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Factors Governing the Design of a Modern Tanker with Special Reference to Machinery

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The paper is written to illustrate principally that the major oil companies, when developing a class of vessel for a building programme, must give much consideration to the trade for which the vessels are intended and apply all the results of collated data in conjunction with the results of experience of the company's technical staff, if the tanker is to be a capable and efficient freighter of the many and varied products of the present-day refineries.

The principal factors governing the design are set down firstly and the author endeavours to show, by referring to these factors throughout the paper, the development of a tanker from the initial stages to the completed vessel giving, as far as the scope of the paper will allow, the multifarious reasons for the decision with special regard to type of both main and auxiliary machinery.

It is not intended to make comparisons between different types of tankers or their propelling machinery. The paper does however make bold to reveal the fact that a tanker which is capable of carrying and handling all grades of petroleum in a safe and expeditious manner must be capable of producing an abnormal amount of auxiliary steam which must influence an owner seriously to consider steam propulsion.

In the development of tankers a stage has been reached where a distinction must be made between tankers produced by the oil industry for the industry and those produced by shipowners purely for chartering to the oil companies for clean or dirty cargoes, or for the carriage of crude petroleum in vessels specifically designed for that purpose.

This paper is intended to deal entirely with the former vessel which is the result of gradual development through the years in order to meet increasing demands for the freighting (without fear of admixture) of the many and varied products from the up-to-date refineries, added to which is the urgent necessity of freighting at the lowest possible cost and for the shortening of inefficiency periods for repairs and overhauls to the absolute minimum.

Shipping lines, maintaining specialized trades, must build their vessels to suit in every way as well as possible the needs of that trade, so similarly must the major oil companies give very careful consideration to the various factors governing the trade for which the type of tanker is intended.

The procedure then is that as much essential information as possible is collected in order to compile complete statistics covering a number of proposals as regards general design until the most attractive size and type of vessel is arrived at. This information is segregated under the following headings:—

(a) The total estimated quantities of petroleum products in bulk which are to be carried per annum.
(b) The quantities of each grade required at each respective port and number of the ports of discharge.
(c) Characteristics of the ports to be served, with special regard to depth of water available and tidal conditions.
(d) Facilities for loading or receiving the various grades.
(e) Meteorological conditions of the trade between ports, such as prevailing winds and currents.
(f) Quantity of packed oil products required to be freighted to any of the respective ports.
(g) Tankage available at receiving ports.

On the foregoing data becoming known, statistics can then be made available which will decide the deadweight of the proposed vessel, also disposition of the cargo tanks, together with the segregation of sections in order to carry the respective grades without fear of admixture. Further the speed of the vessel can be estimated in order that the number of voyages per year can be fixed.

The following brief outline will give some idea of how these statistics are compiled:

Territory	Grade	Supply point	Quantity to be shipped	Tanker tonnage Clean	Tanker tonnage Dirty
				Days required	
U.K.	Gasoline				
	Kerosine				
	Luboil				
	Gas/Diesel				
	Fuel				
	Crude				
Plate	Gasoline				
	Kerosine				
	Gas/Diesel				
	Fuel				
	Crude				
Brazil	Gasoline				
	Kerosine				
	Luboil				
	Gas/Diesel				
	Fuel				
Cuba	Gasoline				
	Kerosine				
	Gas/Diesel				
	Fuel				
Chile	Gasoline				
	Kerosine				
	Gas/Diesel				
	Fuel				

Equivalent to X type vessels for 1 year—Clean..............

Dirty..............

Comparative freighting costs of tankers Curaçao/London or South America

	15,000 D.W.T. Turbo-electric	*Other types*
Value		
Insured for		
Depreciation—per annum		
Interest—per annum		
Running costs—per d.w.t. month (including interest and depreciation)		
Speed—knots		
Consumption—per day		
Bunker price at Curaçao—per ton		
Freight—per cargo ton/Curaçao/London or South America		

The position has now been reached whereby the principal dimensions and coefficients of the hull are calculated; number, disposition and size of cargo compartments; size of machinery spaces; number and sizes of pump rooms, cofferdams, etc.

From these results a fairly close approximation is obtained of the e.h.p. and b.h.p. required to give the selected hull the desired service speed and also that to give the required trial trip speed over the measured mile.

The procedure on embarking on the building of a programme of a class of ships to cover a special trade is as follows:

Draw up a rough specification in line with the findings;
Draw proposed midship section;
Draw proposed outline profile;
Draw proposed general arrangement;
Submit the midship section and general arrangement, also outline profile plans to Lloyd's Register of

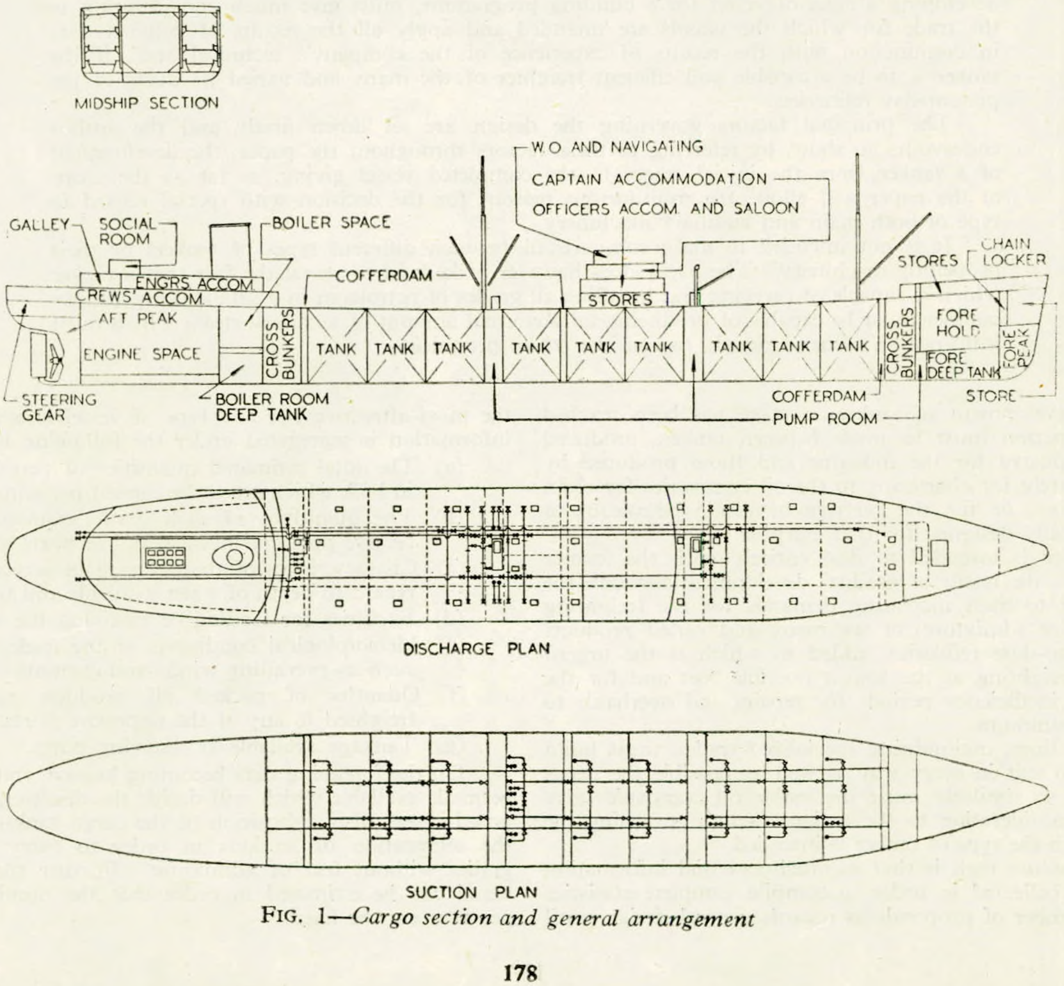


FIG. 1—*Cargo section and general arrangement*

Shipping for their approval as to principal dimensions and general outline.

It will be assumed that all the foregoing considerations (a) to (g) have been satisfied by a tanker coming under the following outline specification:

Length overall...	535ft. 00in.
Length between perpendiculars	510ft. 00in.
Breadth (extreme)	69ft. 00in.
Depth (extreme)	37ft. 00in.
Block coefficient	0·745
Sheer forward	7ft. 00in.
Sheer aft	5ft. 00in.
Height of poop and forecastle	7ft. 06in.
Height of bridge house ...	8ft. 00in.
Height of accommodation deck houses	7ft. 06in.
Estimated draught at 15,000 d.w.t.	29ft. 00in.
giving calculated freeboard of	8ft. 00in.

It is clear from the tentative specification that careful consideration has had to be given to a few paramount features, namely:—

Due to the trade demanding composite cargoes of various grades which must be completely isolated, it has been necessary to segregate the cargo compartments into four distinct sections, with two pump rooms which form isolating cofferdams in themselves.

The size and disposition of each cargo tank has been governed by the requirements revealed under (b) in order that the vessel can serve each port with the respective grades without getting abnormally out of trim and creating difficulties by the unwelcome introduction of compensating ballast.

It will be seen later in the paper concerning the question of the cargo pumping system, that the system has had to be married into the disposition of cargo grades, coupled with the requirements under (d)—loading and discharging facilities.

Referring to the general arrangement it will be seen that as much of the hull, between the forward and after cofferdams, as possible has been utilized for cargo in order to gain the utmost volumetric cargo capacity in accord with permissible deadweight for this size of vessel, allowing for a full cargo of light refined oil of the order of 49/52 cu. ft. per ton. It will be observed that the bunker capacity has been reduced to the limit necessary for the radius of action required.

The space left for machinery certainly gives a measure of concern especially when one considers the fining away aft of this relatively high-speed tanker, and a glance at an outline sketch of the machinery space available will make this apparent (Fig. 3).

CONSIDERATIONS GOVERNING TYPE AND POWER OF PROPULSIVE MACHINERY

Under (a), (b), (e) and (f) it has been calculated that to maintain the supplies to the various ports feeding the respective marketing areas, the vessel must maintain a service speed, port to port, of 14¾ knots. It is therefore necessary to aim at a brake horse-power of the main machinery which will ensure that the speed can be maintained against known adverse currents and winds, and also have a margin to make up for any loss of speed due to inclement weather conditions which may be encountered.

What type of machinery shall be decided upon? It must be made quite clear at this stage that no attempt is here being made to make general comparisons between Diesel and steam turbine machinery, each form of drive is taken on its merits for this particular case. In deliberating on this matter one must bear in mind a distinctive feature of a tanker—the very high steam requirements over and above main engines compared with a dry cargo vessel, so that this factor is coupled with the problem of space available due to:

(a) Propelling machinery right aft;
(b) The extra fining of this ship in line with the required block coefficient.

One must, of course, not neglect the question of weight per b.h.p. and its effect on trimming levers.

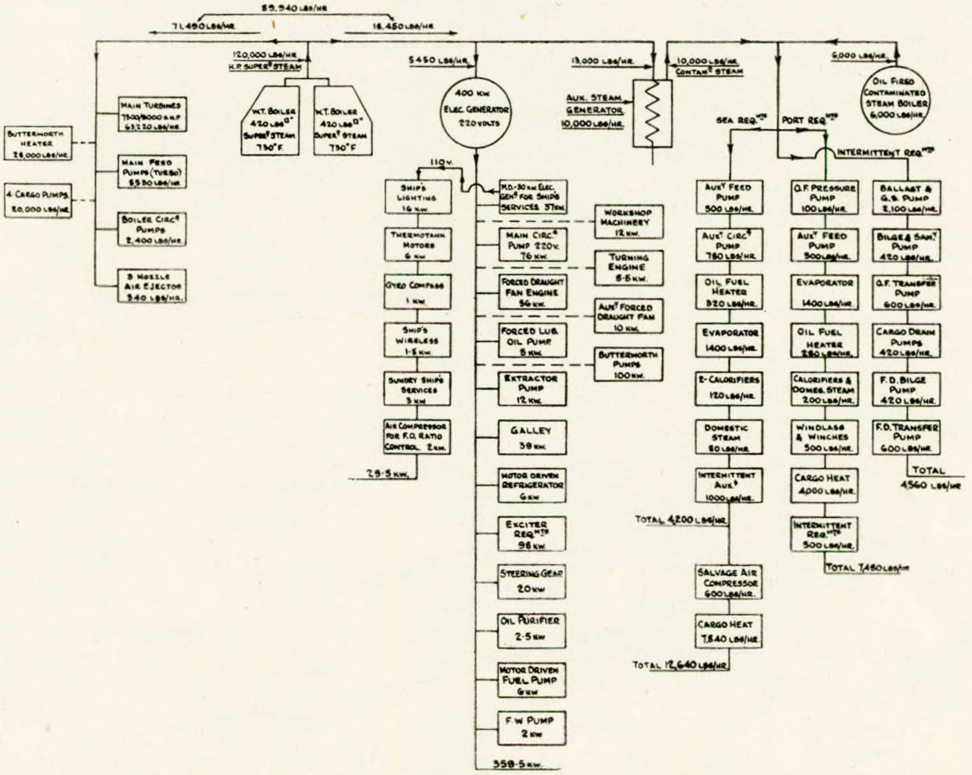


FIG. 2—*Power tree*

Proposed 15,000 d.w.t. tanker. Power requirements at normal service speed at sea and also in port.

Power Distribution

Set down the estimated steam consumptions by compiling what might be termed a "Power Tree" which will give a picture of power requirements and at the same time enable one to decide on the type and source of power for the many ancillary and auxiliary machinery involved.

Fig 2 shows a tree formed tentatively to indicate calculated quantities of steam, clean and contaminated, electrical power including that generated by internal combustion engines.

This clearly illustrates the fact that to make the tanker capable of handling petroleum from the lightest and most volatile grades to the heaviest and most viscous, and also meet the full requirements of service in every way, it is necessary to legislate for a maximum supply of steam, either high-pressure superheated or dry saturated at relatively low pressure, as follows:

(a) 7,840lb. steam per hour heating cargo.
(b) 20,000lb. for say four cargo pumps at 5,000lb. per hour each.
(c) Galley and hot and cold water services, 750lb. per hour.
(d) Accommodation heating and ventilating, 500lb. per hour.
(e) Water heaters for Butterworth tank cleaning system, 28,000lb. steam per hour.
(f) Pumps for Butterworth tank cleaning system, 1,250lb. per hour.
(g) Deck auxiliaries, 2,000lb. per hour.
(h) Electric lighting, 240lb. per hour.

From this it will be accepted (ignoring the question of power for propelling the vessel) that to provide the quantity of steam given above, the number and size of, say, Scotch type boilers would be such that one is forced to the conclusion that it would be profitable to make the main propulsive machinery steam-driven and so interlace the source of power, thus obviating a relatively large steam generating plant for auxiliaries as is the case of a motor tanker, which is practically divorced from the main propelling machinery and only fully in use intermittently.

The question of space can now be given consideration and in tackling this problem it is known, firstly, that given a machinery space of 97 feet in length one can accommodate high pressure steam machinery of the geared turbine type or turbo-electric, but one must see if it is possible to fit in an accepted relatively slow running Diesel engine of 9,000 s.h.p. Some of the well-known single-acting engines (even supercharged), and also the opposed piston type have to be abandoned as it is not possible to get engines of the required power into the space available. One could probably arrange to install a double-acting, two stroke engine in the space available, but the author feels sure that the manufacturers of these engines in this country would not recommend this type of engine for tanker service with full confidence.

The following two example (Fig. 3) will suffice to show that the Diesel engine, the first a 9,000 b.h.p. two-stroke single-acting, and the second, a 6,600 b.h.p. two-stroke opposed piston type, covered too large an area in way of the bedplate connexion to top tank. Also the many and varied steel obstructions at other levels and the fact that deck space had to be found for two Scotch boilers of at least 36,000lb. per hr. capacity (which figure has been shown to be inadequate) make for further difficulties in installing this type of machinery.

The question of weight can be left in abeyance for the present as the author will give a full list of weights of the machinery and boilers decided upon later, which work out at 141lb. per s.h.p. which weight includes all ancillary machinery. This information should satisfy the most ardent Diesel advocate that this weight cannot be approached with slow running Diesels.

