 300 deg. F. (149 deg. C.), preferred by the contributors, was undeniably a safe one, but the acid dew-point was well below this temperature and allowance had to be made for the typical depression of dew-point temperature in Diesel exhaust gases, which was probably due to excess air quantity. In arranging for an inlet fluid temperature of 240 deg. F. (116 deg. C.) with steel elements in circuits, as shown in Figs. 15(h) and 16(c) of the paper, it must be pointed out that this temperature was only applicable when the output was at design conditions. If this designed output was adequate to make the waste heat system self-supporting, under the maximum steam loads imposed under normal ship operating conditions, the effect of the load variations—as shown in Figs. 10, 21, etc. of the paper—was to increase the inlet fluid temperature on load reduction. An added measure of protection was thus available.

It had been found in practice that, where the waste heat unit imposed the nominal back pressure of 4in. w.g. mentioned, silencing and spark-arresting duties were adequately covered. This had been demonstrated with waste heat boilers of smoke tube type and with economizer type units of both staggered and rectangular lattice tube pitching. The rectangular tube pitching was preferred by the author because of the improved facility for inspection of the heating surface.

Although it was accepted that a downward flow of water introduced some difficulties, it was considered desirable to maintain the contraflow principle over as much of the heating surface as was practicable. This, in some measure, compensated for conditions where the engine was operated at power lower than design—a condition not infrequent when operators could only check powers from indicator cards. It also allowed for a margin if gas flow or temperature on trials was less than the design expectation.

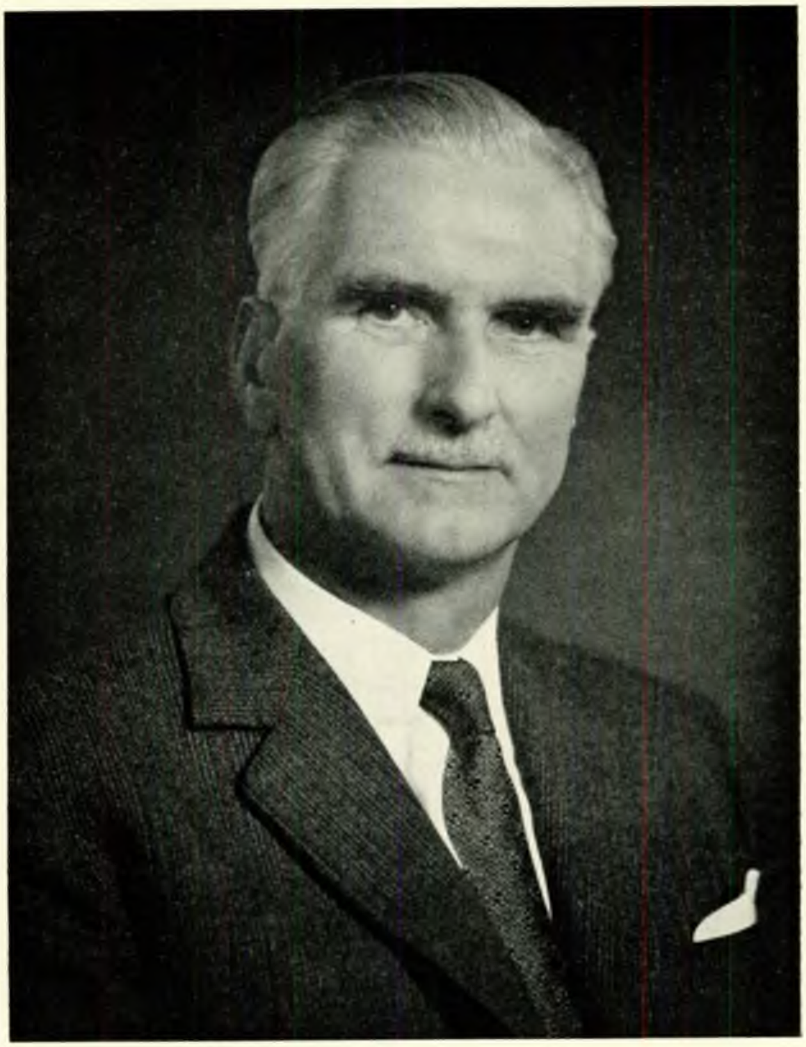
Re-positioning of the feed regulators to place them before the economizer, in the circuits shown in Figs. 15(e) and 15(f), was recommended by the contributors. The author considered that this would not eliminate the hazard from these circuits since, although the amount of pressure built up in the elements under the condition of interrupted feed flow would not be the same, the installations would still be subject to a rapid change of temperature under surging conditions. These changes might be accepted with equanimity in a new installation, but the effect of aging must always be considered.

