The Design and Layout of a 22,000 s.h.p. Tanker Machinery Installation

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The paper discusses the choice of steam condition and the provision of the major auxiliary drives, etc., in the 22,000 s.h.p. machinery plants now being installed in three 65,000-ton d.w. oil tankers.

The terms used in assessing profitability of the capital investment in machinery are advocated as a logical basis for design. The author suggests, however, that foreseeable trends in engine room maintenance and manning problems urge the pursuit of even further simplification of propulsion and auxiliary equipment, if need be at some sacrifice of plant efficiency.

INTRODUCTION

The intent of this paper is to outline an approach which has been made to the design and layout of the steam turbine machinery in a 65,000-ton deadweight tanker.

The treatment is necessarily brief. Those features of the installation which the author believes constitute a departure from the general practice in many recently built tankers are alone considered at any length.

The specification aimed at the provision of nothing exceeding bare adequacy in the number and complexity of auxiliary units, because it was felt that the balance of capital cost, fuel cost and probable maintenance and repair costs, should be biased substantially in anticipation of a continued rise in the cost of repair yard labour throughout the life of the ship.

The following assumptions have been made in assessing the economic logic of the design:

1) Oil fuel cost—£6 per ton.
2) 320 days per annum in the life of a tanker operating Middle East/N.W. Europe are spent under way at sea.
3) Capital charges representing amortization, insurance and service of finance are of the order of 14 per cent of the initial investment. In other words, the expenditure of £1,000 on any machinery item saddles the vessel with an annual capital charge of £140. Where the anticipated effect of such expenditure is an improvement of plant performance it follows that the amount of the improvement must not be less than will yield £140 per annum reduction in operating cost (for example in the fuel bill).

CHOICE OF STEAM CONDITION

There is no completely certain approach to the determination of the optimum steam condition. One might say that for an engine of this power the choice of pressure lay between 600lb./sq. in. gauge and 900lb./sq. in. gauge and the choice of temperature between 850 deg. F. and 1,100 deg. F. The lower limits of 600lb./sq. in. gauge/850 deg. F. would be suggested by the now substantially proven performance of many steam turbine installations built in the last ten or twelve years. The upper limits are extremely difficult to assess. In considering the use of steam at say 1,100 deg. F., the sharp rise in capital cost due to the introduction of austenitic steels for pressure parts subject to the maximum steam temperature is perhaps less difficult to justify than is the acceptance of the relatively unknown degree of risk of severe maintenance trouble which could be encountered with the superheater elements of the boilers.

The pressure was pegged at 600lb./sq. in. gauge because the incidence of pipe joint and valve leakage in recently built new tonnage had not suggested that higher pressure could justifiably be adopted. Conceding that satisfactory experience is now being gained with welded pipe connexions and with valves and fittings manufactured by specialists, it was decided to use those advanced techniques as additional insurance against steam and feedwater leakage rather than to push the design operating pressure to a practical limit.

It is apparent also that in boilers designed for pressures higher than 600lb./sq. in. gauge tube failures due to internal deposits can become much more prevalent unless pre-commission cleaning of the circuit, and subsequent feedwater quality control of an order somewhat higher than the average practice obtaining, is ensured.

Steam temperature was decided at 900 deg. F., principally because it was felt that beyond this figure the onset of corrosive attack on the superheater elements and their supports by vanadium ash, and more serious superheater slagging difficulties, could be expected.

So much, then, for the reduction in fuel rate which could have been won by the adoption of more advanced initial steam conditions. 600lb./sq. in. gauge/950 deg. F. would have improved the fuel rate by about 2 per cent or $2\frac{1}{2}$ tons of fuel a day. A pressure advance to 850lb./sq. in. gauge coupled with 850 deg. F. temperature would have yielded about one ton a day saving. 850lb./sq. in. gauge/950 deg. F. would have reduced the fuel rate by 5 tons a day.

It is well known how rapidly the penalties payable for any sacrifice of reliability can operate to restrict the savings indicated by these estimated improvements in fuel rate. In the complex which goes to make up a seagoing machinery installation there is undoubtedly more scope for real advance by consolidating and rationalizing designs at the present level

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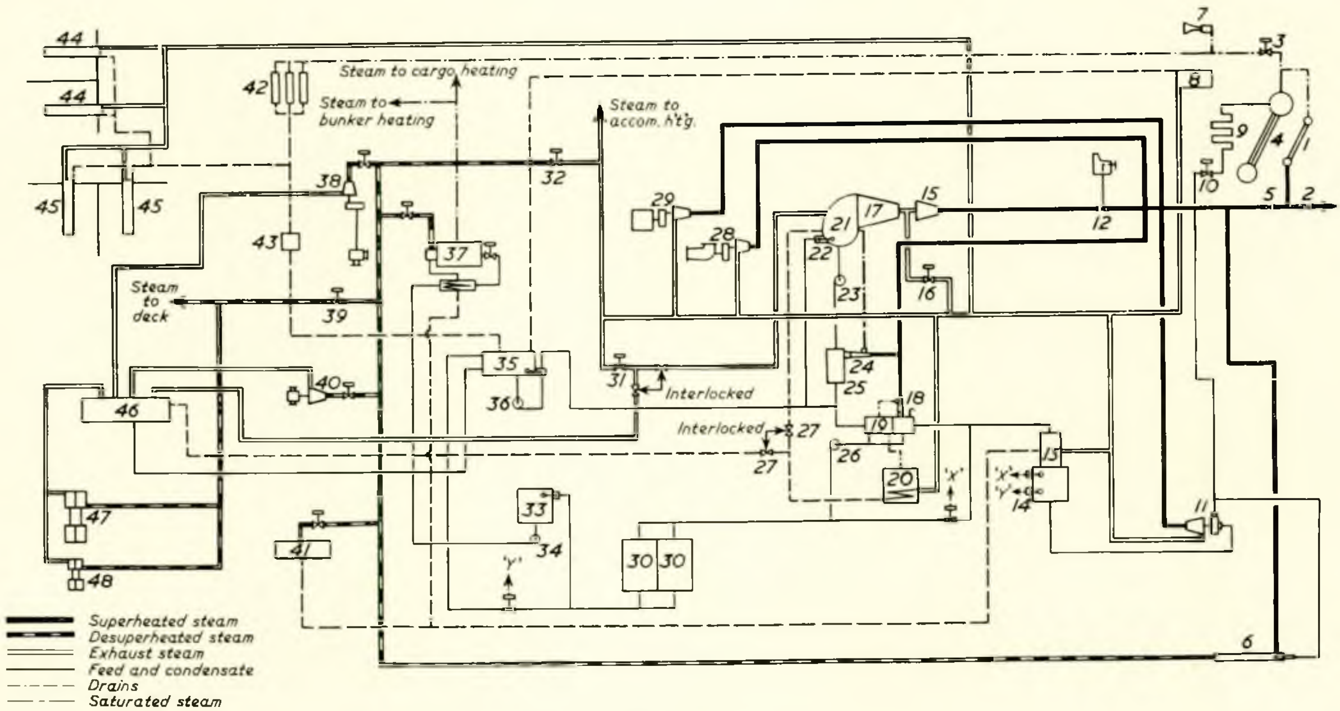

FIG. 1—*Flow diagram*

1) Superheater
2) Steam to sootblowers
3) Saturated steam controller
4) Main boiler
5) Main steam stop
6) De-superheater
7) Whistle
8) Air preheater
9) Economizer
10) Feed regulator
11) Main feed pump
12) Main throttle
13) De-aerator
14) Level controls
15) H.P. Turbine
16) Bleed controller
17) L.P. Turbine
18) Evaporator air ejector
19) Evaporator condenser
20) L.P. Evaporator
21) Main condenser
22) Recirculation controller
23) Main extraction pump
24) Main air ejector
25) Ejector and gland condensers
26) Evaporator condensate pump
27) Evaporator coil drain
28) Main circulating pump
29) A.C. generator
30) Distilled water tanks
31) Surplus exhaust controller
32) Exhaust makeup controller
33) Steam generator feed tank
34) Steam generator feed pump
35) Atmospheric drain tank
36) Drain tank extraction pump
37) L.P. steam generator
38) Cargo pumps
39) Deck steam controller
40) General service pump
41) Sea water heater
42) Oil fuel heaters
43) Condensate filters
44) Sett tank heaters
45) Bunker heaters
46) Atmospheric condenser
47) Cargo stripping pumps
48) Fuel transfer pump

of steam conditions than by moving prematurely into advanced pressure and temperature cycles.

BASIC FEED AND STEAM SYSTEMS

The decisions to adopt non-condensing turbogenerators and a steam turbine drive for the single main circulating pump have fundamental influence upon the final shape of the system, and justification of these decisions is shown later in the paper.

The boiler feed and the main and auxiliary steam systems are shown in the form of a flow diagram in Fig. 1 which has been detailed to include all information likely to prove of interest.

The steam and feed flow quantities for normal sea service operation without cargo heating and without air conditioning are shown in Fig. 2 which has been reduced to the simplest possible form whilst retaining all relevant data. The non-bleed steam rate of the propulsion turbine at 28·5in. Hg vacuum exhaust is based on a heat drop per pound of steam of 477 B.t.u. and a reduction gear efficiency of 97 per cent. Allowances shown for evaporator output and domestic heating steam are based on average figures known to be realistic in practice. The operating cycle of the condensate circulated sea water distiller has onlv minor effect on the nett balance of auxiliary exhaust steam flow since the intake of steam from the exhaust range by the evaporator heating elements is offset by a reduction of heating steam intake by the de-aerator.

Fig. 2 is computed for ship operation in sea water at 60 deg. F. The changes consequent to operation in the tropics are:

a) Increase in turbogenerator loading of about 80 kW due to the onset of accommodation air conditioning and increased machinery space ventilation rate.
b) Increase in circulating pump load in maintaining condenser vacuum and lubricating oil temperature.
c) Increase in main condensate temperature as vacuum drops (this despite the attempted corrective action of item (b)).
d) Some reduction in the exhaust steam consumption of the boiler air heaters as a consequence of the higher ambient temperature in the boiler room.
e) Shut-down of accommodation space heating.

The nett deficiency in auxiliary exhaust heat in Fig. 2 (as represented by the quantity bled from the H.P. propulsion turbine exhaust) will thus reduce under tropical operating conditions, and it will be evident that the efficiency of the cycle will fall off progressively as the condenser vacuum and the bled steam quantity reduce. The approximate revisions to the heat balance are shown in parenthesis in Fig. 2.

It would be possible of course to provide a low pressure bleed point and incorporate an L.P. feed heater in the cycle. The heat balance would then be as in Fig. 3 which again includes the figures for tropical operation. It will be seen that a fuel saving of about 0·54 ton/day is obtained but that when the ship is operating in the tropics the specific fuel rate

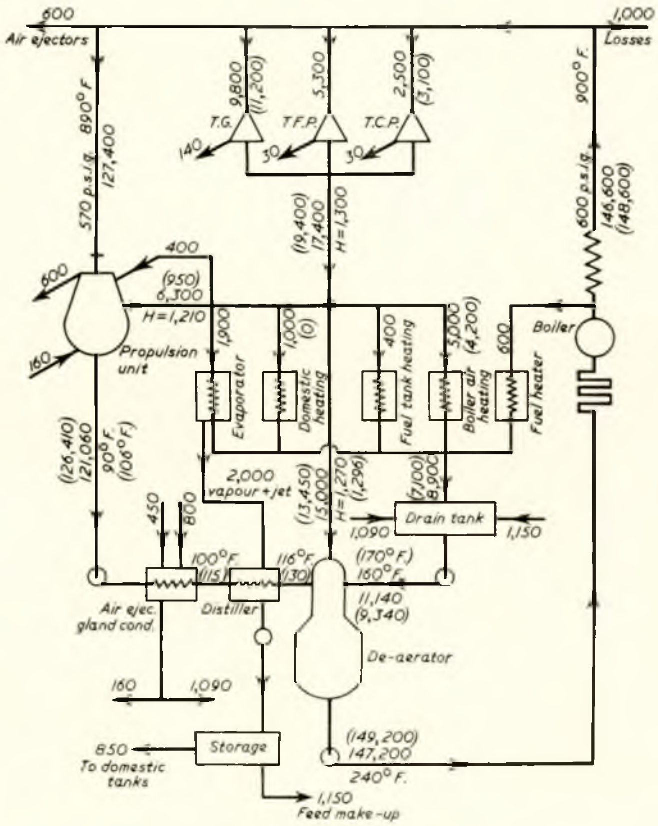

FIG. 2—*Heat balance—cycle as fitted*

21,280 s.h.p. (20,580)
Propulsion turbine non bleed steam rate 5·85lb./s.h.p./hr. (6·17)
Boiler efficiency 87 per cent Vacuum 28·5in. H_g (27·6)
Turbogenerator load 450 kW (530)
Fuel consumption:
11,120lb./hr. = 0·523lb./s.h.p./hr. = 119·1 ton/24 hr.
(11,270) (0·548) (120·7)

is higher than for Fig. 2. Little improvement in plant performance therefore results from the addition of a low pressure heater to the cycle shown in Fig. 2. It could be arranged that the evaporator in Fig. 3 would operate on auxiliary exhaust steam in the tropics. The additional cost and complication in providing the L.P. bleed point, feed heater and dual steam supply to the evaporator would however be only marginally justified.

The pressure level at which the auxiliary exhaust range operates is determined by the temperature which has to be achieved in the de-aerating feed heater. The specific steam rate of the generator, feed pump and circulator pump turbines will alter by about 4 per cent for a 5lb./sq. in. change in designed back pressure in the 30lb./sq. in. abs. region. Thus the higher the exhaust range pressure the greater will be the exhaust steam weight and heat content per pound. It is true that higher exhaust range pressure would give correspondingly higher feed pump suction and combustion air temperatures. The increased steam consumption of the auxiliary turbines, the consequently reduced bleed steam offtake, and the higher cost and weight of economizer equipment (to maintain the same boiler efficiency) combine however to more than offset the feed and air temperature advantage and the efficiency of the cycle will drop. The cycle benefits if the exhaust range pressure is no higher than will allow the lowest safe feed temperature to be developed in the de-aerator.

Experience shows that in the many ships in the author's company operating with 240 deg. F. feed temperature to economizers, corrosion and deposit troubles can be reduced to tolerable levels provided care is taken to ensure that this feed temperature is maintained at all times. Where maintenance of the feed temperature in harbour depends upon substituting a live steam heating source for the bleed steam drawn from the propulsion turbines at sea, the feed temperature for a variety of reasons is frequently not upheld. This problem does not arise with the back pressure generator cycle, where ample and indeed, excessive quantity of exhaust steam for feed heating occurs under harbour conditions.

For the reasons outlined above, the auxiliary turbines were arranged to exhaust at 15lb./sq. in. gauge to obtain 240 deg. F. in the contact type de-aerating feed heater.

MAIN BOILER PLANT

A twin main boiler installation has become virtually standard practice in tankers up to the largest sizes now building and experience has indicated that such an arrangement provides adequate security and flexibility of operation. It was felt that total elimination of economizer and superheater element end leakage in way of their connexions to the headers could be achieved by suitable all welded designs. This has been done and Fig. 4 shows details of the welded superheater employing socket welds securing the elements in the headers. Bifurcation of element ends allowed generous pitching of the header sockets for reliable welding, this with future repairs particularly in mind. Inspection openings in headers were reduced to a minimum.

The cargo discharge performance required called for an installed pumping capacity of about 6,000 tons of 0·9 SG crude oil per hour against a discharge pressure of 150lb./sq. in. gauge

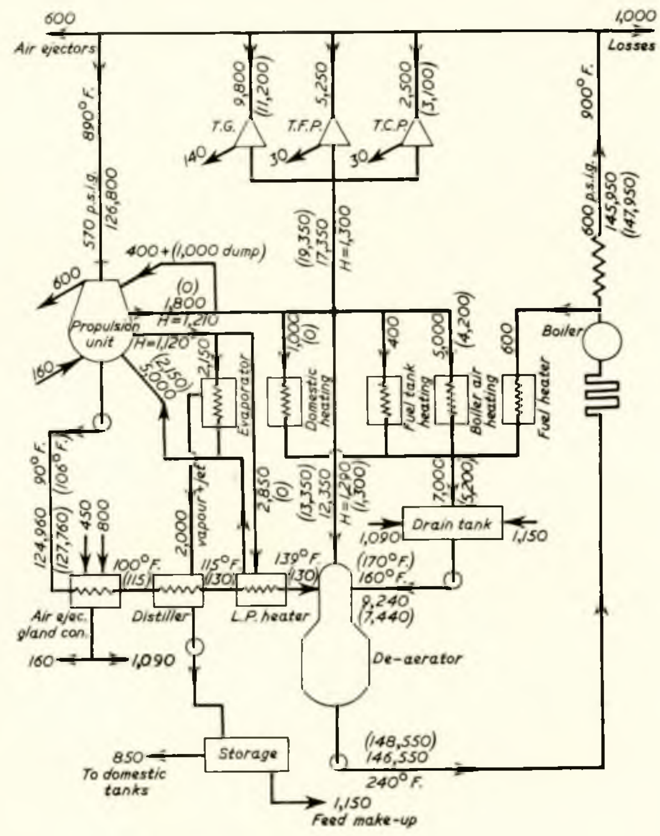

FIG. 3—*Heat balance—with L.P. feed heater added*

21,280 s.h.p. (20,463)
Propulsion turbine non-bleed steam rate 5·85lb./s.h.p./hr. (6·17)
Boiler efficiency 87 per cent Vacuum 28·5in. H_g (27·6)
Turbogenerator load 450 kW (530)
Fuel consumption:
11,070lb./hr. = 0·520lb./s.h.p./hr. = 118·6 ton/24 hr.
(11,220) (0·549) (120·2)

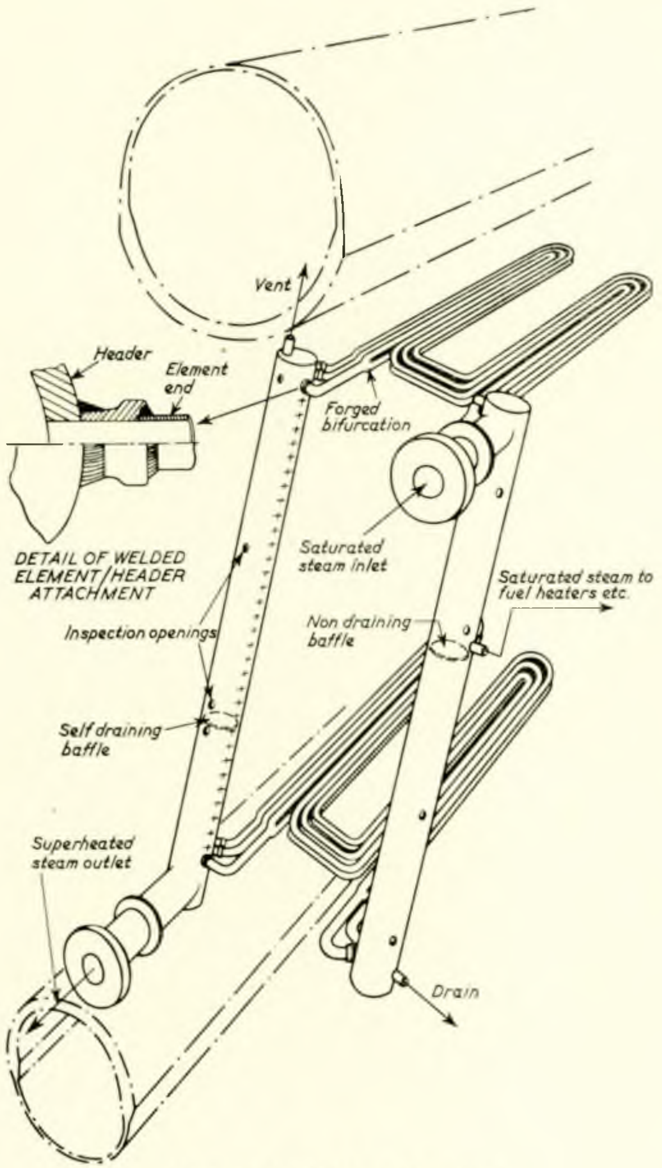

FIG. 4—*Welded superheater*

at the pumps. It was desirable also that in the event of one boiler being shut down, the remaining boiler should be able to maintain a cargo discharge rate very close to the 6,000 tons/ hr. figure. The total desuperheated steam load when the four cargo pump turbines are operating at maximum power is about 100,000lb./hr. The maximum evaporation rating of each boiler is 95,000lb./hr.

Surface type desuperheaters were known from previous designs to approach the limit in terms of their reasonable accommodation within the drums at around 80,000lb./hr. throughput capacity, even when allowing extremely high steam speeds in the desuperheater tubes. Moreover, difficulties of desuperheater tube wastage, particularly at the hot end with internal surface designs were not unknown, and insuring positively against boiler water leakage into the desuperheater through internal joint connexions has always been a problem.

The decision was therefore taken in this case to adopt spray desuperheating equipment and to depend on this equipment entirely for all desuperheated steam requirements. Whatever problems this may have introduced elsewhere it certainly simplified the boiler plant.

The funnel gas temperature and hence the boiler efficiency to be aimed at involves balance of the cost of the flue gas heat recuperation equipment and its weight, bulk and probable maintenance difficulties on one hand and the expected reduction in fuel rate on the other.

The decision to use non-condensing type turbogenerators in this case directed the boiler design toward the adoption of steam air heaters and, in consequence, toward flue gas heat recuperation plant in the form of economizers.

An orthodox arrangement of economizers was provided, studded steel tubes being fitted in the hotter gas zone and steel tubes with cast iron extended fin surface arranged toward the boiler outlet. The proportioning of the surface aims at ensuring a 40 deg. F. temperature rise in the feed water within the cast iron finned section.

Fig. 5 shows the effect of funnel temperature on ship

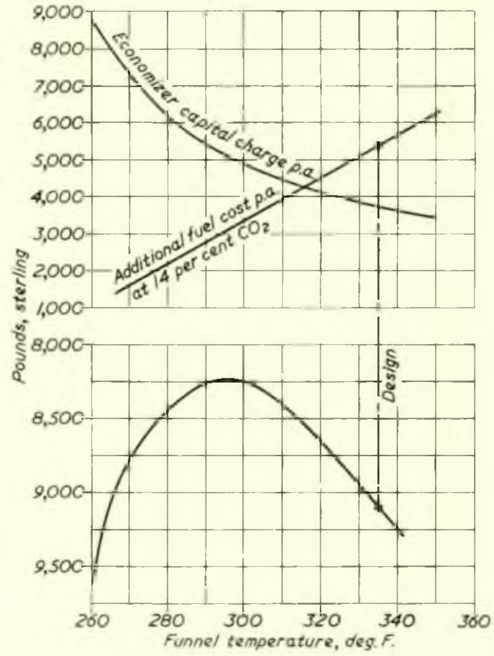

FIG. 5—*Gas exit temperature/operating cost relationship*

operating cost for this particular plant. The economizer capital charges are added to the cost of the fuel heat discarded to the funnel and the lower curve indicates that the gas exit temperature at which the sum of these two factors is a minimum coincides with 295 deg. F.

The significance of the curve, however, lies in the indication of how little, relatively, is the cost penalty incurred by designing for a gas exit temperature higher than the theoretical optimum.

In selecting a funnel gas design temperature of 335 deg. F. at the service evaporation rating of the boilers, it would seem that operating costs are increased by some £880 per annum, which would be avoided if economizer plant of sufficient surface to reduce the funnel gas temperature to 295 deg. F. were installed. It is submitted, however, that the 21-ton saving in economizer installation weight, and the attendant maintenance savings obtained by designing for 335 deg. F., will largely, if not entirely, offset this expense.

Forced Draught Fans

The preliminary study indicated that the fan motor power on each boiler would be about 140 b.h.p. The choice open in a.c. motor speeds for flexibility of control was considered at some length. A.C. motor fan speed variation in the majority of recent steamships tended toward the adoption of pole changing motors, usually giving, say, two speeds approaching any selected pair in the 600/900/1,200 or 1,800 r.p.m. series for 60 c.p.s. power supply. In some cases the motors are wound for two adjacent speeds with the intention that the lower speed fan output matches normal boiler evaporation and the higher speed provides for maximum evaporation. In other cases the motor pole selection may be four and twelve to give low speed, reduced fan power and greater stability of air control under harbour steaming conditions.

Both of these approaches were considered in this case and it was felt that neither offered the complete answer. On the first count the operation of the main boilers in a tanker at maximum rating for prolonged periods is rarely justified and hence the high speed winding is seldom used. In the second place, steam loads in harbour, especially at the discharge terminal, can fluctuate very widely indeed as cargo pump units are cut in or are shut down. The question then was whether it was not worthwhile to avoid the expense and complication of two speed motors and switchgear by adopting single speed fans controlled only by inlet vanes. Fluid couplings were investigated as an alternative to inlet vanes for the purpose of varying fan output. It was apparent that the standard fluid coupling could provide an output speed turn down of about 5:1 and that adjustable radial inlet vanes were capable of providing a greater turn-down of fan output. Admittedly the fluid coupling could offer savings in driving power during conditions of low boiler steaming rates. It was felt however that such consideration was by no means decisive and vane controlled fans driven by single speed four pole motors were decided upon.

The build-up of the estimated air and gas resistance components throughout the boiler plant at maximum evaporation rate is shown below (assumed air inlet temperature to fans 100 deg. F.):—

Air heater	1·4in. W.G.
Boiler casing	0·6in. W.G.
Burners	7·0in. W.G.
Generating and superheater tube banks	1·4in. W.G.
Stud tube economizer ...	1·7in. W.G.
Cast iron gilled economizer	1·7in. W.G.
Uptakes	0·3in. W.G.
Funnel top orifice	1·1in. W.G.
	15·2in. W.G.
Funnel effect	0·2in. W.G.
Total (Clean boiler at 13 per cent CO_2)	15·0in. W.G.

The minimum requirement of fan capacity to meet maximum evaporation rate on the boilers "as new" and with 13 per cent CO_2 combustion would be 27,800 cu. ft./min. against 15in. W.G. It can be argued that fan margin is required only on the head factor to make reasonable allowance for increased flow resistance due to fouling of the gas lanes, and that the "clean" portion of the plant (from the fan intake up to and including the air registers and burner throats) can be regarded as free from any tendency to offer any increase of resistance in service. On the other hand, if no cu. ft./min. margin is included the inability of the boiler to catch up rapidly on a sudden heavy increase in loading may be a serious operational deficiency.

The F.D. fan capacity eventually decided was 32,000 cu. ft./min. against 20in. W.G. This represents a head margin of 140 per cent over the clean resistance of the generating, superheater and economizer tube banks at maximum evaporation rate and with 13 per cent CO_2.

Oil Fuel Burners

Considerable advance has been made in the last decade in the field of oil fuel burning equipment for marine use. From past experience of the author's company it seemed that two aspects of the boiler firing procedure could well be improved. Firstly, there was need to obtain improved combustion efficiency under manœuvring and harbour steaming conditions. It had long been known that boiler fouling could occur very rapidly under harbour conditions where wide load fluctuations and the operation of mechanical atomizing type burners at critically low oil supply pressures are inevitable. The wide load fluctuations are the root cause of operators allowing prolonged operation of burners at pressures hardly high enough to ensure proper atomization. In the absence of notice or an upward change in load the operators justifiably argue that burner orifice capacity must be immediately available if steam pressure is to be maintained.

For this ship, therefore, a burner design was selected which would operate as a straight mechanical atomizing type on full load at sea, and which would operate as a steam assisted atomizing type for manœuvring and port conditions. The wide range feature of the steam assisted burner avoids the basic dilemma of tolerating poor atomization in order to have orifice capacity in hand to meet a sudden upward change in steam demand, and combustion efficiency at low steaming rates should be substantially improved.

The primary function of combustion control equipment is to regulate the oil firing rate in accordance with the load demand on the boiler so that the steam pressure remains constant. Such equipment customarily includes for each boiler a separate fuel oil metering valve and a control element which continuously monitors the fuel and air quantities into each furnace. The purpose of the fuel/air ratio controllers is to effect automatic adjustment of either or both quantities so that the optimum ratio results at any given firing rate.

The air and fuel flows are assessed by the controllers in many cases as functions of pressure drop across an orifice—the furnace throats in the case of the airflows, and the burner spray plates in the case of fuel flows.

Unfortunately, the effective airflow orifice depends upon the number of registers in use, and the effective fuel flow orifice upon the number and also the size of the spray plates working at the particular instant. The fuel/air ratio controllers have no means of anticipating a change in any of these factors and thus cannot by themselves correctly interpret the change in flow quantity/pressure drop relationship which follows any change in either of the effective orifice sizes. Manual bias to the ratio controllers is therefore necessary when the burner set-up is changed, say during manœuvring.

If one adopts the concept that since feed water and fuel oil supplies emanate from single sources and the steam is collected in a common header, a two boiler installation represents simply a twin furnace steam generating plant, then it seems logical that equal firing rates in both furnaces is a condition which should be guaranteed. The positive means of achieving this is to provide a single fuel oil metering valve through which the oil supply to both furnaces is regulated.

In the scheme which has been used in the Shell fleet for the past few years and which is repeated in the class being discussed, all that is asked of the control equipment is that fuel and draught pressure shall respond in the correct direction and in approximately correct magnitude to maintain steam pressure after some change in boiler load. With large changes in steam demand this simplified scheme requires manual trimming to achieve balanced combustion, but since manual adjustment in any case is necessary when air/fuel ratio controls are fitted, little real sacrifice is made whilst a useful reduction of control components is achieved.

PROPULSION TURBINES

The Pametrada double casing H.P. turbine was accepted on the ground that it offered a design having an efficiency comparable if not marginally better than any other marine turbine design available. When it is uncomplicated by primary nozzle sub-groupings, initial stage bypass or steam extraction branches, this turbine should, given the necessary care in construction and installation, provide a fully reliable unit.

The engine specification required that the propulsion turbine and the gear unit should produce 22,000 s.h.p. at 108 r.p.m. with a steam condition of 570lb./sq. in. gauge/880 deg.