From the foregoing it will be gathered that there is now a choice between geared turbines and turbo-electric installations, and there are many advantageous features in both types. However, in considering the following there is a bias towards the turbo-electric form of drive:—

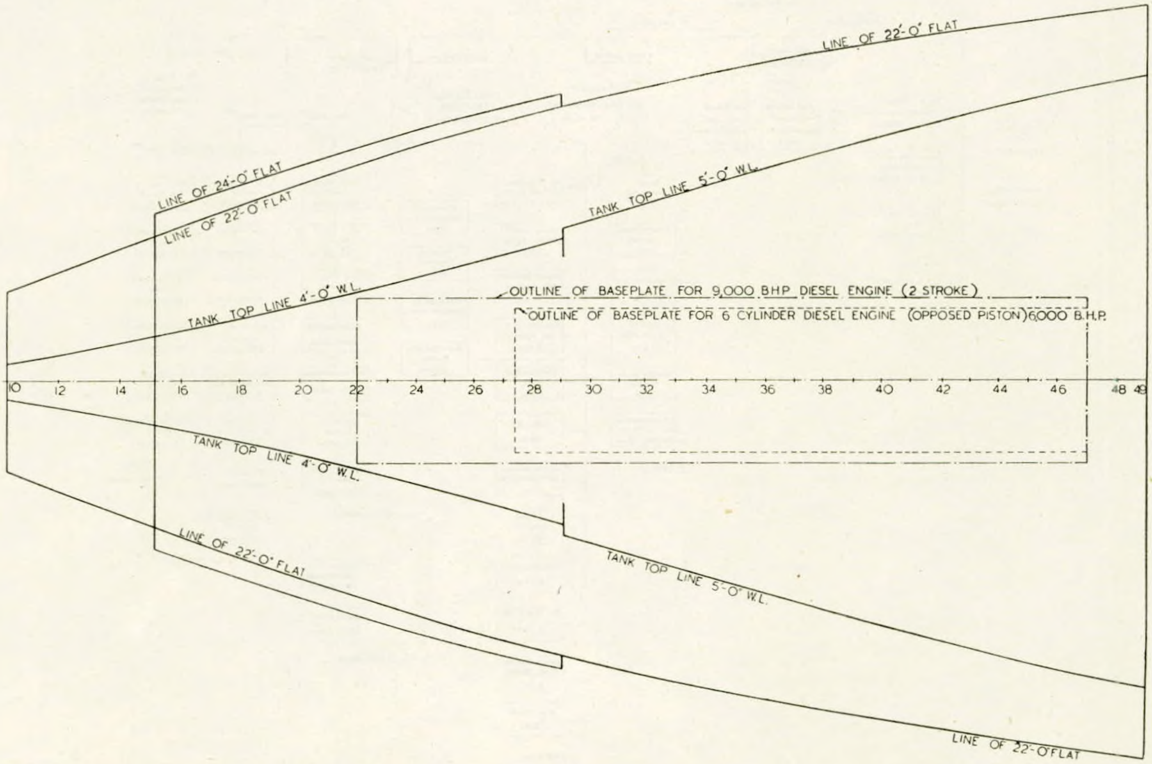


FIG. 3—*Comparison of space occupied by Diesel machinery*

	Geared Turbines	*Turbo-electric*
Efficiency	Overall efficiency stop valve to shaft coupling of: turbine 79·5 per cent, gearing 97 per cent is claimed for both single and double reduction geared turbines of latest design. This, of course, is based on torsionmeter readings at the main shaft and does not account for astern turbine. It must be accepted that the turbine must be cross-compounded making for split up in the drive, say, h.p., i.p. and l.p. This is essential to accommodate the prime movers to the propeller drive in true and rigid alignment.	Overall efficiency stop valve to shaft coupling of: turbine 80 per cent, alternator 96·8 per cent, motor 97·15 per cent, is claimed for this type of machinery and this is based on absolutely reliable electrical readings of h.p. being developed by the propulsion motor. An obvious advantage is that the turbine is a single straight flow unit similar to power station installations, making possible to utilize the best adiabetic heat drop at appropriate revolutions with steam pressures and temperatures. One way rotation of turbo-alternator gives advantage over geared turbine. Very low lubricating oil consumption.
	Weight for 7,200 s.h.p. The average weight of main propulsion units, i.e., main turbines, gearing and condenser is 155 tons.	*Weight for 9,000 s.h.p.* The actual weight of main propulsion units, i.e., main turbo-alternator, condenser and propulsion motor is 178 tons.
Installation	As previously mentioned the machinery must be designed to accommodate the ship. Stools and seatings of turbines and gears must be the best of steel work, above all, rigid. Alignment must be correct to fine limits.	The turbo-alternator being only connected to propulsion motor by high tension cable can be placed as convenient and does not require the amount or type of steel work as support and seating as does the geared turbine. Main propulsion motor seated on top of relatively rigid tank top and is the only unit requiring any measure of alignment on board the ship.
Flexbility		Turbo-alternator can be warmed up by running the machine slowly without moving the propeller and further can be run up to overspeed to test governing and overspeed trips. Full astern power.
Reliability	Established.	Established.

HULL

Ship Design Features

In considering the draught limitations imposed under (c) it will be observed that one has to increase the length to beam ratio to something of the order of $\frac{L}{10} + 18$ which the author feels is reaching the upper limits if one is to achieve satisfactory results, also to ensure that the vessel will not turn out to be a heavy roller in actual seaway. It is well known that the relatively shallow draught beamy tanker is not a kindly sea boat in any sort of heavy weather. Although some recently built American tankers have ratios of $\frac{L}{10} + 24{\cdot}5$, the author's personal reaction is that from several points of view this ratio is too high.

The principal dimensions and dispositions of cargo compartments under (a), (b) and (c) are now known and the design of the midship section can now be outlined, also fore and aft steel plans for presentation to Lloyd's Register of Shipping.

In formulating these plans, past experience is applied and in this connexion a passing reference to some of the steel work troubles in tankers may prove of interest.

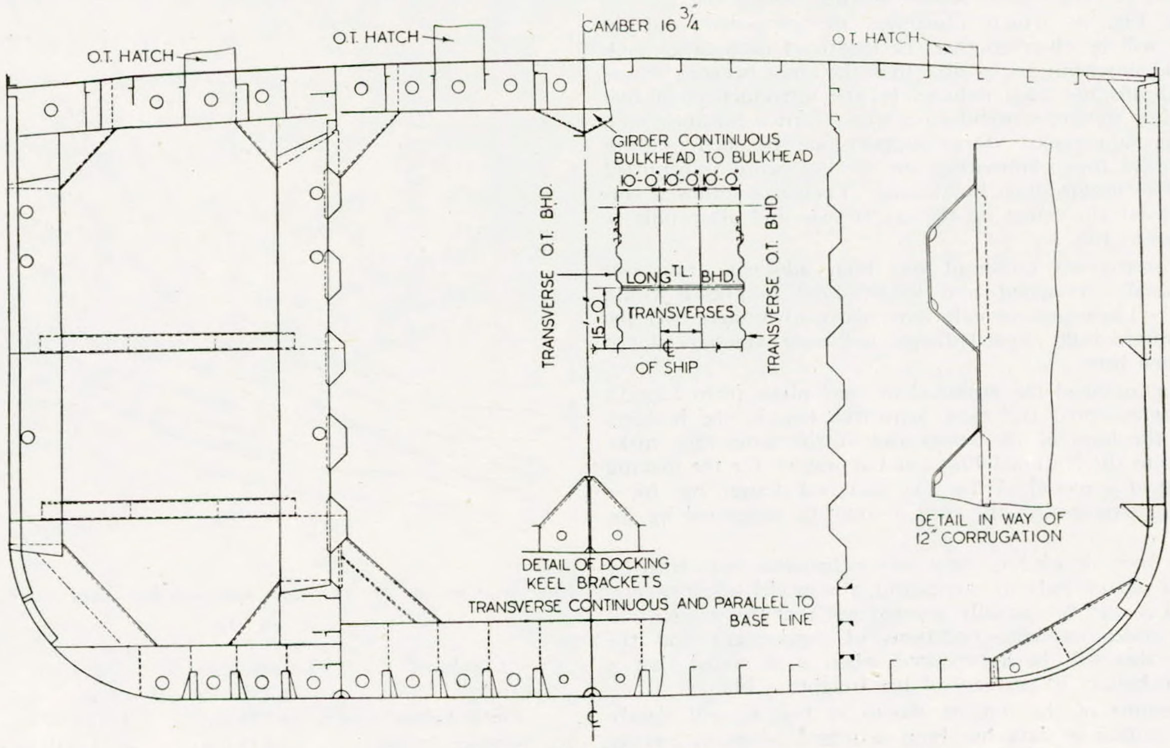


FIG. 4—*Midship section*

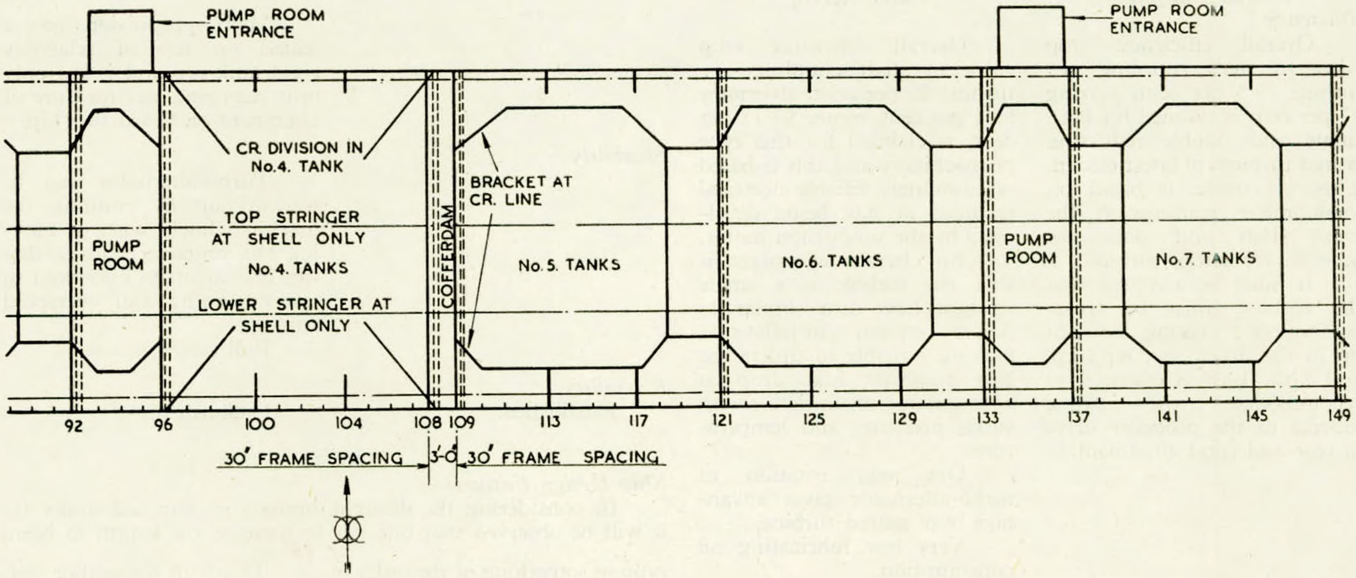


FIG. 5—*Longitudinal profile*

It is now an accepted fact that certain shell, stringer and bulkhead troubles which have been experienced through the years can be attributed to flexing in way of the non-rigid areas, and in this connexion it might be mentioned here that Lloyd's Register specifically set an upper limit on length to depth ratios, this being 14. Should this number be increased the scantlings must be specially considered. They also pay special attention to length of spans between the rigid points.

In past designs there has been a tendency on the designers' part to produce definite hard spots which are not in accord with the large spans and/or areas between these spots. This inconsistency in the distribution of strength has also contributed in fair measure to the rapid internal deterioration of steel work in way of cargo spaces when carrying refined oils.

From Fig. 4, which illustrates the proposed midship section, it will be observed that the length of each cargo tank has been kept within limits and that the span between transverse bulkheads has been reduced by the introduction of two complete deep transverse web frames which form a complete arch ring in the wing tanks. Strut supports are attached to these rigid members thus eliminating any flexing being transmitted from shell to longitudinal bulkheads. They had followed this policy to a certain extent on the centre fore and aft profile as will be seen in Fig. 5.

The corrugated bulkhead has been adopted, transverse being vertical corrugated and longitudinal bulkheads being horizontal. These undoubtedly have many advantages over the normal straight bulb angle stiffened bulkhead which need not be mentioned here.

Having received the approval of steel plans from Lloyd's and having accepted the most attractive tender, the builders formulate the lines of the vessel and at the same time make application to the National Physical Laboratory for the making and testing of a model of the now accepted design, but for a few minor alterations to the runs as may be suggested by the N.P.L.

It has been decided to have very exhaustive tests both by towing and self-propulsion carried out, also model screw experiments, and it may be specially pointed out that this model will be tested under varying conditions of deadweight and the reason for this will be appreciated when it is stated that a tanker is in ballast 50 per cent of her freighting life.

The results of the test, as shown in Fig. 6, will clearly show that sufficient data has been acquired whereby, having decided on the final hull form to give the best efficiency under fully laden conditions, the knowledge gained can now be applied to regulating deadweight and trim at various seasons and on various stages of a ballast passage, having proved the tank experiments against actual full scale trials.

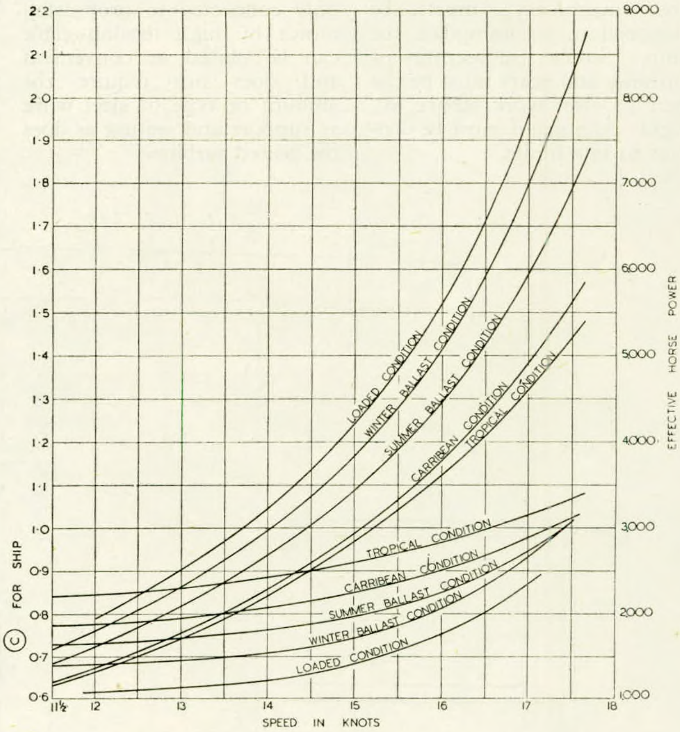


FIG. 6—*Curves of effective horse-power and corrected* © *in ship*

Condition	Displacement	Trim
Loaded	21,770 tons	Level
Winter ballast ...	16,725 tons	4/510 by stern
Summer ballast...	12,835 tons	8·75/510 by stern
Tropical	7,680 tons	10·25/510 by stern

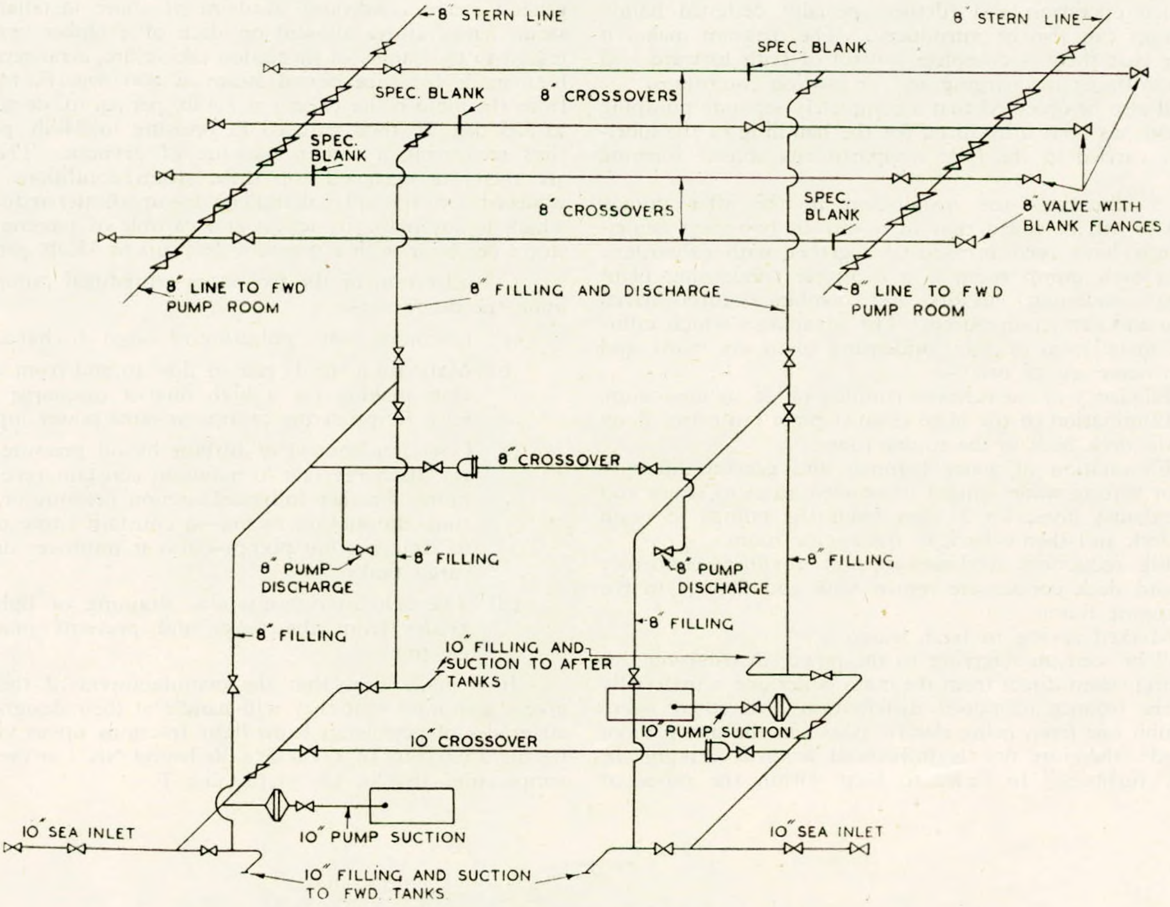


FIG. 7—*Diagrammatic arrangement of cargo piping in after pump room*

Final Hull Form

Length b.p.	510ft. 00in.
Breadth moulded	69ft. 00in.
Draught (designed LWL)	29ft. 02¼in.
Length-breadth ratio	7·39
Breadth-draught ratio	2·365
Displacement	21,839 tons
Block coefficient	0·743
Midship section coefficient	0·971
Prismatic coefficient	0·765
Designed sea speed (service)	14¾ knots
Rough position of centre of buoyancy	0·843 per cent forward of amidships

Propeller

A solid bronze propeller having been designed to absorb efficiently the prescribed horse-power, an exact replica model has been tested at the National Physical Laboratory, the final design being as follows:

Diameter	18ft. 06in.
Pitch (non uniform) mean	12ft. 06in.
Surface on four blades	145 sq. ft.