It was agreed that the maximum economizer duty was to raise feed water to saturation temperature. However, as pointed out in the paper, this could only be achieved for the particular design condition with circuits such as that shown at (h) in Fig. 15. At lower feed flows, the tendency to form steam was inescapable, whereas, with the indirect feed heating method of circuit (h), the feed water could not possibly be raised above the saturation temperature, nor could the temperature of the circulating water be unduly depressed. In addition, the arrangements for feeding the boiler of circuit (h) were quite straightforward and in accordance with methods established during decades, i.e. a straightforward flow through a feed heater to the check valve, without interposing any mixing valves or device which might become subject to maladjustment. The cost differential between indirect feed heating and other methods was only slight when automatic controls of system output were incorporated in the comparison. Controls for indirect feed heating were simple and well tried. With other methods it was usual to have to arrange for some form of gas bypass incorporating a large valve, and ducting and the possible addition of a silencer. An alternative method sometimes proposed was to arrange for isolation of sections of the heat exchanger on the water side; this arrangement had the disadvantage of deposit being left on the tube surface on each occasion when the water had all been evaporated and the cumulative effect would be to interfere with heat transfer.

The author said, in reply to Mr. Howe, that it had been found that engineers of all grades in motor ships rapidly became accustomed to turbo-generator sets and soon preferred them to the conventional Diesel generators. The turbine set ran smoothly and did not require adjustment whilst running, as a Diesel might do. The turbine set also required a steam source and this might cause some uneasiness in the minds of a few engineers unaccustomed to boilers. If the waste heat plant was self-sustaining at sea, the boiler fires would not be alight and the only attention the boiler required was that some water should be available in the drum in order that heat extraction from the gases might be continued. This combination, of a rotary unit to generate power and a safe stationary heat exchanger, seemed to be the essence of reliability and to approach the simplicity of machinery that was possible in the days when all essential units for main engine safety were driven by the main engine, i.e. if the main engine was running, all was well.

As implied on page 410 of the paper, the author agreed that, in low powered vessels or in ships which spent only a small part of their time at sea, there was no case for using a turbo-generator set and Diesel generators provided the most suitable generating plant. In larger ships where one turbo-set was provided, the Diesel generators could provide standby generating capacity in the most economical way. For example, a class of 62,000 d.w.t. motor tankers on order were to be fitted with one 600 kW turbo-generator and two 342 kW, 1,200 r.p.m. Diesel generators. The ships described in Appendix A of the paper formed an exception to the general rule since they had two 550 kW turbo-sets and one 240 kW, 1,200 r.p.m. Diesel set; in this instance however the cargo pumps were electrically driven and the whole installed generating capacity was required if cargo was being discharged at the full rate. Such conditions did not apply in other ships where there was usually 100 per cent standby generating capacity available at most times.

The maintenance costs given in the paper did not include that for the waste heat recovery plant since it was considered that these costs would equate with those for the waste heat boilers which would otherwise be installed to supply heating services only.



A. LOGAN, O.B.E.

ALEXANDER LOGAN, O.B.E.

Mr Alexander Logan is the elder son of the late Robert Leitch Logan, who was a chief engineer in the Federal Steam Navigation Co., and later superintendent engineer for Mitchell Cotts and Co. He served his engineering apprenticeship with R. and H. Green and Silley Weir, Blackwall Yard, and attended classes at West Ham Technical College and the L.C.C. School of Engineering, Poplar.

In 1924 he went to sea as a junior engineer with the Royal Mail Steam Packet Company, serving in their steam and motor ships. After gaining his Ministry of Transport certificates of competency, he joined the Anglo-Saxon Petroleum Company and continued seagoing service with that company until 1936 when he was appointed an assistant engineer superintendent, engaged in supervising new buildings in Odense, Denmark and Monfalcone, Italy.

During the war he had a long spell in Malta where besides being responsible for ship and engine repairs he assisted in the operation of captured enemy ships. On his return to this country he was stationed first on the North East coast and later in the Mersey area.

In 1945 he became superintendent engineer in charge of repairs and maintenance of the British Shell Fleet and later, with Mr Hugh Armstrong supervising new building and Mr W. J. Brown in charge of repairs, he succeeded Mr John Lamb as Manager of the Marine Technical Department. On the formation of Shell Tankers Ltd., in 1953 he was made Technical Director and, from March 1959 until his retirement in June 1963, was on the board as Deputy Managing Director of that Company.

Throughout his career Mr Logan has taken an active interest in the affairs of the Institute of Marine Engineers and, in 1958, for his paper on "Corrosion Control in Tankers", he was awarded the Denny Gold Medal. He was Chairman of Council in 1949 and elected a Vice-President in 1953. For many years he has been one of the Institute's representatives on the Technical Committee of Lloyd's Register of Shipping and, until his retirement from Shell Tankers, was a member of the Council of the U.K. Chamber of Shipping. He represented that body on H.M. Minister's Nuclear Committee and on the Ministry of Transport's committee on oil pollution.

At present besides being Deputy Chairman of Beldam Asbestos Co. Ltd., he is a member of the Fire Research Board, Department of Industrial and Scientific Research, and is Chairman of the American Bureau of Shipping British Technical Committee.

He was appointed an Officer of the Order of the British Empire by Her Majesty The Queen in the Coronation Honours.