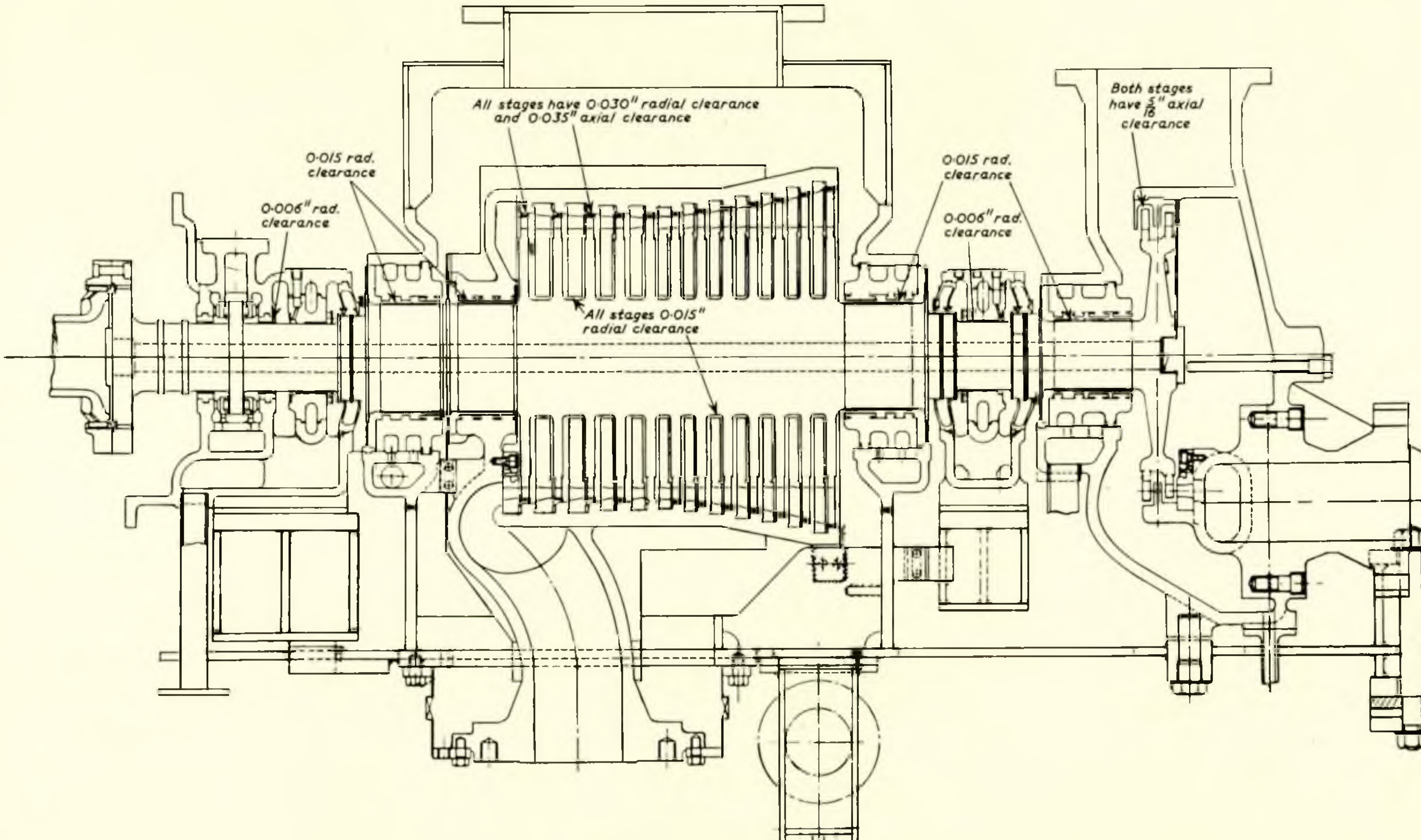

FIG. 6—*Sectioned view of the H.P. turbine*

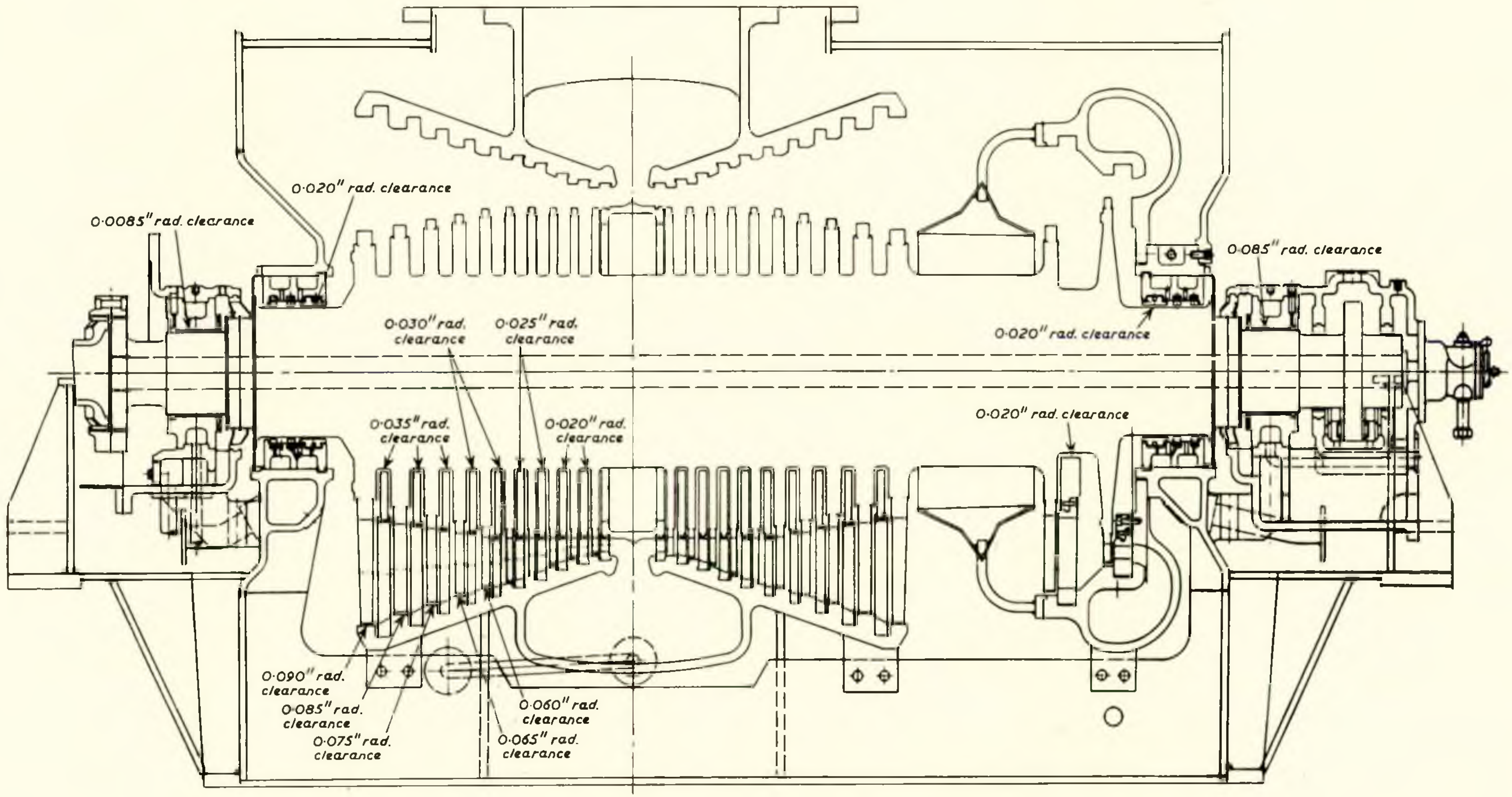

FIG. 7—*Sectioned view of the L.P. turbine*

F. at the H.P. turbine inlet, with a main condenser vacuum of 29in. Hg and when no steam is bled from the H.P. exhaust to supplement the auxiliary exhaust range.

Figs. 6 and 7 show sectioned views of the H.P. and L.P. turbines. Spring back casing gland and diaphragm labyrinths are fitted throughout the H.P. turbine and spring back glands are provided at each end of the L.P. turbines. Details of turbine bearings and gland/labyrinth clearances are also shown.

In assessing the relative performances of contending turbine designs, it is, as in all such comparisons, essential that account is taken of how the performance is reached. An example of this is the effect of the astern elements on the ahead running efficiency. It will be noted that in this particular design, in common with widely adopted British practice, H.P. astern element is carried on an extension of the H.P. ahead rotor. This allows the development of astern power on both rotors and its transmission through both gear trains; astern power up to about 60 per cent of the full ahead power can thus be provided. If astern power of this order is essential, it must be appreciated that its provision entails a significant loss of performance when running ahead. The power dissipation in astern turbines as a percentage of the full ahead power, when running ahead, has been variously estimated at 1 per cent to 1½ per cent in each element, or say a total of 1½ per cent power loss in a design having astern provision on the L.P. rotor only and twice this amount for a design as illustrated in this case. When it is remembered that in addition to windage loss on the astern wheels and blading, the inward gland sealing leakage to the H.P. astern cylinder in this case can only find its way to the condenser via the L.P. astern nozzles it is not difficult to believe that the total loss for this design could be as high as 650 h.p. The penalty for carrying two astern elements for 60 per cent astern power in lieu of one element for about 40 per cent astern power may amount in this case to an additional fuel consumption of some 1·2 per cent or 1·4 tons of fuel a day.

It had been found in earlier double casing H.P. turbine designs, that any discrepancy in the inner casing locating key settings and clearances led inevitably to heavy rubbing of rotor packings and special care was taken to ensure that the horizontal and transverse key clearances were strictly in accordance with the design. The loose elements of all keys were phosphated to ensure against binding due to the accumulation of corrosion debris.

Considerable thought had been given in previous ships regarding the philosophy of providing separate groupings of the primary nozzles for part load operation and/or a bypass around the initial blading stages for maximum power running.

The accepted principle in providing nozzle sub-groupings is that they improve the performance of the turbines by promoting efficiency at part loads. It is essential, none the less, that a clear understanding be agreed as to the precise operating power at which the machinery is to have maximum efficiency and it is obviously important, also, that maximum efficiency must refer to the entire installation and not merely to the propulsion turbine itself.

Provision of a bypass around the initial expansion stages to obtain the maximum power rating is inherently inefficient and is difficult to justify for commercial use, unless for very limited periods of operation.

If one examines the relative steam rate performance without and with nozzle sub-groupings, it is found that for this particular design the specific steam rate at 75 per cent of designed full power (achieved by throttle control) increases by some 5 per cent. If a 75 per cent power nozzle group were provided the steam rate increase would be about 3 per cent when operating on this group alone.

It is admitted, then, that the part load performance of the propulsion turbine itself is enhanced by the use of a subdivided primary nozzle plate. The saving is rather marginal, and in the author's view, hardly justifies the additional complication in the double casing turbine design, unless considerable periods of reduced power operation are envisaged.

It has lately become apparent, however, that some flexibility in ship operating speed, whilst still maintaining maximum possible efficiency, is desirable to meet the wide fluctuations which can occur in the oil transportation market.

The problem in providing such a dual speed ship lies in avoiding mediocrity of performance not merely at one of the speeds, but at both. If the requirement is the faculty to operate at a speed of, say, 10 per cent below that speed for which the hull and its power plant were basically designed (i.e. the speed anticipated as applying over a majority of the ship's existence), the machinery design can be directed toward

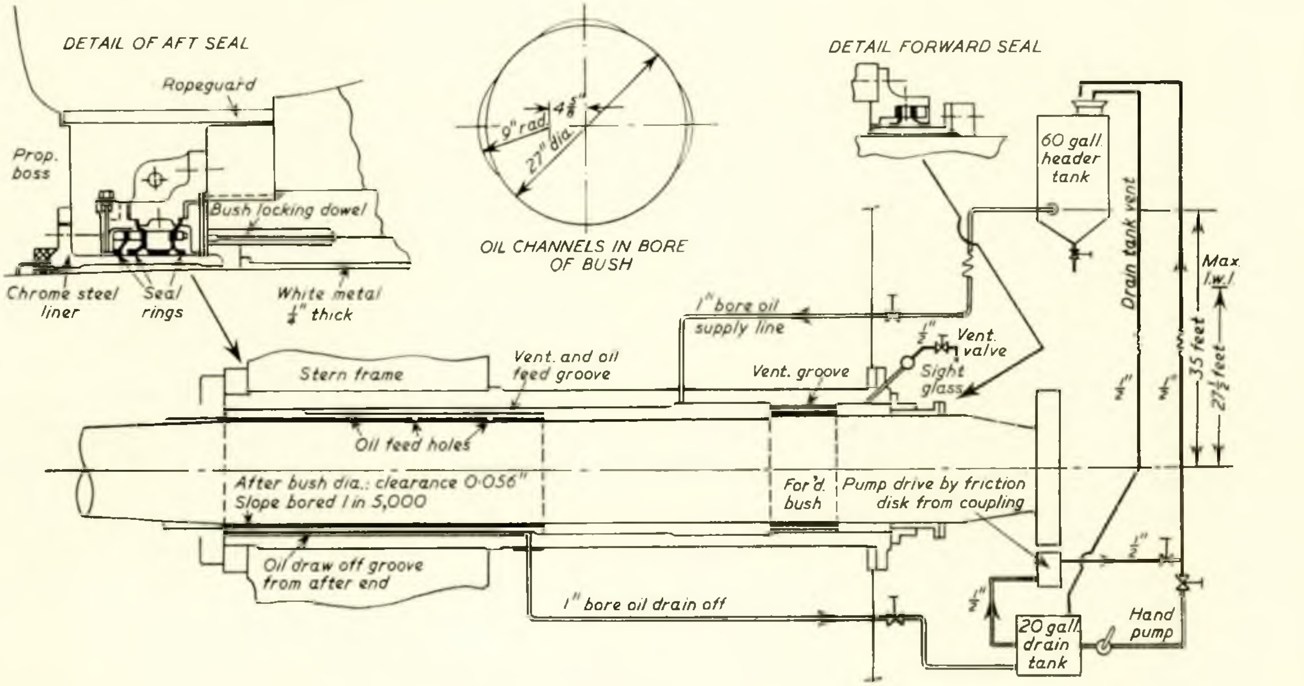

FIG. 8—*Sterntube and lubricating arrangement*

high part load performance. The basic hull form unfortunately cannot be filled out to take maximum advantage of the slow steaming period. If, on the other hand, a speed increase above basic design of at least 10 per cent is called for, it is evident that the machinery installed will need to have a power rating considerably higher than that which would propel a suitably fined hull form at a basic plus 10 per cent speed. Many components of the "boosted" installation, by virtue of the built-in power reserve, will operate at the "normal" condition considerably off peak efficiency. It appears then that the dual speed ship involves compromise both in machinery design and in hull form.

Gearing

The reduction gearing is designed for *K* values of 90 in the primaries and 80 in the secondaries. 60/40 teeth with 16 deg. flank angle, 6/10in. pitch for primaries and 8/10in. for secondaries were adopted. Cast light alloy casing and coupling covers were adopted to reduce noise transmission and to provide against the corrosion which can so easily begin on gear elements due to condensation on the inside of steel covers. A glanded design of attachment of the primary flexible coupling casings to the primary gearboxes was adopted so that no thrust reaction between the gearboxes and the turbines could occur to adversely affect alignment. The main propulsion thrust is taken on a Michel block located aft of the gear case. Shafting is to Lloyd's Rule size for 22,000 s.h.p. at 108 r.p.m.

The adoption of an oil lubricated sterntube was decided upon after very successful results had been obtained in two 7,500 s.h.p. installations using the same oil seal design. It was felt that experience on high power single screw installations indicated that the wear-down inevitable with sea water lubricated bearings should not be accepted for these ships. It was anticipated also that vibration would be minimized by the elimination of the excessive lateral freedom which the screw shaft in a water lubricated bearing exploits. Fig. 8 shows the tailshaft and stern tube arrangement. The small coupling driven oil circulating pump is perhaps a refinement although it should ensure that fresh lubricant is at all times moving in the vicinity of maximum load carrying area of the bushing.

MAJOR AUXILIARIES

The turbogenerators, boiler feed pumps, main circulating pump and the boiler F.D. fans represent a substantial component of the auxiliary plant cost and it is obvious that their matching to the cycle requirements involves a careful balancing of first cost and efficiency.

Turbogenerators

The cycle adopted was conceived around the use of non-condensing generator turbines because it appeared evident from a study of the relative costs of condensing type and non-condensing type sets and their respective impacts on the cycle performance, that the better investment return would be obtained by using non-condensing, or back pressure type sets.

If condensing sets of equal kW rating had been fitted, the inclusion of an L.P. feed heater and operation of the evaporator from the same low pressure extraction belt on the L.P. turbine would have been logical. The resultant additional

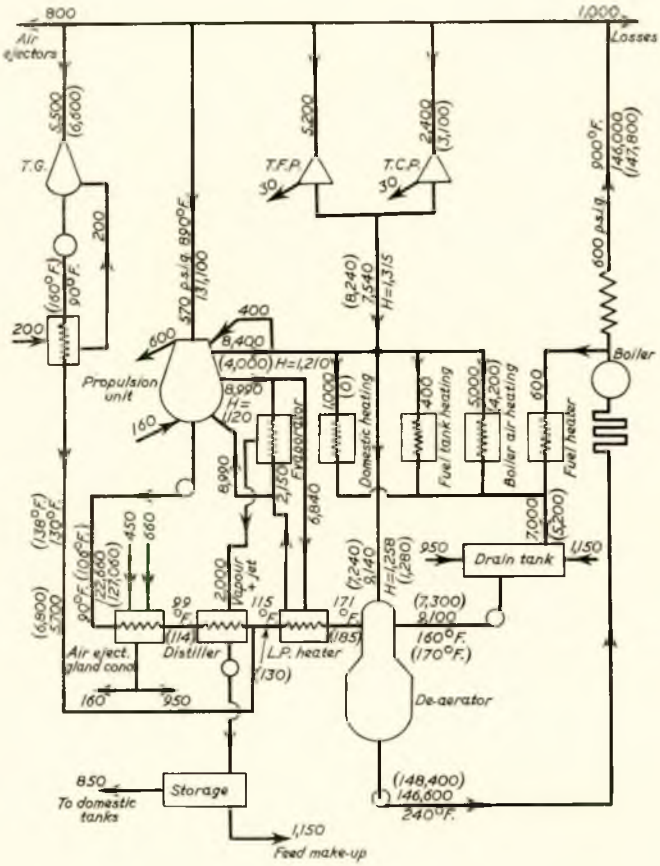

FIG. 9—*Heat balance—condensing type generator turbines and L.P. heater*

21,280 s.h.p. (20,580)
Propulsion turbine non-bleed steam rate 5·85lb./s.h.p./hr. (6·17)
Boiler efficiency 87 per cent Vacuum 28·5in. H_g (27·6)
Turbogenerator load 450 kW (530)
Fuel consumption:
11,075lb./hr. = 0·520lb./s.h.p./hr. = 118·6 ton/24 hr.
(11,210) (0·546) (120·1)

TABLE I SUMMARIZING FIGURES 2, 3, AND 9.

Fig. No.	Cycle PSIA=lb./sq.ins abs.	60 deg. F. sea temperature Vacuum 28·5″ Hg. T.G. load 450 kW.			85 deg. F. sea temperature Vacuum 27·6″ Hg. T.G. load 530 kW.			Overall average fuel rate. (Fig. 3 corrected to 20,580 s.h.p. at 27·6″ vacuum) lb./hr.	Extra capital cost of plant £	Fuel saving *per annum* £	Capital charges *per annum* £
		s.h.p.	Fuel lb./hr.	Fuel rate lb./s.h.p./hr.	s.h.p.	Fuel lb./hr.	Fuel rate lb./s.h.p./hr.				
2	Back pressure T.G. 38 PSIA bleed only } As fitted	21,280	11,120	0·523	20,580	11,270	0·548	11,195	0	0	0
3	Back pressure T.G. 38 PSIA bleed and 10 PSIA bleed	21,280	11,070	0·520	20,460	11,220	0·549	11,175	2,500	410	350
9	Condensing T.G. 38 PSIA bleed and 10 PSIA bleed	21,280	11,075	0·520	20,580	11,210	0·546	11,142	16,500	1,200	2,300

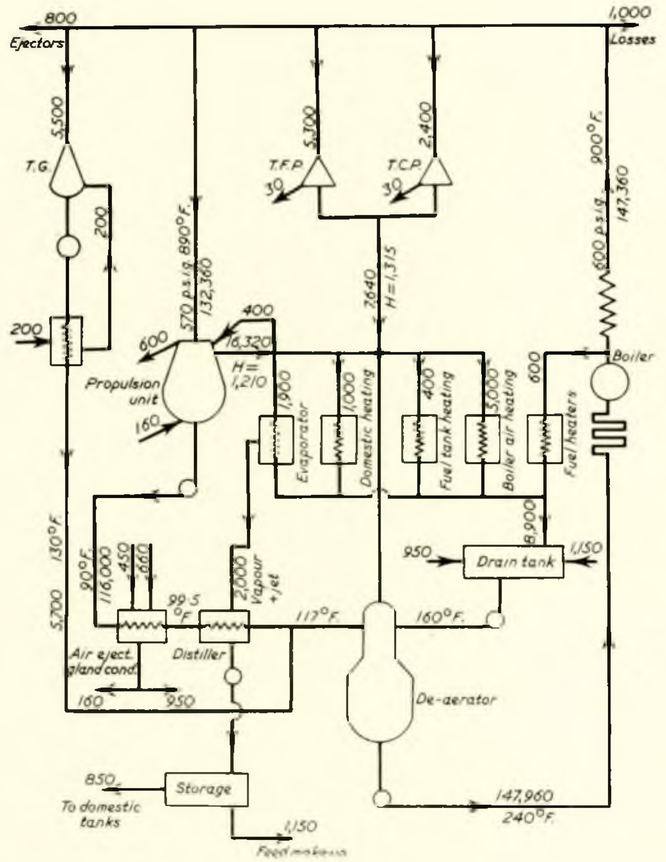

FIG. 10—*Heat balance—condensing type generator turbines*
21,280 s.h.p.
Propulsion turbine non-bleed steam rate 5·85lb./s.h.p./hr.
Boiler efficiency 87 per cent Vacuum 28·5in. Hg
Turbogenerator load 450 kW
Fuel consumption:
11,177lb./hr. = 0·525lb./s.h.p./hr. = 119·7 ton/24 hr.

capital expenditure would have been about £16,500 and a direct increase in machinery weight of some 26 tons would also have resulted. Fig. 9 shows the heat balance with 60 deg. F. and 85 deg. F. sea temperature conditions applying.

The nett effect on overall efficiency is as shown in Table I. For this ship, the £16,500 additional investment in non-condensing type generator sets and a low pressure feed heater would yield an annual fuel cost saving of the order of £1,200, which fails to satisfy the criterion of economic efficiency which has been adopted.

As a matter of interest, Fig. 10 shows the basic scheme revised only by the substitution of condensing type generators, i.e. no L.P. bleed service is provided. The fuel rate has increased by 0·6 ton a day, in spite of the additional capital investment of about £14,000 and some 23 tons additional machinery weight.

Feed Pumps

The main boiler feed pump drive can be arranged, in descending order of efficiency.

a) From the gearbox of the turbogenerator.
b) By independent electric motor.
c) By independent steam turbine.

a) Suffers from the high premium charged by the turbogenerator manufacturers. The possibility of pump seizure in the event of temporary loss of suction must also be reckoned with. In the absence of a suitable automatic clutch interposed between the power take-off pinion in the gearbox and the pump coupling, the only answer to this is the adoption of increased pump internal clearances. Pump efficiency in that case is inevitably lost to the extent where the loss all but completely cancels out the original gain derived from the more efficient prime mover. Even so, the scheme remains attractive provided the cost is not significantly more than (c). This can be the case in composite generator/feed pump sets where the ratio of feed pump power to generator power is not too high. In large installations feed pump power tends to become a larger proportion of the combined load on the set and this undoubtedly results in a heavier and more expensive reduction gearbox from which the pump drive is taken. The cost of scheme (a) tends therefore to outstrip (c) in higher powered installations.

b) Involves in this instance a 350 b.h.p. a.c. motor for each feed pump. The experience of the author's company with large 2 pole motors and switchgear in a previous 7,500 s.h.p.-tanker class was not encouraging. As in scheme (a) special protection against pump seizure in case of loss of suction is necessary since the motor when off-loaded has no overspeeding tendency.

Motor driven feed pumps would require uprating of both turbogenerators from 600 kW in scheme (c) to at least 850 kW. The cost picture would be:

Scheme (*b*)		*Scheme* (*c*)	
2 × 850 kW generator sets ...	... £56,000	2 × 600 kW generator sets ...	... £48,000
2 Electrofeeders	... £10,500	2 "high efficiency" type turbo feed pumps ...	... £12,500
Additional switchgear	£2,200	Steam and exhaust pipework	... £600
	£68,700		£61,100

The independent turbine driven pump (i.e. scheme (c)) eventually decided upon has a steam rate of 5,300lb./hr. at the service feed rate of 147,200lb./hr. The pump horsepower at this load is 260. The effect of employing an electric motor driven feed pump in lieu of a steam turbine driven pump, assuming equal pump efficiency, would amount to a reduction of about 320lb./hr. in total superheated steam generation. This yields a nett cycle efficiency gain of 0·22 per cent. It can be taken therefore, that providing electrofeeders would have cost £7,600, they would have saved 0·24 tons of fuel a day or £460 per annum. Again, according to the method of assessment which has been used, this would not be economic.

The disappointingly small gain which would derive from using electrofeeders in this case results from the replacement of the reduced auxiliary exhaust heat in the de-aerator by 38lb./sq. in. abs. bleed steam with an extraction factor of 0·482.

Electrofeeders would show to better advantage in Fig. 3 where the heat replacement could be made in the L.P. heater. The capital charges on the bigger generators, etc., remain higher than the fuel saving, however, and the electrofeeder consequently cannot earn its place.

The electrofeeder could improve Fig. 9 sufficiently to justify 850 kW condensing type turbogenerators but it is important to recognize that this does not mean that the economic position of Fig. 9 in relation to Fig. 2 is in any way strengthened.

Main Circulating Pump

The estimated maximum steam load on the main condenser was 126,000lb. of exhaust steam per hour or about 115 × 10^6 B.t.u./ hr. Designing for 10 deg. F. temperature rise in the cooling water means a cooling water demand of 19,200 gall./min. and a pump capacity rating of 22,000 gall./min. was selected to include for main oil cooler circulation.

Previous experience had shown unsatisfactory flow conditions at entry to a ship side circulating water inlet when this is arranged with inward flow normal to the hull. A 38,000-ton, 16½-knot class with a 3 : 1 ratio of clear intake grid area to pipe bore suffers a loss of head at entry of about 8ft. when the ships are operating at full power. This is not

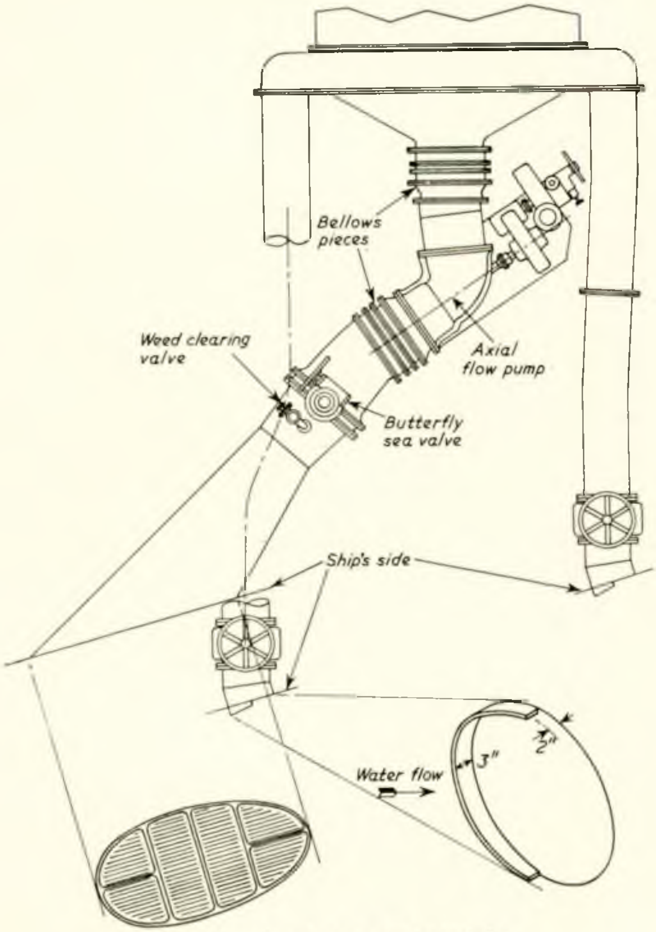

FIG. 11—*Arrangement of sea inlet*

exceptional to average practice continuing today in many modern steamships and it is obviously undesirable that pump power should be required to re-create this head.

An arrangement of sea inlet shown in Fig. 11 was developed in this case with the object of reducing entry loss to the minimum. At the same time, loss of head at the overboard discharge connexions on the ship side was studied and the fitting of lips to the forward edges of the overboard discharge orifices was decided after consideration of the work published on the effect of similar lips fitted in a destroyer of the U.S. Navy[1].

The assessment of the total resistance head in the main condenser circulating system, including entry and discharge losses, indicated that this should be as low as 6ft. when the cooling water flow was at normal rate.

The full astern manœuvring condition had to be kept in mind however. The condenser heat load is about 100×10^6 B.t.u./hr. when operating at full power astern, and of course no benefit derives in this circumstance from the particular orientation of the circulating pump intake. In order to ensure therefore that adequate circulation would be maintained when operating astern, a total pumping head capacity of 15ft. was provided.

A choice existed between the centrifugal and axial type pumps. Suitable axial designs could maintain efficiency over a much wider range of operating r.p.m., and, hence, throughput and total head, than a centrifugal pump. Since reduced r.p.m. operation was anticipated as the normal condition, an axial flow pump was selected.

The possible prime mover for the axial flow main circulating pump was:

1) Drive from the main propeller shafting (with suitable provision for manœuvring)
2) A.C. electric motor
3) Independent steam turbine

The scheme studied for the first concept involved a low speed fluid coupling and a clutched steam turbine and was found to prove hopelessly bulky and excessive in first cost.