Cargo Pumping

As has already been decided under (b), this ship must be a four section vessel (as shown in a previous Fig.) and one must now give attention to the number and type of cargo pumps most suited for the handling of the component cargoes and consistent with governing conditions as regards the type of power to drive these pumps. One must also plan the pumping arrangement so that complete and safe isolation may be achieved as between grades and at the same time one must be able to avail oneself fully of the vessel's intake and discharging capabilities to maintain a high rate of loading and discharging.

Figs. 7 and 8 show the pumping arrangement inside the after pump room diagrammatically, from which it will be observed that the aim has been to have at least two valves

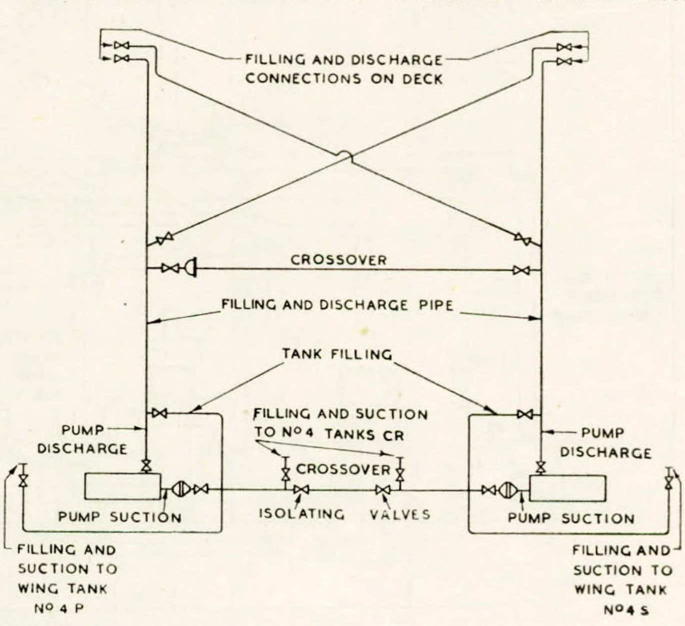


FIG. 8—*Diagrammatic arrangement of lubricating cargo oil piping in after pump room*

against each operation and further specially designed handy blank fittings can also be introduced. The diagram makes it quite clear that there is complete control of both forward and after section under discharging and/or loading conditions.

It will also be observed that a completely separate pumping arrangement has been introduced for the handling of the lubricating oils carried in the four compartments abreast forming N.4 tank.

Fig. 9 illustrates the installation in the after pump-room and it will be noted that turbo-driven two-stage centrifugal pumps have been introduced together with exhausters. Further in each pump room is a complete condensing plant comprising condenser, ejectors and combined turbo-driven circulating and extraction pump. The advantages which influenced the installation of this condensing plant are many and varied—to name one or two:—

Efficiency of the turbines running on 25 inch vacuum.

Elimination of the huge exhaust pipes returning along the deck back to the engine room.

Elimination of water hammer and general difficulty of getting water caused by condensation in steam and exhaust lines, up 37 feet from the pumps to main deck and thence back to the engine room.

Big reduction in dimensions of auxiliary condenser and deck condensate return tank and fittings in the engine room.

Marked saving in fresh water.

It will be seen on referring to the power distribution tree that by using steam direct from the main boiler one is materially assisting the balance of power distribution. Accepted safety rules prohibit one from using electric power in the pump room as designed: therefore one is influenced towards driving the pumps by turbines. In order to keep within the range of general steam conditions available at shore installations, also steam temperatures allowed on deck of a tanker, with special regard to the danger of insulation taking fire, arrangements have been made for superheated steam at 800 deg. F. to be taken from the main boiler drums at 500lb. per sq. in. de-superheated to 505 deg. F. then reduced in pressure to 240lb. per sq. in., thus recovering a certain measure of dryness. The turbines are therefore designed for these steam conditions. This is achieved by specially designed de-superheater-reducing unit which is automatic in action and capable of passing 30,000lb. steam per hour with a pressure drop up to 350lb. per sq. in.

Consideration of the two-stage centrifugal pump is based upon the desire to:—

(a) Eliminate heavy pulsating of cargo discharge lines.

(b) Maintain a steady rate of flow to and from the pump, thus making for a high rate of discharge compared with reciprocating pumps at same power input.

(c) Governor control of turbine by oil pressure when set for discharge rate to maintain constant revolutions of pump whether full head suction pressure or draining, thus eliminating racing—a constant cause of damage to reciprocating pumps—also it improves draining of cargo tanks.

(d) The exhauster pump aids draining of light-fraction grades from the tanks and prevents pumps from gassing.

It is made clear that the manufacturers of these pumps give a guarantee that they will handle at their designed output all grades of petroleum from light fractions up to viscous oils having a viscosity of 3,000 secs. Redwood No. 1 at the pumping temperature, that is, 130 to 140 deg. F.

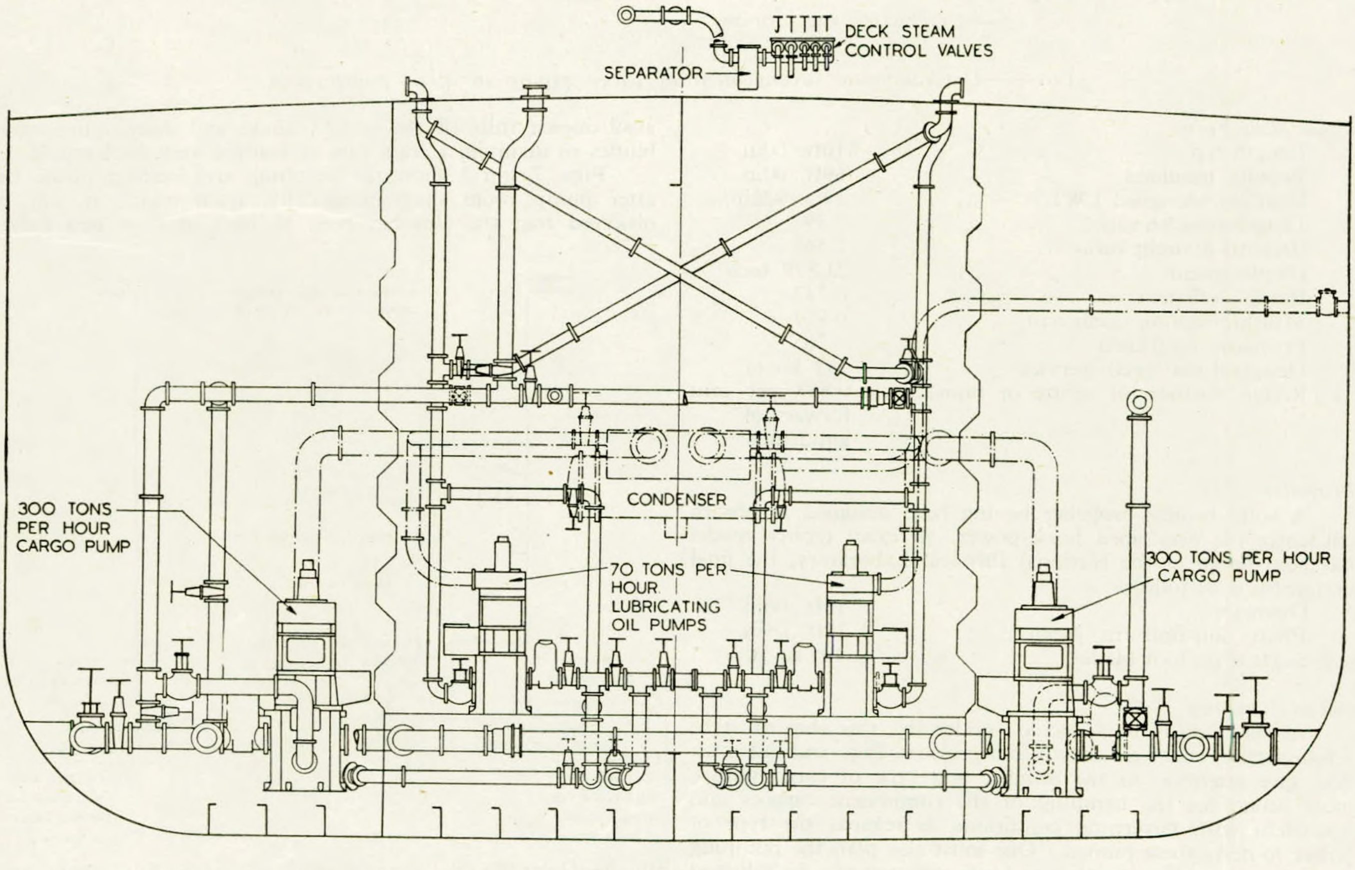


Fig. 9 (*a*)—*Arrangement of pump and pipelines in after pump room*

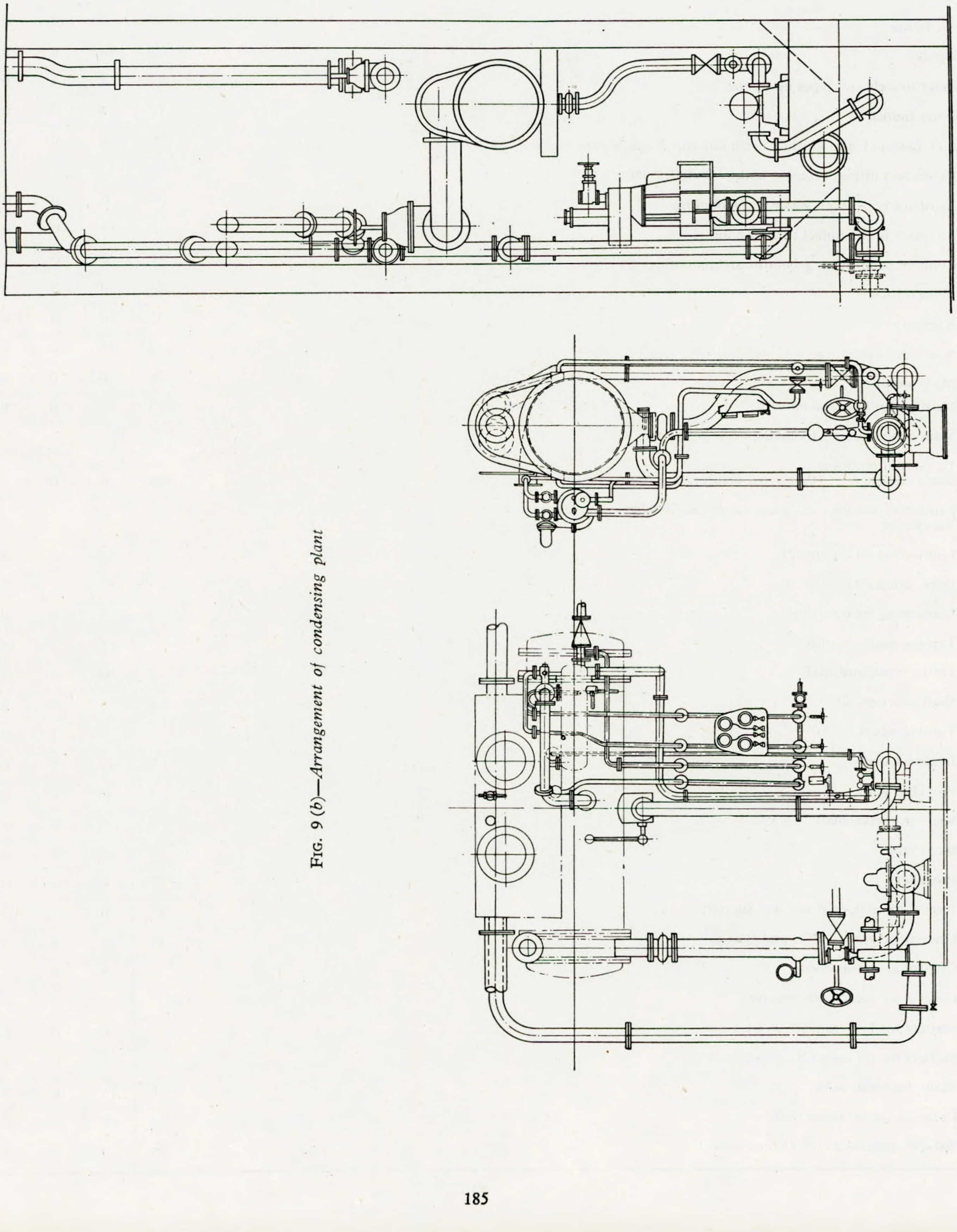


FIG. 9 (*b*)—*Arrangement of condensing plant*

TABLE 1 WEIGHTS OF

Description	Tons	Cwt.	Qrs.	Lbs.
2 main boilers and circulating pumps (with water)	108	0	0	0
		15	6 tons	
Air heater	48	0	0	0
Uptake	2	10	0	0
Inner funnels and pipes up funnel	13	2	1	0
Outer funnel	11	4	0	0
F.D. fans and cold air duct (each fan unit 2 tons 9 cwts. 0 qrs. 14 lbs.)	9	0	0	0
Valves and fittings on main boilers (included above)	—	—	—	—
Auxiliary boiler with water and mountings	36	0	0	0
Auxiliary boiler funnel to casing top	2	15	0	0
Contaminated steam generator and mountings	11	2	0	0
Main turbine	34	0	0	0
Alternator	32	0	0	0
Alternator cooler	2	10	0	0
Main motor	76	0	0	0
Main motor fans and coolers	5	0	0	0
Main condenser — Including W.S. pipes main cir.				
Turbo feed pumps Drain cooler — Pumps, de-aerator, de-aerator condenser	90	with 0	water 0	0
Extraction pumps Air ejector — etc. evaporators and distillers, feed heaters				
Lubricating oil pumps (2)		12	0	0
Lubricating oil coolers (2)		18	0	0
Lubricating oil centrifuge		8	0	0
Turning gear complete		8	3	10
Thrust block and shaft	10	10	0	0
Shaft bearings (2)	2	3	0	0
Turning wheel		9	0	0
Intermed shaft	6	10	0	0
Propeller shaft	9	0	0	0
Spare propeller shaft	9	0	0	0
Stern tube	4	12	2	0
Propeller	17	0	0	0
Engine room floor plates and supports	12	0	0	0
Underfloor pipes, valves and fittings	17	5	0	0
Control air compressor		5	0	0
Control air compressor receiver		5	3	0
Starters for F.D. fans (2) 3 cwt. each		6	0	0
Starters for oil units (2) 2½ cwt. each		5	0	0
Main discharge valve	1	10	0	0
Daily service oil tanks (full)	3	13	0	0
400 kW. generators (2) 13 tons each	26	0	0	0

MACHINERY				
30 kW. generators (2) 1-17-2 each	3	15	0	0
100 kW. diesel generator	9	3	0	0
Boiler fuel oil tank	4	10	0	0
Auxiliary condenser	2	10	0	0
Air pump		18	2	0
Auxiliary feed heater	1	0	0	0
Observation tank	1	6	0	0
Heat exchanger		3	0	0
Diesel oil centrifuge		3	0	0
Switchboard	3	0	0	0
Diesel fuel oil tanks (2)	11	5	0	0
Distilled water tanks (full)	52	0	0	0
Lubricating oil gravity tanks (2)	8	18	0	0
Lubricating oil storage tank	6	0	0	0
Cascade filter	7	10	0	0
Auxiliary feed pumps	1	12	1	0
Transfer pump	2	16	0	0
Auxiliary circulating pump	1	6	0	0
Air receiver for 100 kW. generator		4	0	0
Air compressor for 100 kW. generator		10	0	0
Oil burning units	1	8	0	0
Auxiliary boiler fan		5	0	0
Pulsometer pump 100 kW. generator		8	0	0
Evaporator feed pump		5	0	0
Main circulating pump starters (2)		10	0	0
Extraction pump starters (2)		4	0	0
150 gallon condensate tank	1	10	0	0
Renovating tank		6	0	0
100 kW. generator daily service tank		8	0	0
100 kW. generator fresh water tank		2	0	0
H.T. cubicle and control desk etc.	2	10	0	0
Head tanks for boiler circulating pumps (2)	3	3	0	0
Spare armature	1	12	0	0
Spare boiler circulating pump	1	12	0	0
30 kW. generator starters (2)		4	0	0
Steam feed and exhaust pipes and fittings	25	0	0	0
100 kW. generator pipes and fittings	1	10	0	0
Circulating and discharge pipes and fittings	18	0	0	0
Lubricating oil pipes and fittings	3	0	0	0
Cables and electrical gear not listed, approximately	10	0	0	0
Machinery weight =	794	11	0	10

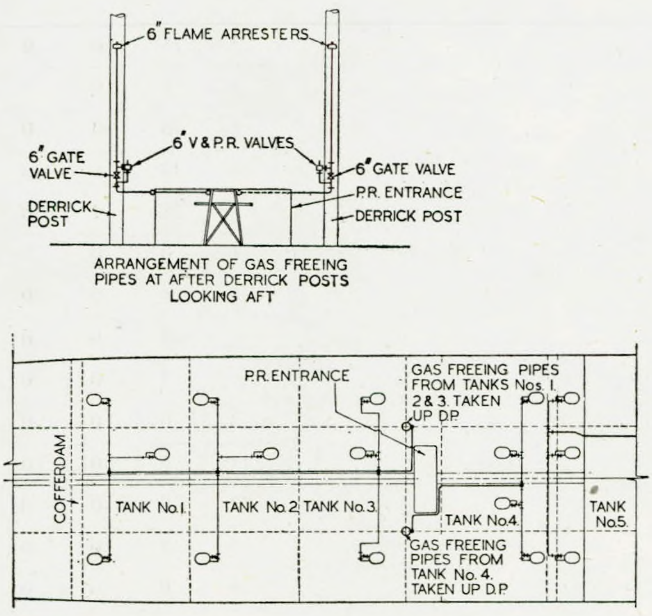


FIG. 10—*Venting. Typical gas-freezing arrangement*

Cargo Venting

It will be appreciated that if the vessel is to be capable of carrying the lightest and most volatile grades, special attention must be given to the venting of the cargo compartments, and in this connexion safety standard regulations are set as issued by the Panama Canal Authorities and all tankers passing through the Canal must comply with Grade A, B or C according to the cargo being transported through the Canal.