The power required at the pump coupling on full rating was estimated at 140 b.h.p. and consideration of a.c. motor drive involved an additional 100 kW on the rating of each turbogenerator set at some £3,200 increase in cost. The relative inflexibility of speed and output control with the motor drive, unless complication in the form of a slip ring or multi-winding motor is used, weighs against such an arrangement. It was apparent that the configuration of the main sea inlet led to the condition where full power astern operation of the main engine would dictate the maximum circulating pump power, since the axial flow pump power requirement rises steeply with increase of total pumping head.

The a.c. motor driven axial pump involved a difficult compromise in the selection of intermediate operating speeds since the pumping power at any one speed could vary from manœuvring to full away conditions; this was one problem. Ensuring that the low speed would not result in excessive circulation in cold sea water conditions, unless some flow bypass or recirculation system is adopted, was another. A high speed motor with a fluid coupling and a gearbox interposed between the motor and the pump could give adequate speed variation, but the possibility of motor overloading would remain unless the motor is a single speed unit rated for maximum conditions.

The infinitely variable speed of the turbine drive, with steam nozzle control for 50 per cent, 75 per cent and 100 per cent turbine power was a much more comfortable proposition. Should the pumping head increase suddenly at any particular speed setting, the r.p.m. will merely drop, with no danger of complete shut-down.

Fig. 2 indicates the circulating pump turbine running at about 80 per cent and 100 per cent of its full power. It is probable that these figures are pessimistic in that adequate condenser circulation should be achieved with somewhat lower pump power since long periods of operation in sea water at less than 60 deg. F. must be envisaged. It is suggested that 50 per cent power nozzle operation is a reasonable year-round average for the circulating pump, making the average steam consumption about 1,700lb./hr.

Comparing this with a two-speed motor driven pump, whose lower speed is selected to match 50 per cent power (i.e. 70 b.h.p.), the additional generator turbine steam requirement would be about 1,260lb./hr.

The nett saving in total superheated steam generation would be about 190lb./hr., equivalent to a £275 per annum saving in fuel cost.

The conclusion is that the superior efficiency of the multi-stage generator turbine over that of the single stage circulator pump turbine is of little relevance when the average load factor on the installed capacity of the unit is of low order. For circulating pumps, where, by reason of the wide seasonal and geographical variations in sea temperature, prolonged period of operation at low power can occur, the turbine drive offers a reduction in first cost expenditure sufficient to more than compensate the modest increase in fuel rate. Indeed it is not unlikely that unless more than two speeds are available, the motor driven pump power in moderate to low temperature sea conditions will be excessive and result in a marginal increase of overall fuel rate.

Layout—Machinery Space Length

A successful machinery layout obviously requires that adequate space be available. At the same time space grossly in excess of adequacy will not lead to the best overall ship design.

A feature in our large crude oil carriers built in the last few years has been the isolation of a certain portion of the tank space into a "ballast only" category. The aim here is to provide ballast space which will be void when the ship is loaded with cargo and which can be ballasted concurrent with cargo discharge through a pump and pipeline system completely divorced from the cargo pump and pipe system. This allows important saving of time spent on the berth. It also allows the ship to come alongside the loading terminal and commence loading operations immediately, the ballast discharge commencing as soon as moorings are made fast.

The void tank spaces in the ship's midbody result in significant reduction of hull bending stress levels in the loaded condition. Scantlings can be lightened to exploit this position and important savings in hull steel cost and weight result.

The portion of tank space allocated to clean ballast must obviously be limited to that figure which will permit cargo of the lowest specific gravity envisaged being contained in the remaining spaces in quantity sufficient to put the ship to her marks.

Assuming then that the overall dimensions and the hull form are prescribed and that bunker space and pumproom length have been determined, minimization of machinery space length will directly improve the main hull design by allowing a maximum of clean ballast space and by enabling the tank length/overall length ratio to be increased.

It is logical to reckon machinery space length, as applied to single pumproom tankers, as measuring from the aft peak bulkhead to the aftermost cargo bulkhead. The main turbine/reduction gear unit has to be sufficiently forward to allow accommodation of the main gear casing within the narrowing sections toward the stern; the remaining length requirements are determined principally by consideration of the space demands of the cargo pumps and their prime movers and the pumproom piping and valves, etc.

Vertical type cargo pumps allow a reduction in machinery space length and elevate the driving turbines and their reduction gear units to a more accessible level in the engine room. The bulkhead seal on the drive shaft is less exposed to flooding danger in the event of a serious leak and bilge accumulation in the pumproom.

In setting up the location of the boiler plant and the basic units of the installation, it is desirable that the main control station be close to the boiler firing platform. The boiler plant is the primary source of operations in any steamship and the activity on the firing floor, particularly during manœuvring, should always be under the close observation of the senior watchkeeper. Remote indicating instruments have proved their reliability, but they must always remain a second choice to the provision of direct vision facilities if the latter can possibly be accomplished.

The turbogenerators and electrical switchgear and metering panels form the basis of auxiliary power for control of the ship. Supervisory facilities should be as direct as possible from the immediate vicinity of the engine room control station.

Piping

For every item of main and auxiliary equipment, consideration of its function, its particular supervision requirements and the consequences of its eventual location upon piping and wiring runs, should be assessed carefully so that the best balance of all factors involved will indicate the most suitable location. A clear concept of pipe system runs, and not merely the major systems, must be formed before the positioning of machinery units and the orientation of the boiler plant is fixed. Only by so doing can pipeline lengths be minimized, and the minimizing of piping is in itself a useful criterion of the success of the layout as a whole.

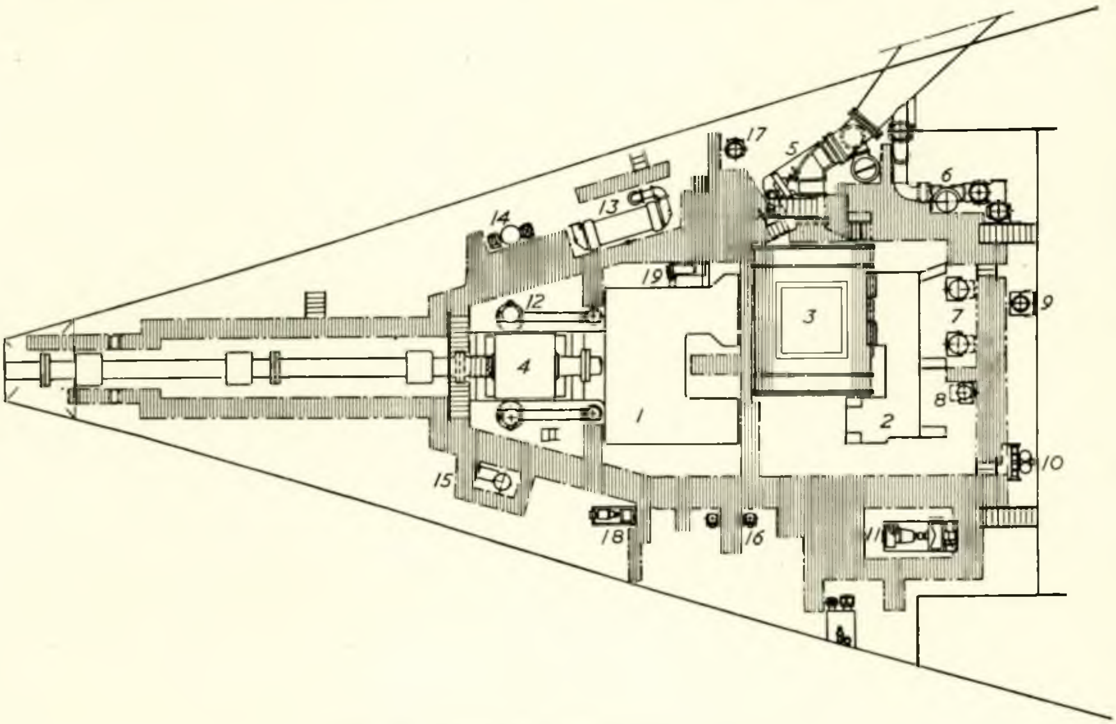

FIG. 12—*Machinery layout—lower level*

1) Main gearcase seat
2) Forward turbine seat (incorporating drain tank)
3) Condenser
4) Thrust block
5) Main circulating pump
6) Auxiliary circulating pump
7) Extraction pumps
8) Drain pump
9) Oil fuel transfer pump (electric)
10) Oil fuel transfer pump (steam)
11) General service pump
12) Lubricating oil pumps
13) Lubricating oil coolers
14) Lubricating oil filter
15) Lubricating oil purifier
16) Sea water service pumps
17) Bilge/air conditioning supply pump
18) Auxiliary bilge pump
19) Evaporator supply pump

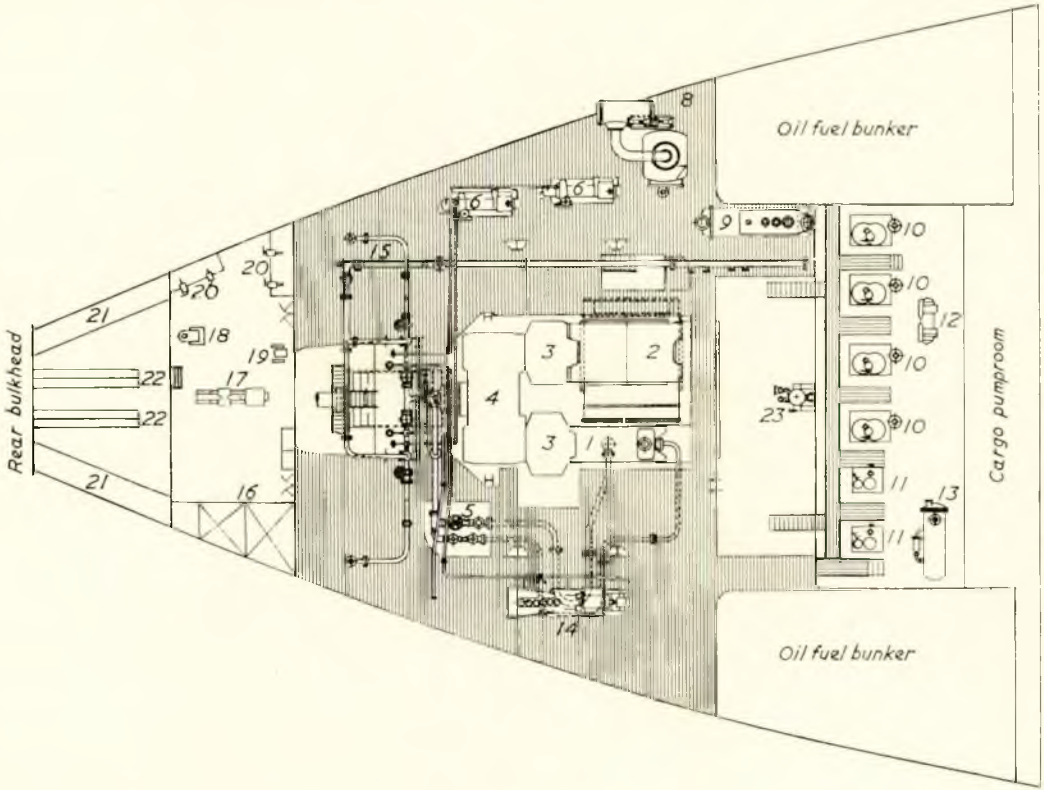

FIG. 13—*Machinery layout—intermediate level*

1) H.P. Turbine
2) L.P. Turbine
3) Primary gearcase
4) Secondary gearcase
5) Manœuvring valves
6) Main feed pumps
7) Evaporator
8) Evaporator brine/fresh water pumps
9) Auxiliary condenser
10) Cargo pumps
11) Ballast pumps
12) General service compressors
13) Sea water heater
14) Diesel generator
15) Desuperheater
16) Lubricating oil storage tanks
17) Lathe
18) Drill
19) Grinder
20) Work bench
21) Spare gear storage
22) Storage bins
23) Air ejector

Important pipe ranges and reducing valve stations frequently present tiresome problems of access and maintenance to operating staff. This is inevitable if such fittings are clipped closely to bulkheads in a manner suggesting that their total concealment was the primary aim. Reducing valves and all steam and feed distributing ranges should rather be set up in prominent locations which allow access from all sides for their operation and maintenance.

Access

If ease of movement throughout the machinery space and in way of each unit and all important pipe and valve ranges is properly provided for at the design stage, the installation work in the shipyards will correspondingly benefit and an altogether superior job in terms of general workmanship should certainly result. How often it happens that high pressure joints leak in service because the more inaccessible flange bolts were not properly hardened up in the erection stage!

There is seldom any real necessity for frequent breaks in the machinery space operating levels. These invariably lead to awkward movement which can be fatiguing in a large engine room and in extreme cases can be downright dangerous. Direct, straightforward access throughout the entire engine and boiler space can only be achieved however if some imagination and foresight is exercised in the early stages of planning.

The layout developed for these ships is shown in Figs. 12, 13, 14 and 15 and a list of the engineering equipment fitted is given in an appendix to the paper.

The main boiler plant is located aft and the wide beam has allowed transposition of the boilers from the conventional handing. This brings both fuel burner groups toward the centre of the firing floor and improves the weight distribution on the boiler flat. The driving turbines of the cargo and ballast pumps are mounted in a recessed flat formed by the stepped after bulkhead of the main pumproom. A total machinery space length of 130ft. which includes a pumproom length of 27ft. and 100,250 cu. ft. of oil fuel storage capacity has been provided.

The machinery is accommodated in three main levels. The upper flat contains the boiler plant and the main control station. The turbogenerators and all major switchgear occupy the starboard side with the condensate circulated sea water distiller, the fuel pressure pumps and heaters and the steam/steam generator unit to port. The steam supplies to these units, and their exhausts or drains, are piped directly through the flat in an attempt to obtain a logical and clean lined arrangement throughout the control level.

Steam is taken off the boilers at the lower ends of the forward facing superheater headers and piped immediately down through the firing floor to a distribution range located aft of the main gear case. The steam lines to the propulsion unit therefore take a direct route to their connecting points on the turbine inlets. The manœuvring valves are operated from the control console directly above them. Superheated

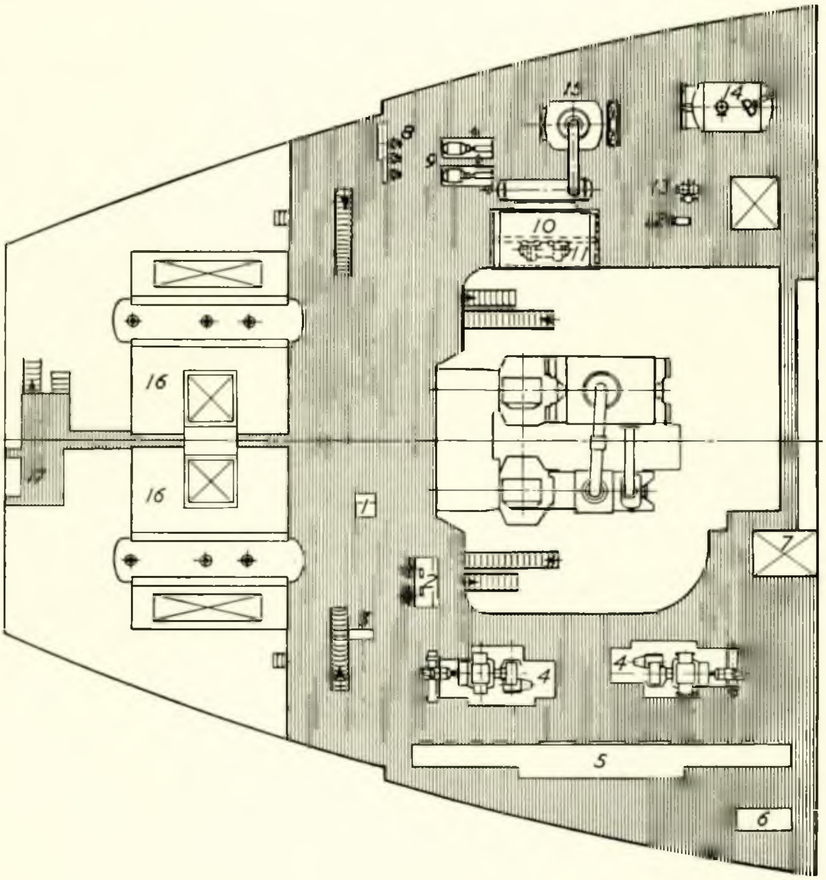

FIG. 14—*Machinery layout—control level*

1) Log desk telephones
2) Manœuvring console
3) Combustion control panel
4) Turbo alternator
5) Main switchboard
6) Sub switchboard
7) Lift shaft
8) Oil fuel heaters
9) Oil fuel pumps
10) Lubricating oil gravity tank
11) Combustion control compressor
12) Steam/steam generator feed tank
13) Steam/steam generator feed pumps
14) Steam/steam generator
15) Evaporator
16) Boilers
17) Boiler chemical test station

steam is piped from the distribution range separately to each turbogenerator and feed pump, to the circulating pump and air ejector and to the desuperheater.

The intermediate flat surrounds the propulsion turbine set. The cargo pump turbines are located at the extreme forward end. The boiler feed pumps, atmospheric auxiliary condenser and the sea water circulated distiller are on the port side and the Diesel generator is on the starboard side directly below the main switchboard.

The engine room floor on the port side is dominated by the horizontal turbine driven axial flow main circulating pump. The auxiliary circulating pump and the main oil coolers are also on the port side. A complete bottom platform is dispensed with in favour of generous width walkway gratings where necessary. This exposes a maximum of the tank top bilge area to constant inspection.

Habitability

Alert and efficient watchkeeping cannot be expected in an engine room which is too hot, too noisy or inadequately lighted and there is need to re-appraise present standards of habitability in machinery spaces. Impetus has been given recently toward improvement of these standards by the adoption of full air conditioning of accommodation spaces. An increasingly critical attitude on the part of engine room crews toward the environment in which they spend their duty hours is inevitable and in many instances justified.

Some radiant heat into working spaces is unavoidable in high temperature steam plants. Even so, exposed hot metal parts such as valve bonnets, etc., are obvious evils which should not be tolerated.

High humidity level often results from poorly designed turbine gland seal arrangements, freely vented drain tanks and the like. In almost every case these vents can be piped into waste mains and run outside the space.

Boiler casings present large areas of space heating surface even although the face temperature of these casings may not be high. If the casing temperature cannot be reduced to only a few degrees above the boiler room ambient by ducting the combustion air from the F.D. fans around the internal casing and arranging the steam air heater immediately before the air entry to the burner fronts, then at least additional insulation on the exterior of the casings should be considered.

A total ventilating air rate of 27 changes per hour is provided for the machinery spaces in this design and the installed vent fan power totals 114 horsepower. It may be that in future the pumping of large volumes of frequently humid air into machinery spaces will have to be replaced by some form

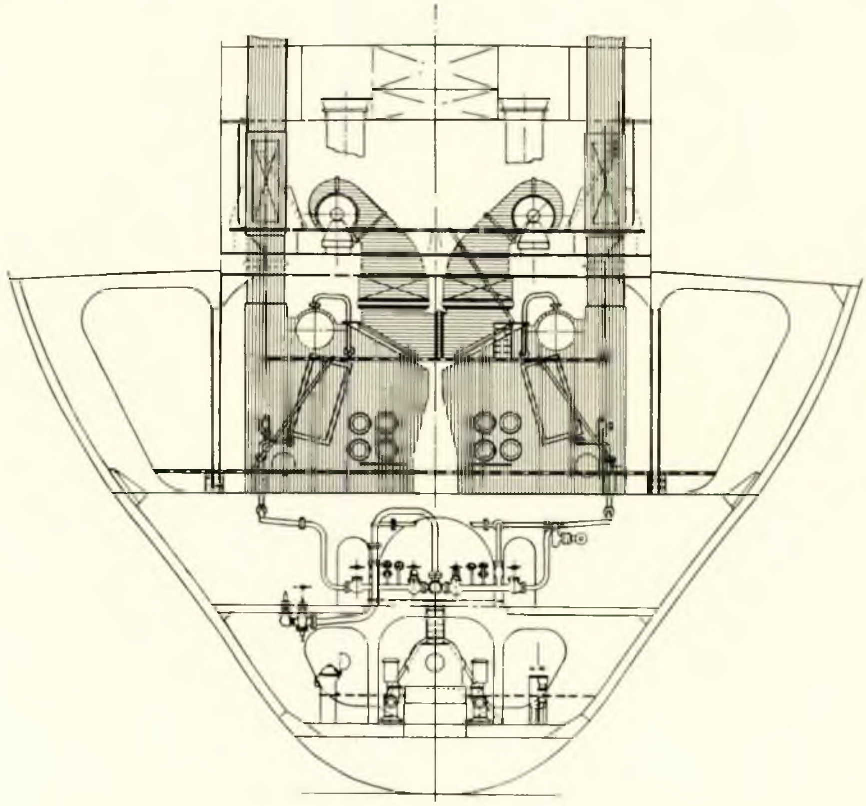

FIG. 15—*Machinery layout—looking aft on boilers*

of mechanical cooling with a minimum of fresh air supply, and it may even prove that the reduction in fan power in so doing will largely offset the additional power required by the cooling units.

Among the machinery units which contribute largely to noise level are the propulsion and generator turbine gearboxes. Air noise from forced draught and ventilating fans, generator cooling fans and air compressors can also be troublesome.

In this particular design, an attempt has been made to reduce gear noise by fitting sound insulating material around the main and auxiliary turbine gearboxes. The turbogenerator cooling air ducts and the forced draft fan intakes are shrouded by noise attenuation chambers. An average noise level of something less than 80 decibels, at least outside the immediate vicinity of gearing units, is aimed at.

FUTURE POSSIBILITIES

It may be of interest finally to consider some possible changes, consisting mainly of further pruning, to the design which has been described in the paper.

Propulsion Unit

An argument can be made for the provision of one astern turbine element only, say on the L.P. rotor. The resulting simplification of the H.P. rotor and the elimination of part of the power loss would be immediate and tangible gains.

The application of full reverse power in an effort to stop the ship from full speed pre-supposes the propeller's ability to absorb this power, and it is known from experience that flow conditions into the propeller during the initial stages of an emergency stop, limit the astern r.p.m. which can be usefully applied before cavitation and loss of astern thrust occur. The power absorption limit of the propeller under such conditions is probably between 35 and 40 per cent of the full ahead power and this can usually be obtained from a two-stage astern turbine on the L.P. rotor.

On the other hand the astern turbines in the design described should enable superior manœuvring performance of the ship at the slow speeds obtaining under docking conditions. Here the propeller can operate astern more efficiently and higher astern power should result in better acceleration when required.

Propeller R.P.M.

The improvement in propulsion efficiency resulting from a reduction in r.p.m. from 108 to say 100 would be perhaps a little better than one per cent. The increased cost of the gearing and the propeller would be set against this but it appears that a nett benefit would result from adopting lower r.p.m. However, large heavyweight propellers present real problems of transport and handling in the dock.

Generating Sets

The provision of an auxiliary or emergency Diesel set in addition to the two 600 kW turbo sets in the present scheme follows existing convention and the Diesel generator undoubtedly is of value in facilitating "dead ship" starting and in drydock. A further vessel now building for the author's company will be equipped with a total electric power generating outfit of one steam turbine generator and one Diesel set of equal (i.e. ships full load) rating. The Diesel set is regarded as the standby unit and the ship would be considered operational for the passage back to a home port in the event that the turbine generator became a casualty.

The nett capital cost saving in a repeat ship of the class considered in this paper would be £14,000, using a 600 r.p.m. pressure charged Diesel and is recommended as further astringency in the design.

Evaporator Plant

Provided any of the vital parts such as the heating elements, the brine and distillate pumps and the salinometer control can be quickly replaced by spares in the event of failure or overhaul necessity on the parts in use, it is reasonable that only one evaporator plant be installed. A capital cost saving of £8,500 would result.

Steam for Cargo Heating Coils

The steam/steam generator plant fitted in this design, and in many present day tankers, is a heavy and costly aggregate of heat exchangers, piping and control devices, calling for its full share of maintenance. Cargo and fuel tank heating coil jointing techniques have much improved in recent years and if satisfactory observation facilities are provided, an occasional minor contamination in the returning drains should not be difficult to cope with.

It is agreed that the dangers of oil contamination of boiler feed and heating coil material pick-up by acid condensate and its return to the boilers in the drains, require completely effective safeguards. The capital cost saving of about £13,800 in dispensing with the steam/steam generator plant and its associated steam and feed pipe systems, however, is worth weighing against alternative means of protecting the main boiler tubes against the hazards of feed contamination.

Air Conditioning Coolant Source

It has been shown that a shortcoming of the back pressure turbo generator cycle is the tendency for the build-up in tropical operating conditions of exhaust steam surplus to the offtake by the de-aerator, air heater and evaporator, due in part to the electric motor drive power for the air conditioning refrigerant compressors.

If means of providing accommodation space air cooling from the energy available in the auxiliary exhaust steam can be found, the benefits would be:

a) No requirement for additional kW rating on generators to power refrigerant compressors.
b) Smoothing out of exhaust steam flow quantity by making air conditioning load subtractive from this quantity in the tropics instead of additive.
c) Opening made for fitting of an L.P. feed heater.

Such means could be either

1) A turbine driven, centrifugal freon compressor taking steam from the exhaust range and exhausting to the main condenser at sea. 15lb./sq. in. gauge steam supply—atmospheric exhaust in harbour.)
2) Thermo compressors operating on exhaust motive steam in a vacuum refrigeration plant.

Either scheme would derate both generators by at least 50 kW; (2) would be more expensive than (1) but would have the advantage of mechanical simplicity.

ACKNOWLEDGEMENT

The author wishes to thank the Management of Shell Tankers Limited for permission to publish this paper, and to thank his colleagues in the Technical Division for their help in its preparation.

REFERENCE

1) Dudley, S. A. "Flow Characteristics of Main Condenser Scoop Injection System Based on Ship Board Tests". Read at New England Section, S.N.A.M.E., 16th May 1958.

Appendix

Air Conditioning Equipment

Amidships unit capacity 250,000 B.t.u./hr. one 20 h.p. motor driven "Freon 12" compressor running at 860 r.p.m. After unit capacity 725,000 B.t.u./hr. one 60 h.p. motor driven "Freon 12" compressor running at 860 r.p.m.

Air Conditioning "Freon 12" Condenser Circulating and Bilge Pump

One horizontal centrifugal pump, bilge duty 174 tons/hr. at 50ft. head, circulating duty 90 tons/hr. at 90ft. head. Speed 1,750 r.p.m. Priming by steam ejector.

Generator—Diesel

One 6 cylinder four stroke Diesel running at 1,200 r.p.m. direct coupled to a 125 kW 440 volt a.c. generator. Radiator cooling, 24 volt battery start.

Generator—Turbine

Two back pressure multi-stage turbines driving 600 kW 440 volt a.c. generators through single reduction gearing.

Atmospheric Drain Tank Extraction Pump

One vertical centrifugal pump, capacity 35,000lb./hr. against 80lb./sq. in. gauge at 1,750 r.p.m.

Air Ejector—Main

One single-element, two-stage ejector, capacity 70lb./hr. of air from 28·5in. Hg vacuum to atmospheric pressure.

Air Ejector—Emergency

One single stage ejector, capacity 66lb./hr. of air from 27·5in. Hg vacuum to atmospheric pressure.

Ballast Pumps

Two steam turbine driven vertical centrifugal pumps, capacity 2,000 tons/hr. against 92ft. total head at 1,000 r.p.m.

Boilers

Two variable superheat boilers with economizers and steam air preheaters, design basis each boiler:

Evaporation rate, normal ...	80,000lb./hr.
Evaporation rate, maximum ...	95,000lb./hr.
Steam pressure at superheater outlet	600lb./sq. in. gauge
Steam temperature at superheater outlet	900 deg. F.
Feed inlet temperature	240 deg. F.
Boiler tube heating surface ...	8,957 sq. ft.
Superheater heating surface ...	1,950 sq. ft.
Stud tube economizer heating surface	1,443 sq. ft.
Cast iron gilled tube economizer heating surface	5,645 sq. ft.
Flue gas temperature leaving cast iron gilled tube economizer at normal evaporation	335 deg. F.
Number of oil burners	4
Furnace volume	1,071 cu. ft.

Bilge Pump—Auxiliary

One horizontal screw displacement pump, capacity 90 gal./min. against a total head of 60ft. at 1,160 r.p.m.

Capstan—Cable

One steam turbine driven anchor cable capstan rated 62 tons at 40ft./min. Emergency drive from port warping capstan gives pull of 62 tons at 19ft./min.

Capstans—Warping

Four steam turbine driven warping capstans, rated 16 tons at 100ft./min.

Cargo Pumps

Four steam turbine driven vertical centrifugal pumps, output 1,900 tons sea water per hour against 345ft. total head running at 1,200 r.p.m.

Circulating Pump—Auxiliary
One vertical centrifugal pump, capacity 4,000 gal./min. against 25ft. head at 1,150 r.p.m.

Circulating Pump—Main
One turbine driven horizontal axial flow pump, capacity 22,000 gal./min. against 15ft. head.

Combustion Control
Simplified control including master controller fuel oil control and forced draught fan control. Variation of air/fuel ratio is by manual bias of the forced draught control signal.

Compressor—Combustion Control
Two compressors, twin cylinder air cooled Vee, capacity 30 cu. ft./min. at 1,160 r.p.m. reservoir pressure 55lb./sq. in. gauge.

Compressor—Deck
One 2-cylinder air cooled vertical compressor, capacity 115 cu. ft./min. reservoir pressure 100lb./sq. in. gauge driven by 1,800 r.p.m. four-cylinder air cooled Diesel engine.

Compressor—General Service
Two compressors twin cylinder air cooled Vee, capacity 25 cu. ft./min. at 1,160 r.p.m. reservoir pressure 100lb./sq. in. gauge.

Condenser—Auxiliary
One "U" tube atmospheric condenser. Cooling surface 2,560 sq. ft. 1in. O.D. × 14 S.W.G. cupro-nickel tubes welded into cupro-nickel tube plate.

De-aerator
One de-aerating contact feed heater rated 190,000lb./hr. of feed from 100 deg. F. to 240 deg. F. and reduce oxygen content to 0·01 c.c. litre.

Desuperheater
One spray type rated 80,000lb./hr. from 900 deg. F. to 600 deg. F., spray supply from feed pump 750lb./sq. in. gauge at 240 deg. F.

Emergency Fire Pump
One pump, capacity 37½ tons/hr. against 150ft. head driven by hydraulic motor at 2,750 r.p.m. Motive power from hydraulic pump driven by deck compressor Diesel.

Engine Room Lift
Lift boiler flat level to upper poop and boat decks rated one ton capacity.

Evaporators
One single effect plant with sea water circulated distiller and air ejector, capacity 60 tons a day.
One single effect plant with condensate circulated distiller and air ejector, capacity 40 tons a day.
Each plant fitted with brine pump and distillate pump, driven by common motor.

Evaporator Sea Water Supply Pump
One horizontal centrifugal pump, capacity 400 gal./min. against 50ft. head at 1,750 r.p.m.

Extraction Pumps—Main
Two, two-stage free suction head pumps, capacity 160,000lb./hr. against 65lb./sq. in. gauge from vacuum of 28·5in. Hg.

Foam Fire Fighting Installation
Foam system installed to protect boiler room, cargo pump room, or main deck and cargo spaces. Capacity 16,000 gallons of foam. Bottled CO_2 gas pressure source.

Forced Draught Fans
Two single speed fans with inlet vane control, capacity 32,000 cu. ft./min. of air at 100 deg. F. against 20in. W.G. at 1,750 r.p.m.

Feed Pumps—Main
Two steam turbine driven horizontal three stage centrifugal pumps, capacity 240,000lb./hr. against 740lb./sq. in. gauge at 4,750 r.p.m.

Feed Pump—Steam/Steam Generator (Electric)
One horizontal centrigugal pump, capacity 6,000lb./hr. against 180lb./sq. in. gauge at 3,500 r.p.m.

Feed Pump—Steam/Steam Generator (Steam)
One direct acting reciprocating pump, capacity 45,000lb./hr. against 180lb./sq. in. gauge.

Domestic Fresh Water Pumps
Two positive displacement pumps. Capacity 4·75 tons/hr. each against 45lb./sq. in. gauge. Controlled by pressure switch, speed 1,750 r.p.m.

Fuel Transfer Pump—Forward
Two vertical rotary displacement pumps, capacity 40 tons/hr. against 110lb./sq. in. gauge at 274 r.p.m.

Fuel Transfer Pump—Aft (Electric)
One vertical triple screw pump capacity 36 tons/hr. against 50lb./sq. in. gauge at 1,140 r.p.m.

Fuel Transfer Pump—Aft (Steam)
One vertical duplex reciprocating pump, capacity 100 tons/hr. against 50lb./sq. in. gauge.

Fuel Burning Installation
Two rotary screw type pumps, capacity 15,000lb./hr. of fuel against 450lb./sq. in. gauge at 1,150 r.p.m. feeding four pressure jet burners per boiler. Three 50 per cent capacity fuel heaters.

General Service Pump
One steam turbine driven horizontal centrifugal pump, capacity 1,170 gal./min. against 200lb./sq. in. gauge at 3,550 r.p.m.

Lubricating Oil Coolers
Two coolers arranged for single or parallel operation, capacity of each (working in parallel) to reduce 15,000 gal./hr. of oil from 135 deg. F. to 110 deg. F. with sea water at 90 deg. F.

Lubricating Service Pumps
Two vertical screw displacement pumps each rated 548 gal./min. against 45lb./sq. in. gauge at 1,150 r.p.m.

Lubricating Oil Purifier
One centrifugal purifier, capacity 500-800 gal./hr.

Propeller
One "Nikalium" five-bladed propeller, mean pitch 17·33ft., 23·0ft. diameter, blade area 267 sq. ft.

Refrigerating Machinery
Two vertical twin cylinder "Freon 12" compressors each operating on one space but capable of maintaining the temperature in both spaces. Coolers are fan blown battery type. Electric defrosting fitted in freezing room unit. The ice capacity of each unit is 0·85 ton.

Steam/Steam Generator
One generator and drain cooler capacity of 45,000lb./hr. of steam at 150lb./sq. in. gauge.

Steering Gear

Four ram electro-hydraulic with two variable delivery pumps driven by 100 b.h.p. motors.

Stripping Pumps

Three vertical duplex, capacity each 200 tons/hr. against 140lb./sq. in. gauge

Sea Water and Fire Service Pumps

Two vertical centrifugal pumps, capacity 200 gal./min. against 115ft. head at 1,750 r.p.m. Maximum head available 128 feet.

Tank Cleaning Sea Water Heater

One heater rated 1,000 gal./min. from 50 deg. F. to 180 deg. F.

Tensioning Winches

Five two-cylinder, horizontal, totally enclosed steam winches with pull of 80,000lb. from seventh layer of rope on drum.

Discussion

CDR. E. TYRRELL, R.N. (Member), in opening the discussion, said that Lloyd's List had recently stated, that, British shipowners considered that Shell Tankers had done more to improve the technical quality of British ships, since the war, than any other single body. He was sure that everyone present would agree that Mr. Main had demonstrated, by his paper, that he had more than played his part in this achievement.

Only large organizations could afford the intensive research and development necessary for technical success. This, if properly applied, would achieve a better and cheaper product which would earn money, in excess of that originally spent. The work undertaken by Mr. Main and his colleagues had obviously been expensive, but his paper proved that it had saved his company money.

But what happened to the small shipowner? He had neither the money nor sufficient qualified staff to carry out an investigation of the type undertaken by Mr. Main. Nor were there in this country powerful consulting naval architects and marine engineers, such as Gibbs and Cox, George Sharp and John J. McMullen and Associates, who operated in the United States. These consultants had ready at their fingertips, or could readily obtain, most of the information required by the small shipowner, and could sell it to him more cheaply than he could obtain it by making his own investigations.

It was right that the shipbuilder should accept full technical responsibility for the ships he built, but in the U.S.A. and, to a lesser extent, on the Continent, the turbine designer and manufacturer, directly or indirectly, accepted responsibility for the machinery installation. He did this, either by advising the shipbuilder how best to meet the owner's requirements, or by negotiating with the owner direct.

The turbine builders each sought to convince the owner that their machinery was the best bargain. Detailed costing and economics of many, often hundreds, of possible installations drawn from heat balance calculations, were often necessary to obtain the most advantageous set of machinery. This work was difficult, if not impossible, by normal methods, and computer programmes had been developed for the more common marine thermodynamic cycles. These made it possible quickly to obtain the fuel rate and capital cost of scores of possible machinery installations. Mr. Main might have obtained even better solutions to his problems, had he used the computer techniques of the major turbine specialists. The advantages of the use of these facilities by the small shipowner were obvious.

Mr. Main had assumed that the cost of repair yard labour would continue to rise, throughout the life of the ship and that capital and maintenance costs must be kept down by simplification of the machinery installation. Yet he had made two basic decisions which adversely affected these premises. On page 279 he had given his reasons for selecting 240 deg. F. feed temperature. If he had selected 280 deg. F. to 290 deg. F. feed to the economizers, he would have gained in capital cost and in boiler maintenance charges. Experience showed that 240 deg. F. feed caused overmuch corrosion and deposit troubles with steel economizers and that it was necessary to fit cast iron primary economizers, as had been done in this case. With 280 deg. F. feed, satisfactory results were obtained with steel economizers, which were cheaper and lighter than cast iron.

Mr. Main had also observed, on page 279, that in this case the maintenance of the feed temperature depended on substituting a live steam heating source for the bled steam, drawn from the propulsion turbines at sea and that the feed temperature, for a variety of reasons, was frequently not upheld. Bled steam for 280 deg. F. steam could usually be obtained from the crossover pipe, so that no expensive bled steam belt was needed in the turbine. A reliable and automatic supply of live steam could be fed to the de-aerator by servo air-operated valves, of the type which Mr. Main accepted for automatic combustion control. These he rather illogically rejected for feed temperature maintenance.

The turbines illustrated in Figs. 6 and 7 must be expensive. Mr. Main had shown on page 284 that he was doubtful of the value of the H.P. astern turbine and this was expensive both in first cost and in fuel.

The H.P. turbine had 10 stages, which was more than some competitive designs had. More stages cost more, and gave rise to large diaphragm gland leakage, as a longer rotor had to have a larger diameter, to keep up the whirling speed.

Although it had not been stated in the paper, he believed that this H.P. turbine had built-up diaphragms which were thicker than welded ones at the same strength. This again increased the rotor length, diaphragm gland diameter and hence the leakage. Some people maintained that built-up diaphragms were not satisfactory for steam temperatures above 850 deg. F. Perhaps Mr. Main would remark upon whether the omission of welded diaphragms contributed to the decision to limit the steam temperature to 900 deg. F. The internal parts of any double casing turbine were difficult to reach. He had witnessed maintenance work carried out on the inlet nozzles of the H.P. turbine in the *Caltex Edinburgh*. The outer and inner top half casings, the H.P. astern casing and the rotor had had to be lifted before it was possible to reach the nozzle group.

There had been instances of trouble with the internal gland of double casing H.P. turbines. The glands tended to close in at the horns, and he would be glad to know if this trouble had been eliminated.

In his opinion, there was little justification for the extra cost, complexity, weight and space of the double-flow L.P. turbine, as the horsepower could be obtained with a single flow design. Although diaphragm gland leakage was of little importance compared with that in the H.P. turbine, the large diameter gland required for the long rotor had an adverse effect on the water rate of the turbine.

Mr. Main seemed to have a somewhat expensive and complex set of turbine machinery, which incorporated features adversely affecting the steam consumption. Yet on page 281 he said that the turbine "was accepted on the grounds that it offered a design having an efficiency comparable with, if not marginally better than, any other marine design available". Perhaps Mr. Main could say whether, in fact, any other designs were considered for installation in this country.

On the Continent and in the U.S.A., the shipowner could, during the design of any turbine machinery installation, satisfy himself that the indicated fuel rate would be met, as this was usually guaranteed under financial penalty. It would be interesting to know in this case whether the fuel rate had been guaranteed. Financial guarantees, to be effective, must be large enough to offset at least half the losses the owner may sustain from the increased fuel consumptoin, during the life of the ship and it was known that guarantees for land sets anyway, had been accepted on a basis of 20 per cent repayment for every one per cent deficiency in fuel rate, i.e. the turbine manufacturer received nothing for his machinery if the fuel rate was five per cent above the guaranteed figure.

He would add that he had been given non-bled steam rates of 5·79 and 5·71 for comparable turbines from two other designers. These were figures which they were prepared to guarantee with the onset of financial penalties at 2½ per cent and complete rejection of machinery at 5 per cent above the guaranteed rate.

DR. R. S. SILVER, D.Sc., M.A., B.Sc. (Member) said that Mr. Main had produced a paper of very great interest and competence on a very important subject. The clear statement, that capital charges must be assessed at 14 per cent, was a very valuable guide to designers, who required to evaluate the extent to which improvements in the performance would actually be profitable. Combined with the stated fuel price of £6 per ton, and after allowing for boiler efficiency, this meant that an expenditure of £1,000 had to give a reduction in heat requirement of 110,000 B.t.u./hr. at least to be worthwhile.

There were many things in the paper to which he would like to refer, but he proposed to confine his remarks to the fuel rate aspects. Mr. Main had given a comparison of several different systems and it might be useful to discuss the reasons for the differences which he had found. To do so he would, for convenience, use the method of energy balance comparison which he had put forward in 1959.* He had been a little more modest about it before, but when he heard people still advocating the use of computers for something which could be done very simply once the algebra had been attended to, it did seem to him worth repeating. It might be useful, also, to confirm Mr. Main's results by the method referred to, which could be operated without troubling to do any balance diagrams other than the first. It enabled one to point to the essential nature and problems of improvement.

TABLE II.—PARTIAL HEAT REQUIREMENT COEFFICIENTS FOR CONDITIONS GIVEN BY MR. MAIN

(Evaluated from Formulæ in Ref. 1)

1) L.P. Bleed at 10 lb./sq. in. abs.	$\frac{\partial H}{\partial b_{10}}$	=	− 149 B.t.u./lb.
2) Back pressure generator	$\frac{\partial H}{\partial T_g}$	=	+14,400 B.t.u./hr. kW.
3) Condensing generator	$\frac{\partial H}{\partial T_g}$	=	+16,300 B.t.u./hr. kW.
4) System electric pumps on back pressure generator	$\frac{\partial H}{\partial T_g}$	=	+13,210 B.t.u./hr. kW.
5) System electric pumps on condensing generator	$\frac{\partial H}{\partial t_g}$	=	+15,230 B.t.u./hr. kW.
6) Turbo-feed pump	$\frac{\partial H}{\partial t_p}$	=	+14,820 B.t.u./hr. kW.
7) Circulating pump turbine	$\frac{\partial H}{\partial T_p}$	=	+16,650 B.t.u./hr. kW.
(3) − (2)		=	+1,900 B.t.u./hr. kW.
(4) − (6)		=	−1,610 B.t.u./hr. kW.
(2) − (7)		=	−2,250 B.t.u./hr. kW.

Table II gave the values of the relevant partial heat coefficients, calculated from the data given by Mr. Main. One could assume that the system shown in Fig. 2 had already been calculated and balanced. Then from Table II, without calculating in detail any of the other systems, one could deduce as follows, slide rule accuracy being quite sufficient for the deductions:

1. Mr. Main's Fig. 3:
Inclusion of an L.P. Heater bleeding at 10lb./sq. in.
The heat requirements would be reduced by 149 B.t.u. for every pound of steam bled to the heater. Thus, if one took 5,000lb./hr. the heat requirement would be reduced by 745,000 B.t.u./hr. Allowing for a boiler efficiency of 87 per cent and a calorific value of 18,500 B.t.u./lb., this is 46·4lb./hr. of fuel or 0·5 tons/day. Thus the result obtained in Mr. Main's Fig. 3 system was obtained immediately.

2. Mr. Main's Fig. 10:
Condensing Turbo-generator.
From Table II it was seen immediately that the condensing turbogenerator was less economic than the back pressure set. Its partial heat requirements exceeded those of the back pressure set by 1,900 B.t.u./hr kW. Hence for 450 kW. the system requirement would be increased by 855,000 B.t.u./hr. i.e. 0·575 tons/day on the fuel rate. Mr. Main's balance in Fig. 10 agreed closely with this at 0·6 tons/day extra.

3. Mr. Main's Fig. 9:
Condensing Turbo-generator with L.P. Heater.
Since $\frac{\partial H}{\partial b_{10}}$ was negative, one could offset the increased requirement of the condensing set by increasing the L.P. bled steam. In Fig. 9 Mr. Main had used 8,990lb./hr. of L.P. steam, which would therefore reduce the system requirement of Fig. 10 by 149 × 8,990 = 1,340,000 B.t.u./hr., which was equivalent to 0·9 tons/day. Mr. Main showed a reduction of 1·1 tons/day.

4. *Feed Pumps*
Table II showed that electric feed pumps on the back pressure generators had a partial heat requirement of 13,210 B.t.u./hr. kW. as against 14,820 B.t.u./hr. kW. for the independent turbine pumps, i.e. a benefit of 1,610 B.t.u./hr. kW. Thus with a pump load of 260 h.p. (194 kW.) the reduction in system heat requirement with an electric driven pump was 313,000 B.t.u./hr. This corresponds to about 0·21 tons/day fuel saving, rather less than the figure of 0·24 tons/day quoted by Mr. Main.

One noticed from Table II also that the electric pump would be actually worse than the turbo-feed pump if the generators were condensing, since the partial requirement was then 15,230 B.t.u./hr. kW.

It would be seen that the use of the method of partial heat requirement coefficients greatly facilitated comparisons. By starting only with one complete heat balance, in this case Fig. 2 one could arrive at the total thermal effects for any other system very quickly and without doing any other complete balance.

It was now worth discussing the reasons for some of the effects, and the practical pointers which they gave. Why was the condensing set worse than the back pressure set? Why was so little benefit obtained from the electric pump?

It could be shown that

$$\text{back pressure } \frac{\partial H}{\partial T_g} = 2{,}080 + \frac{7{,}240}{\eta_b} \text{ B.t.u./hr. kW.} \qquad 1)$$

$$\text{condensing } \frac{\partial H}{\partial T_g} = \frac{10{,}550}{\eta_c} \text{ B.t.u./hr. kW.} \qquad 2)$$

In these equations η_b and η_c were the adiabatic efficiencies of the respective turbines, i.e. actual heat drop divided by adiabatic heat drop. From the data given by Mr. Main it appeared that he had assumed the same adiabatic efficiency, of approximately 58 per cent for both cases, which was no

* Proc.I.Mech.E., 1959, Vol. 173, p. 297.

doubt justified. One saw that to have the condensing set give better results than the back pressure set, it would require an adiabatic efficiency η_c' such that

$$\frac{10,550}{\eta_c'} < 2,080 + \frac{7,240}{\eta_b}$$

With η_b = 58 per cent, η_c' would need to be greater than 73 per cent. Thus to offset the benefits of feed heating by the back pressure set, a very much higher efficiency was required for a condensing set.

Turning next to the feed pump, it could also be shown that

$$\frac{\partial H}{\partial t_p} = 210 + \frac{7,200}{\eta_p} \qquad 3)$$

$$\frac{\partial H}{\partial t_g} = 300 + \frac{7,590}{\eta_b} \qquad 4)$$

where again η_p is the adiabatic efficiency of the feed pump turbine.

Thus, the slight benefit of the electric pump was due to the higher adiabatic efficiency of about 58 per cent for the generating turbine, as compared with about 50 per cent for the feed pump turbine. However it started with a disadvantage in the numerator, so that it had to have at least 53 per cent adiabatic efficiency to be equivalent to a feed pump turbine of 50 per cent.

As remarked earlier, Mr. Main's figures meant that to be worthwhile, £1,000 expenditure had to reduce heat requirements by 110,000 B.t.u./hr. at least. Now, for a pump load of 260 h.p. (194 kW.) the partial requirements due to the feed pump would be

$$40,700 + \frac{1,395,000}{\eta_p} \text{ B.t.u./hr.}$$

With η_p = 50 per cent this was 2,830,000 B.t.u./hr. In order to save 110,000 B.t.u./hr., it followed that the feed pump turbine adiabatic efficiency had to be 52 per cent, i.e. a reduction of four per cent in steam consumption.

Thus, for a turbo-feed pump approximately £4 per normal load horsepower of extra capital cost would seem to be worthwhile, provided it saved more than four per cent in pump steam consumption. Conversely, an excess of four per cent in steam consumption seemed to be acceptable, provided it reduced the capital cost by more than £4 per normal load horsepower. He would be glad to know whether Mr. Main agreed with these figures.

On the circulating pump one saw from Table II that the adoption of the turbine drive meant an extra 2,250 B.t.u./hr. kW. in the heat requirements as compared with electric drive using the back pressure generator. On 50 per cent power this meant 157,000 B.t.u./hr., which was equivalent to about 130lb./hr. of superheated steam generation. This was rather less than calculated by Mr. Main, but it reinforced his conclusion as to the low penalty for the greater adaptability of turbine driven pumps to circulating water requirements, for ships passing through a wide range of sea temperature.

To conclude, it was worth noting that, all the variations in auxiliaries, which were discussed in the paper, only varied the heat requirements by about one million B.t.u./hr. The total heat requirements were about 180 million B.t.u./hr., of which the requirements due to the propulsion turbine were slightly over 160 million. Thus, the possible benefits, due to auxiliary changes, were of the order of 0·6 per cent of the requirements due to the propulsion turbine. This was merely a repetition and reinforcement of what they all knew already, namely that a very minor change in propulsion turbine operating efficiency could offset or dwarf any savings due to changes in auxiliaries.

One circumstance occurred to him as worth comment. It was interesting as well as amusing that the author had gone to an oil-lubricated stern tube, i.e. he was now using oil for something which was traditionally water lubricated, while turbo-feed pump makers had produced water lubrication for a duty which was traditionally done by oil.

PROFESSOR H. LAMERIS (Member) congratulated Mr. Main on his very detailed lecture which, he said, gave an interesting and well-founded report about all the viewpoints which had to be considered when compositing a new layout for an important tanker machinery installation. Everyone would be grateful to Mr. Main and his company for the paper.

The chosen steam pressure of 600lb./sq. in. and steam temperature of 900 deg. F. were, at the moment, on the conservative side and could be increased to 850lb. and 950 deg. F., with a consequent gain of three per cent on the fuel rate and only a small increase in investment and weight. He fully agreed with Mr. Main that the reliability of the installation was predominant, but he did not believe that there were any risks that might be expected.

He again agreed with Mr. Main that for reasons of simplicity, in other words, reliability, weights and investments, the back pressure turbo-alternator was the right solution, provided that the exhaust steam, at all reasonably foreseen loads and load-hours, had not to be dumped into the condenser, resulting in excessive heat running to sea.

Therefore, he thought it would be advisable to increase the temperature in the de-aerator with decreasing load, thus at the same time protecting the economizer against SO_3 corrosion and fouling, at the penalty, of course, of a somewhat lower boiler efficiency at part loads. This could be easily arranged by increasing the setting of the back pressure reducing valve to the condenser. With an increase of the de-aerator temperature from 240 deg. F. at full load to 280 deg. F. at about 1/3rd load, no exhaust steam had to be dumped into the condenser.

In his opinion, this was an essential addition to the back pressure system, in order to combine load flexibility and fuel economy, with the additional advantage of less corrosion in the economizer. Of course, the suction head of the feed pump must be laid out for the most adverse condition.

The author's arguments about the choice of layout of the economizer had been very interesting, but he would like to know whether the author had also considered the combination with a small gas-airheater, thus reducing the funnel temperature to say, 295 deg. F. which would improve the boiler efficiency by about 0·9 per cent, equalling, in this ship, a gain in fuel consumption of about 1·1 ton/day.

Mr. Main had mentioned that dual atomizing burners would be used. His question was: would not the danger of SO_3 corrosion at low loads be promoted with these steam atomizing burners. While admitting lack of practical experience with the foregoing dual atomizing burners, in his opinion mechanical atomizing burners with return flow combined with shut-off possibility, had an advantage over the proposed burners as they could remain in position even when they were not atomizing. This gave at the same time the possibility of a fully automatic operation of the burners, depending on the boiler load, so that no boiler operator was needed; the boiler combustion, the number of burners in operation, and, of course, the feedwater, being automatically controlled. These burners, and this arrangement, were supplied by his company. The automatic control of both boilers was centralized on the engine room control panel. In this way an easier control was achieved, with a saving of personnel.

He would agree with the author that a nozzle-controlled H.P. turbine had its advantages in meeting varying transport market circumstances and as these could not be foreseen, the H.P. turbine could be better arranged with such control valves. The extra cost and maintenance were very small.

When 60 per cent of the ahead power had to be available for astern power there was no alternative to accepting an H.P. astern turbine, and its subsequent loss in efficiency when running ahead. A controllable pitch propeller would overcome these difficulties and, in fact, do away with the complete H.P. and L.P. astern turbine and the accompanying losses. The advantages were then: (1) the running speed of the propulsion turbine could be adjusted to that most effective in any given service or weather condition and held constant at the most efficient level; (2) as the astern turbine became redundant, higher steam temperatures could be employed and the ahead turbine would remain at a more constant temperature during manœuvring; (3) intermediate reheat could easily be introduced; (4) automatic turbine control was also easier, so the way to the fully automatically controlled engine room was open.

His company had at present on the design board controllable pitch propellers up to 20,000 s.h.p. while, of course, it had in the course of the years already supplied several propellers of this type, though to lower outputs.

The double flow L.P. turbine had a very long bearing distance. From a reliability point of view he would prefer, for this output, a single flow turbine with a shorter length.

The feed pump had a 25 per cent larger capacity than the maximum boiler load, the main circulating pump 15 per cent and the forced draught fan also 15 per cent. These figures were not uncommon, but maximum boiler load was already about 25 per cent higher than was needed for full power ahead. Therefore, when the ship was moving slower than maximum speed, these machines would run heavily overloaded with exhaust steam, so very careful consideration was necessary.

It was very interesting to learn that for the future only one turbo alternator would be considered. What was the author's opinion about substituting for the rather heavy and noisy Diesel mentioned, an emergency gas-turbo alternator?

Did not the author think that the oxygen content reduced in the de-aerator to only 0·01 cc/1. e.g. 14_{μ} gr./1., was a rather high figure for a high pressure boiler?

MR. E. G. HUTCHINGS, B.Sc. (London) (Member) said that the overall design of a marine steam plant was always a compromise and, given identical data, it was unlikely that two independent persons would come to the same conclusions. Possibly the approach to the subject was, in the long view, more important than the solution for a specific vessel and he entirely agreed with the author's approach to this complex problem. Generally speaking, he would agree also with the author's decisions, but would offer the following comments.

He would suggest that the whole feed cycle and boiler plant were affected, not so much by the decision to use non-condensing turbogenerators, but more by the use of a design of double casing turbine, which incurred a heavy penalty in initial cost if additional bleed points were considered.

The author had stated that mild steel economizers were acceptable, providing the water entered at 280 deg. F. or more. An increase in pressure of the exhaust range to 35lb./sq. in. gauge, giving a feed temperature of 280 deg. F., together with a low pressure feed heater would result in a significant improvement in efficiency. It was his impression that with certain other designs of turbine, such a cycle could be arranged without incurring an excessive cost penalty. The higher feed temperature would permit the use of mild steel, throughout the economizer, so that a funnel temperature of 345 deg. F. could be obtained without extra boiler cost, despite the higher feed temperature and there would be a reduction in the weight of the boiler plant.

The incidence of pipe joint and valve leakage, particularly in a marine plant, was affected far more by the choice of steam temperature than by the pressure, but he agreed with the choice of 600lb./sq. in. with the temperature chosen, due to the greater care necessary with feed water treatment and initial cleanliness of the system above this pressure.

On the question of temperature, he felt that one could go higher than 900 deg. F. without serious risk of vanadium attack on the superheater tubes, but it was true that the higher the steam temperature, the greater would be the maintenance due to deposits on the superheater tubes and vanadium attack on the superheater supports. The problem of superheater deposits could be greatly reduced if the fuel were washed to remove sodium, so that sodium vanadates were not formed. This approach had been receiving more attention in recent years and it would be interesting to hear the author's views on this point.

The author had stated that, with a feed temperature as low as 240 deg. F., care must be taken to ensure that this figure was maintained at all times. In this he was absolutely right and the point could not be emphasized too strongly. This was another reason for adopting a higher feed temperature. Below 240 deg. F., two problems occurred, namely, slagging and corrosion of the economizer. By increasing the feed to 280 deg. F., a significant margin was given on both these problems. However, to maintain the boiler efficiency without increasing the cost, it was necessary to use mild steel economizers throughout, so there was still only a small margin before corrosion became likely.

To select a gas temperature, some 40 deg. F. higher than the optimum shown in Fig. 5, was sound for the reasons stated and also because the extra fuel cost, when operating at part loads, would be an even smaller proportion of the fuel bill at that load.

The author's approach to forced draught fan margins was a new one to him, but it was quite realistic and had a lot to recommend it, when the resistance of the oil burners represented a large proportion of the total draught. The author should not, however, overlook the fact that the fan margin was to some extent an ignorance factor. It was not at all unknown, for a vessel to require in practice, a higher evaporation than the maximum specified for the boilers. This was no criticism of the turbine manufacturer: it could come from the auxiliaries or some unexpected domestic load put on at the last moment. If the boiler evaporation in this case had been under-estimated by 5 per cent, then the total (clean boiler) resistance at its maximum evaporation would be increased to 16·5in. and the impressive head margin of 140 per cent reduced to 66 per cent. On his own calculations, he could only make the margin 104 per cent: perhaps this was a misprint. He would also suggest that a margin should be included for the fouling of the steam airheater. He considered that the absolute margin allowed by the author was adequate, but would emphasize that if this approach were adopted, the margin on the resistance of the gas lanes must be at least 100 per cent and not the customary 20 to 30 per cent usual, when the margin was applied to the total resistance.

He had reached now the point of disagreement with the author. He would not agree that considerable advance had been made in oil burning for marine installations in recent years. It was certainly true that there was a much wider choice available today, but apart from the development of very wide range spill burners, there was little that was new. Certain companies had recently developed steam atomizers, but his own company had had high efficiency steam atomizing burners in use for many years. These burners offered important advantages for the cost of a little water, namely a very high combustion efficiency, giving CO_2 figures consistently above 14 per cent, a wide range with simple control and constant spray angle, without the use of very high oil pressures and, above all, boiler cleanliness was greatly improved with a consequent reduction in maintenance. In regard to what had been said about steam atomizing burners promoting SO_3 corrosion, the fact was that the steam atomizer was an advantage since it permitted a reduction in the amount of oxygen present and thus the amount of SO_3. Certain steam atomizers only utilized steam for assisting atomization at low outputs, but his own opinion was that the pure steam atomizer was to be preferred, as the combustion efficiency was higher and the use of steam for atomizing throughout the trip resulted in a much cleaner boiler.

He would add that the adoption of a well-designed and proved steam atomizer would permit the fan to be designed confidently on a basis of 14 per cent CO_2, thus reducing the design pressure by 14 per cent, the design volume by nearly 7 per cent and the fan motor power by something in excess of 20 per cent, with consequent reduction in the electrical load on the vessel.

His company had frequently advocated steam atomizers for marine boilers, but until recently, these had usually been rejected due to the water consumption. It was interesting to note that some owners had used this type of burner for some years and a recent series of six tankers, fitted with steam atomizers, were obtaining a fuel consumption below 0·51b./s.h.p./hr. on trial, with a steam temperature of 850 deg. F. and a very simple feed cycle.

The author's approach to automatic combustion control was based on the assumption that regular specialized attention was not available. If this were so, he would agree with his decisions, but would suggest that an alternative approach was worthy of consideration. If one faced up to the problem and provided personnel in the ship, whose specific task it was to check and maintain the various automatic systems, then automatic control could with confidence be taken to a far higher degree than at present. The prizes offered by this approach were, first, more efficient operation and, in the long run, when confidence had been established, the number of engine room personnel could be reduced and routine maintenance carried out at sea.

He wondered whether the suggestion to use a smaller quantity of colder air for ventilation was wise, from a health point of view. Despite regulations and any mechanical arrangements made to prevent it, it was likely that personnel would continue to find some way of getting a strong blast of air onto their bodies. Particularly after sweating excessively, this could be more injurious to health if the ventilating air temperature was reduced.