Venting is designed to meet the highest grade, thus enabling one to carry volatile cargoes having a Reid's pressure of 14lb. and over.

Fig. 10 shows the arrangement which it will be agreed makes for the maximum of safety, yet allows flexibility when required for isolating against admixture by direct intertank overflow or gas intrusion and condensing.

The flame arrester should be specially mentioned as it meets the full Buxton test for flame arresting and is not merely a spark arrester of the gauze wire type.

Fig. 11 will show that the arresting is done by units forming a spiral annular space of prescribed depth and made from silver steel ribbon.

Masts and Derricks

The results achieved by adopting what are known as "goal post masts" in a tanker are:

(1) Elimination of a set of samson posts in way of the after discharge position.
(2) Utilization of the masts as eduction vents from pump room aft.
(3) Very good leads for tank venting pipes.
(4) Good lead of jack stays for windsail venting.
(5) Excellent arrangement of derricks at the fore dry cargo hatch.
(6) Unimpeded view ahead from navigating bridge.

Accommodation

It is profitable policy for both owners and crew that accommodation in a tanker should be of a very high standard, and this will be apparent when one realizes that a tanker during her 320 days' service per averages 250 days at sea. Further, during the vessel's relatively short stay in port the work of either discharging or loading is carried on continuously, and the terminal at which the vessel normally lies is generally many miles from the nearest town. All these factors tend to influence members of the crew and even officers to leave the vessel. Accommodation of a very high standard is therefore good policy.

Amenities

In step with the foregoing the officers and crew are made as comfortable as possible by mechanical ventilation-heating system; large and roomy tiled bathrooms in which running hot and cold fresh water service is laid on; hygienic toilet rooms; cooled drinking water; clean, airy, hygienic messrooms; kit and other arrangements; portable swimming bath, etc.

PROPULSION AND AUXILIARY MACHINERY

Provisional examination of the type of propelling machinery has led one to adopt turbo-electric drive (Fig. 12). Twin-screw propulsion has been considered but as there is a present-day tendency towards much higher power per shaft coupled with the accepted loss of efficiency of something in the order of 6-9 per cent for bossing, etc., in a twin-screw vessel, it was finally decided that a single screw would be adopted.

The advantages of relatively high steam temperatures and pressures are well known and it has been decided to use steam

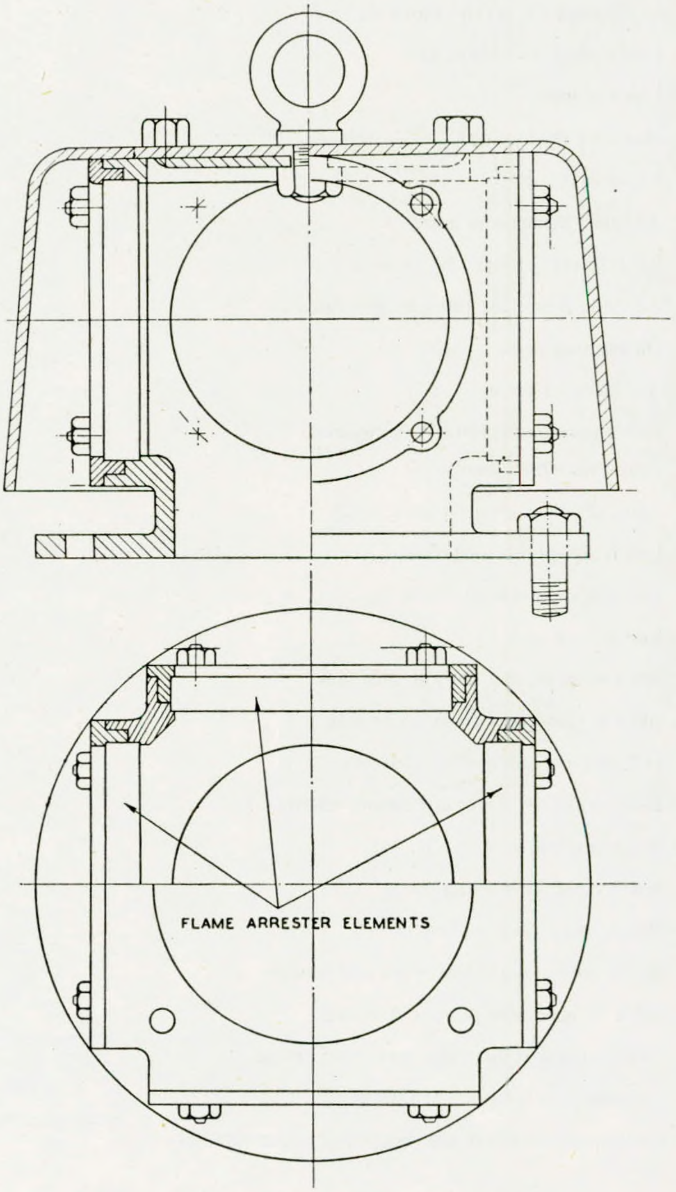
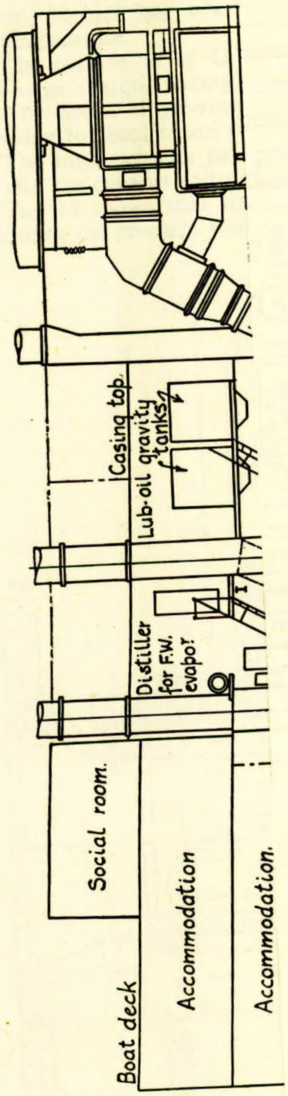


FIG. 11 (*a*)—*Flame arrester*


Casing top.
Lub-oil gravity tanks
Distiller for F.W. evapor
Social room.
Accommodation
Accommodation.
Boat deck

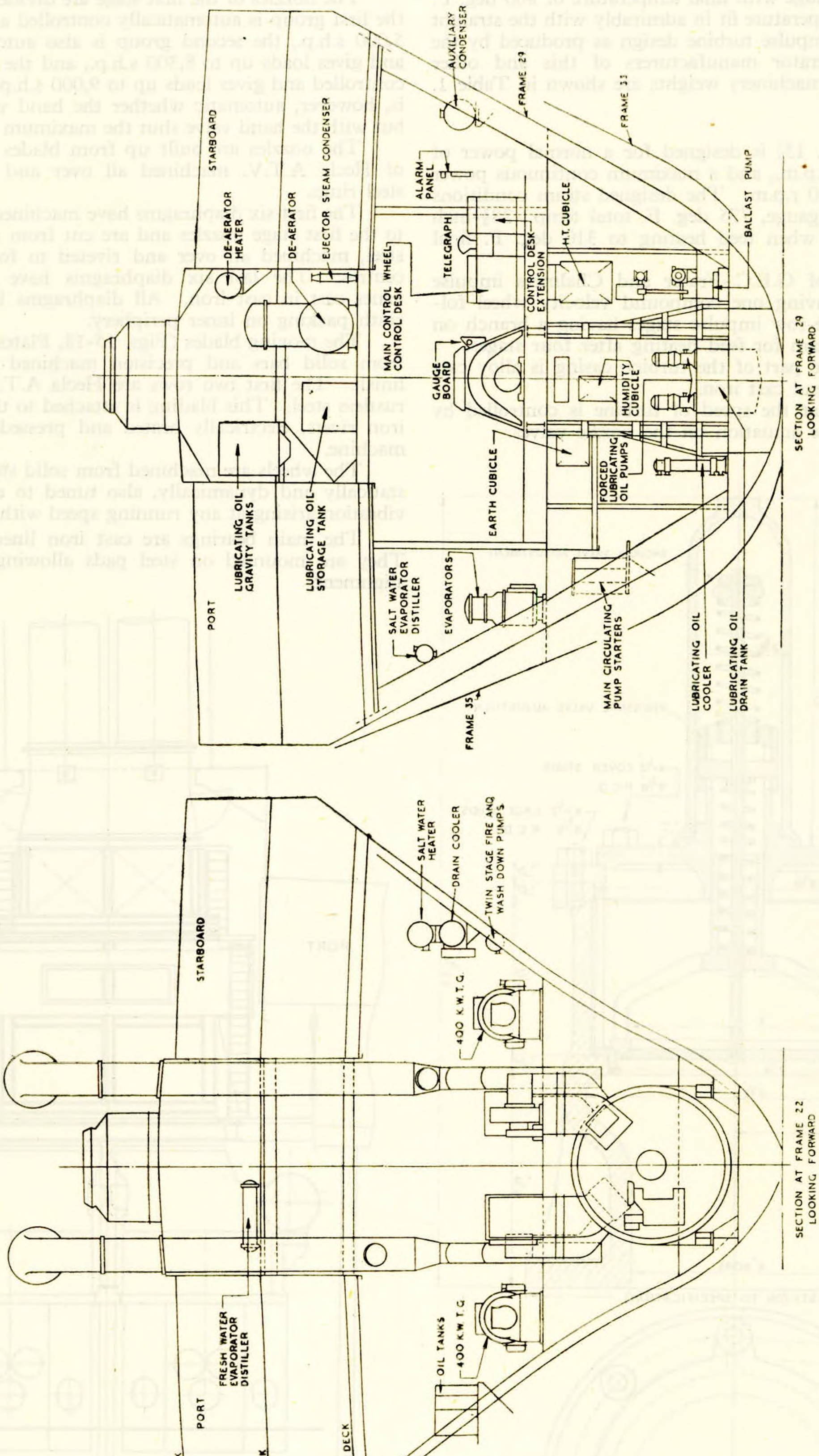


FIG. 12 (*e*)—*Sections at frames 22 and 29*

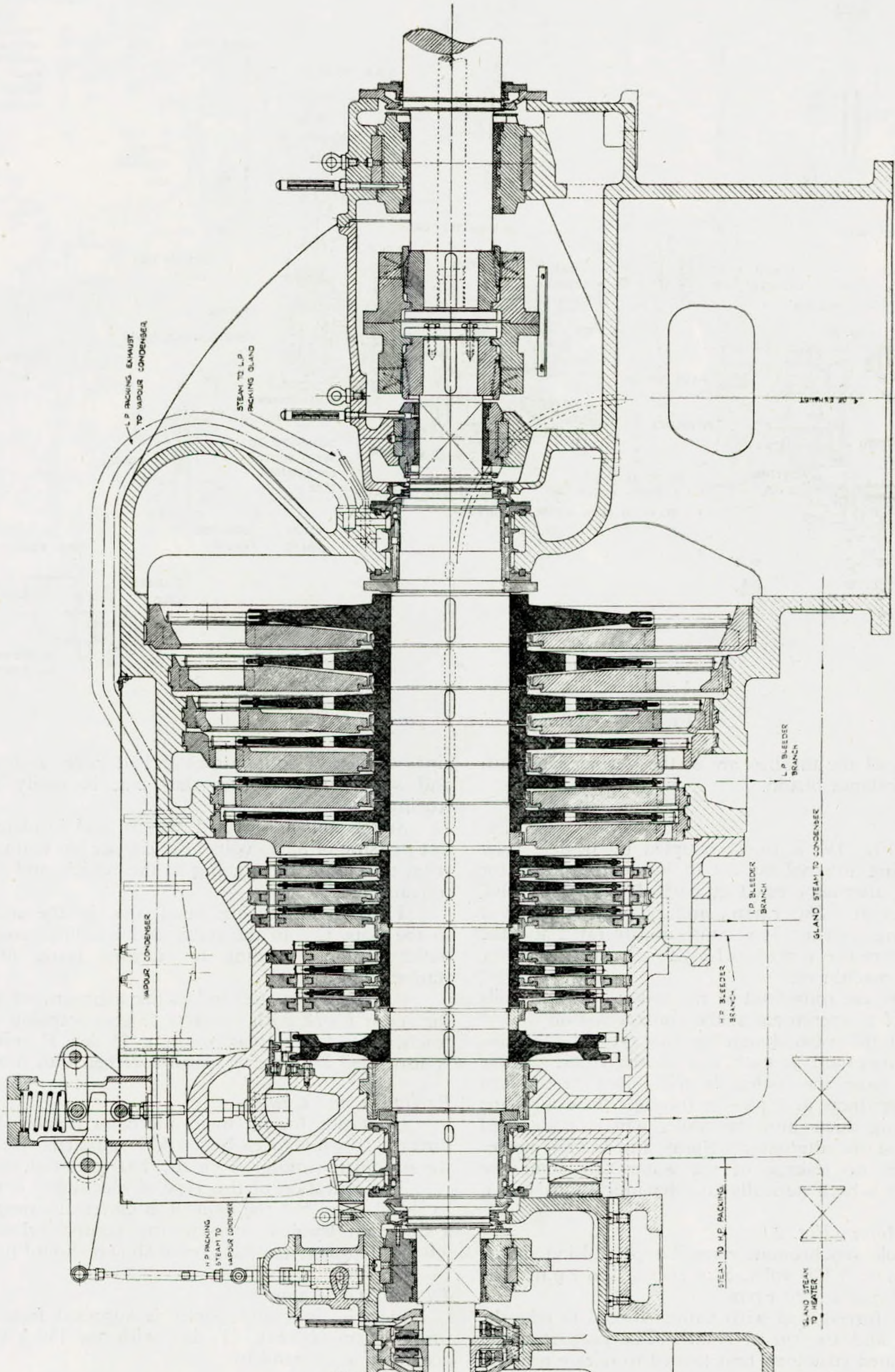


FIG. 13—*Turbine in section*

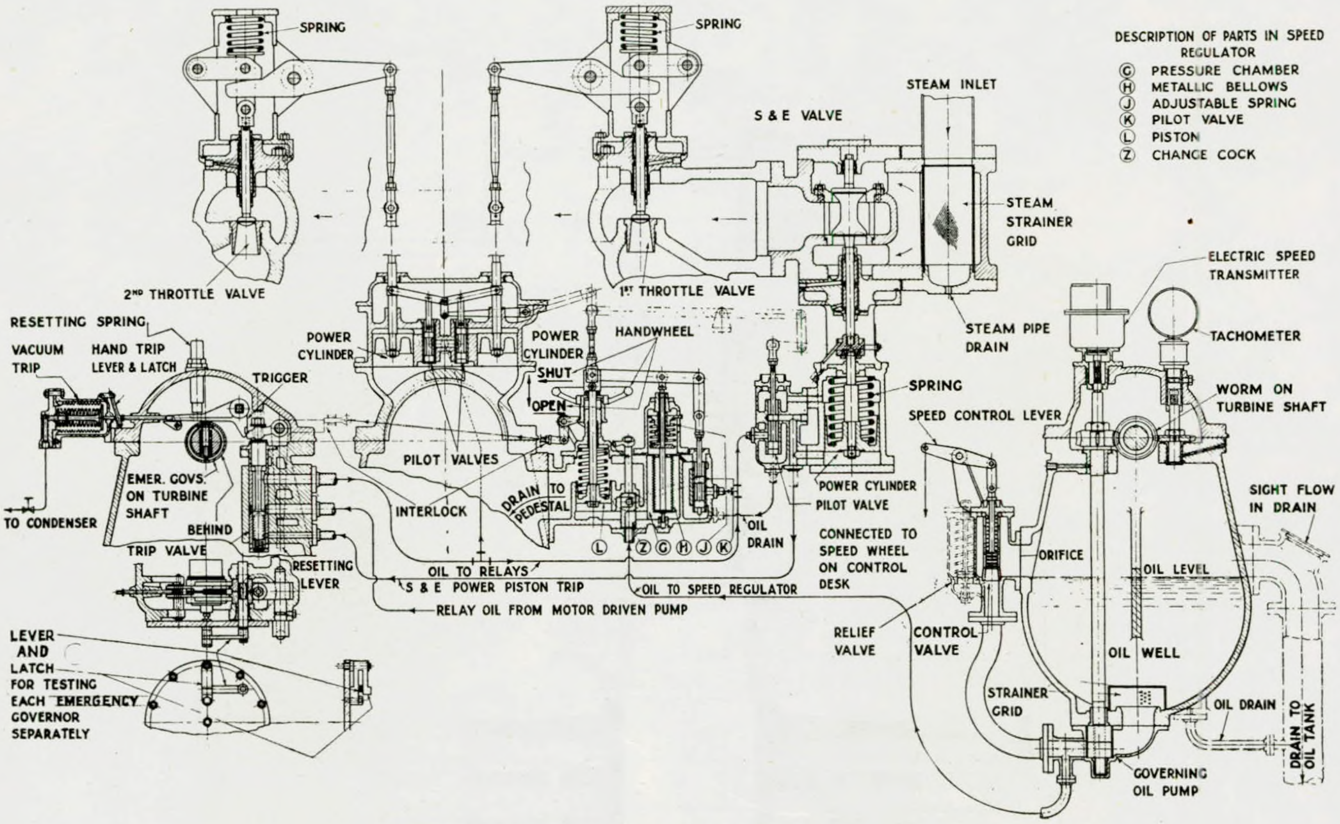


FIG. 14—*Expanded view of governing gear*

Packing glands of the turbines are of the Vernier labyrinth type together with balance piston.

Alternator

The turbine (Fig. 19) is direct coupled by flexible claw type coupling having internal collars to locate the alternator rotor axially to an alternator rated at 6,930 kW., 3,320 volts, 3-phase, 51·6 cycles at 3,280 r.p.m., and is provided with a closed circuit cooling system. It is continuously rated and the maximum temperature rise is within Lloyd's modified rules for electric propulsion machinery.

Thermo-couples are embedded in the stator iron and coils for the recording of temperatures at the control station.

The rotor is of the usual design for this type of machine, with cooling fans mounted at each end of the rotor. These fans draw cool air from the coolers in the closed circuit into the rotor ventilating ducts and pass it through these air gaps and stator ventilating ducts into the cooler which is enclosed within the seating of the alternator. Spray and leakage baffles are fitted to ensure no leakage of sea water can enter the machine. The rotor is both statically and dynamically balanced.

Main Propulsion Motor (Fig. 20)

The salient pole synchronous reversible propulsion motor is rated at 9,000 s.h.p., 3,320 volts, 51·6 cycles, 126 r.p.m., the power being 8,300 s.h.p. at 120 r.p.m.

The shaft is of forged steel with flanges forged to take the magnet wheel hub and the thrust shaft. The magnet wheel hub is a steel fabricated structure, heat treated to release residual stresses, the rim being of a section suitable to carry the flux and ample enough to withstand the stresses imposed by the magnet wheel poles and coils. The magnet wheel poles are of solid steel with steel shoes, connected together with heavy copper strips. These shoes, of course, provide the heavy torque required when reversing. The magnet pole coils are wound with copper strips on edge and insulated to obtain good creepage distances to deal with the heavy voltage induced on reversing. Slip rings consist of four gunmetal half rings and staggered joints and so constructed that they can be easily dismantled for trueing.

Stator coils are formed, wound and insulated to withstand test pressure of 8,000 volts. Six copper bar connexions are taken from the stator, three going to the cubicle and three to current transformers.

Thermo-couples are fitted, one in the stator iron, three in the slots, two in the stator end windings and one in the air outlet, distance reading instruments being fitted at control station.

A mercury-in-steel indicating thermometer is also fitted on the stator frame with contacts to give warning when outlet air reaches 160 deg. F. maximum, or 75 deg. F. minimum. There is also fitted a humidity alarm together with indicator lights.

Excitation and Control

Excitation for the main alternator is taken from an exciter directly coupled to each 400 kW. auxiliary generator and for the propulsion motor from the 200-volt ship supply.

The operation of this type of machinery is too well known to detail here, but the control is extremely simple, there being a direction lever, a manœuvring control wheel and a speed wheel. However the slide being shown should prove of interest.

400 kW. Generator

Electric light and power is supplied from two 400 kW. turbo-generators (Fig. 21) d.c., with one 150 kW. Diesel-driven generator as a stand-by.

Steam Flow

Steam from the main boilers is only used for the main turbine, 400 kW. turbo-generator, turbo-feed pumps, reduced pressure and temperature steam for cargo pumps, turbo-boiler circulating pumps, turbo-auxiliary steam circulating pumps, steam generator and Butterworth tank cleaning heater. All the steam required for usual ship and tanker service which is likely

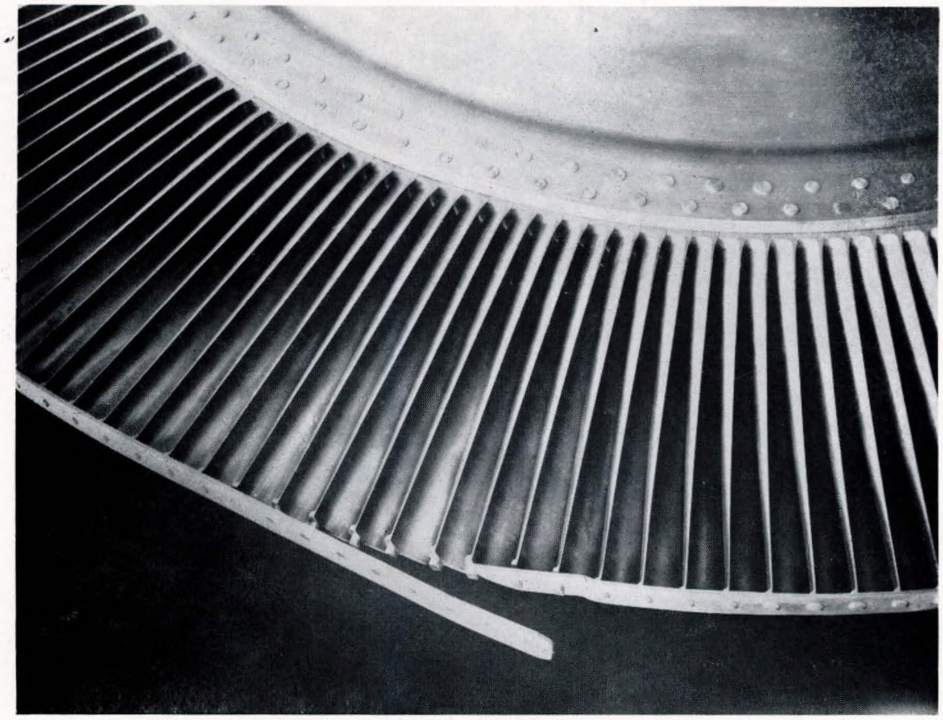


FIG. 15

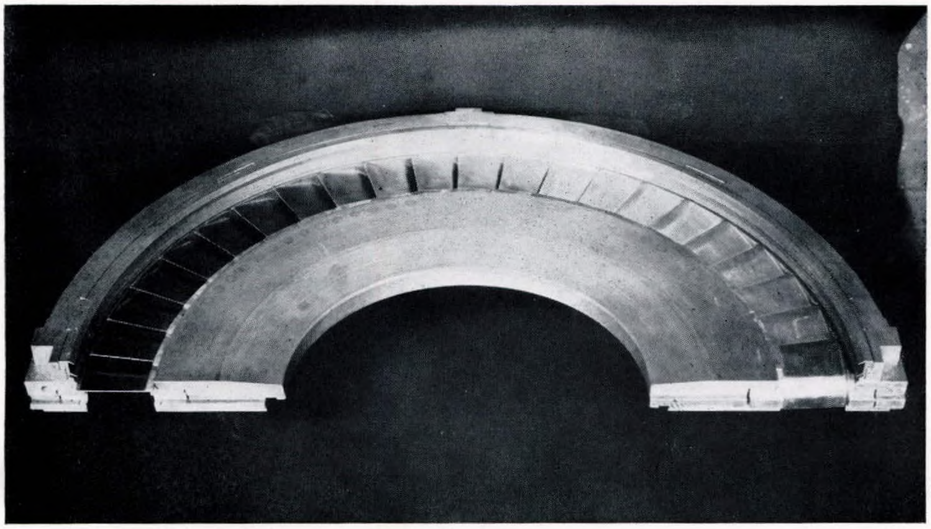


FIG. 16

Plate 2

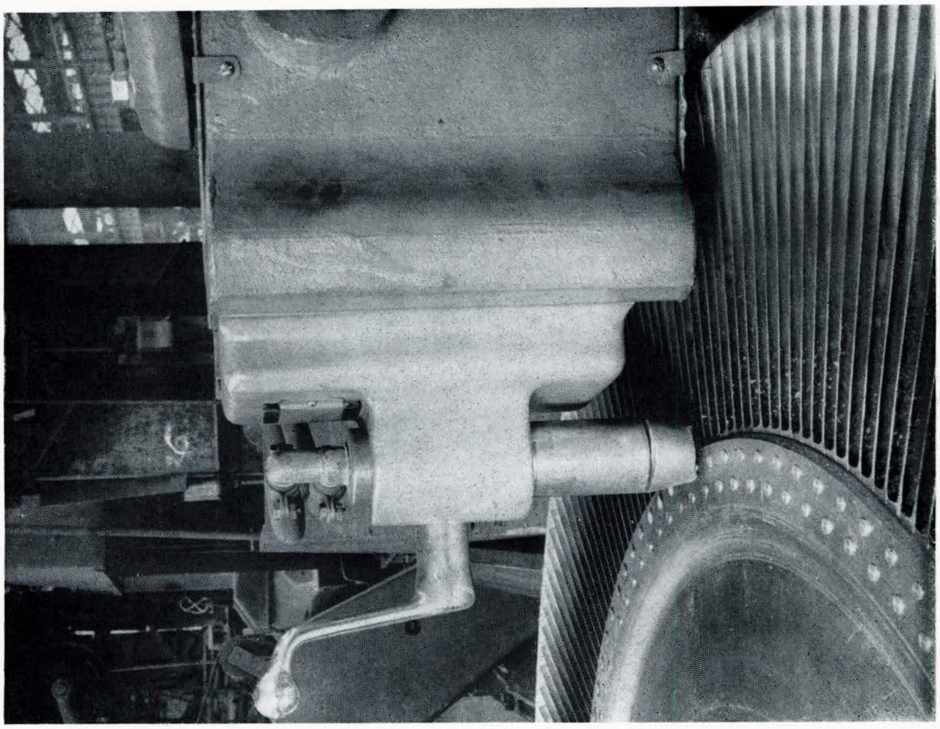


FIG. 18

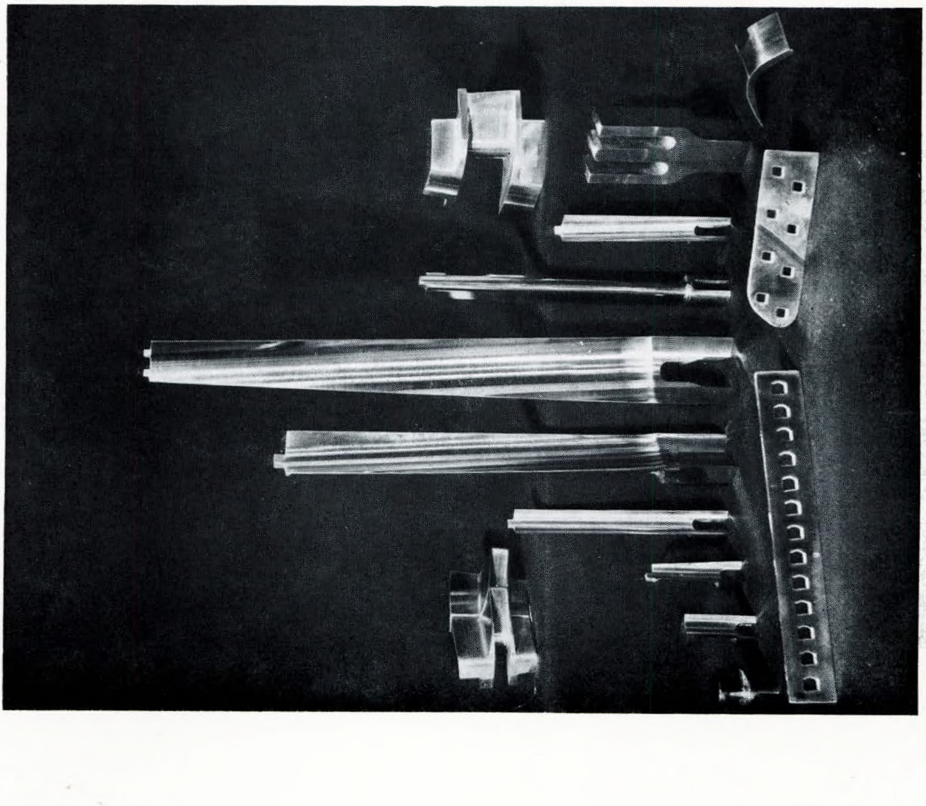


FIG. 17

to become contaminated, such as boiler fuel heating, reciprocating pumps and machinery in the engine room and cargo heating, is completely isolated from the main high pressure relatively clean system. The importance of this isolation in the case of a tanker need not be enlarged upon as purity of the main boiler water is paramount if one is to operate this tanker.

It will be observed that the highest possible elimination of air from the closed feed circuit has been aimed at by installing a very well designed and most efficient de-aerator. This unit working in conjunction with a first-stage feed heater taking exhaust steam from feed pump exhaust and a second-stage heater taking bled steam at 40/50lb. per sq. in. gauge pressure from the fourth stage of the main turbine, gives a feed temperature entering the economizer of 310 deg. F. at 8,300 s.h.p.

An interesting innovation is introduced in this ship in the form of a mixed pressure arrangement (Fig. 22) on the 400 kW. generator sets. This is a balancing assembly on the generator turbines whereby when the auxiliary range pressure exceeds 1lb. per sq. in. gauge a valve is opened to a low pressure stage of the turbine and simultaneously the high pressure steam to nozzles is reduced in proportion in order to maintain a constant r.p.m. of the generator and also constant pressure in the exhaust range.

This has proved a very valuable asset as it is found on a tanker of this class that the exhaust steam quantities vary considerably, also there are certain conditions which arise when one finds an excess of about 4,000lb. of steam per hour in the auxiliary range, and the only alternative would be to release to the main condenser—when in port a most uneconomical procedure and may be attended by a certain amount of risk of distortion of turbines.

Boiler water levels and feed pump control is fully automatic coupled with several safety cut-out devices as emergencies.

The main condenser is of two-flow type, size 9,000 sq. ft. As it has long been the author's opinion that condenser tube trouble is closely allied to condenser and circulating pump design, special attention has been given when designing this condenser to bring turbulence down to a minimum by ensuring that:—

(1) The rate of flow through the tubes should not be more than 6-7ft. per sec. when at full power.

(2) Adequate capacity of water ends and avoiding all sharp angles or reactionary obstructions with a view to the utmost uniflow effect.

(3) Large and uniflow entry from main centrifugal pumps.

(4) Relatively slow running centrifugal pumps designed to generate the least possible agitation.

Main Steam Generating Plant

The plant consists of two La Mont marine boilers having forced circulation in the following details:—

Evaporation normal full power	60,000lb. per hour
Working pressure superheater outlet ...	500lb. per sq. in.
Design pressure ...	560lb. per sq. in.
Steam temperature at stop valve	800 deg. F.
Feed water temperature to economizer ...	310 deg. F.
Quantity of water circulated	746 g.p.m.-360,000lb. per hr.
Differential pressure ...	23lb. per sq. in.
Combustion chamber volume	648 cu. ft.
Heat release rate (gross)	122,000 B.Th.U.s per cu. ft. per hour
Estimated combustion chamber temperature	2,325 deg. F.
Gas temperature at airheater outlet ...	325 deg. F.
Gas temperature at economizer outlet...	528 deg. F.