Finally, he would say that he had found the paper of great value in their general studies. He admired the author's courage in setting down, in such detail, the reasons behind his choice of this machinery.

Mr. M. L. Ryall, B.Sc. (Associate Member) said that he had read Mr. Main's paper with particular interest, as he had shown himself a lucid exponent of, what might be called, the "simplicity school" of tanker machinery operators, who did not consider that the additional fuel saving, to be obtained by improving the overall efficiency of a cycle, with these steam conditions, by up to 5 per cent, was worth the extra capital cost or the increased maintenance charges. To judge from the number of ships in other tanker fleets, in service or building, which did incorporate a more complex cycle to improve overall efficiency, there were several tanker operators who would not agree with Mr. Main's conclusions. Now, who was right?

This, of course, was a complex problem, and it was not his purpose in this discussion to take sides. He would, however, disagree with Commander Tyrrell's assertion, that Mr. Main's choice would have been made easier with the use of an electronic computer. He had some limited experience in the use of computing, in the steam field, and would say that some problems which the tanker owner faced today were not soluble on the computer. Such problems were:

i) Will it become more and more difficult to get engineers to go to sea?
ii) Will the standard of these engineers decline?
iii) What is going to be the cost of unreliability in ten or fifteen years' time?
iv) What is likely to happen to fuel costs in the foreseeable future?

An electronic computer, useful tool though it was, could provide no answers to these problems. At least, it could be said that Mr. Main's arguments for his choice were in most cases fair and logical. However, the premises, on which the arguments were based, must in some cases be matters of individual opinion.

While there was much food for discussion in the paper, he proposed to confine his remarks to three items, namely, main turbines, circulating pumps, and turbo-alternators.

It was gratifying, in view of the brickbats which had been hurled in the direction of Pametrada, both officially and unofficially, in the last few months, to learn that the Association had at least one satisfied customer, and the wide experience of Mr. Main's company made his remarks worthy of respect in this direction.

Several technical points in connexion with the turbine design were raised in the paper and in the discussion by Commander Tyrrell, which were misleading. Firstly, it was not true that a double flow L.P. design was necessarily less efficient than a single flow L.P. design. The gain in efficiency, obtained through having a smaller diameter shaft in the single flow design, was offset by the loss in efficiency, caused by external gland leakage at the inlet end of the single flow design. There was no external gland leakage at the inlet to the double flow design of turbine. Thus, from the aspect of turbine efficiency, the type of L.P. turbine shown in Fig. 7 was as good as a single flow design. Furthermore, a single flow design would not, as claimed by Commander Tyrrell, have proved much cheaper. Single flow L.P. turbines at this power necessitated more extensive use of twisted and tapered blades at the exhaust end and these larger blades required more expensive materials, more complicated blade roots and more expensive rotor material. The reduction in cost in reducing the length of the rotor and casing was, to some extent, offset by the increase in overall diameter. However, it was worth noting that, the standard range of L.P. turbines now being developed at Pametrada would all be of the single flow type up to 25,000 s.h.p., that a single flow Pametrada L.P. turbine developing a maximum of 24,700 s.h.p. was at sea, and that single flow L.P. turbines of Pametrada design were currently being built to develop up to 26,500 s.h.p. The main reason for the designer's current preference for single flow design was the anticipated greater reliability of the shorter rotor. When it was remembered that, the design shown in Mr. Main's paper was first conceived over four years ago, it would be seen that the shipowner's choice of L.P. turbine was a reasonable one.

Commander Tyrrell drew attention to the fact that the turbines, shown in the paper, did not have welded diaphragms, which he claimed would have shortened the turbines. In fact, the turbines would have been no shorter if welded diaphragms had been fitted, as the strength properties of welded diaphragms and the brazed type shown were similar. The designers would now show a preference for welded diaphragms, however, mainly because these were readily replaceable and could be made with more accurate nozzles.

Mr. Main had discussed astern power requirements and the effect of the windage of the astern wheel, running ahead, on the overall turbine efficiency. The author's estimate of a possible 650 s.h.p. loss due to the astern turbines was not backed up by experimental evidence. The design estimate of total astern windage loss at 28¾in. Hg vacuum was 65 s.h.p., i.e. only about 0·3 per cent of the ahead service power. This estimate was supported by the well known work of Kerr, by the recent tests on the Continent by Suter and Traupel and by recent air turbine and steam Curtis wheel tests at Pametrada. Thus the most that could be expected to gain in efficiency, through a reduction in astern provision, by omitting the overhung portion on the H.P. rotor, as was only about 0·1 per cent. The designers' view was that admission of steam at over 850 deg. F. to the L.P. cylinder was not a good thing, from the distortion aspect and that astern power availability, on both cylinders could be useful, should one cylinder break down. His opinion was that the very large size and consequent inertia of this class of ship warranted the astern power which was specified.

Turning now to the subject of circulating pumps, it would appear, from the author's remarks, that there was a strong case for the steam driven pump, providing it was fitted with nozzle control and the type of inlet and overboard discharge arrangement described. He however, still felt that there was little to choose between steam and electric drive for this pump. The author had pointed to a £3,200 increase in generator cost if an electric pump were fitted, but he had not mentioned that an electric pump would probably be some £2,000 cheaper than the steam driven pump adopted. He would submit that the ability to vary the speed of the pump in the case of the steam drive was not an important consideration. It almost always paid, to go for as high a vacuum as was attainable, when running the machinery at sea, and it was only when, by so doing, the vacuum exceeded about 29in. to 29·2in. Hg that any fall off in overall efficiency began. The matter was referred to in his contribution to Messrs. McAlpine and Paterson's paper in 1960.*

Mr. Main had shown that the design of the circulating water inlet and discharge adopted, had reduced head loss in the circulating system, and so reduced both price and consumption of the pump. He could not help wondering what adverse effect this might have on the ship's resistance, however, was the author sure that, what he was gaining in machinery performance, was not being lost in hull resistance? Tests to answer this question would, he suggested, be difficult to carry out conclusively.

He agreed with the author's conclusions on the merits of the back pressure generator, relative to a condensing set in a ship of this type. Was it not possible to carry the case for simpler generators a stage further, without introducing an expensive, large, Diesel set with its costly maintenance requirements? He would suggest, that large savings in first cost would be available, if the T/A turbine was made a single stage Curtis wheel, taking steam from an H.P. bleed at 150-200lb./sq. in. abs., and exhausting back to the main engines at the transfer pipe. The alternator turbine would then be of very simple construction. The casing would only have to stand 25lb./sq. in. gauge and there would be no high temperature parts, as the bled steam supply would be at only some 650 deg. F. The rotor could be overhung, as in some cargo pump turbines and only the simplest glands would be required to deal with the low wheelcase pressure.

During manœuvring, supply to the alternator would be from the saturated range at a reduced pressure and the alternator would exhaust through an orifice to the main condenser. In harbour, the alternator would exhaust to the auxiliary condenser. Control should not prove any more difficult than with the old, much more complex, mixed pressure condensing sets.

The idea of this simple type of alternator turbine had not been put forward previously, to his knowledge, in Britain, though it was being offered in the U.S.A. There seemed to him to be no practical objection to passing steam into, as well as from, the main engines.

He estimated roughly that about £4,000 per set, for 600 kW units, could be saved relative to the cost of the back pressure sets adopted in the paper.

The efficiency of cycles, incorporating simple Curtis wheel turbo-alternators such as are described above, had recently been investigated at Pametrada. It appeared that not only would these alternators cost less, but they would also result in a cycle with at least the same overall efficiency, as a similar cycle, employing a normal multi-stage back pressure set. This was due partly to the reheating effect, on the main engine steam, of passing back into the main engines the comparatively hot exhaust steam from the alternators, which increased the available heat drop in the main engines and partly to a small corresponding reduction in L.P. exhaust wetness.

Such an alternator would be rather extravagant in harbour, but the comparatively short period of time when it would operate at harbour conditions made this of small importance. It was perhaps superfluous to point out that, with allowance for passing steam into the main engine at the transfer pipe, problems of exhaust dumping under tropical conditions ceased to exist.

He would be interested to hear what the author's view would be on the possible adoption of such simple turbo-alternators.

The author was to be congratulated on presenting such a stimulating paper.

MR. G. VON FEILITZEN said that being makers of main propulsion steam turbines, as well as several types of marine auxiliaries, the De Laval Ljungstrom Company had had reasons

* McAlpine, T. and Paterson, I. S. 1960. "Recent Developments in Pump Auxiliaries for Ships", Trans.I.Mar.E., Vol. 72, p. 229.

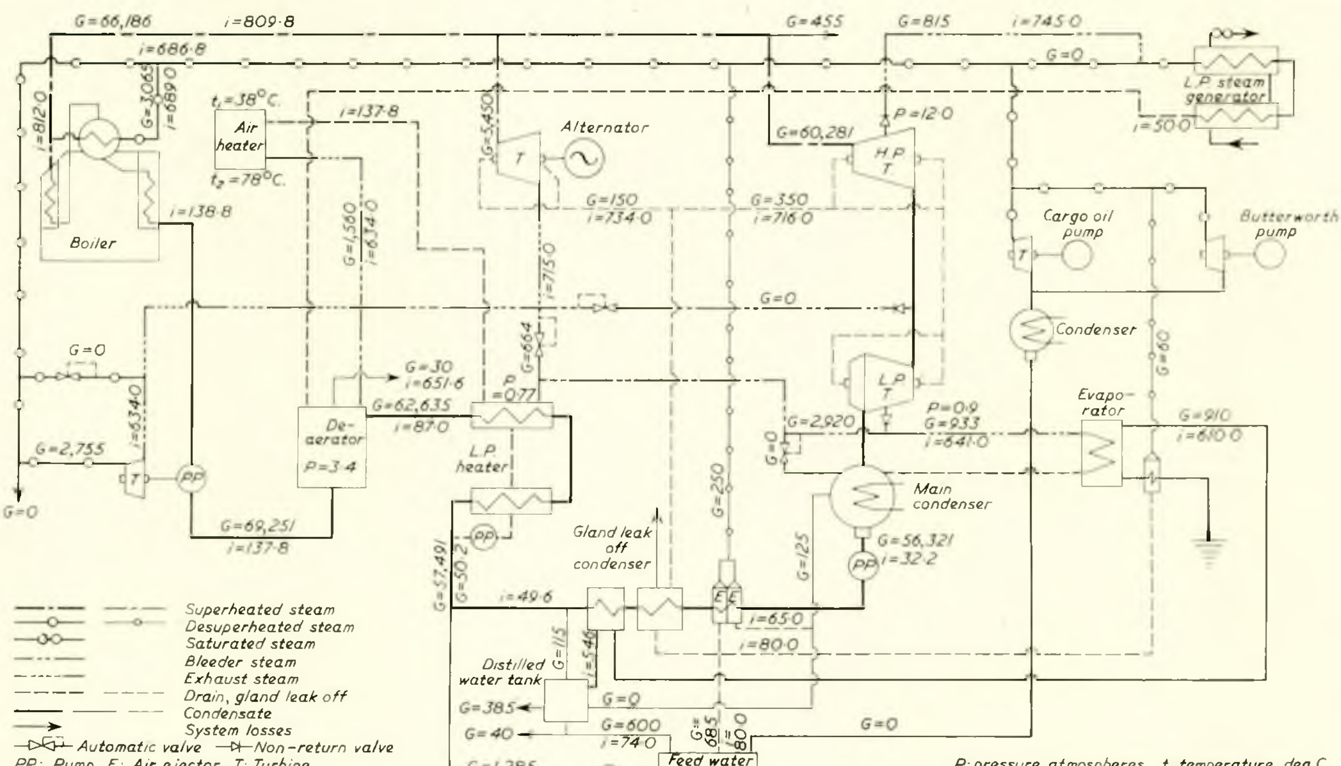

FIG. 16—*Heat balance for De Laval 22,000 s.h.p. propulsion machinery*

to make studies similar to the one presented by Mr. Main. On many essential points their practical experience and theoretical studies agreed with the results found by Mr. Main. They wished to support his aim for simplicity and reliability rather than for the highest possible efficiency through complexity and high maintenance costs. There were, however, some discrepancies between the statements made by Mr. Main and their practice.

With a somewhat modified heat cycle, the performance of the machinery could be considerably improved. The most advantageous cycle (Fig. 16) included a bleed point on the L.P. turbine to feed the evaporator and an L.P. preheater. This was standard practice for De Laval turbine installations, whereby the following gains were made:

a) improved cycle efficiency without serious complications and for a small investment
b) reduced last stage area and consequently decreased turbine first cost
c) possibility of dumping surplus steam from the auxiliary exhaust range into the L.P. heater, instead of dumping to the main condenser
d) increased steam flow to the H.P. turbine and consequently a rise in turbine efficiency due to decreased ventilation losses.

The optimized cycle employed the L.P. heater at condensate outlet temperature of about 185 deg. F. and the de-aerator at about 280 deg. F., thereby also avoiding the more expensive cast iron construction of the economizer. It should be noted that this de-aerator temperature required a bleed point pressure in the crossover pipe, which could not easily be reached with a double-flow L.P. turbine.

Another difference was that they used a bleed on the H.P. turbine to feed the L.P. steam generator. The de-superheater was arranged internally in the boiler. This had given no difficulties on existing installations. As far as the efficiency of the main turbines was concerned, they had calculated with the same non-bleed steam rate as had been given for the Pametrada unit shown. The non-condensing turbogenerators specified by Mr. Main seemed, however, to have too conservative steam rate figures. The De Laval standard back pressure units had a steam rate roughly 10 per cent lower and the condensing types about 13·5 per cent lower. The same applied to the De Laval boiler feed pumps for which they had specified a 14 per cent better overall efficiency.

They had chosen two-speed electric motor drive for the main circulating pump. No difficulties, of the kind mentioned by Mr. Main, had occurred in engine rooms equipped with this system, not even during full astern manœuvring conditions, which could be maintained without limitation in time.

For the main boiler plant they assumed a boiler efficiency of 88 per cent based on high heating value. Any reasonable efficiency could be reached, depending on the degree of investment, but for this type of cycle, 88 per cent was considered the best choice.

Summarizing their calculations, they had found that the fuel rate at a sea temperature for the cooling water of 60 deg. F. was 0·503lb./s.h.p./hr. and under tropical conditions, with the cooling water temperature at 85 deg. F., 0·525lb./s.h.p./hr. The difference between their figure and Mr. Main's amounts to about four per cent, which could be referred to improved boiler efficiency, about one per cent, improved efficiency on major auxiliaries about 1·5 per cent and improved cycle arrangement 1·5 per cent.

In order to show the application of bleed points, on both the H.P. and L.P. turbines, sections through a De Laval standard H.P. and L.P. turbine for 22,000 s.h.p. were displayed in Figs. 17 and 18.

The H.P. turbine was for the single casing type which facilitates alignment. The turbine had nine stages and a speed of 5,000 r.p.m. which contributed to low manufacturing costs and high efficiency. There was no astern element in the H.P. cylinder.

The L.P. turbine rotating at 3,045 r.p.m. was of the single flow type with seven stages. The astern turbine at the forward end consisted of one Curtis plus one single stage, which would develop 50 per cent of maximum ahead power. If higher astern output was required, the astern turbine could be changed to two Curtis wheels.

He did not agree with Mr. Main about nozzle sub-groupings. One made a considerable loss, by using throttling at part load operation. It was stated in the paper that a loss of 5 per cent existed, when operating at 75 per cent load,

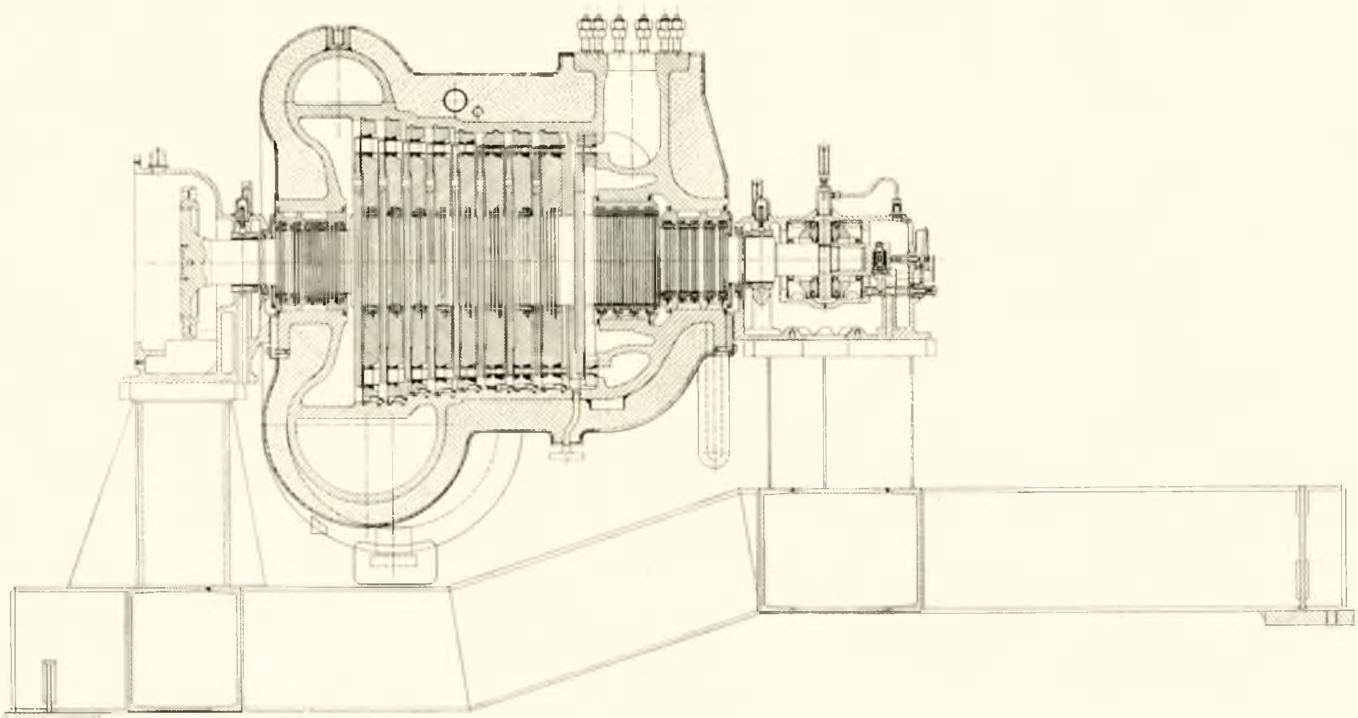

FIG. 17—*Longitudinal section through H.P. turbine in De Laval 20,000 s.h.p. main propulsion unit*

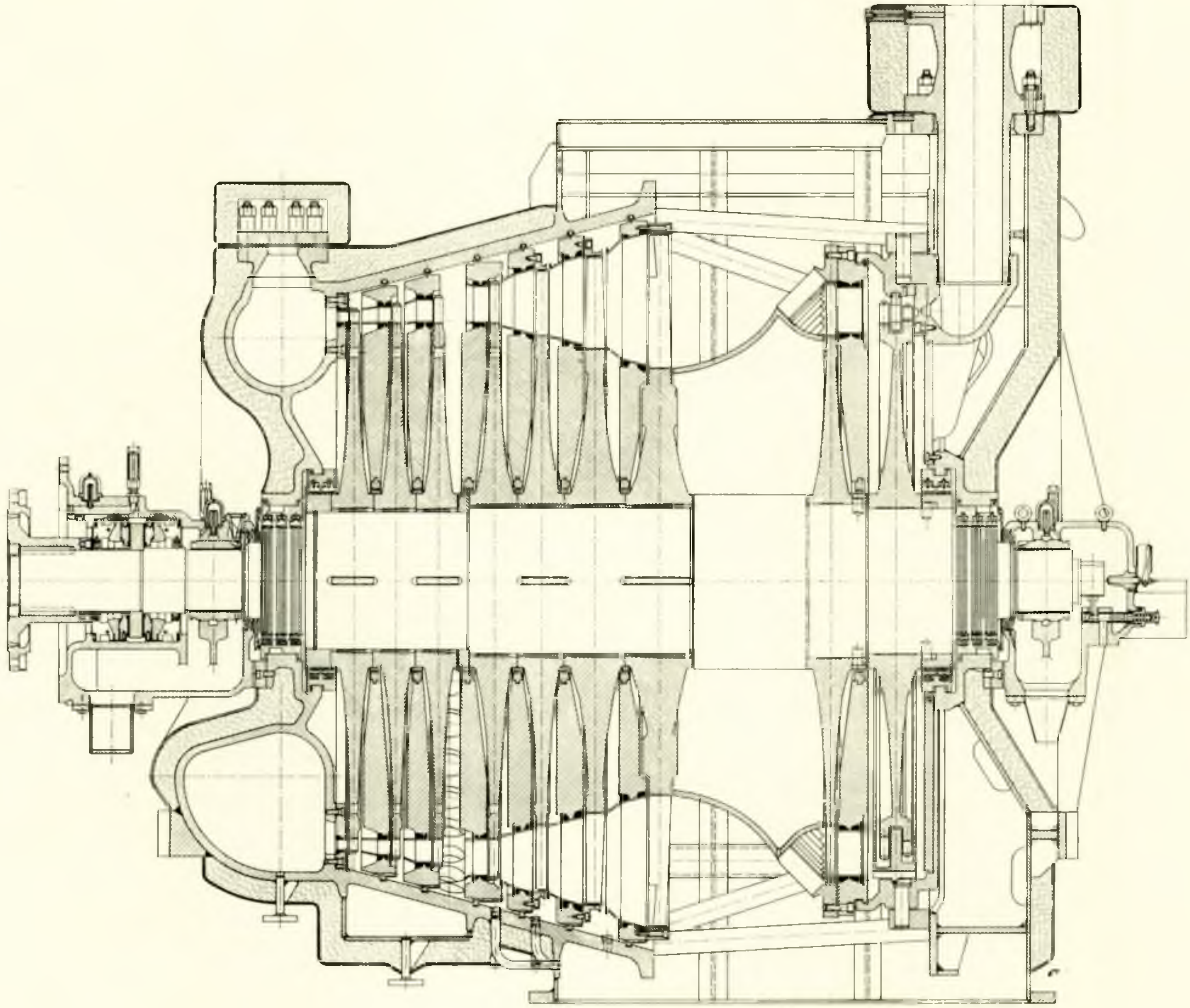

FIG. 18—*Longitudinal section through L.P. turbine in De Laval 20,000 s.h.p. main propulsion unit*

by throttling only and 3 per cent by having a nozzle sub-group designed for 75 per cent load. On De Laval standard turbines, which normally were designed for operation with peak efficiency in the range of 85-95 per cent load, the increase in specific steam rate was only 1 per cent, when operating at 75 per cent load with a corresponding nozzle group. Nozzle sub-grouping was a very simple way, at least with single casing H.P. turbines, to improve economy at part load operation, but if one absolutely wanted to avoid sub-grouping, a better method might be to regulate the boiler pressure and eliminate throttling losses on the main unit in that way.

MR. A. T. MACKENZIE said that they had heard a great deal about condensing and back pressure turbo-alternators, but practically everyone had agreed that the back pressure turbine was the winner. Much of this depended, of course, on the basic assumptions that were made. The first basic assumption was as to the number of days at full power steaming, a second was fuel cost and a third was depreciation allowance. The assumption would not necessarily be the same for all owners, or for ships on other trading routes. It was interesting to examine the effect of varying two of these assumptions, assuming fuel costs to remain constant.

The figure of 320 days per annum full power steaming for the Middle East/N.W. Europe service would seem to be somewhat optimistic, unless the assessment of effectiveness was made over the first seven or eight years of the ship's life. If the assessment was made over a longer period covering, say, four special surveys, a figure of 300 days per annum might be more realistic.

It should be borne in mind that any increase in the number of days spent at reduced power steaming, manœuvring, canal passages, port time, etc., would reflect adversely on the fuel rate where back pressure alternators were fitted.

It was, of course, entirely up to the owner concerned to decide how he wished to evaluate his fuel savings against depreciation allowance, but he would like to attempt to show that a straightforward equation of fuel savings against depreciation did not always provide the correct answer.

He had often been asked, during general conversation, why it was that some tanker owners selected the simplest possible installation, while others appeared to favour systems of the utmost complexity. The answer, he would submit, was dependent upon the value placed upon initial cost and fuel savings and even more largely upon the depreciation period allowed.

In designing the machinery installation for a large and valuable vessel, the assessor would undoubtedly consider many dozens of alternative heat cycles. For the present discussion however he would like to discuss two alternative designs, the one being a "simple" cycle, as adopted by Mr. Main, and the other a "complicated" cycle, e.g. employing three bleed points, condensing turbo-alternators and a split economizer feed heating system. Both cycles were designed upon 600lb./sq. in. gauge, 900 deg. F. at the superheater outlet, for the eminently practical reasons suggested by the author. In the case of the simple cycle the heat balance was virtually identical with that in Fig. 2 with the exception that the designed vacuum condition was based on a sea temperature of 75 deg. F. (the statistical average of sea temperatures on the Middle East/N.W. Europe service) and more liberal allowances were made for losses and evaporator make-up. The fuel rate evaluated at 0·525lb./s.h.p./hr. (cf. 0·523 for Fig. 2). Referring to Table I, however, one found that the fuel rate for Fig. 2 evaluated at the average overall annual figure of 0·544lb./s.h.p./hr., when considered over the course of the whole year.

The complicated system, based on identical steam inlet and condenser conditions, but with condensing turbo-alternators, instead of back pressure turbo-alternators, showed an average annual fuel rate of 0·503lb./s.h.p./hr. This represented an annual saving of £18,000 per annum (if one used the 320 days per annum full power steaming quoted by the author) for an additional capital expenditure of £30,000 on a builders' estimate.

Whatever period of amortization, or whatever other parameters were chosen, it still appeared that the additional capital cost of the "complicated" installations as compared with Fig. 2 was recovered in 1·7 years and that seemed to be good business.

It had always appeared to him, however, that the final economic criterion, in assessing whether a change in machinery design was justified, was the overall effect upon the freight rate of the projected ship. In this connexion, it might be stated that an increase of £30,000 or thereabouts was barely noticeable, when referred to the cost of the ship as a whole, but a reduction of 10 per cent, in the daily fuel consumption, could produce a substantial fall in the daily operating costs and hence, in the cost per ton of oil carried.

Any increase in maintenance requirements and operational complexity which were involved in the adoption of a more elaborate system should also be taken into account. He would suggest that this was an aspect which need not concern them seriously, in the comparison between the two systems considered, provided that the turbine design selected was able to provide the necessary bleed points without mechanical complication which adversely affected reliability. The rest of the elaboration consisted solely of heat exchangers and pipework which, given satisfactory standards of material and workmanship, should not cause a substantial increase in the maintenance commitment. Operationally, such a system was largely self-regulating and experience had proved that it did not put any undue burden on ships' engineers in this respect.

Mr. A. C. Hutchinson, B.Sc.(Eng.) (Member) said that he proposed to confine his remarks to a single detail of the paper. This was the choice of pressure. Everyone looking for higher efficiency took it for granted that temperature increases were a good thing, but not pressure increases. Just over two years ago he had been in exactly the same situation as he was in now, taking part in a discussion, before the Institute, of a very similar paper by the late Commander Bonny* on an investigation of the optimum machinery for a large steam turbine tanker.

During this discussion, where he had found himself very much in the minority, he had used arguments taken from another Institute paper: "The Use of High Pressure Steam in Marine Installations", by Mr. Norton†, to argue, rather contrary to Mr. Norton's opinion he thought, that there was much more scope for cycle improvement by increasing pressure than there was by increasing temperature. This was all printed in the Transactions and he would not waste time by going over it again.

Since then, in thinking about the case against increasing pressure, he had come to the conclusion that there were two kinds of argument. There were "king" arguments, real solid objections to increase of pressure. There were also unreal, phantom arguments against increased pressure which dwindled or vanished when one looked them squarely in the eye.

One of the real, or "king" arguments was boiler water chemistry and because of this he would be grateful for a more detailed statement from the author, as to the mechanism whereby internal boiler deposits became much more prevalent at higher pressures. Another "king" objection he believed to be the bending stresses in turbine blades, but he did not think that these were beyond the competent steam turbine designer.

A good phantom argument was centred round pipe flanges. The author believed in laying this phantom or spectre by welding up joints. He himself suspected that some of the trouble derived from British Standard Specification No. 10, for flanged pipe joints.

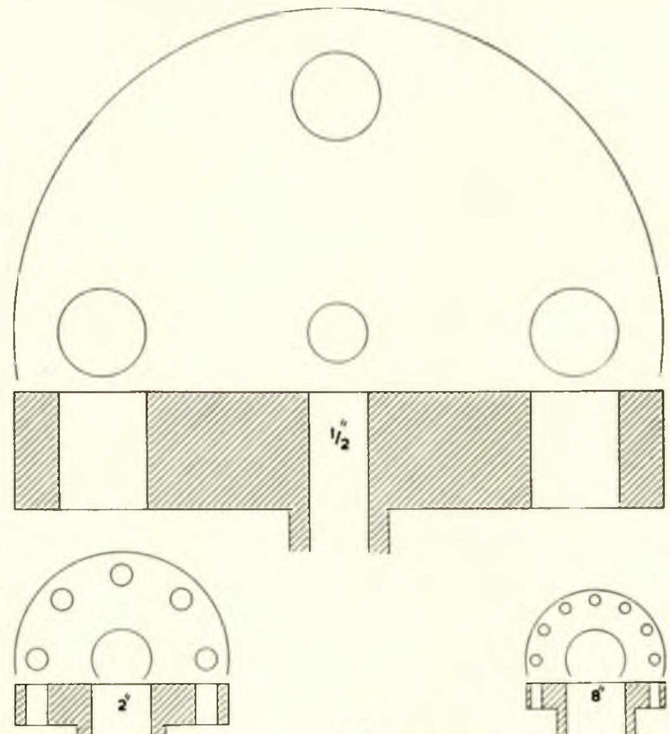

Fig. 19—*Steam pipe flanges from Table T, B.S.10, drawn with a common bore diameter to show their dissimilarity of design*

Fig. 19 showed three flange designs taken from Table T of that specification, all re-drawn to have a common bore diameter. Table T covered steam pressures up to 1,400lb./sq. in. In central station machinery with very large flows, even with high steam pressures, the pipe diameters were large and one could use an efficient flange design like the 8in. flange at the bottom right hand corner. The smaller the steam flow, however, and the smaller the pipe under consideration, the more inefficient the flange design became and the dimensional advantages of using dense steam were concealed by grosser and grosser flange designs. The extreme of absurdity was reached with the ½in. flange which had to be used for gauge pipe connexions.