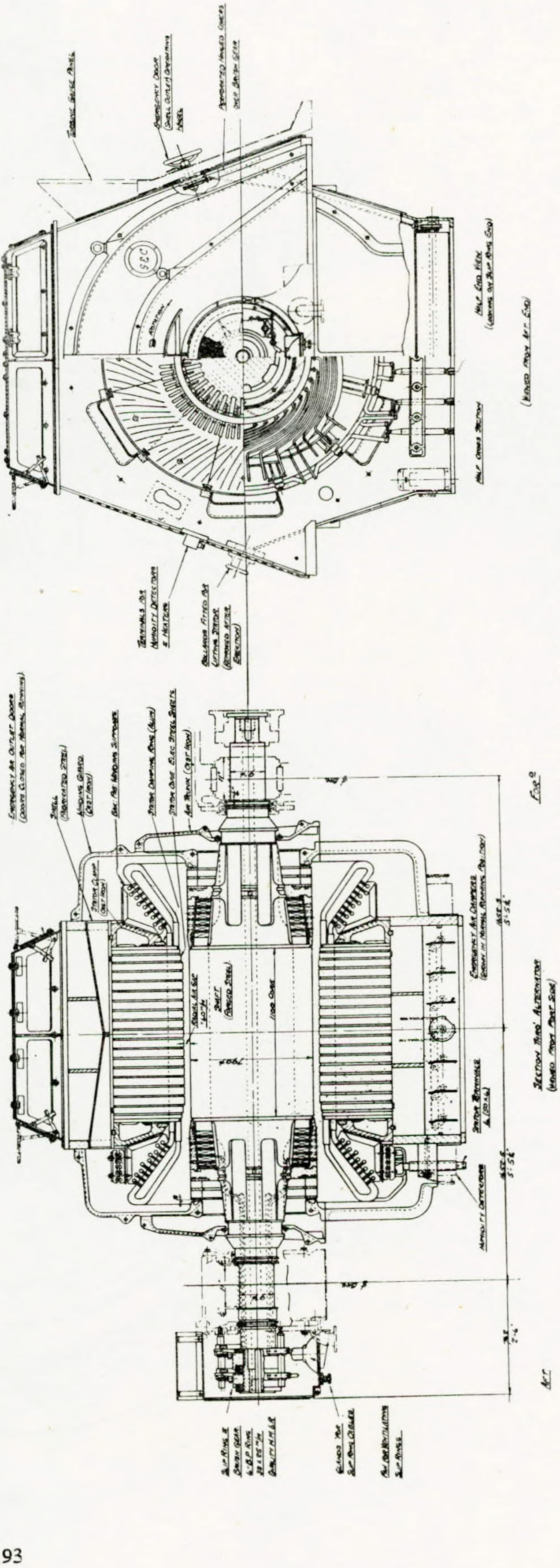


FIG. 19—Section through alternator

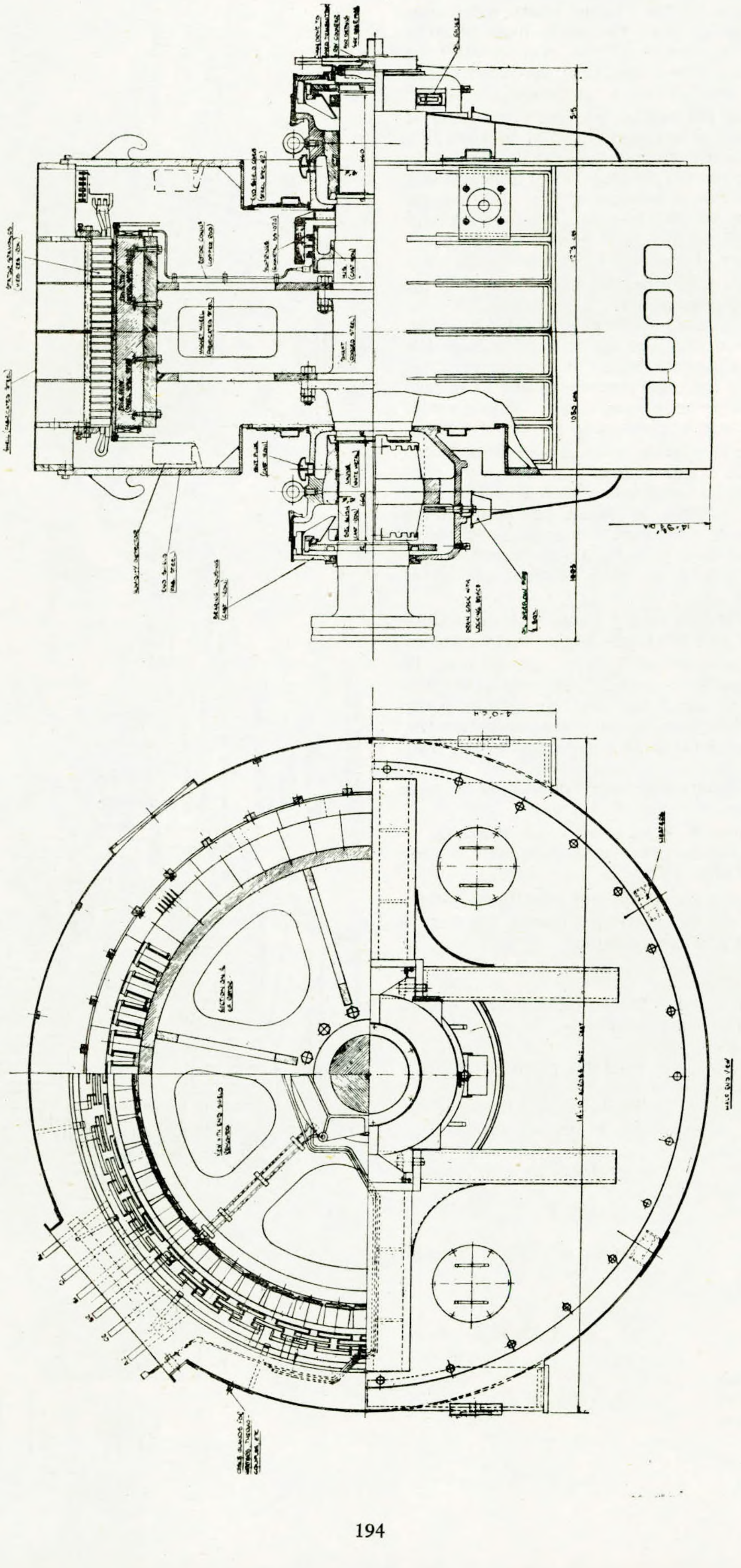


FIG. 20—Sectional arrangement of propulsion motor

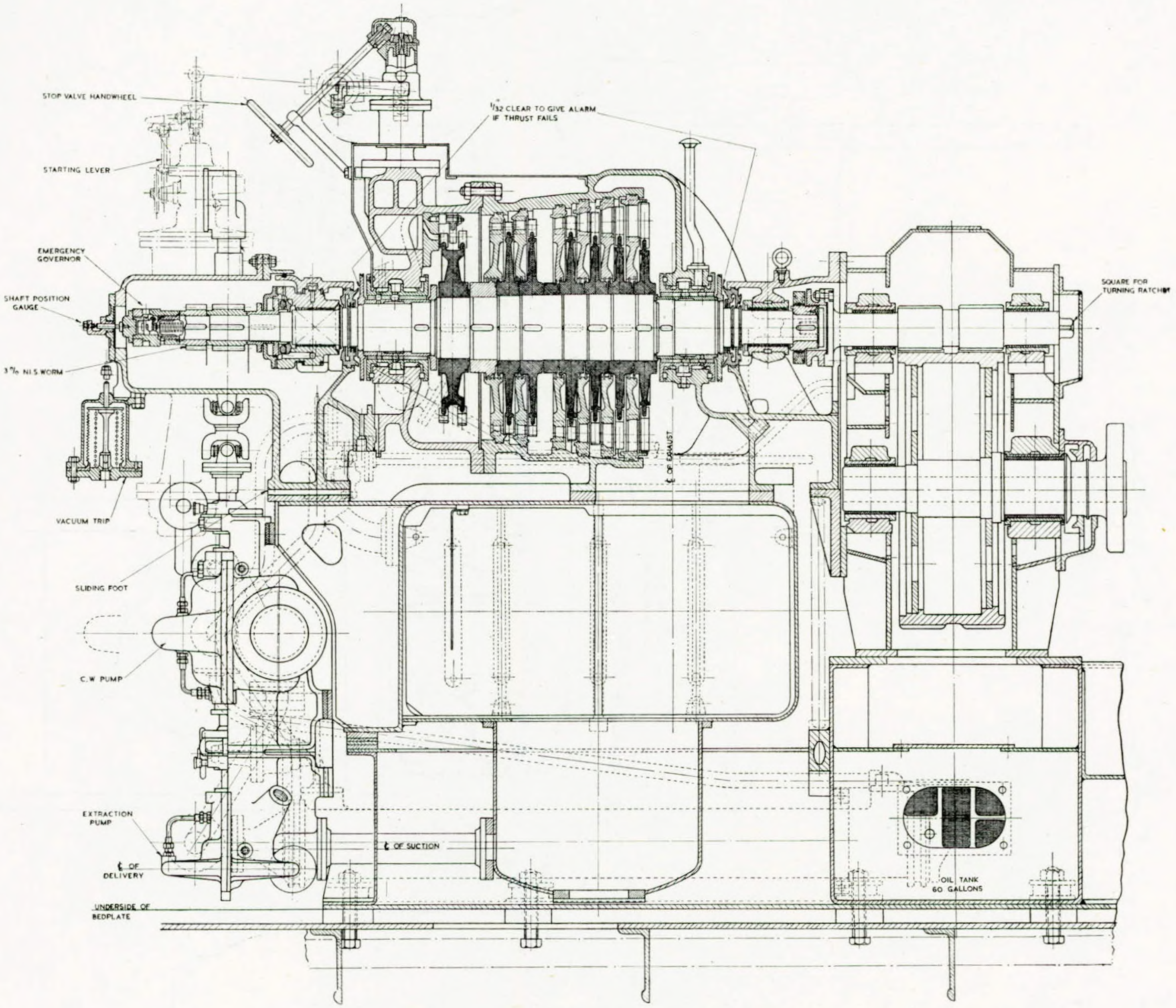


FIG. 21—*Section through turbine and gears*

Air temperature at inlet of airheater ...	80 deg. F.
Air temperature to burners	296 deg. F.

Heating surfaces

Boiler	3,100 sq. ft.
Superheater	1,550 sq. ft.
Economizer	1,740 sq. ft.
Air heater	6,715 sq. ft.

Oil burners (5 Wallsend Howden) pressure atomization

Weights

Weight of boiler, superheater and economizer dry	50·3 tons
Weight of boiler, superheater and economizer under steaming conditions including water	54 tons

These boilers (Fig. 23) have many features in common with ordinary types of water-tube boilers having drums, headers and tubular heating surface. There is, however, a fundamental difference and that is, whereas the usual water-tube boilers depend upon natural displacement for their circulation, the circulation of the La Mont boiler is positive by means of a turbo-driven circulating pump having a differential in pressure of something in the order of 35lb. per sq. in. gauge. The amount of water going through each tube is regulated to give a prescribed speed, i.e., ranging from 2ft. per sec. at inlet to 15ft. per sec. on entering the drum. This is considered an important feature when one considers the limiting of height of the water-tube boilers experienced in ships' plant, which height, of course, affects in no small measure natural circulation. Further forced circulation enables the designer to lay out his heating surface in the manner most effective for rapid heat transfer (Fig. 25).

It will be observed that the position of the superheater tubes makes it impossible to have a condition whereby the gas temperature can be lower in way of the superheater tubes than that of the gas in way of the second evaporator nest. As will be noted, special baffling in the steam water drum has been carefully considered and applied, i.e., annulus, hood, steam take-off, perforated and surge baffles have been fitted. These baffles

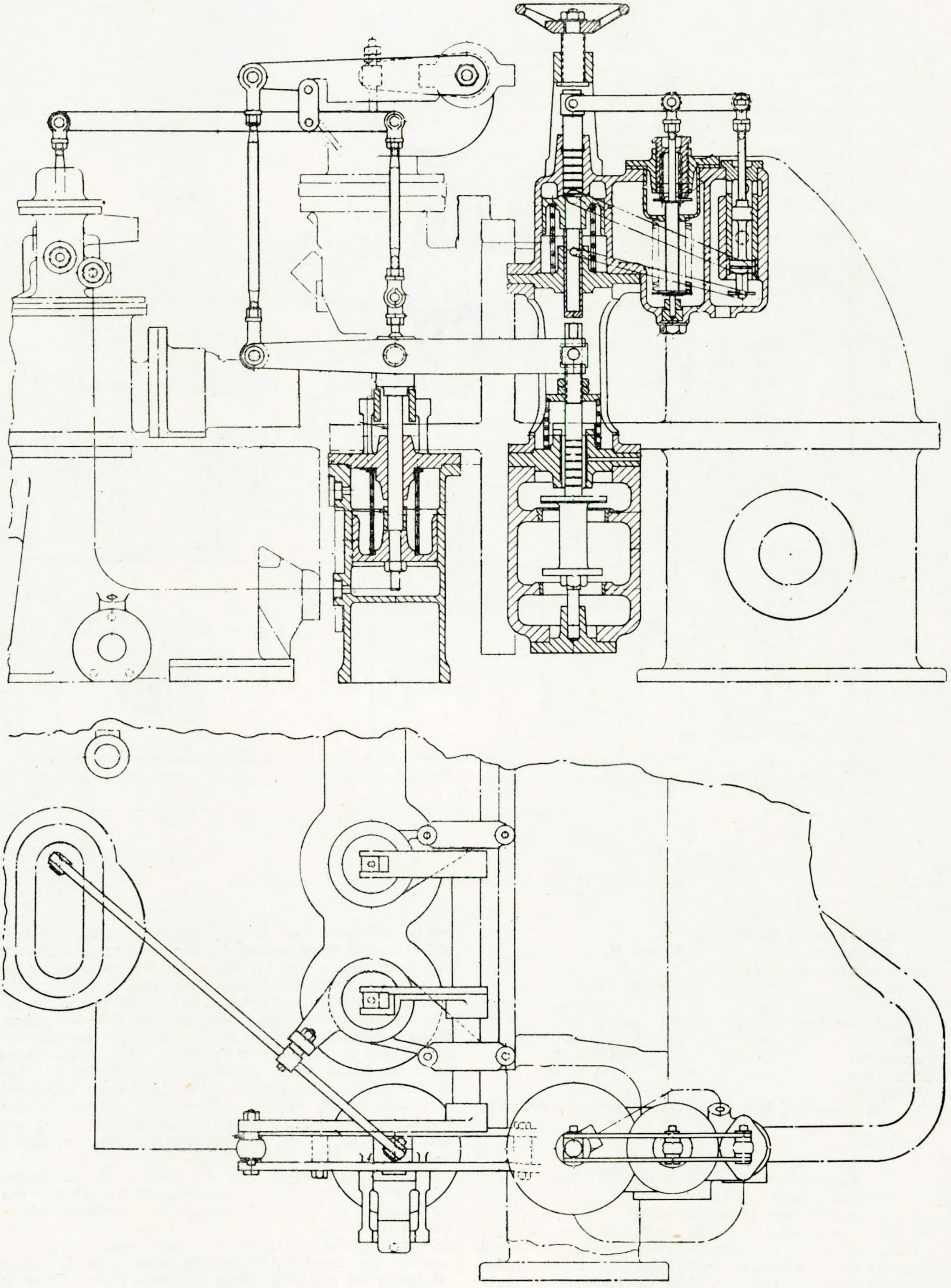


FIG. 22—*Mixed pressure control (arrangement of governing gear)*

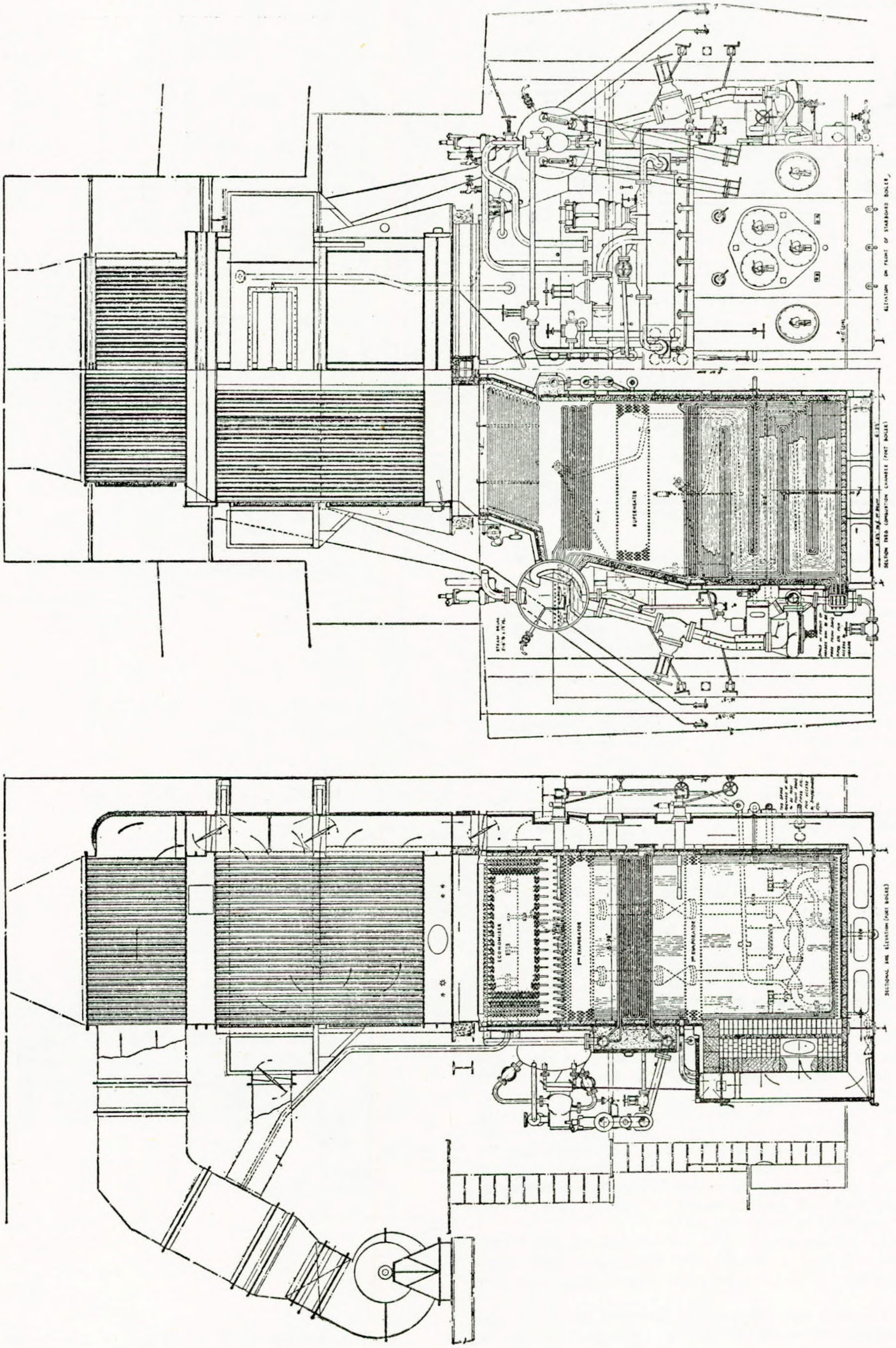


FIG. 23—*Arrangement of boilers*

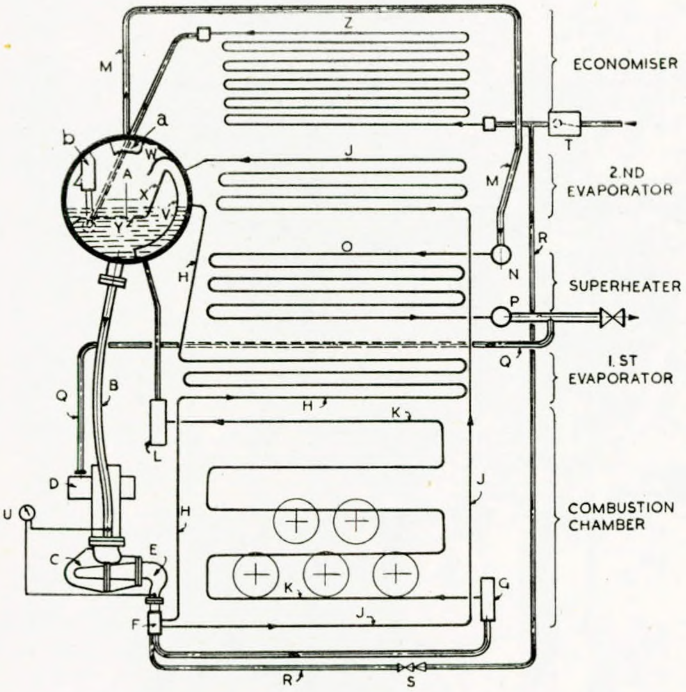


FIG. 24—*Diagrammatic arrangement of La Mont boiler circulation*

A	Steam and water drum
B	Suction pipes
C	Circulating pump
D	Turbine
E	Breeches pipe
F	Main distributor header
G	Rear wall distributor header
H	1st evaporator tubes
J	2nd evaporator tubes
K	Rear wall tubes
L	Rear wall collector header
M	Saturated steam pipes
N	Superheater inlet header
O	Superheater tubes
P	Superheater outlet header
Q	Steam to circulating pump
R	Economizer recirculating pipe
S	Economizer recirculating valve
T	Feed regulator
U	Differential pressure gauge
V	Annulus baffle
W	Hood baffle
X	Spill plate
Y	Perforated baffle
Z	Economizer tubes
a	Steam take off baffle
b	Feed boxes

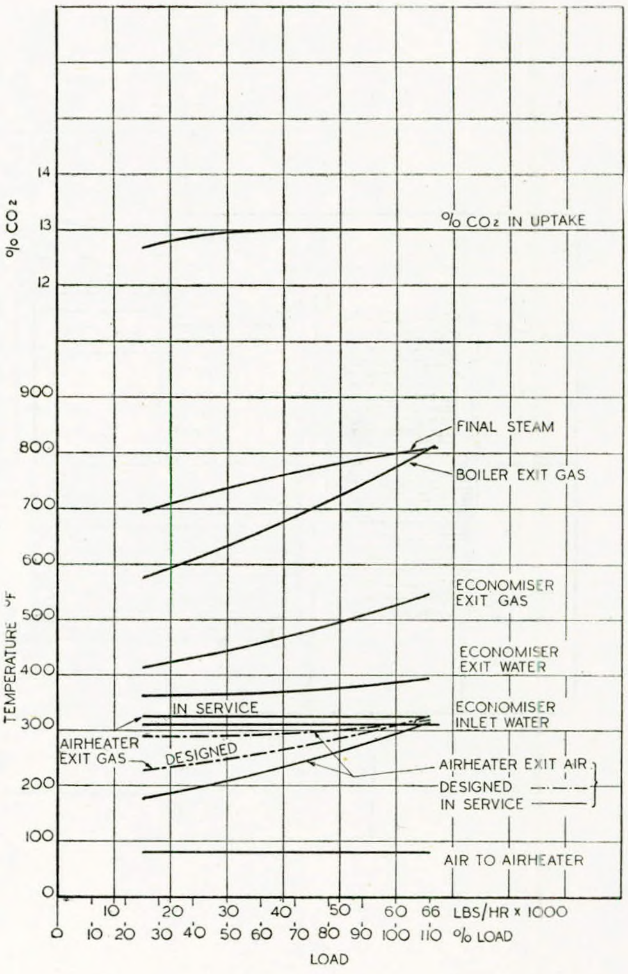


FIG. 25—*Technical data of boiler*

were designed as a result of considerable experience in order to ensure dry, clean steam.

Very little brickwork is necessary as the furnace side back wall and floor are water cooled, only the front wall is firebrick lined to provide sufficient residual heat in the refractories to ensure satisfactory ignition of the incoming oil fuel.

Fig. 26 shows the design performance of the boilers over a wide range of loads. The author thinks it will be accepted that the compact design of these boilers with their relatively high evaporation rate makes for a valuable saving of space and weight, which has helped very materially in the design of this lay-out.

With machinery of this class one cannot overstress the importance of control of the superheated steam. As will be seen in Fig. 26, a simple but very effective means of controlling the temperature has been introduced. This is by introducing into the gas stream at the most appropriate point a controllable quantity of tertiary air thus reducing the temperature of the gases before reaching the superheater.

The boiler is surrounded by a pressurized air jacket that is changed by the forced draught fan which practically eliminates the possibility of combustion gases leaking into the engine room.

A very low exit gas temperature has been aimed at and achieved even at normal ratings and an interesting feature in

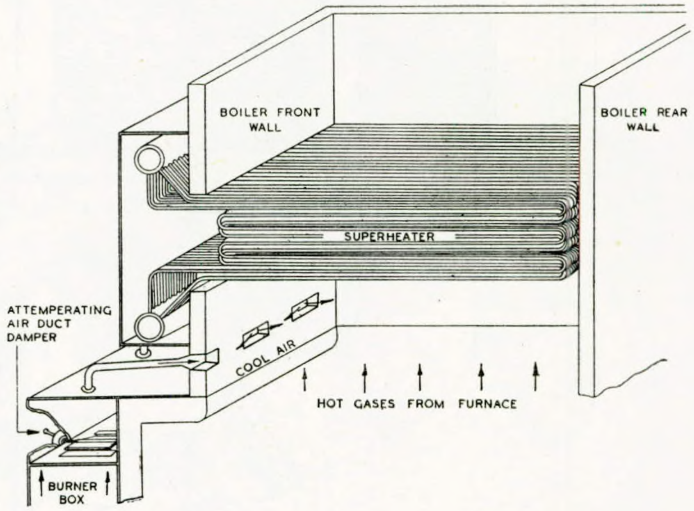


FIG. 26—*Attemporating air supply to superheater*

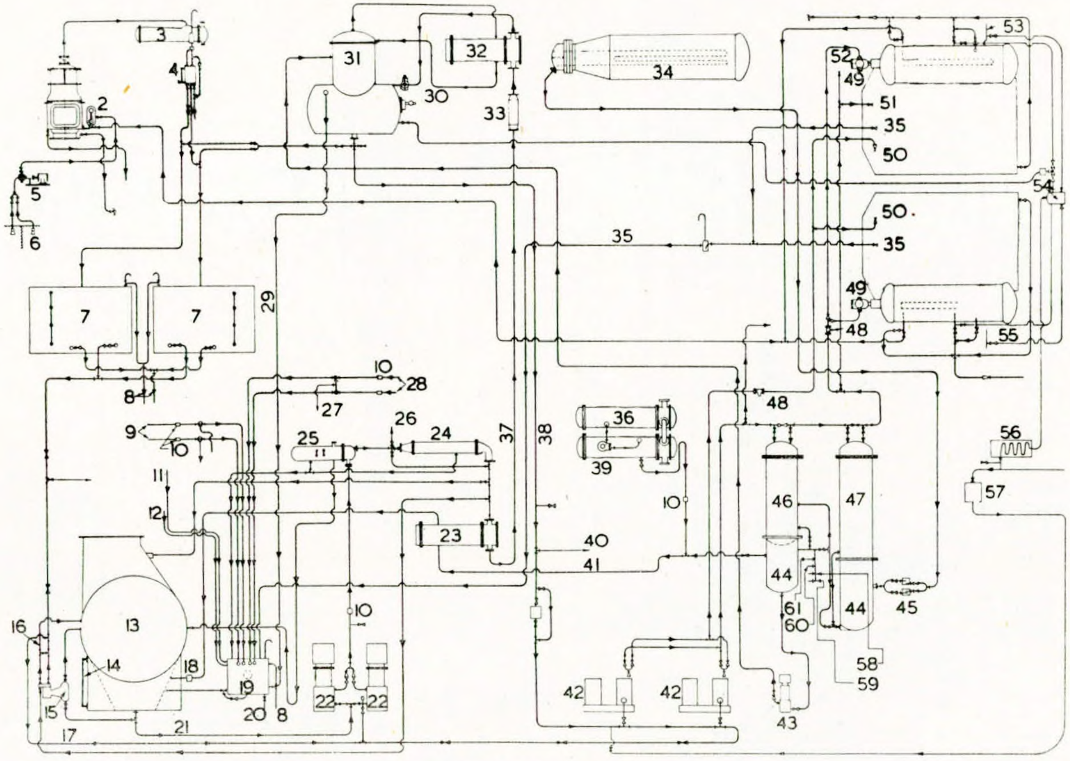


FIG. 28—*Main feed diagram*

1—Fresh water evaporator. 2—Feed regulator. 3—Fresh water distiller. 4—Test tank. 5—Fresh water evaporator feed pumps. 6—Suctions from port and starboard fresh water double-bottom tanks. 7—Distilled water. 8—Overflow. 9—Condensate return from 400 kW. turbo-generators. 10—Salinometer. 11—Drain from gland vapour condenser. 12—Condensate from gland vapour condenser. 13—Turbo-alternator condenser. 14—Water level gauge. 15—Closed feed valve. 16—Hand operated by-pass. 17—Emergency feed suction. 18—Trap. 19—Condensate tank. 20—Drain to bilge. 21—Condensate suction. 22—Extraction pump. 23—Drain cooler. 24—Gland steam heater. 25—Two-stage air ejector. 26—Condensate to gland vapour condenser. 27—To bilge. 28—Condensate returns from cargo pump rooms. 29—De-aerator overflow. 30—Control valve. 31—De-aerator. 32—De-aerator heater. 33—Ejector steam condenser. 34—Contaminated steam generator. 35—Superheater drains. 36—Two-stage washdeck heater. 37—Condensate to de-aerator. 38—Feed pump suction. 39—Steam trap. 40—Distilled water (hot) to chemical dosage mixing tank. 41—Emergency drain to condenser. 42—Turbo-feed pump. 43—Feed heater drain pump. 44—Flash chamber. 45—Steam traps. 46—No. 1 feed heater. 47—No. 2 feed heater. 48—Feed filter. 49—Feed regulator. 50—Cold feed to boilers. 51—Feed water to de-superheater forward deck steam. 52—Feed water to de-superheater for steam to washdeck heater. 53—Circulation pump suction. 54—Pressure reducing vessel. 55—Circulation pump suction. 56—Cooler. 57—Densometer. 58—Turbo-steam chest drain. 59—Drain from bled steam pipe line. 60—Ejector steam pipe drain. 61—Drain from main steam manifold.

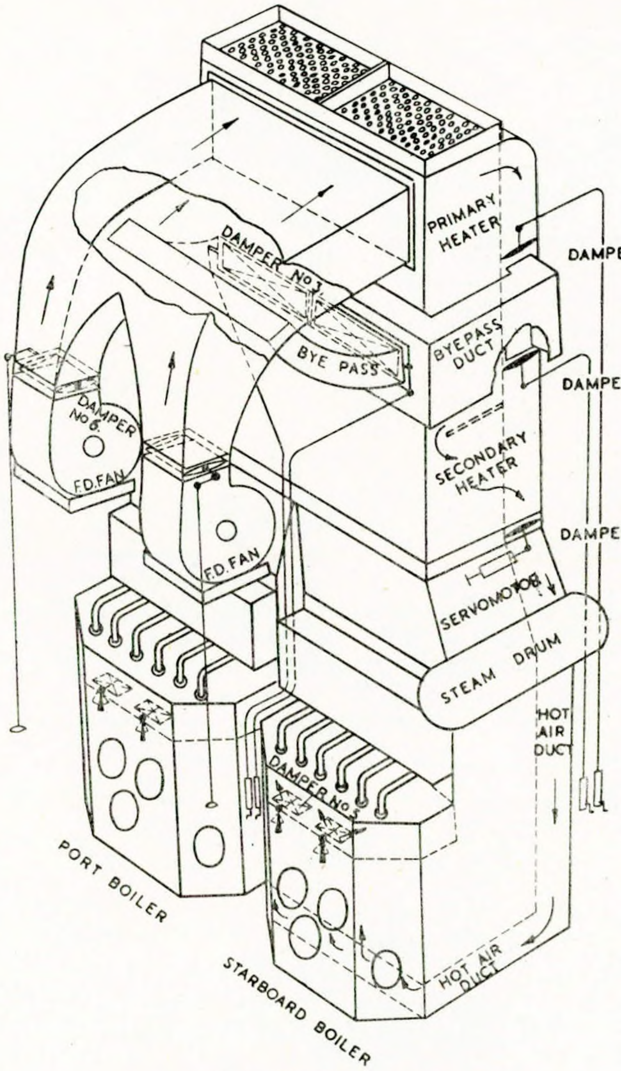


FIG. 27—*Air dampers*

Port boiler	Starboard boiler	Dampers
No. 1 P	No. 1 S	Primary heater isolating
No. 2 P	No. 2 S	Temperature control
No. 3 P	No. 3 S	Bye pass
No. 4 P	No. 4 S	Forced draught balance
No. 5 P	No. 5 S	Attemperating air
No. 6 P	No. 6 S	F.D. fan isolating