Another example of the conspiracy to deprive the engineering world of benefit from high steam pressure was shown in Fig. 20. This was a picture of himself sitting on top of his employer's new 1,500lb./sq. in. test boiler. His left hand was on a steel casting which weighed 14 cwt. and his right hand was

* Bonny, A. D. 1958. "An Investigation into the Optimum Machinery Installation for a Large Steam Turbine Tanker", Trans. I.Mar.E., Vol. 70, p. 361.

† Norton, E. 1958. "The Use of High Pressure Steam in Marine Installations", Trans.I.Mar.E., Vol. 70, p. 261.

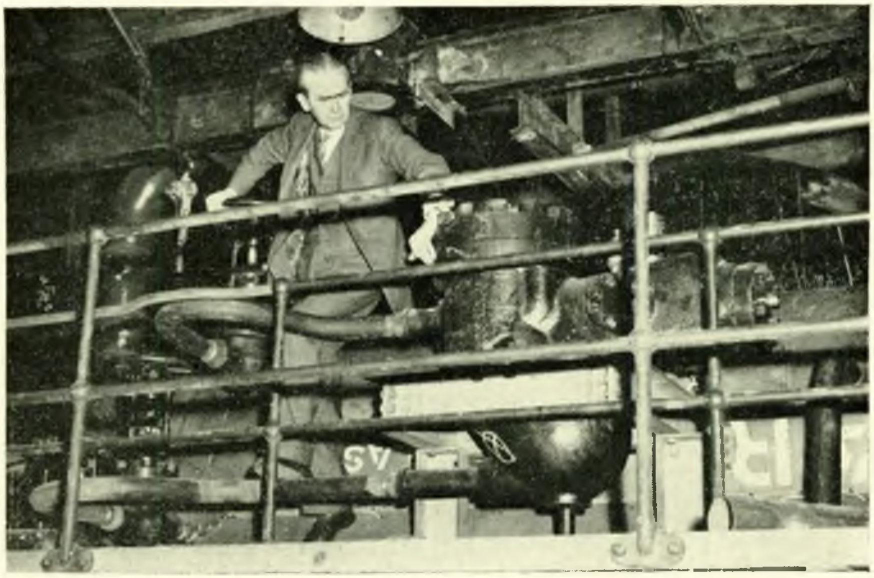

FIG. 20—*An example to illustrate the large size of conventional steam fittings*

on the wheel of a 2in. stop valve. The casting was the high and low water-level alarm fitted to the boiler drum. Its function was to blow two whistles. Together with the two stop valves and the connecting pipes, the total assembly represented nearly two tons of steel, or an average of approximately one ton per whistle!

It seemed to him that, having regard to this kind of existing steam practice, the author was fully justified in sticking at 600lb./sq. in. He would only like to suggest that if advancing steam conditions were being considered, what might be called "the Baker line" of temperature increases had its limitations and there was really a strong case for re-examining steam pressure increase.

MR. J. NEUMANN, B.Sc.(Eng.) (Associate Member) complimented the author upon advocating a properly designed circulating water (C.W.) inlet, which could be used to provide some of the head, necessary for overcoming the resistance of the C.W. system. As predicted in the model experiments of Hewins and Reilly* and confirmed by a number of full scale ship trials, the effective pressure head produced by such an inlet varied approximately with the square of ship speed. In his experience it was generally considered that, with careful design, this pressure head could be produced without detrimental effect on the ship resistance and power required to propel the ship through water. The combination of such C.W. inlets and turbine-driven axial flow circulating pumps had been used successfully for many years in ships of the Royal Navy.

Similarly, he agreed that the provision of a lip to the forward edge of the overboard discharge would reduce the C.W. system resistance, and this had also been confirmed in full scale trials. The use of such a lip could be recommended to effect a cure in a completed ship which, for one reason or another, was seriously short of C.W. and hence suffered loss of power due to a low vacuum.

However, the fitting of a lip resulted in an increase of the ship's resistance, depending on the height of the lip, and therefore when the use of a lipped discharge was considered as a design measure and not as a cure, the additional ship resistance must be weighed against any saving in circulating pump consumption. Present indications were that the saving in pump consumption would need to be paid for several times over, in extra steam to the main engines, to compensate for the increased ship resistance caused by the lip.

* Hewins, E. F. and Reilly, J. R. 1940. "Condenser Scoop Design", Trans.S.N.A.M.E., Vol. 48, p. 277.

He noted that a butterfly sea valve was shown in the main C.W. inlet, but a sluice valve in the outlet, and he would be interested to learn the author's reason for this.

In the paper it was implied that the main gearing had been situated as far aft as possible within the narrowing sections towards the stern. Since the boilers were also aft, it appeared at first sight that the author had found the answer to the question—boilers aft or main engines aft—by arranging both these components aft. However, examination of the small-scale plans suggested to him that the main gearing was in fact forced forward by the deep frames, to an extent sufficient to allow for accommodation of the boilers aft. It would be of interest to learn whether this surmise was correct, or have some alternative explanation from the author.

Mr. Neumann noted that the handing of the boilers was such that the burners were conveniently positioned near amidships. However, this resulted in the uptakes being outboard, and he presumed that twin funnels were envisaged. Did this location of the uptakes result in any penalty to the arrangement of accommodation, as compared with a conventional common uptake design?

From the accent on good performance at reduced ship speeds, corresponding perhaps with 70-75 per cent of the design full power, it appeared that the tanker joined a number of other types of ships, including passenger liners, in which part load performance was of consequence. On this basis, there seemed little doubt that an increasing number of ships would take advantage of the improved part load performance offered by nozzle group control of main engines. In naval vessels, emphasis on good performance at low powers often led, in addition, to the provision of special auxiliaries for use at part loads, but he was not suggesting that such arrangements would be required in merchant ships, where the lower end of the power range of interest was unlikely to fall below, say, half power.

With regard to the flow diagram, Fig. 1, he noted that steam for bunker heating and accommodation heating was taken from the exhaust range, at about 10lb./sq. in. gauge with the drains returned, via filters, to the atmospheric drain tank. Was it an innovation to employ such low pressure steam for these duties, or had the author found such pressures to be adequate in practice?

The steam supply to the reciprocating pumps was shown from the de-superheated steam line, with the exhaust from these units led via the atmospheric condenser to the atmospheric drain tank. It would be of interest to learn whether this was a normally accepted arrangement and what special measures, if any, had been taken in the design to avoid contamination of the exhaust steam and of the feed system by pump lubricant.

MR. A. NORRIS (Member) said that after a quick perusal of the paper he was somewhat puzzled by Fig. 5. The author stated that operating costs were increased by £880 per annum, as a result of the gas exit temperature being above the optimum point, but a saving of 21 tons in economizer installation weight was effected. The curve marked "Additional fuel cost p.a." in Fig. 5 showed an increase in fuel cost of approximately £2,200 p.a. which, on the basis of £6/ton given on page 277, would seem to indicate that over 360 tons of additional fuel were burnt. The additional weight of bunkers required should be offset against the 21 tons economizer weight saving claimed, when assessing the extra weight of cargo which could be carried on a loaded voyage. While much depended on the number of voyages per annum, on the data given it was difficult to see how a significant weight saving—i.e. an increase in cargo carrying potential—could be claimed.

On page 288 it was stated that vertical type cargo pumps allowed a reduction in machinery space length. He had looked at such claims on several occasions and, bearing in mind the higher cost of vertical pumps, could not see any overall advantage in using this type. In addition, at least one builder had examined it on behalf of his company and could see very little difference. The length required for vertical pumps was shown on Figs. 12 and 13 of the paper, but the two figures seemed to be drawn to different scales and if one took, as the focal positions, the after bulkheads of the wing bunkers, this would be seen. The claimed reduction depended on where one measured the engine room length. If it was measured at the flat level—and it was only here that one was looking for space in turbine ships these days—the saving was not so apparent. As the author had said in the paper, it was the distance between the two extreme bulkheads that was the criterion, but it could be seen from Fig. 13 that a substantial length was taken up from what was apparently the after bulkhead of the pump room above this intermediate level, to the similar bulkhead located further aft below this level. He did not think any more space was taken up by the horizontal cargo pumps, bearing in mind that whatever type was used allowance had to be made for piping. It might well be a matter of different practice obtaining in different companies. His own company had used horizontal cargo pumps with great success for many years. Perhaps this influenced one's thinking accordingly.

Another point that had rather interested him was what appeared to be an arrangement of the steam piping, shown in Fig. 13 and in the elevation on page 291, in which the piping ran down through the boiler room floor. It made for a nice arrangement, but the possibility of having bilge water swilling about in the boiler room and possibly getting onto the steam pipes had to be considered. There were many ways of overcoming it. One could build a box higher than the boiler room flat and so on. It would be interesting to learn which of the alternatives had been thought most suitable.

Reference was made on page 291, in discussing future possibilities, to the provision of one full duty Diesel generating set and one full duty turbo set as an alternative to the present installation. With this arrangement, an adequate amount of this Diesel fuel would have to be carried at all times, for use in the event of a breakdown of the turbo-set, and this amount would be included in the usual safe margin of bunkers allowed for a passage. Diesel fuel was rather more expensive than boiler fuel and if any passage should be extended, to an extent where it was necessary to burn the Diesel fuel under the boilers, it would be costly. There would, presumably, also be the complication of a purifier and a separate Diesel oil system.

MR. C. R. KIRSCHBAUM said that he had enjoyed the paper greatly. Such papers were very useful in helping to decide the choice and shape of machinery installations for the future. Upon studying the preprint he had found that Mr. Main's design did not differ greatly from his own. Therefore he had only some remarks of minor importance to make. He would like, first, to ask the author if the choice of two electrically-driven circulating pumps, each of 50 per cent of the maximum design capacity, would not give a better solution. The efficiencies of these electrically-driven pumps were better than the efficiency of the turbo-circulating pump. Besides, the lower quantity of exhaust steam made it more probable that one would use an L.P. feedheater, which improved again the total efficiency of the cycle. The price of two smaller electrically-driven pumps would be equal to or lower than that of a turbo circulator.

Secondly, he would agree with the author in his choice of a one-speed motor for the boiler fan. However, much attention had to be spent on the tightness of the vanes, as sometimes these leaked as much as 30 per cent of the maximum air quantity. In fact, this caused sometimes, a bad air-fuel ratio at low loads.

A drawback of the one-speed motor driven boiler fan was that the auxiliary Diesel generator had to take this higher electric load at starting up condition, or when cases of blackout occurred. As long as the Diesel generator could take this higher load, he could not see much against the proposition.

He felt that the H.P. astern turbine was not worth the larger investment, the loss in efficiency, and the complication that it would certainly produce. He believed that an astern turbine in the L.P. casing would give the normal amount of astern power, necessary for this type of ship.

He would like very much to hear the opinion of the author on the necessity for more automation and simplification of machinery installation, in ships that would be built in the near future. It was the duty of the designer to have his designs ready for the day, when the quantity and quality of marine engineers was not so high as at present.

Items which could be improved upon included: the elimination of the cleaning of burners and evaporators; simplification of the water washing of the boiler; a 100 per cent automatic boiler control and a simplification at the main switch board which made it fully "foolproof".

Correspondence

MR. A. BELL, B.Sc.(Mech. Eng.) (Associate Member) in a written contribution, said that to utilize the exhaust from the back pressure turbogenerators Mr. Main had adopted steam air heaters. This had led him to assume that this choice was limited to economizers for flue gas heat recuperation, hence his choice of final feed water temperature had been severely restricted.

But the use of steam air heaters did not necessarily preclude the use of gas air heaters. Even with the high outlet temperature from the steam air heater of 230 deg. F., a suitable gas air heater could be designed, to operate in series with the steam air heater and a small economizer, at an annual charge lower—for gas outlet temperatures above 318 deg. F.—than that for Mr. Main's economizer only scheme. Fig. 21 showed, superimposed on Mr. Main's Fig. 5, the annual charge for a gas air heater and economizer designed for an air inlet

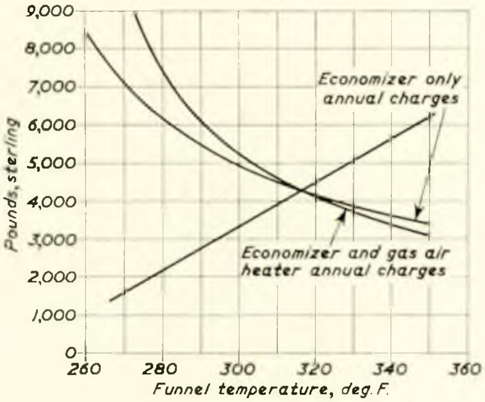

FIG. 21—*Comparison of annual charges: economizer only (from Fig. 5) and economizer and gas air heater (with steam air preheat to 230 deg. F.)*

temperature to the gas air heater of 230 deg. F. and a feed temperature of 240 deg. F. These charges included the appropriate allowance for the increased fan capital and running costs due to the higher draught loss.

A lower steam air preheat, before the gas air heater, would increase the advantage of this arrangement, but would necessitate alterations in the steam cycle and would make direct comparisons with Mr. Main's installation difficult.

Several speakers, during the discussion, raised the question of economizer corrosion with a feed temperature of 240 deg. F. The controlling factor for both economizer and air heater corrosion was the minimum metal temperature. With a feed temperature of 240 deg. F., the minimum economizer metal temperature would not be much higher than 250 deg. F., whereas for the steam air heater/gas air heater combination, with steam air preheat to 230 deg. F. and a gas outlet temperature of 335 deg. F., the minimum metal temperature would be 290 deg. F.

With a higher feed temperature, an all-steel economizer could be used with no risk of economizer corrosion, the change to all-steel construction offsetting the increase in economizer cost due to the higher feed temperature.

An increase, from 240 deg. F. feed temperature to 360 deg. F., would give a reduction in fuel consumption of approximately two per cent*. Thus, a reduction of two per cent in the fuel consumption could be obtained, for little more than the cost of the additional feed heaters and with less risk of corrosion.

MR. G. B. HALLEY, M.B.E., B.Sc. (Member) wrote to say that with such a large proportion of time spent under way at sea it was evident from Table I that the back pressure generator was more economical than the self-condensing set, despite the fuel saving with the latter. Incidentally did the fuel saving given in the table take into account harbour working, when the advantage lay heavily in favour of the self-condensing set?

The problem with back pressure sets was the embarrassing abundance of exhaust steam. This was an especial drawback when the feed rate was reduced, for example in port or when steaming at reduced speed and it made the addition of an L.P. bled point of dubious value, as illustrated in Fig. 3.

It would be a great advantage if the auxiliary exhaust could be cut down and the bled steam rate increased. One obvious way to do this would be to increase the efficiency of the auxiliaries and it was thought that there was scope here. Another way, which he believed the author had already investigated, would be to drive the main circulator from the auxiliary exhaust range.

It would be interesting to see what the result would be of increasing the auxiliary exhaust pressure, to give a feed temperature of 280 deg. F. instead of 240 deg. F. It was considered likely that this would increase the cycle efficiency and increase the bled steam requirements, thus making the introduction of the L.P. heater more attractive, whilst the economizer could be entirely of the stud tube type. Another advantage would be the greater permissible fall in feed temperature, before the onset of severe economizer corrosion and deposit troubles, although admittedly this was not likely to occur with the back pressure generator cycle. Perhaps it would be found that, with the increased feed temperature, the self condensing generator could earn its place.

It was often argued that twin two-speed M.D. main circulators were more economical than the turbo circulator, favoured by the author. It was, therefore, interesting to read the convincing arguments put forward in the paper in defence of the latter.

Only one atmospheric drain tank extraction pump was fitted, the main extraction pumps acting as standby. In harbour, discharging cargo, one main extraction pump would be on its normal duty and the other would be acting as the drain tank extraction pump, because of the large pumping load. There would thus be no standby for either duty, under this condition.

The S.S. generator load under normal steaming had been practically eliminated, but would it not be more reliable to use this for supplying steam for oil fuel heating and reciprocating auxiliaries, in view of the risks otherwise involved in contamination of the main feed?

Had the author considered fitting submersible pumps for the lubricating oil service?

It was difficult to see why a steam driven instead of an M.D. pump was selected for the standby fuel transfer pump.

This was a most interesting and informative paper and the author was to be congratulated on the way he had covered his subject so concisely and yet so fully.

* Bonny, A. D. "Modern Marine Steam Turbine Feed Systems", Trans.N.E.C.Inst.E.S., Vol. 73, Part 5.

Author's Reply

Mr. Main, replying, said that in the interests of brevity he would confine his remarks to an attempt to answer the salient points raised at this stage. These appeared to centre on the choice of feed temperature selected and the design of the turbine units.

If the cycle had been designed for 280 deg. F. feed temperature the steam consumption of the auxiliary turbines would have been increased due to the higher back pressure against which they were required to operate. On the other hand the absorption of feed heating steam would be higher still and would justify the introduction of an L.P. heater. In this case the increased offtake of bleed steam would of course have the effect of requiring a higher designed H.P. exhaust pressure to maintain power balance on both rotors. The improvement in cycle efficiency using an L.P. heater was likely to prove marginal however.

It had to be remembered that although a higher rate of auxiliary turbine throttle steam incurred no immediate loss provided all the exhaust was absorbed in the feed heaters, the recycling of this greater auxiliary heat quantity through the boiler plant resulted in additional losses proportional to the boiler efficiency.

On the question of the design of the propulsion turbine units he would stand by what he had said concerning the choice made. At the time when the design had been selected the steam rate according to his information had been comparable with, or better than, that given by other leading designers. The alleged inferiority of the Pametrada design insofar as reliability was concerned had not, in his company's experience, been proven.

In regard to generator and feed pump turbines, again he would beg that comparison was not being made between 1961 models and those designed four or perhaps more years back. There obviously had been a tightening up and improvement in auxiliary unit designs recently offered which had in some cases brought the laggards up into line and made the position far more competitive and he would attempt to answer more fully in writing.

Mr. Main in his written reply thought that in reviewing the contributions made to the paper it seemed that the feed temperature selection, bearing in mind its fundamental influence on plant cost and cycle efficiency, was the most important criticism. There were two major repercussions to the adoption of 280 deg. F. feed temperature as opposed to 240 deg. F.

The first one concerned the cost of the economizer plant. He had checked with the economizer manufacturers and the following figures had emerged: assuming that the boiler efficiency (i.e. the funnel temperature) and the F.D. fan power (i.e. draught loss) were to remain unchanged, the "all-steel" economizer plant for 280 deg. F. feed inlet temperature would have 48 per cent more surface, 23 tons more weight, and would cost 8 per cent (i.e. some £2,100) more than the part cast iron gilled/part steel economizer plant which was fitted.

The second factor was the influence of the higher feed temperature upon the cycle efficiency and this was best discussed on the basis of a further heat balance diagram. Fig. 22 showed Fig. 3 modified to operate with 40lb./sq. in. gauge back pressure on the auxiliary turbines. Fig. 16 used the same L.P. feed heater duty rating as Fig. 3 in the paper. The specific steam rate of the generator turbine had been increased by 16 per cent and the steam rates of the feed pump and circulator pump turbines had been raised by 20 per cent. These figures were arrived at by offsetting the increase in steam rate due to the lower available heat drop per pound by a reasonable increase in the adiabatic efficiency of the turbines deriving from the fact that from a turbine design point of view the lower heat drop was more manageable.

It was appreciated that the L.P. bleed quantity for 60 deg. F. sea condition could have been increased in Fig. 22

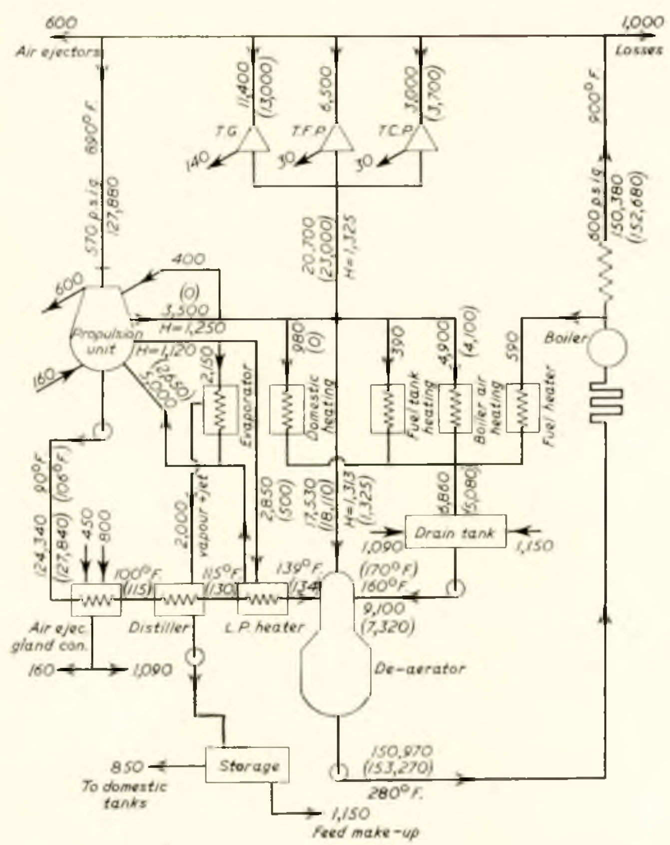

FIG. 22—Heat *balance with L.P. feed heater added and with feed temperature of 280 deg. F.*

21,280 s.h.p. (20,590)
Propulsion turbine non-bleed steam rate 5·85lb./s.h.p./hr. (6·17)
Boiler efficiency 87 per cent Vacuum 28·5in. Hg (27·6)
Turbogenerator load 450 kW (530)
Fuel consumption:
11,040lb./hr. = 0·519lb./s.h.p./hr. = 118·3 ton/24hr.
(11,200) (0·545) (120)

provided that a corresponding reduction was made to the heat quantity taken from the higher stage bleed point. It would be seen however, that tropical operation reduced the higher stage bleed to zero and the L.P. bleed to a mere 500lb./hr. and the problem thus involved optimization of the L.P. heater capacity to operate between these limits.

Fig. 22 showed that its average of temperate and tropical operating conditions called for about 11,120lb./hr. fuel consumption compared with 11,195lb./hr. for the cycle fitted and this amounted to a possible annual saving of £1,540. Offsetting the £350, capital charges for the L.P. bleed arrangements and £300 capital charges for the additional economizer surface reduced the saving to £890 per annum, and maintenance costs would inevitably further cut this figure down.

The author agreed that the De Laval system described by Mr. von Feilitzen enabled the dumping of auxiliary exhaust steam to be delayed to the limit of the heat absorbing capacity of the feed heaters and that given better efficiencies in the turbine driven auxiliaries a larger proportion of L.P. feed heating became possible. There was admitted advantage with the 40lb./sq. in. gauge back pressure system in that this higher pressure exhaust steam would give better performance on fuel and domestic heating services. On this point he would answer Mr. Neumann by saying that 15lb./sq. in. gauge exhaust steam was being used here for the first time in his experience for these duties and whilst it was felt that 15lb./sq. in. gauge would be successful, 40lb./sq. in. gauge would obviously provide greater margin against inadequency.

Commander Tyrrell's observations concerning the propulsion turbine design had, for the most part, been answered already by Mr. Ryall. The author was not aware of any basic inability of built up diaphragms to take somewhat higher steam temperature. Although welded diaphragm manufacturing techniques had hardly been sufficiently perfected to justify their incorporation in this design, there seemed no doubt that they would become widely adopted in future. The tendency for casing glands to close in on their horns was by no means peculiar to those fitted in double casing turbines. This was almost always the result of quenching of the bore of the gland shell by wet sealing steam. The basic trouble could only be avoided by a properly designed gland seal system. Further refuge could be taken from the danger by adopting spring back gland labyrinths.

Dr. Silver's contribution had been a most interesting addition to the paper. He was correct in suggesting that an increase of 4 per cent in feed pump steam consumption would be justified if the capital cost could be reduced by not less than £1,000. If, as in this ship, two identical turbo feed pump units were fitted, each unit needed only to be £500 cheaper to make the additional 4 per cent steam consumption acceptable. Information now available on the efficiency of the recently developed water lubricated two stage pumps made by Dr. Silver's firm suggested that they could more than satisfy that requirement, and it was very good to learn that such improvement had been achieved.

The author agreed with Professor Lameris that steam conditions must continue to be advanced. Too much conservatism could only stagnate development. It was impossible however to obtain quantitative information from which the optimum steam condition for a ship could be derived. One could proceed on personal judgment and this would frequently be faulty. An assessment could be formed of the manufacturers' and shipyards' ability to design, construct, and install a higher pressure system and this assessment could be unduly pessimistic. The fact remained that the cost of any failure continued to have increasingly serious commercial consequence as the pace and complexity of the shipping business increased.

Professor Lameris had suggested that the exhaust range pressure should be allowed to rise when the reduction in load on the engine tended to result in a build-up of exhaust steam surplus to that which could be absorbed in the feed heater, etc. To a limited extent this was allowed in the system described in the paper by setting the exhaust range pressure controller so that the dump valve did not open until a pressure of about 18lb./sq. in. gauge had been reached. The difference in heat intake at the de-aerator—Fig. 22 (280 deg. F. feed) and Fig. 3 (240 deg. F. feed) is about 44 per cent. The author could not see however, that with the main condensate flow reduced to $\frac{1}{3}$, all dumping of exhaust would be avoided by allowing the exhaust range pressure to rise to about 40lb./sq. in. gauge. The combined auxiliary turbine exhaust already provided somewhat more than the de-aerator heat intake at full power in the cycle fitted. The exhaust heat quantity from these turbines would increase by about 22 per cent when the back pressure was raised from 15 to 40lb./sq. in. gauge. If the de-aerator heat intake was increased by about 44 per cent due to the higher temperature rise to 280 deg. F. and then reduced by about 65 per cent due to a reduction of this order in the condensate flow, there would be an exhaust surplus of about 7,000lb. of steam per hour. In regard to Professor Lameris' question on a small gas air heater presumably he meant a gas air heater in series with, and taking preheated air from, the steam heater. A difficulty here was the complication in ductwork. It was likely also that the heater would be rather large in view of the relatively low temperature difference available. The F.D. fan power would be increased and additional soot blower equipment and maintenance involved. The total allowable capital cost of these additions could not afford to be more than about £7,000 per boiler to support the figures suggested by Professor Lameris.

The author had been grateful for an opportunity recently to see the Werkspoor return flow oil fuel burners and agreed they were attractive. If the steam assisted burner did not contribute directly to a reduction of the boiler fouling problem then a lot of the advantage at present hoped for was lost. SO_3 corrosion was not in the author's opinion necessarily more likely with steam assisted atomization.

It was indeed a step toward the eventual complete automation of the boiler firing process if there was no need to withdraw burners not in use at reduced loads. This required some provision for cooling the idle burner tip as protection against the radiant heat of the furnace. The Werkspoor return flow burner appeared to provide an admirable solution here although the cooling flow in this instance was the return flow and at maximum output on the remaining burners this was necessarily at a minimum.

Regarding the single turbogenerator in future ships with an internal combustion drive on the standby set, the author agreed that a simple open cycle gas turbine unit was attractive and if the cost was acceptable he would favour it.

The figure of 0·01 c.c./litre oxygen content for the de-aerator performance was the minimum performance guaranteed by the manufacturers and it was true that this would be high for modern practice. In fact it was anticipated that the design employed would achieve much higher de-aeration in service and would certainly be not more than 0·005 c.c./litre.

The author welcomed Mr. Hutchings' remarks although in regard to the use of 280 deg. F. feed temperature he would refer again to the figures which had been carefully estimated by the economizer manufacturers. The washing of boiler fuels to remove dissolved sodium was being tried in one of the vessels of the author's company. This installation had only been working for a few months however and no conclusive results were available on its performance. It was absolutely essential that every possible effort was made to eliminate the boiler fouling problem and the work on fuel washing, and indeed on any other promising avenue of approach, was likely to become intensified.

The author was grateful for Mr. Hutchings' correction of the figure given in the paper for F.D. fan capacity margin. This should indeed have been 104 per cent and not 140 per cent.

Negligible air heater fouling, even in the close pitched fin design used, had been experienced in existing ships. Mr. Hutchings had brought out very clearly the argument in favour of the steam assisted oil burner and the savings consequent to enabling designs to be based on 14 per cent Co_2 combustion.

As far as combustion control equipment was concerned it was the author's view that no matter how certain the eventual advent of full automation, the maintenance and servicing facilities for such complete equipment on board a tanker were not established today. In this general respect the design described in the paper had made use of automatic equipment as far as was supportable in the existing circumstances. The introduction of complete automation need not be regarded as a sudden "break through". There was already more of it than many people perhaps appreciated.

Whilst it would be very wrong to obstruct the introduction of automation as a principle it was difficult to imagine a more painful, and less rewarding exercise than an attempt to automate equipment and systems which were basically unsuited or insufficiently developed. The automatic controls themselves had to be sufficiently rugged for sea service and their installation at the shipyard very carefully carried out if the need for a complement of instrument technicians aboard equally numerous to the present complement of engineers was to be avoided.

Mr. Ryall presumably would not suffer unduly from complacency in apparently having retained one satisfied customer. Perhaps it had been certain of the Association's manufacturing retailers who had been largely responsible for the recent popularity of the competing brands of propulsion turbines.

Design estimates for power loss in astern turbines appeared to differ very widely indeed, and it was difficult to reconcile Mr. Ryall's figure of 0·1 per cent for the H.P. astern wheel power loss with the general feeling that there was a really urgent need for an alternative reversing mechanism for marine steam turbines. An estimate of the astern turbine loss obtained from the manufacturers of a set of turbines in some other ships in his company's fleet had been given as 0·6 per cent of the ahead power. This design utilized an astern element on the L.P. rotor only and the manufacturers advised that the addition of a separate H.P. astern element to raise the astern power by a further 10 per cent might double the total windage and disc friction loss.

Mr. Ryall's figures, presumably, and those quoted above were for standard 28½in. Hg vacuum conditions. The H.P. astern cylinder gland sealing steam flow through the L.P. astern turbine could well be greater than Mr. Ryall had allowed. The turbine builders frequently insured against rubbing of the H.P. astern cylinder gland (where running alignment could admittedly be difficult) by providing considerably more clearance than the designer may have deemed to be adequate. Taking this and the effect of reduced vacuum in warm sea conditions into account, the suggestion of 1 to 1½ per cent loss in each element may not have been so very much out of court. It appeared that the figure of 650 h.p. used in the paper had been too high however, and the author was grateful for Mr. Ryall's correction on this point.

Mr. Ryall's contention that an electric motor drive for the main circulating pump would be some £2,000 cheaper was not apparent in the cost studies made at the time. The provision of a multi-speed motor drive would have involved an expensive extra cost for switchgear and it was considered that output control of the pump in either case was the only valid basis of comparison.

It was agreed that turbine operation with up to 29in. Hg condenser vacuum could be economic provided no significant undercooling of the condensate took place. Despite the claims of the condenser designers it had not been found possible in practice to avoid this undercooling in cold sea water unless the cooling water flow was reduced.

The single stage Curtis wheel generator turbine mentioned by Mr. Ryall would obviously require an entirely reliable device to divert its exhaust from the H.P./L.P. crossover to a condenser in the event of an emergency stop of the main engine. The alternative supply of reduced pressure desuperheated steam would have to be immediately available at all times. The necessary automatic devices and the steam line drainage problem should be relatively easily solved however, and the system was worthy of very close consideration.

The Diesel set mentioned in the paper combined the functions of an alternative electric power generating source in the event of breakdown of the turbine set (whether this was a multi-stage unit or a single-stage unit operating on bled steam) and the necessary auxiliary power source for dry dock and "deadship" starting use.

The author was indebted to Mr. von Feilitzen for his remarks, which provided a most useful comment on all the major aspects of the cycle described in the paper. He had the greatest regard for the technical achievements of Mr. von Feilitzen's company and wished to acknowledge that the introduction of the back pressure cycle in tankers had been mainly due to their clear lead.

The figures claimed for auxiliary unit efficiency were almost too good to be true and he thought perhaps that the De Laval generator turbine would not be more than 7 per cent better than the one referred to in the paper if comparable part load performance was considered. Even so, this represented performance of a very high order of efficiency.

Mr. Mackenzie had suggested that 300 days per annum at sea would have been a more realistic basis for assessment of ship performance. If this were so, his argument advocating a higher efficiency cycle was, as the author saw it, diminished rather than strengthened.

He was not clear as to the comparison which was made with the cycle yielding a fuel rate of 0·503lb./s.h.p./hr. 75 deg. F. average sea temperature for Middle East/N.W. Europe did not allow the inference that average condenser vacuum on this sea route was 28½in. Hg, if it was agreed that some limitation of condenser vacuum in cold sea water was essential. Even so if this was used as a basis of comparison, the cycle fitted should also be credited with an average operating vacuum of 28½in. Hg since the condenser could achieve this with 75 deg F. sea temperature. A fuel rate of 0·523 should then be compared with 0·503 and this reduced the possible fuel saving to less than half of Mr. Mackenzie's figure of £18,000 per annum.

The increased capital cost of the 0·503lb./s.h.p./cycle was more likely to approach £50,000. The additional effect of the extra machinery weight and maintenance costs would remove most of any apparent savings achieved.

Mr. Mackenzie's feeling that his £30,000 was barely noticeable in the cost of the ship as a whole was not an uncommon one, and perhaps something like 0·7 per cent of the capital cost took a bit of finding. The implied suggestion however, that capital cost should be measured in some different currency from that in which the ship's earnings were assessed, left the author a little uneasy.

The author thanked Mr. Hutchinson for his amusing and very valid criticism of the standard flange tables. There was much in what he said and something could be learnt from American friends in this particular respect.

As to his request for a detailed statement on the mechanism of boiler deposits at higher pressures than 600lb./sq. in. gauge, the author felt it would be as well if someone better qualified than he could undertake this task. He would say that boiler water treatment specialists had confirmed that this phenomenon was no phantom.

Replying to Mr. Neumann regarding the effect of the overboard discharge lips he would say that although the net effect on fuel economy may be against their use in a fast naval vessel it was unlikely that in large tankers where the major component of the hull resistance was skin friction, the same was unlikely to apply.

The reason for using a butterfly type sea water inlet was mainly because it offered a valve capable of being closed manually in much less time than a sluice valve in the event of say a burst C.W. line. The compactness of the butterfly valve was also of considerable advantage and in fact the overboard discharge valves had subsequently been changed to this type.

The relative positions of the main gearcase and the boiler plant in this particular layout followed orthodox practice insofar as the facility for a "straight lift" of the main gearwheel out of the ship had been retained. It was doubtful whether such facility was really necessary particularly if the headroom available below the boiler flat would allow the wheel to be moved forward as a preliminary to removing completely. The gearcase could then be moved considerably further aft (at least 6 feet) with a corresponding reduction of machinery space length. The problem of accommodating auxiliary machinery would naturally become more acute, and this of course added impetus to the argument for further reduction of auxiliary plant components considered at the end of the paper.

The twin uptake arrangement resulting from the particular boiler handing was possible only in the circumstances of the wide beam available for crew accommodation.

No lubrication in the steam spaces of any reciprocating machinery was arranged. The operating time of these units was extremely limited and a reasonable degree of independence upon lubricants liable to cause feed contamination could be had by employing suitable cylinder and piston ring materials.

The paper did not claim, as Mr. Norris had suggested it had, that a *pro rata* increase in cargo carrying capacity was exchanged for the missing 23 tons of economizer plant. It could of course be so on very short voyages, but the overall effect on the Persian Gulf/Europe run was likley to be negligible. The recoup of the £880 per annum net extra cost between cost saving on the economizer plant and paying the fuel bill was expected to derive mainly from reduced maintenance costs.

The effective length of the pumproom was measured at the level of the cargo pumps themselves and the after bulkhead of the pumproom at this level was the line shown at the extreme right side of Fig. 12. If the pumps were the horizontal type, the driving turbines would foul engine room units 9 and 10.

The reduction claimed in engine space length was, as was stated, in the dimension from the aft peak to the aftermost cargo bulkhead. For equal pumproom lengths measured at pump level there was no doubt that the stepped construction of the aft pumproom bulkhead and the use of vertical pumps allowed the cargo bulkhead to move aft by that distance which represented the length of horizontal cargo pump turbines.

The main superheated steam pipes were led through the boiler flat via tubular, open trunks which extended to the height of the firing floor.

In reply to Mr. Kirschbaum's question on circulating pumps, the figures in the paper compared the single turbine driven unit with a single (two speed) motor-driven unit. Twin motor-driven pumps each of 50 per cent capacity would not be cheaper and although there was admittedly some security advantage, this scheme would not otherwise appear to offer any significant benefits.

Mr. Kirschbaum made a good point in regard to the starting of the F.D. fan. Provision was made so that one of the boiler room ventilating fans could discharge into the forced draught ducting to expedite steam raising with only the Diesel generator available. The Diesel generator would not be able to start either of the F.D. fans.

It was not easy to put one's views on the very wide subject of automation and simplification in a few words. One thing which would be obvious was that there was nothing basically difficult in achieving simplification. To achieve simplification with a tolerable sacrifice of efficiency had been the real purpose of this design.

To be able to accept simplification in the sense that reduced standby equipment, etc., was provided, required that one had to be satisfied that the reliability of the equipment installed was adequate to provide the necessary standard of security.

The author felt that wider, and eventually, complete application of automation to propulsion machinery and boilers was inevitable in future designs. He believed this would result from the very logical commercial pressure which dictated that a constant effort must be made to cheapen the cost of doing the job, and this in face of a continuing rise in the cost of the individual labour involved. The equipment which is to be automated, and the automatic devices themselves must obviously be developed to a high standard of reliability if this were to be achieved.

Replying to Mr. Bell, the author felt that his case for series steam heating and gas heating of the combustion air had been well put. No doubt the maintenance difficulties which had been encountered with gas air heaters taking air direct from the F.D. fan had soured many shipowners against gas air heaters in any shape or form. If it could be proved that an arrangement incorporating an adequate degree of steam preheat could be installed for less cost and without undue complication of F.D. and gas ductwork; there was undoubtedly a place for such arrangement.

Mr. Bell presumably intended the use of a regenerative type gas air heater. If a tubular, recuperative design was used the minimum metal temperature at the air inlet end might be less than his quoted figure of 290 deg. F. and consequently give rise to corrosion troubles in that locality.

Mr. Halley asked if the fuel consumption in harbour had been taken into account in comparing condensing type and back pressure type turbogenerators. The answer here was that harbour consumption had been ignored mainly because of the difficulty in making any realistic assessment. When the cargo pumping equipment was in use, the turbogenerator exhaust could be fully absorbed. The operating conditions at the loading terminal created the maximum exhaust surplus but this was normally for a period less than 24 hours duration.

The comparison between a 240 deg. F. and a 280 deg. F. feed temperature cycle could only be fairly made on the assumption that an L.P. heater would be fitted in the latter case. If the installed cycle, i.e. Fig. 2, had its feed temperature raised to 280 deg. F., the economic performance would actually be inferior to that which was at present anticipated.

Submerged type lubricating oil pumps were not seriously considered for this particular design but this did not infer any rejection of the submerged pump principle. Provided that a reliable extended shaft drive between the pump and the motor which would raise the motor out of the bilge could be arranged there would seem to be advantage in being able to dispense with oil suction piping.

Appendix

S.S. *Serenia*, the first ship of the class, left the builders' shipyard on 16th June en route to drydock on Merseyside.

No difficulty was experienced with the machinery during this passage although the presence of substantial launching brackets aft and some fairly stiff weather in the Pentland Firth made any accurate estimate of fuel performance difficult.

A series of rapid and severe engine manœuvres, including a maximum astern power run of 20 minutes duration, was carried out at the River Mersey Bar prior to docking. The H.P. and L.P. turbines were opened up for inspection in the dock and were found in excellent condition although a slight rub had occurred on the H.P. astern cylinder gland and certain of the H.P. ahead shroudings had touched lightly on their axial clearance edges.

The tailshaft was pulled and the white metal bushings in the sterntube found in good shape. Fig. 23 shows a view into the main bushing from the outboard end, the tailshaft being not yet fully withdrawn. Bearing marks are evident and although the marking in the foreground, which is the extreme after portion of the bearing, appears somewhat hard, there was no sign of distress in the form of wiping or crushing or cracking of the white metal. No work was performed by way of bearing surface adjustment or easing. The upper and port side oil grooves can be discerned by the light bearing marks and the hand tool marks (done prior to fitting). A puddle of lubricating oil is visible below the tailshaft end.

FIG. 23—*Looking into the main tailshaft bush from the after end. The upper and port side oil grooves can be discerned by the light bearing and hand tool marks (done prior to fitting). A puddle of lubricating oil is visible below the tailshaft end*

During trials off the west coast of Scotland following the drydocking a check on the fuel rate was made over a 30 hr. run. The settling tanks, carefully calibrated in the shipyard, indicated a fuel consumption of 10,450lb./hr. A displacement type integrating flow meter gave 10,880lb./hr.

Power was measured by Maihak and Siemens Ford torsionmeters which were in very close agreement throughout at 20,900 s.h.p. on 106·7 r.p.m.

The engine builders would be perfectly entitled to claim that they had demonstrated a specific fuel performance of 0·5 lb./s.h.p./hr. The fuel meter used had on previous calibration checks consistently proved accurate well within 1 per cent, however, and a careful analysis of all observed data and correction factors, etc., would be required to indicate where the 4 per cent discrepancy in measured fuel rate lay.

Some further observations could be made at this stage:

1) The bleed from the propulsion turbines was considerably higher than design for Fig. 2 conditions and was in the region of 8,500lb./hr.
2) The main circulating pump turbine steam consumption was about 350lb./hr. The head across the pump was measured at 3½ft.
3) The turbo generator loading throughout the trial averaged 443 kW (measured on a recording type kW meter).
4) The sea temperature was 55 deg. F. and the domestic heating load was negligible.
5) The steam inlet condition at the H.P. turbine was 560lb./sq. in. gauge/895 deg. F. for 600lb./sq. in gauge/900 deg. F. at the superheater outlets. The

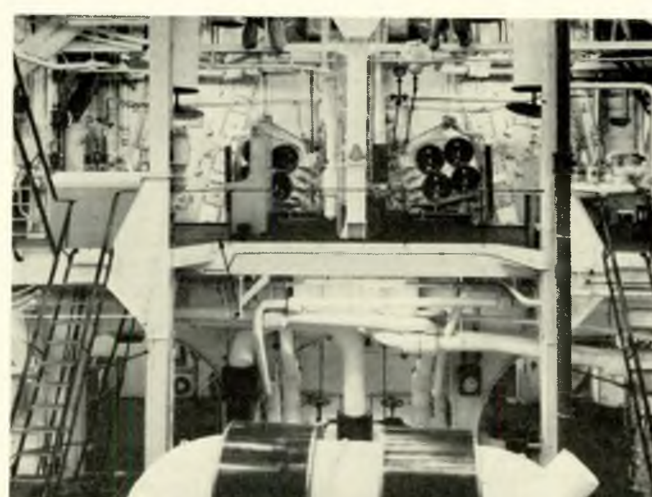

FIG. 24—*Looking aft across the propulsion unit to the manœuvring console, automatic combustion control and boiler fronts. It will be noted how the transposition of the boilers has grouped the burners towards the centre and taken the superheater outlets to the outboard side as shown in Fig. 15*

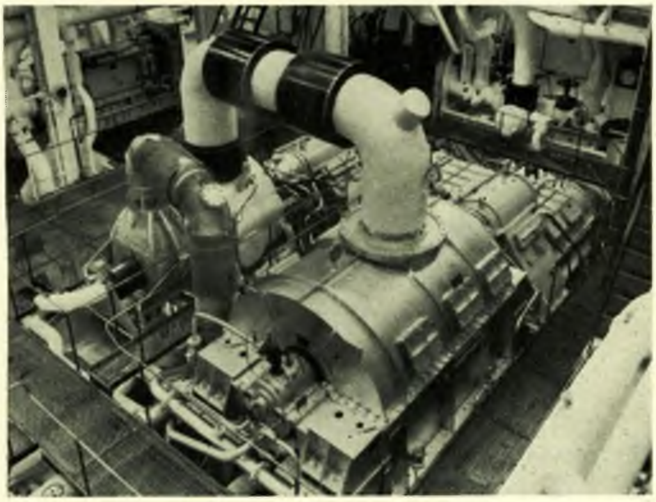

FIG. 25—*Showing the main propulsion plant with the auxiliary Diesel alternator in the left background and the superheated steam distribution manifold in the right background. The open nature of the layout is well illustrated in this view*

FIG. 27—*Showing the manœuvring console and combustion control panel with turbo-alternator and main switchboard in the background*

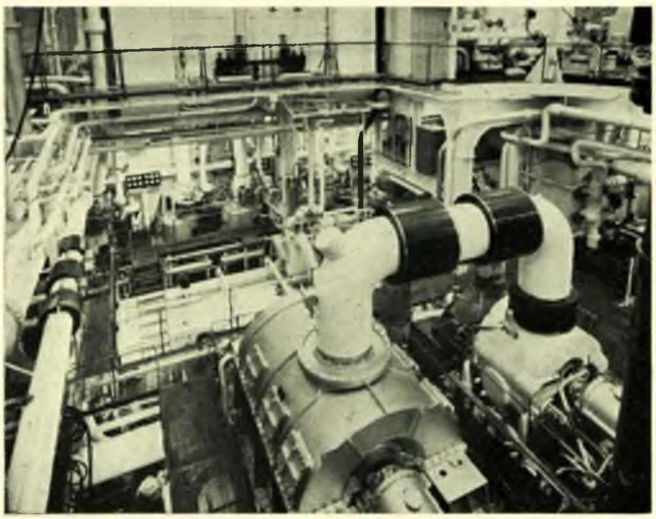

FIG. 26—*Showing the main turbine and looking forward into the cargo pump turbine flat. The external desuperheater runs up the left side of the photograph*

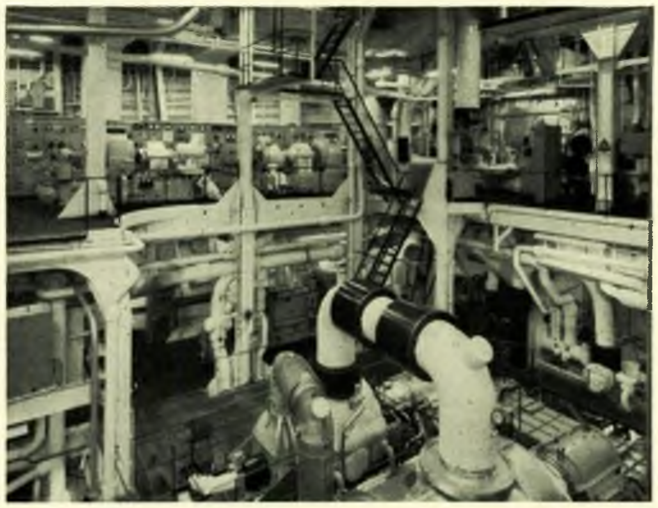

FIG. 28—*Looking to starboard where the main alternators and switchboard are situated. The Diesel alternators and combustion control panel may also be seen. The high level of lighting and absence of dark spots is apparent in this photograph*

main condenser vacuum was held at 28·5in. Hg throughout by adjustment of the main circulating pump speed. The overboard discharge temperature was 77 deg. F. and the condensate undercooling appeared to be of the order of 4 deg. F.

6) Co_2 at boiler outlet averaged between 13·7 per cent and 14 per cent.

It can be confidently anticipated that the "all purposes" specific fuel rate in temperate sea water would be within the figure of 0·523lb./s.h.p./hr. allowed for this installation.

If a mean was taken between consumption measured by settling tank soundings and oil flowmeter (and this would seem an unduly optimistic procedure) the basic temperate sea water fuel rate would be 0·510lb./s.h.p./hr. and indeed this figure appeared to be closely substantiated by the results of the ship's first weeks in service.

The Development of Heavy Duty Marine Diesels During the Past Five Years*

W. KILCHENMANN (Member)†

Discussion at the Meeting of the Eastern U.S.A. Section in New York, on Monday 13th March, 1961

MR. NICHOLAS BACHKO said that Mr. Kilchenmann was to be congratulated for a well written and informative paper. The subject of Diesel propulsion for large ships was not overwhelmingly popular on that side of the ocean and if Mr. Kilchenmann's paper received critical discussion, it was hoped that he would understand that it was the Diesel powered ship and not himself that engendered the painful comment.

Mr. Bachko for his own part, read and re-read the paper with avid interest and mixed feelings, because it brought to mind the many long hours of hard work he had spent on Diesel engines when he first went to sea. The paper made reference obliquely to the work involved, but some of the problems needed to be experienced to be appreciated.

It was somewhat of a shock to realize that many of the characteristics, peculiar to the machinery design in his first Diesel ship twenty odd years ago, were being regarded today as new developments. The engines to which he referred had a bore of some 760 mm., burned bunker C fuel, and produced over 1,500 b.h.p. per cylinder.

They had all the problems touched on in the paper, plus a few more:

1) Excessive wear of cylinders and piston rings.
2) Heavy carbon deposits on injectors, scavenge ports, etc.
3) Heavy sludging and contamination of crankcase oil.
4) Water leakage from piston cooling lines in the crankcase.
5) Corrosion of bearings and journals.
6) Contamination and sludging deposits in the scavenge belt with consequent fires.
7) Unusually heavy maintenance requirements of fuel separators, due to constant operation on hot bunker C.
8) Burning out of braking valves.
9) Fracture and failure of fuel oil pumps, and more besides.

As for the difficulty of handling tools and spare parts in repairs of these large engines, he recalled his absolute astonishment at the "miniature" size wrenches and small sledge hammers on his first steam turbine ship.

He did not mean to bring this discussion around to the relative merits of Diesel propulsion, but was truly amazed at the relatively small degree of progress associated with the Diesel engine at a time when other propulsion systems were being rapidly advanced. Steam turbines, even while enjoying a definite advantage in lower maintenance and in higher degree of reliability, were still being refined to insure an even better characteristic than they now attained. At the same time the overall plant efficiency was being improved to a point where this propulsion system was challenging the Diesel in the very theatre where the Diesel had enjoyed unchallenged acceptability —i.e. Europe.

In the United States the wide disparity between boiler fuel costs and Diesel fuel costs, plus the high Diesel maintenance cost, precluded the resurgence of the Diesel. It continued to tempt its backers, of whom there were many, and each generation of ships must have its Diesel power experiment, but invariably the Diesel experimental vessels ended in the "ghost fleet".

If they could accept the higher cost of Diesel fuel, they would install the large powered aircraft type gas turbine which enjoyed very low first cost, very light weight and proven reliability in the time that it had been operational, with estimates of life and reliability being revised upward with the passing of each day. Also, that engine had efficiencies which showed specific fuel rates at high powers and improved cycles, challenging to the Diesel.

They also had a proven reliable gas turbine for marine application which challenged the Diesel on all counts, including the burning of heavy boiler fuels. In addition, this unit, in a newly proposed cycle, had very definite prospects of surpassing Diesel specific fuel rates, while burning heavy boiler fuels. This engine challenged all propulsion systems on weight and cost and offered some remarkable power margins, at no extra cost, when operating in cool ambient temperatures. The disadvantages of fuel washing and treatment, and of power reversal, were capable of reliable solution, in time. Incidentally, the first of these disadvantages was common to all fossil fuel fired plants.

LIEUT. G. D. KINGSLEY, JR., U.S.M.S. said that it was with a great deal of pleasure that he discussed Mr. Kilchenmann's paper, especially in view of the fact that it put forth such strong arguments in favour of the large, slow speed, high horsepower Diesel engine to compete with steam propulsion. The audience might understand his feeling in this regard when they realized that the Diesel programme at the United States Merchant Marine Academy had to be geared to the demands of the American shipping picture, in which Diesel power was confined principally to river and harbour craft. The engines used aboard this type of vessel ranged from the small high speed units to the fairly respectable horsepowers

* The full text of this paper and the discussions in Canada are published in the September issue of the Canadian Supplement.

† Technical Director, Diesel Engine Department, Sulzer Bros., Switzerland.

found with intermediate speed engines. It pleased him a great deal to see that, with the possibilities of ship propulsion described by Mr. Kilchenmann's paper they might yet see the day of large Diesel engines in ships sailing under the American flag.

The paper had outlined very well the major areas of development and improvement that were required to bring high output engines into being. He would like to make a brief comment on those areas. They could find no fault with the discussion on welding, and although in the United States they had never constructed engines of the size discussed in Mr. Kilchenmann's paper, welded construction in varying degrees had been satisfactorily employed by builders of small high speed and intermediate horsepower engines since the middle-thirties. With improved welding and annealing techniques, and the many advantages of lighter weight offered, they could see why Sulzer had developed and extended those processes for their large slow speed engines.

With reference to the topics "Turbocharging of large two-stroke cycle engines", and "Further increase of output by adoption of large cylinder bores", it was impossible, in the time allotted, to give a satisfactory discussion on the wealth of information that the author had put forth in his paper. As a teacher of Diesel engineering, he could say that the paper would be recommended reading for cadets in the future.

In view of the short time available, he would like to comment about a part of the paper, on which there might be a slight difference of opinion, and that concerned the term "heavy fuel". On several points he agreed with Mr. Kilchenmann; namely, that for the utilization of boiler fuel, Diesel engine development had gone about as far as it could go. He also concurred heartily with the necessity of cooling nozzle tips. He also felt that some form of separation between crankcase and combustion area was necessary. In this regard he favoured the diaphragm and packing gland about the piston rod, rather than the oil scraper ring and sludge chamber that was used by some engine manufacturers.

However, he would like a little clarification of the term "boiler fuel". It had been his experience that the term "boiler fuel" referred to anything from a relatively easily handled 600 sec. Redwood number 1 blend of distillate and residual, up to and including 5,000 and 6,000 sec. crudes and residuals. It was also his understanding that the so-called heavy fuel burning engines utilized 600 to 1,500 sec. blends of distillate residuals. He might offer, as an example, a recent survey of some 1,600 ships conducted by one of the major oil companies, where only six vessels used fuels in excess of 3,500 sec., and 36 vessels used fuels in excess of 1,500 sec. Further, he was under the impression that a common method of preparing Bunker C fuels was to supply to the bunker tanks, an additive which was primarily a sludge disperser. The fuel was then delivered to the settling tank, from whence it was delivered first to a heater, then a separator, and finally to the day tank. The final stages of fuel preparation took the fuel from the day tank, passed it through terminal heaters, and then filters, from which it was delivered to the pump, injectors and the combustion chamber. He had gone through this rather lengthy and basic description of the fuel handling process with the hope that later Mr. Kilchenmann might shed some light on the following:

1) In order for the Diesel propulsion plants to compete right down the line with steam propulsion plant, in the United States, the use of low grade boiler fuels was an economic necessity. Did Mr. Kilchenmann think that the use of low grade fuels was feasible in his large output engines?
2) Were there better methods of handling bunker fuels than those outlined?
3) Could the author, from his experience, say if it were correct to assume that the average large slow speed, high powered engines were using fuels varying from 600 to 1,500 sec. Redwood number 1, and if this assumption were correct, was there a definite point where increased maintenance cost precluded the use of heavier or true Bunker C fuels, even at their lower cost?

He wished again to thank Mr. Kilchenmann, Mr. Speer and Mr. Thomas for the opportunity to participate in the activities of the Institute of Marine Engineers.

Mr. W. S. Henry said that Mr. Kilchenmann had presented a most interesting and timely paper which was a credit to the famous name of Sulzer. The thoroughness, with which his company had developed and evidently proven the soundness of their RD engines, gave evidence of why such large horsepowers as 24,000 b.h.p. in Diesel engines could be achieved.

The modern, large, slow turning, two-stroke Diesel engine was very different from the heavy, cast-frame engines of the past and, no doubt, the cost of the modern engine reflected the considerable refinements developed for those engines in all their parts. The use of welding, turbo-supercharging, the ability to burn heavy fuels, and the use of progressive engineering concepts and materials had all helped make the large-powered two cycle Diesel engine possible.

He did not wish to make a lengthy economic comparison between the type of engine described in the paper and the modern steam turbine. However, he would suggest that such a comparison would show that, except for long runs, a vessel powered with an American steam turbine would have a slight overall economic advantage over a comparable Diesel driven vessel, both burning heavy oils at sea and, further, a Diesel vessel burning distillage fuels would be definitely non-competitive.

He considered that a well designed and proven large modern Diesel could have certain operational advantages, such as steaming radius, manœuvrability and integration of machinery, leading towards pilot house control and automation of the vessel. In contrast, credits for present-day American-type steam turbines would include, among other advantages, superior reliability, minimum maintenance costs, minimum port time requirements, and capacity for developing maximum horsepower continuously with no adverse effect on maintenance or fuel economy.

One of the most important problems facing operators of large powered Diesel vessels was the attitude of crews towards maintenance on the large engines. This condition was aggravated by increased wear of parts and repairs encountered, due to burning heavy oils. Also, for the same reason, the manning of such vessels under the American flag would be next to impossible. The heavy fuels for the Diesel vessel would have to be selected or blended, and both steam and Diesel plants would require the onboard treatment, of washing of fuel.

The J. J. Henry Co. Inc., had supervised the construction of a number of large Diesel bulk carriers recently, in Japan and Germany, but the engine powers were all under 10,000 b.h.p. Inasmuch as Diesel engine powers of more than twice that figure were presently being developed, his company were anxious to learn all they could about actual operating results with such machinery, using heavy oils. Also it was of interest to note that the author's firm had developed special tools and facilities for maintenance and overhaul work. The use of hydraulic tools by qualified engine repair men or crew would aid greatly in routine engine work.

He would again like to compliment the author on a very fine technical paper, which would help to bring American marine engineers up to date on the subject of large modern heavy-duty marine Diesels.

Dr. John J. McMullen, B.S., M.S. (Member) said that he did not have a prepared contribution, but he felt that being an old time friend of Mr. Kilchenmann, in Switzerland, he should come to his defence. In fact, he had suggested that the author should deliver a paper in the United States, on the development of heavy-duty marine Diesel engines during the past five years. The reason why he felt that a paper of this type would be interesting to a group of engineers in the

United States was based on the attitude which was typified by the previous three contributions. One of the things he feared was that the engineer in the United States was not thoroughly familiar with the development of the Diesel engine, during the past five years. He wished to add that he did not believe that there was any possibility of a large slow-speed Diesel engine being installed in an American ship but, on the other hand, it should be obvious that all of the people who were installing heavy-duty slow-speed engines were not wrong.

The basic point was the economics of the entire problem. Briefly, the fuel costs in a foreign ship represented the highest single item of operating expense and it was for this reason that emphasis had been placed on fuel economy, rather than on wages and capitalization, which were the highest items in an American ship. In other words, the entire economic picture, as far as foreign operators were concerned, was quite different from that for United States operators.

It was hard to tell which was the cause and which was the effect, but it was also true that abroad, the selection of Diesel engines was far greater than anything they had had in the United States. For example, there was only one marine Diesel engine available in the United States, whose horsepower was in excess of 6,000 s.h.p.; whereas in Europe the number of Diesel engines available, he would guess offhand would approach eighteen or twenty. In addition, the shipyards, which were building Diesel engines abroad and which had licenses to construct such engines, were quite numerous. This again came into the economic picture, because the shipyards would much prefer to supply a ship with a slow speed Diesel engine, built by themselves, than with any other type of equipment, which they would necessarily buy as an outside purchase.

Basically, however, the fact was that during the past five years the advance made by the slow speed Diesel engine was really astonishing. If, five years previously, someone had mentioned the fact that it would be possible in five years time to order a Diesel engine of 20,000 to 25,000 s.h.p., he personally would never have believed it. It was for that reason and that reason alone that he intended to listen to Mr. Kilchenmann when he mentioned the fact that five years from then the horsepowers might be in the range of 35,000 to 40,000.

He wished to congratulate Mr. Kilchenmann on the paper, and at the same time to use this occasion to say that as far as American cargo ships were concerned, he intended to agree with Mr. Bachko that the next real advance in the United States was in the field of gas turbines.

Author's Reply

Mr. Kilchenmann replied that Mr. Bachko's discussion represented a very strong attack against all those who liked Diesel engines. Anybody reading it out of context would think that the many shipowners in Europe, who put Diesels into their ships, were rather foolish. He, therefore, wanted to give a few figures taken from Lloyd's statistics, about the proportion of Diesel and steam propulsion in various ship sizes ordered and under construction during the past few years. In the range of 100 to 2,000 gross register tons, 80 per cent of the ships had Diesel engines in 1954 and the figure had risen to almost 100 per cent by 1960. In the range of 2,000 to 6,000 gross register tons, 80 per cent of the ships had Diesels in 1954, while 90 per cent had Diesels in 1960. Looking at the figures for 8,000 to 10,000 tons, there were 50 per cent Diesel ships and 50 per cent steam ships in 1954, whereas in 1960 there were 85 per cent Diesel ships and only 15 per cent steam ships. The audience were probably still more interested in higher ranges.

In the range of 10,000 to 15,000 tons, there were 60 per cent Diesel ships and 40 per cent steam ships in 1954, and in 1960 there were 80 per cent Diesel and only 20 per cent steam. Higher still, in a range of 15,000 to 20,000 tons, the picture began to change, because until recently Diesels large enough for such ships were not available. Thus in 1954 only 20 per cent of these ships had Diesels; yet in 1960 the figure had risen to 50 per cent. Going higher, 20,000 to 25,000 tons, there were only 10 per cent Diesels in 1954, but in 1960 the figure had risen to 40 per cent. In still higher ranges the picture admittedly looked less favourable for the Diesel, because of the reason just mentioned. In the range of 25,000 to 30,000 tons there were no Diesel ships at all in 1954, but in 1960 the proportion was already 10 per cent. Above 30,000 tons, there were no Diesels at all until 1957; but at the present time already 10 per cent of this upper range had Diesels. These figures were, he thought impressive and would show that the Diesel, far from being challenged in Europe by the steam turbine, as stated by Mr. Bachko, was taking an ever inceasing share of the ship propulsion field.

Mr. Bachko mentioned, in his contribution, an engine which 20 years before was developing 1,500 b.h.p. per cylinder and he contended that very little development work had taken place in the past 20 years, when comparing present engines with that one. If that engine really had a bore of 760 mm. and developed 1,500 b.h.p. per cylinder, it must have been either a double-acting engine or an opposed piston engine. Double-acting engines had disappeared and nobody would now want to build them. They were too complicated, so that he would not be surprised if Mr. Bachko had experienced trouble with them. If, on the other hand, his engine were of the opposed piston type, then, in the author's opinion, an opposed piston design represented the most complicated way of building an engine. There were too many cranks, too many connecting rods, too many of everything in that engine type. Thus, he saw little point in comparing a complicated engine of 20 years ago with the very simple engines now being built; as shown by his paper, the difference was tremendous. Mr. Bachko mentioned various specific points which he would like to take up. For instance, excessive wear of cylinders and piston rings, when burning Bunker C fuel; these days, excellent wear figures of ·002in. per 1,000 hours were frequently obtained. His company considered ·002in. to ·004in. good cylinder liner wear figures. That was possible with any kind of heavy fuel up to 3,500 to 4,000 sec. Redwood, provided the engine used one of the high alkaline lubricating oils now made available by the oil companies. When the cylinder liner wear figures were from ·004in. to ·008in. per 1,000 hours, they were considered acceptable, but subject to improvement. Wherever wear was higher than ·008in. per 1,000 hours, something was definitely wrong and should be checked.

Mr. Bachko's next point was "heavy carbon deposits on injectors, scavenge ports, etc.". The paper specifically mentioned the improvements made to avoid such difficulties. It showed how trumpet-shaped carbon deposits on the injector nozzles and other deposits in the ports were avoided. The effect of the piston underside, i.e. an increase of air pressure in the scavenging receiver, prevented exhaust gas from flowing back from the cylinder through the scavenge ports. Thus, the scavenge ports remained 100 per cent clean. In older engines there were two rows of scavenge ports, with non-return valves fitted to the upper row. Every time the piston came down, exhaust gas expanded in the non-return valves and the ports thus became heavily carbonized. That was no longer possible. On the exhaust side, very efficient cooling of the metal surrounding the ports, together with modern lubricating oils, resulted in a clean port area. Deposits in the exhaust ports were negligible.

A further point was heavy sludging and contamination of crankcase oil; this was the reason why his company made a complete separation between cylinder and crankcase. The stuffing box itself was connected with the atmosphere; any gas which leaked through and any entrained dirt, were blown out by scavenging air to atmosphere, so that it could not enter the crankcase.

Mr. Bachko's next complaint referred to water leakage from piston cooling lines in the crankcase; that was really a very serious problem with the older type engines, which had water cooled pistons. The author had tried to show with diagrams, the advantage of water-cooling with regard to maintaining a low piston temperature. He had also stated that only with his company's new arrangement, which completely separated the telescopic tubes for the piston cooling water from the crankcase, did they consider it safe to re-introduce water cooling.

Corrosion, etc., were all problems connected with the contamination of the oil and the water leakage just discussed.

Time prevented him from referring in full to the last few points of Mr .Bachko's discussion.

Mr. Bachko had mentioned a wide disparity between boiler fuel cost and Diesel fuel cost. He seemed not to be aware of the fact that Diesels currently used exactly the same fuel as boilers. Even for manœuvring, the Diesel used boiler oil. This was not new. Many shipowners had burned boiler fuel in their Diesels for quite a few years.

Mr. Bachko proposed to use aircraft type gas turbines for marine applications. The author, without going into details, could only wish him luck.

He was grateful to Lieut. Kingsley for his contribution. The first problem he had raised was again whether boiler fuel could be used in Diesels. The answer was most emphatically "yes". The question no longer represented a problem.

Lieut. Kingsley's description of boiler fuel preparation, for its burning in Diesels, was entirely correct and fully corresponded to what, in the author's experience, was done in ships. There was no better way of doing it.

As to fuel viscosity, most ships used fuel with 1,500 to 1,700 sec. Redwood. That appeared to be general practice, although much heavier fuels could be used, as the following illustration would show: on one occasion his company had run, on their test bed in Winterthur, an engine operating on a very heavy fuel. The engine ran only in the daytime and was stopped after normal working hours. The next morning, one of the test engineers dismantled a 30 foot length of fuel piping between fuel pump and injector. The fuel used the previous day had a viscosity of 5,000 sec. Redwood. In the morning, engine and fuel were, of course, cold. The engineer connected the length of piping to the outlet of a pump and thus put pressure on the fuel still located inside the pipe. After a few seconds, a black mass with the consistency of toothpaste started slowly dripping out of the other end. That was the kind of fuel which could now-a-days be used in engines. Of course, such fuel must be heated, whereby the viscosity came down to around 27 centistokes, and it must be heated under pressure. He knew of quite a number of ships which used fuel with 3,500 sec. Redwood, and the wear figures for their engines, were just as good as for those which operated with lighter viscosities.

Mr. Henry had asked a question concerning the economies of Diesels versus steam plants in ships. The author had a few figures, received not from an engine builder, but from a shipbuilder in Scotland, who built both Diesel and steam ships. The figures related not just to the main power plant, but to complete engine rooms, including all auxiliaries. The first referred to a ship with 7,500 s.h.p. Diesel power, alternately 7,500 to 7,850 steam turbine power. Taking the price of the complete engine room with a Diesel as 100 per cent, the corresponding price for a steam turbine was 116 per cent. A second example mentioned 15,000 h.p. for a Diesel plant, 15,500 h.p. for a steam plant. Again taking the Diesel plant as 100 per cent, the cost of the steam plant was 111 per cent.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 14th February 1961

An Ordinary Meeting was held by the Institute on Tuesday, 14th February 1961 at 5.30 p.m., when a paper entitled "The Design and Layout of a 22,000 s.h.p. Tanker Machinery Installation" by Mr. J. B. Main (Member), was presented by the author and discussed.

Mr. W. R. Harvey, O.B.E. (Chairman of Council) was in the Chair and 173 members and visitors were present.

In the discussion which followed eleven speakers took part.

A vote of thanks to the author, proposed by the Chairman was greeted by acclamation. The meeting ended at 8.30 p.m.

Eastern United States Section

A meeting of the Eastern United States Section was held in New York on the 13th March 1961, when a paper entitled "The Development of Heavy Duty Marine Diesels During the Past Five Years" was presented by Mr. W. Kilchenmann (Member).

Mr. P. C. Speer, Honorary Secretary of the Section was in the Chair and Dr. J. J. McMullen (Member), Lieut. G. D. Kingsley, Jnr., U.S.M.S. and Mr. W. S. Henry took part in the discussion which followed the presentation of the paper.

On Tuesday, 14th March Mr. Kilchenmann re-presented his paper to the United States Merchant Marine Academy, King's Point.

Election of Members

Elected on the 18th September 1961

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Minoo Dhunjishaw Aibara
Harold Duncan Aimer
Leonard Balls
John Beldam Booth
Leslie Budd, Eng. Lieut., D.S.M., R.N.
Duncan Campbell, Capt. R.N., B.Sc.(Glasgow)
George Arthur Davies
Ascencio Anthony de Freitas
Ian Ralph Edwards, Cdr., I.N.
Ralph William Edwards, Cdr., R.C.N.
Haydn John Elworthy
Frank Entwistle
Artur Frenkel
Reginald Thomas Green, Lieut. Cdr., R.A.N.
Herbert Leonard Honess
Harry Jagger, Lieut. Cdr., R.N.
Garden Lawson
William Nicholas Makepeace
William Milne
Raymond Kennedy Mowat
John Gosman Brodie Muir
Derek Seymour Northcote
Jack David Owen
Matthew Atkinson Pallett
Edward William Phillips
Robert George Raper, Capt., R.N.
Colin Rawden, Cdr., R.N., B.Eng.(Sheffield)
George Soens
Douglas Thompson
Leslie James Western
Edward Hall Wilson
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William Arthur Barnes
Brian Bell
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Ronald Guthrie Bowmaker
Ronald William Bridgeman
Lawrence Trevor Brown
Peter John Brunt
Cyril Casmil Burges
John Clark Button
Justin Carasco
Franco Antonio Casa, Dott. Ing.
Surash Chandra Chaturvedi
Douglas Stephen Cook
Ian Alexander Currie
Cecil Edward Curry
Sasadhar Datta
Philip Douglas
Rex Joseph D'Souza
Stephen A. Fleming
John Andrew Flett
Humphrey Kynaston Gibson
Jatinder Nath Goswamy
Thomas Hodgson Govenlack
Richard Neville Hanson
Clive Edmund Holmes
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John Alan Lewis
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Kenneth William Lutas
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James George Milne
Grev Nadirshaw Mody
Stanley Morris
Raymond Muddell
Peter James Munday
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John Oliver
Kenneth Stanley Orchard
George Rex Phillips
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Alfred James Weall
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Pierre Leon Manuel
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Robert Archer
Satyapal Bhardwaj, Sub. Lieut., I.N.
John Garry Beaumont, B.Sc.(Durham)
John Henry Brown, Jnr.
Campbell Crawford Gawn
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A. C. Verma, Sub. Lieut., I.N.
David Graham Walters, B.Sc.(Eng.) (London)

STUDENTS

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Dhirendra Kumar Hajela, B.Sc.(Agra)
Douglas Roy Maxwell McBride
William Michael Stuart Parks
Peter Stringfellow
Amarjit Singh Vijan
Vinod Kumar Vohra

PROBATIONER STUDENTS

Jagish Chander
Dhir Chand Sharda

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

Harvey Addison
Clifford Somerville Harnett
Duncan Lyon
Benjamin Corbett Pester, Lieut. Cdr., R.N.Z.N.
C. S. Sundaram
Charles Herbert Bradwell Watson

TRANSFER FROM ASSOCIATE TO MEMBER

Matthew Dunnachie
James Whyte McCowatt
Edward Clark Ranson

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

John Cochrane
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Frederick Cunningham Turnbull

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Donald Henry Andrew
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Surendra Singh, Lieut., I.N.
John Strachan

TRANSFER FROM GRADUATE TO ASSOCIATE

Francis Hugh Ferguson

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER

James Leak

TRANSFER FROM STUDENT TO GRADUATE

David Warwick Dean
Garneth Cyril Alexander Ellis
Peter John Hambling
John Keith Wilson

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE

Peter Harvey Bamforth
Nicholas Bill Knowlton
Ian Robert Jamieson
James Brown Ramsay Shearer

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

Derek Anthony Hancock
Allan McLeod Hodgson

OBITUARY

Sir William Crawford Currie, G.B.E.

An appreciation by Sir Donald F. Anderson (Past President)

The immediate impression which Willie Currie made on everyone whom he met was that of a modest, kind, endearing man, most thoughtful for others. This was a true reflexion of his character, and only became stronger as one knew him better.

I worked under him in various capacities for over 20 years, and as a much younger man I was constantly astonished by his readiness to listen to the young, and his wish to let them have their say. No one could have been less of a dictator than he—no one could have filled his chair with less pomp or more consideration for others.

It is not surprising that with this character he should have evoked such loyalty and respect from those who worked in the companies under him. Whether at sea or ashore, whether at home or abroad, they all regarded him with unbounded affection, and he took immense trouble to get to know them.

Willie Currie was a tiger for work, and however long the day had been he never left the office without a bundle of papers to read at home. With an excellent memory and an apparently unlimited capacity for absorbing information, he had an immense knowledge of the detail of his business. He had little time for outside interests, but shooting, rugger, cricket, and those areas, such as India, where he had lived and worked as a younger man, always kept their attraction for him.

The P. and O. Group was the centre of his life. His thoughts and activities revolved around it, and he had the good fortune to be supported in this by Lady Currie, who took part with him in so many of his shipping and company activities, as well as making her individual contribution to many others.

To me—and a thousand others—Willie was a unique character. Those who served him will never think of him without warmth, for he was that sort of a man.

Sir William Currie, G.B.E., Chairman of the P. and O. Steam Navigation Company and of the British India Steam Navigation Company from 1938 to 1960, who was President of the Institute in 1945, died on 3rd July 1961 at the age of 77. One of the leading and best-loved figures in British shipping, Sir William was convalescing after a serious operation when he suffered a relapse and died.

Born in Calcutta on 4th May 1884, William Crawford Currie was almost born into shipping, for his father served in the Indian office of the British India Steam Navigation Company for 28 years and became a director. He was educated at Glasgow Academy, Fettes and Trinity College, Cambridge, where he gained a Rugby Blue. He inherited his father's love of India but he came to Britain at an early age, and it was not until he was 26 that he returned to the land of his birth.

After leaving Cambridge in 1906, he joined David Struthie, a firm of chartered accountants in Glasgow. He served his articles with them and qualified four years later. The call of India must have been insistent for he left immediately for Calcutta to become an assistant in the first Lord Inchcape's firm of Mackinnon, Mackenzie and Co., managing agents for British India Steam Navigation. He became a partner in this firm in 1918 and by 1922 was the senior managing partner. He soon began to take a prominent part in the public life of India. He was Sheriff of Calcutta in 1921-22 and was also elected to the Bengal Legislative Council, on which he served until he left India in 1925. A year before that he was elected president of the Bengal Chamber of Commerce and also president of the Associated Chamber of Commerce of India, Burma and Ceylon. During his last year in India he was appointed member of the Council of State for India and received a knighthood.

The second half of Sir William's business life opened on his return to the United Kingdom in 1926. He became a partner in Gray, Dawes and Company, the London agents of British India Steam Navigation. In the following year he became a member of the Imperial (now Commonwealth) Shipping Committee, on which he sat for four years. He was elected president of the Chamber of Shipping in 1929, when he was a director of James Nourse Ltd.

Soon after Lord Inchcape's death in 1932 he was elected a director of the P. and O. and later one of the managing directors. He became chairman in 1938 when Lord Craigmyle resigned and also of the British India Steam Navigation Company. The P. and O. lost nearly a million and a quarter tons of shipping, but even in 1945 it had enough ships to resume at least skeleton services. The reconstruction of the fleet was one of the most astonishing achievements in the records of British shipping.

From the outset of the Second World War until it ended, Sir William was a member of the Advisory Council of the Ministry of War Transport. In 1942 he became the director of the Liner Division at the Ministry, where he remained until 1945. In addition to his work in connexion with shipping, he served as a member of the executive committee of the British Red Cross Society and Order of St. John from 1942 to 1947 and also on the Red Cross Prisoners of War Committee from 1943 to 1946.

After serving his term as President of the Institute for 1945-46, Sir William became, in the following year, High Sheriff of Buckinghamshire. From 1946 until 1948 he was chairman of the British Liner Committee which co-ordinates all shipping matters between London and Liverpool. He was elected Prime Warden of the Worshipful Company of Shipwrights in 1949.

Besides his directorships of P. and O. and B.I., Sir William was a deputy chairman of Williams Deacon's Bank, an extraordinary director of the Bank of Scotland, chairman of the Marine and General Mutual Life Assurance Society, a director of the Suez Finance Company and of William Cory and Son Ltd.

He was chairman of the honorary committee of management of the training ship *Worcester*, a trustee of the National Maritime Museum, a member of council of King George's Fund for Sailors, a past president of the Seafarers' Education Service and an honorary member of the Honorable Company of Master Mariners. Sir William was elected a member of the Baltic Exchange in 1947. He became an underwriting member of Lloyd's in 1952 and underwrote marine and non-marine risks under the agency of Messrs. Gray Dawes Mackay and Co.

Sir William was created G.B.E. in 1947 and was appointed a Commander of the Legion of Honour of France in 1953. He was appointed an Honorary Captain, R.N.R., in February 1960. He took a lively interest in the local affairs of Dinton, the Buckinghamshire village where he lived for many years, and was vicar's warden of the parish church, besides holding many other offices in the neighbourhood.

He married in 1914 Ruth Forrest, daughter of C. S. Dods, by whom he had two sons, the elder of whom, Captain William Mackinnon Currie, 45th Cavalry, Indian Army, was killed in Burma in 1944.

Sir Westcott Stile Abell, K.B.E., M.Eng.

Sir Westcott Stile Abell, who was President of the Institute in 1924, died on 29th July 1961 at the age of 84. An eminent naval architect and constructor and a former Chief Ship Surveyor of Lloyd's Register of Shipping, his career was an example of triumph over misfortune which would have discouraged many lesser men.

At the age of 20 he lost his right hand and suffered very serious injury to his throat when he was lighting fireworks to celebrate Queen Victoria's Diamond Jubilee. Yet he made so rapid a recovery, teaching himself to write with his left hand, that he was able to continue his studies at the Royal Naval College, Greenwich and to pass out at the head of his year.

Westcott Stile Abell, the eldest son of Thomas Abell of Exmouth, was born on 16th January 1877, and educated at West Buckland School and the Royal Naval Engineering College, Keyham, Devonport, before proceeding to the Royal Naval College, Greenwich.

Joining the Royal Corps of Naval Constructors in 1900, he was soon appointed to the staff of the Chief Constructor at Devonport Dockyard but after some months there was called to the Admiralty to assist at the inquiry on the stability of the royal yacht, *Victoria and Albert*, and became a member of the Department of Naval Construction. From 1904 until October 1907, he was professional private secretary to Sir Philip Watts, Director of Naval Construction. At this time he was intimately concerned in the celebrated Committee on Designs appointed by the future Lord Fisher.

In October 1907 he was appointed junior Lecturer in Naval Architecture at the Royal Naval College, where he remained until December 1909, when he was selected by the University of Liverpool to be the first occupant of its Chair of Naval Architecture. While lecturer at Greenwich, and during his professorship, Sir Westcott contributed important papers to the Institution of Naval Architects. In 1913 he was appointed a member of the Committee of the Board of Trade to investigate the application of the Merchant Shipping Act to the question of the load line. He was chairman of the technical sub-committee, and the experience he gained and his familiarity with all matters concerning shipping caused him to be selected by the Committee of Lloyd's Register to fill the post of Chief Ship Surveyor.

This work began in May 1914, and on the outbreak of war, his assistance was sought by the Admiralty mainly to supervise the construction of the vast number of auxiliary craft required by the Navy. He resigned his professorship at Liverpool, being succeeded in the chair by his brother, Professor T. B. Abell. He served on a special Admiralty committee to consider the practicability of building submarine merchant ships during the time of the submarine menace. In 1916 he was appointed technical adviser to the future Lord Maclay, Controller of Shipping, and a member of the joint committee to allocate the distribution of steel for the purposes of Government Departments. Subsequently he became a member of the Shipbuilding Advisory Committee to assist in carrying into effect the mercantile shipbuilding programme.

After the war many questions arose which called for the ripe experience of Sir Westcott, especially at the Board of Trade for the classification of ships and other important matters. He served on the Board of Trade Load Line Committee in 1927 and was the British delegate at the International Conference on the Safety of Life at Sea two years later.

He retired from his position of Chief Ship Surveyor to Lloyd's in 1928 and for the next 13 years was Professor of Naval Architecture at Armstrong College, Newcastle upon Tyne.

Sir Westcott, who was created K.B.E. in 1920, was in addition to his Presidency of the Institute in 1924, Master of the Worshipful Company of Shipwrights in 1931, President of the Devonshire Association in 1933, and President of the Smeatonian Society of Civil Engineers in 1941.

Although President of the Institute so long ago, Sir Westcott always maintained an interest in the Institute of Marine Engineers and followed its development keenly. He had been a full Member since 1916.

Among his publications were "The Safe Sea", published in 1932, "The Shipwright's Trade" in 1948 and "The Ship and Her Work".

In 1902 he married Beatrice, daughter of J. W. Davenport, by whom he had one son and three daughters. His wife died in 1953. His son is Mr. T. W. D. Abell (Member), former Chairman of the Scottish Section and now Managing Director of William Doxford and Sons (Engineers) Ltd.

William Blackwood Johnstone, O.B.E.

An appreciation by D. D. McGuffie

Mr. W. B. Johnstone, son of the late Mr. William Johnstone, Lloyd's Register principal surveyor at Greenock, died on 20th August 1961, at the age of 67 while on holiday He was educated at Allan Glen's School, Glasgow, and served his apprenticeship at an engineer with A. and J. Inglis Ltd., Pointhouse. Thereafter he went to sea to gain further experience.

In 1915, during the first world war, he joined the firm of Alexander Stephen and Sons as assistant manager in the Repair Department. This position Mr. Johnstone held until 1928 when he was appointed manager. He joined the board of Directors in 1947 and was appointed an O.B.E. in 1951.

He was a member of the Ministry of Transport Select Committee on Oil Pollution and a past member of Lloyd's Register Technical Committee. In 1953 he was elected Chairman of the Dry Dock Owners' and Repairers' Central Council and he was past president of the Clyde Shiprepairers' Association. He was also a member of the Glasgow Chamber of Commerce and the Royal Institution of Naval Architects. He was recently appointed president of the Clyde Shipbuilders' Association and was a member of the Executive Committee.

His activities in the Scottish Section of the Institute of Marine Engineers are well known and for the past two years he held the position of a Vice-President of the United Kingdom.

Mr. Johnstone, during his early years, was a very keen and active member of the West of Scotland Cricket Club, and a founder member of the Clyde Cruising Club, Dinghy Section.

PATRICK NESSAN CRONIN (Associate Member 16330) was born on 1st December 1922. He was indentured between 1943-47 with George Watt Ltd., Dublin and from 1947-48 with the Commissioners of Irish Lights at Dun Laoghaire. In 1948 he began his seagoing career with the Blue Star Line, sailing finally as second engineer in both steam and motor vessels. In 1953 he obtained a First Class Ministry of Transport Combined Steam and Motor Certificate and took up employment with Insurance Engineers Ltd., Dublin, as engineer surveyor. He remained with this firm until 1955, when he joined The Irish National Insurance Co. Ltd. in a similar capacity. Two years later, Mr. Cronin obtained a post, again as engineer surveyor, with the Irish Fisheries Board and from there he was appointed an industrial inspector with the Department of Industry and Commerce of the Irish Government in 1958.

Mr. Cronin was elected an Associate Member of the Institute in 1955. He died of a heart attack in his office at the Department of Industry and Commerce on 25th January 1961 at the age of 38 years.

THOMAS GREY BOYS (Member 18424) was born at Sunderland, Durham, in 1901 and served his apprenticeship with Sunderland Forge and Engineering Co. Ltd. from 1917-22. During the period of his indenture he attended Sunderland Technical College for part-time instruction classes. From 1922 he acted as an electrical tester for two years with that company's test bed. Between 1924 and 1927 he served as a seagoing electrical engineer with the British India Steam Navigation Co. Ltd., prior to joining the Shaw Savill and Albion Co. Ltd. on 28th May 1928.

He served as second electrical engineer aboard the *Coptic* for one year, then joined the shore staff of Shaw Savill, serving in various capacities until September 1950 when he was appointed assistant superintendent engineer (electrical). Since that date he had been responsible for electrical maintenance and staff appointments for the company's fleet, and, in addition, all the electrical work for the 15 new vessels built since his appointment, including the *Southern Cross,* and the *Northern Star,* which is now building.

Mr. Boys was elected a Member of the Institute on 2nd January 1957 and had represented the Institute on the British Standards Institution Committee for Radio Interference Suppression and Marine Installations, since February 1959.

RICHARD HENRY BROWN (Associate 8876) died in March 1961 while on sick leave. Born on 30th December 1909, he served his apprenticeship with Smith's Dock Co. Ltd. of North Shields between 1926-1931. He entered the B.P. Tanker Co. Ltd. as a junior engineer in July 1932 having commenced his seagoing career about a year previously, with the Baltic Trading Co. Ltd. His first appointment as a chief engineer was in m.v. *British Vigour* in August 1947 and he continued serving in that rank in various B.P. motor ships until his decease. During his 29 years at sea with the company, which included the full span of World War II, Mr. Brown served in no fewer than 27 Diesel tankers. He was involved in enemy action while attached to m.v. *British Science* as second engineer when that vessel was sunk by hostile aircraft in the Mediterranean in April 1941.

Mr. Brown held a First Class Board of Trade Motor Certificate and had been an Associate of the Institute since 1939.

JOHN ERIC COLLIER (Probationer Student 23206) was accidentally drowned at Easter 1961. He was educated at Fareham County Secondary School for Boys, Hants., between 1955-60 and had obtained the General Certificate of Education in Mathematics, Physics, History and Human Biology at Ordinary Level. Apprenticed to the B.P. Tanker Co. Ltd. on their apprentice engineers' training scheme, he had commenced the Ordinary National Diploma course at Poplar Technical College in September 1960, and had a further year of study, one year's workshop training and 18 months seagoing experience to undergo at the time of his tragic death.

John Collier was elected a Probationer Student of the Institute in December 1960. He was in his eighteenth year.

JAMES LEWIS (Member 21867) died on 10th May 1961 aged 36 years. His apprenticeship was served at Cammell Laird and Co. Ltd., Birkenhead, with whom he spent five years. Following this he was employed by Shell Refineries for about three months and then went to sea for five months with the Pacific Steam Navigation Co. Ltd. He joined the B.P. Tanker Co. Ltd. (then British Tanker Co. Ltd.) as sixth engineer in September 1947 and served until June 1959 when he was forced to leave the sea owing to ill-health.

During his time with the company he successfully gained a First Class Ministry of Transport Motor Certificate with Steam Endorsement and was promoted to chief engineer (motor ships) in August 1956. Immediately prior to leaving B.P., Mr. Lewis was serving as second engineer on one of the company's supertankers.

He was elected to full Membership of the Institute in 1959.

FREDERICK JOHN POTTS (Member 10714) was born on 30th May 1904. After a five-year apprenticeship at the Walker-on-Tyne yard of Swan, Hunter and Wigham Richardson Ltd. from 1920-25, he went to sea. Between 1927 and 1935 he served for various periods with the Ellerman Hall Line, the Canadian Pacific Steamship Co. Ltd. and the Union-Castle Mail Steamship Co. Ltd. In 1935 he obtained his Ministry of War Transport First Class Certificate and took up employment as senior draughtsman with Foster Wheeler Ltd. Between 1936-40 he acted as technical adviser to Drew and Clydesdale Co. Ltd., London and from 1940-43 as chief designer and works engineer with the Walker Crosweller Co. Ltd., Cheltenham. In 1943 Mr. Potts joined the Chaseside Engineering Co. Ltd. of Enfield as works manager. From 1944 onwards Mr. Potts was sales manager for the Diamond Blower Co. Ltd. in London, a position he held until 1952 when he formed his own business of F. J. Potts and Partners at Wembley, Middlesex.

Mr. Potts died suddenly on 10th August 1961. He had been a full Member of the Institute since 1946 and was also a Member of the North East Coast Institution of Engineers and Shipbuilders.

PIETER VAN STRIEN (Associate 22796) died suddenly at sea on 20th July 1961, while sailing through the Suez Canal and was buried the following day in Cairo. At the time he was employed as second engineer in *London Victory,* a vessel of London and Overseas Freighters Ltd., a company he had joined three months previously. He was educated at Amsterdam Technical College, passing out in 1929 with a Diploma B, issued by the Netherlands authorities. He was apprenticed to Ducroo and Brown, Weesp, Holland from March 1929 till July 1930 and went as assistant engineer to the Holland-America Line from September of that year to December 1932. Thereafter Mr. van Strien served with various Dutch shipping companies till 1937, when he became plant engineer with N.V. Holland-Syndicate at Amsterdam until 1940. In October 1940 he was commissioned in the Royal Netherlands Navy as sub-lieutenant (E) and served throughout the Second World War in Dutch submarines, being demobilized in September 1945 with the rank of lieutenant (E).

Mr. van Strien then returned to the Holland-Syndicate for eight years, where he was placed in charge of the production unit. In 1954 he worked for nine months as an engine fitter at H.M. Dockyard, Portsmouth and in September of that year he went to the Gold Coast (as it was then) to join the Ankobra River Power Company as maintenance engineer and acting superintendent. On 6th December 1958 he was transferred, within the same group, to the Ariston Gold Mines as installation engineer; there he had responsibilities for the installation of hoists and Diesel engines, etc.

Elected an Associate of the Institute in October 1960, Mr. van Strien held a Second Class Ministry of Transport Combined Steam and Motor Certificate. He was 54 years of age.