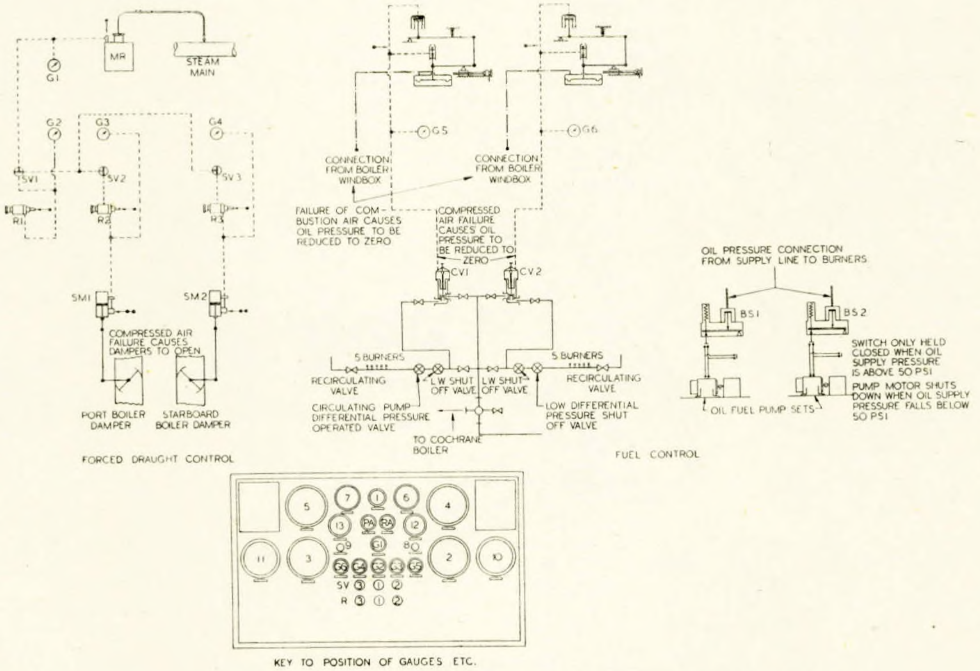


FIG. 29—*Diagram of Hagan fuel burning control*

G.1—Master control. G.2—Forced draught dampers—unified manual control. G.3—Forced draught damper control—port. G.4—Forced draught damper control—starboard. G.5—Oil valve control—port. G.6—Oil valve control—starboard. PA—Power air. RA—Relay air. 1—Oil fuel pressure. 2—Superheater pressure—port. 3—Superheater pressure—starboard. 4—Boiler pressure—port. 5—Boiler pressure—starboard. 6—Economizer pressure—port. 7—Economizer pressure—starboard. 8—Differential pressure—port pump. 9—Differential pressure—starboard pump. 10—Differential pressure—port. 11—Differential pressure—starboard. 12—Smoke density—port uptake. 13—Smoke density—starboard uptake. MR—Master regulator. SV.1, 2, 3—Switching valves. R.1, 2, 3—Compensating relay. SM.1, 2—Servomotor. CV.1, 2—Oil control valve. BS.1, 2—Bellows switch.

the air heaters is that they are in two sections—a normal hot section and a separate relatively small cold section. The purpose of this is to localize the inevitable corrosive action of combustion gases on cold airheater tubes by reason of the metal temperature being frequently above or below the dewpoint of the gases. With this arrangement the corrosive action is limited to one relatively small part of the airheater consisting of short tubes which are made readily replaceable; further this small section can be readily by-passed in the event of its becoming defective without materially affecting the performance of the plant; further the cold end section of the airheater can be by-passed when under low loads or raising steam when the gas temperatures are unduly low. For similar reasons provision is made whereby the economizer is automatically recirculated with hot boiler water under stand-by and low load conditions, thus maintaining the economizer metal temperatures at all times well above the dewpoint of the gases. The importance of this is not generally appreciated by marine engineers. It is well known that one important effect of an economizer is to stabilize the boiler water level as the temperature of the feed water entering the boiler is with an economizer much nearer to the temperature and density of the boiler water and thus the recirculation conditions of the boiler are not so badly affected as they would be if relatively cold water of relatively high density were suddenly introduced into the boiler drum. Apart from other reasons an economizer is extremely helpful from this point of view but unless some provision is made to maintain the economizer metal temperature always higher than the dewpoint of the gases, then external corrosion will undoubtedly be met.

Fig. 27 shows very clearly the provision made for by-passing either primary or secondary air heaters or both, whilst the plant is in operation. It also shows the attemperating air dampers to which the author has referred earlier.

As referred to earlier considerable care has been taken to ensure that any contaminated steam services will have their own independent feed system, whilst the feed system for the high pressure boilers is an entirely closed cycle (Fig. 28) in no place connected either directly or indirectly to any contaminated service. The make-up water is from double evaporated and distilled sea water, the whole of the feed water for the main boilers being de-aerated.

Considerable attention has been paid to chemical conditioning of the boiler water for which purpose chemical injection vessels are provided which enable a solution of the necessary chemicals to be made and injected directly into the boiler circu-

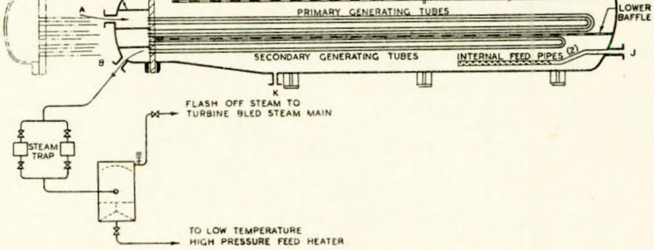


FIG. 30—*Steam/steam generator*

A	Generating steam inlet
B	Generating steam outlet
C	Safety valve
D	Pressure gauge
E	Steam to whistle
F	Saturated steam outlet
G	Manhole
H	Safety valve
J	Feed inlets
K	Drain

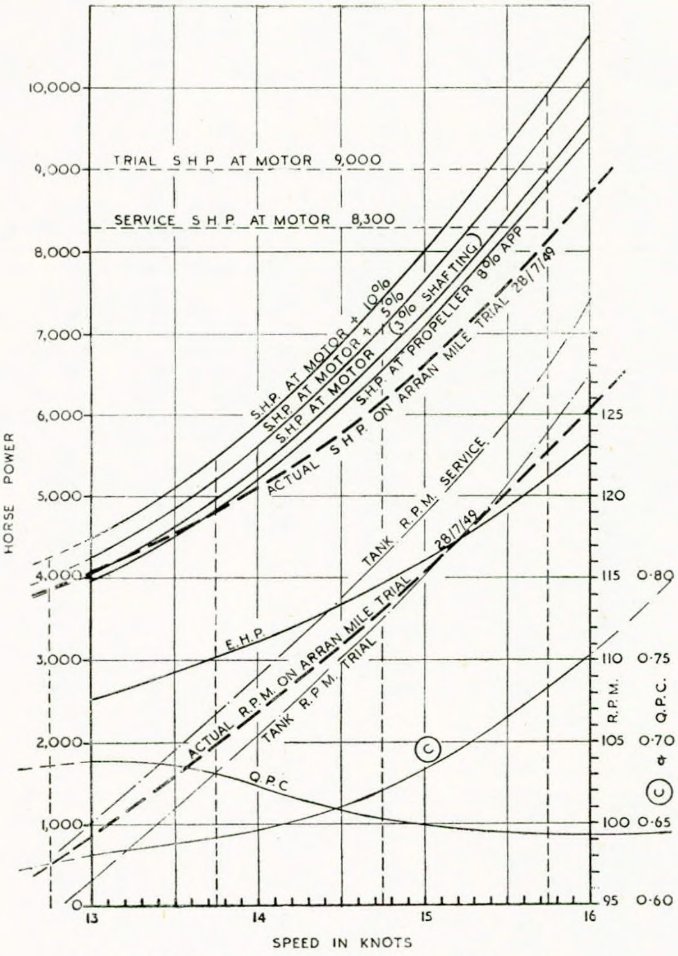


FIG. 31—*Estimated trial powers (based on tank results)*

Dimensions: 510ft. 0in. b.p. × 69ft. 0in. × 37ft. 0in.
Displacement: 21,770 tons total on 29ft. 0in. draught (even keel)
A tank condition load:
Block coefficent = 0·742
Pris. coefficent = 0·762
Midship area coefficent = 0·972
L.C.B. (excluding C.R. stern) = 4·40ft. f
L.C.B. (including C.R. stern) = 4·50ft. f

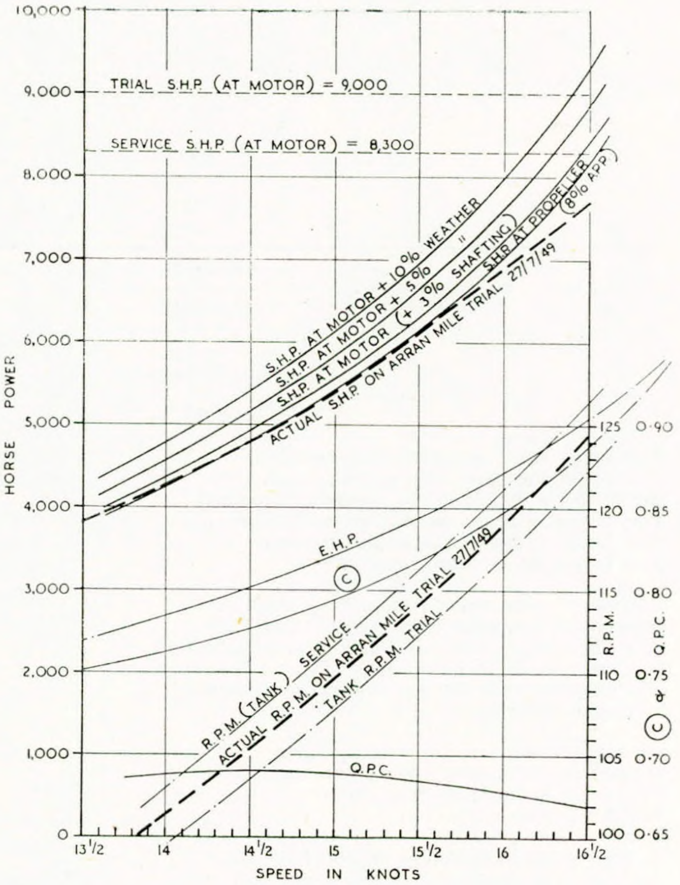


FIG. 32—*Estimated trial powers (based on tank results) summer ballast condition*

Dimensions: 510ft. 0in. b.p. × 69ft. 0in. × 37ft. 0in.
Draught: 18ft. 1½in. b.k. (mean)
Displacement: 12,835 tons (total)
Trim by stern: 8ft. 9in.
Level trim coefficients: Cb = 0·704; Cp = 0·735; Cm = 0·956;
L.C.B. = 6·94ft. f.

lation system. Provision is made whereby should excessive density be reached in the boilers it can be reduced, whilst still retainng the heat by passing continuous blowdown from the boilers to the fresh water evaporator.

Another interesting feature of the chemical conditioning of the feed and boiler water is the provision of a controllable continuous blowdown from either boiler to the de-aerator. This permits a small amount (about 2 per cent) of alkaline boiler water to be injected into the feed water. As is well known, condensate is invariably slightly acidic, its *p*H value being generally about 6·7. By returning a small amount of boiler water into the feed the *p*H of the feed water can be brought up to the alkaline side and maintained at about 8.

Another feature of the feed and boiler water control is that there is in the engine room a central water control station to which all sampling points are brought. Thus samples of feed and boiler water from every position can be obtained at this point by simply opening clearly marked sampling cocks; there is thus no danger of having samples mixed up and recording wrong results. Furthermore, all analyses can be carried out readily on the spot. The complete daily routine analyses governing the feed water on both main boilers can be easily carried out in 30 to 35 minutes.

In actual practice no difficulty whatever has been experienced in carrying out the water analyses and effecting the necessary chemical additions.

Automatic oil fuel/air combustion control has been introduced to the pressure atomizing oil burning system. This is on the well-known principle of steam pressure governing fan pressure (Fig. 29) which in turn gives fuel pressure at the burner by means of damper control plus restriction of fuel supply. The diagram shown here illustrates the system which is in no way complicated. Electric motor driven rotary displacement pumps are introduced with a simplex reciprocating steam pump as stand-by.

Special attention has been given to the soot blowers: these are of a high standard of manufacture, carefully positioned to be most effective, also screened from flame impingement. The air heaters, however, are fitted merely with blowers designed to sweep the top tube plates and thus prevent an accumulation of carbon at this point with its consequent fire hazard.

Contaminated steam generator (Fig. 30) as previously mentioned, supplies steam at 150lb. per sq. in. to all auxiliaries and services likely to contaminate the exhausts and the supply is completely isolated from the high pressure steam. The unit is designed to generate 10,000lb. of steam per hour when supplied with steam at 500lb. per sq. in. and 500 deg. F. through the coils. The returns are led through steam trap to a flash off chamber which is an integral part of the second-stage feed heater. A small Cochran boiler of 6,000lb. per hour capacity at 150lb. sq. in. is installed to meet certain conditions of service.

TABLE 2 MACHINERY

Condition	Caribbean 27/7/49			
Draught	For. 9 ft. 6 in. Aft. 18 ft. 5 in.			
Run Number	North 1	North 2	North 3	North 4
Time B.S.T.	17·01	17·59	18·48	19·33
Port boiler steam, pounds per square inch	490	490		490
Port boiler superheat, degrees Fahrenheit	795	805		790
Port boiler economizer inlet, degrees Fahrenheit	200	260		290
Port boiler air heater outlet, degrees Fahrenheit	290	280		285
Port boiler funnel gas, degrees Fahrenheit	350	365		380
Port boiler fan wind box, inch water	3	3·1		3·4
Starboard boiler steam, pounds per square inch	490	490		490
Starboard boiler superheat, degrees Fahrenheit	860	840		825
Starboard boiler economizer inlet, degrees Fahrenheit	200	260		280
Starboard boiler air heater outlet, degrees Fahrenheit	250	300		330
Starboard boiler funnel gas, degrees Fahrenheit	360	315		320
Starboard boiler fan wind box, inch water	2·8	3		3·3
Oil fuel temperature, degrees Fahrenheit	270	280		270
Oil fuel pressure, pounds per square inch	170	125		180
Feed water meter, pounds per hour	52,500	70,000		82,500
Number of fires each boiler	3–24	3–24		4–24
Turbine steam pressure, pounds per square inch	480	460		460
Turbine steam temperature, degrees Fahrenheit	778	780		770
1st throttle pressure, pounds per square inch	230	380		420
2nd throttle pressure, pounds per square inch	45	70		120
After 1st stage, pounds per square inch	39	70		141
H.p. bled steam, pounds per square inch	15	28		55
L.p. exhaust, pounds per square inch	1	2		2
Vacuum, inch Hg.	29·3	29·2		28·7
Feed temperature at heater outlet, degrees Fahrenheit	194	252		278
Feed pump steam pressure, pounds per square inch	468	470		465
Feed pump discharge pressure, pounds per square inch	640	655		645
Feed pump exhaust pressure, pounds per square inch	12	13½		13½
Oil relay pressure, pounds per square inch	50	50		50
Oil governor pressure, pounds per square inch	34	31		25½
Oil bearings pressure, pounds per square inch	10	9¾		9¾
Alternator tachometer, r.p.m.	2,470	2,950		3,450
R.p.m.	90·5	106	116·5	126
S.h.p. desk meter	3,000	4,800	6,200	7,700
Thrust, tons	29·7	42	49·7	57·5
Speed, knots	12·081	13·698	14·938	16·085
Mean speed of north and south runs	12·932	14·648	15·726	16·772
	Contaminated steam generator working. Port 400 kW. set running. Port excitation distiller running.			

Notes: Oil fuel temperature pyrometer reading 48 degrees Fahrenheit high.
Burners 24 tips in use Nos. 3 and 4 fires steaming each boiler.
Contaminated steam generator supplying steam to:—Ballast pump.
Bilge pump.
Auxiliary feed pump: mono air pump.
Domestic purposes.
Heating coils in bunker tanks.
Oil fuel unit heaters.

TRIALS DATA

Summer ballast 27/7/49				Winter ballast 28/7/49				Loaded 28/7/49			
For. 13 ft. 6 in. Aft. 23 ft. 2 in.				For. 20 ft. 6 in. Aft. 25 ft. 7 in.				For. 28 ft. 6 in. Aft. 29 ft. 0 in.			
North 1	North 2	North 3	North 4	North 1	North 2	North 3	North 4	South 1	South 2	South 3	South 4
05·34	06·45	11·32	12·20	05·56	06·49	07·36	08·22	14·25	15·26	16·11	16·58
495	497	499	498	490	494	498	496	492	490	485	480
760	760	810	780	790	795	800	788	820	810	815	780
220	210	275	282	215	218	275	288	245	250	220	275
235	235	280	290	225	242	220	220	270	215	220	235
330	340	370	380	320	330	355	370	350	350	360	400
3	3·1	3·3	3·4	3·7	3·5	3	3·5	3·2	3	3·2	3·3
495	498	498	495	490	492	496	498	492	490	485	480
815	810	830	805	845	830	845	825	835	810	820	810
250	210	272	278	220	220	270	282	240	250	265	275
255	250	262	265	223	240	260	270	240	265	270	300
310	310	335	348	335	310	305	310	320	280	300	320
3	3	3·2	3·5	3	2·8	3	3·5	3·6	3·6	4	4
280	260	275	270	280	280	260	260	280	275	265	265
230	180	140	175	173	170	125	168	150	180	180	190
49,000	59,500	71,600	80,500	50,000	60,000	70,500	86,000	52,000	61,000	79,000	86,000
3–24	3–24	4–24	4–24	3–24	3–24	4–24	4–24	3–24	3–24	4–24	4–24
450	460	465	445	460	468	458	458	455	452	450	440
750	748	772	762	770	765	778	760	785	760	772	763
365	365	420	400	280	350	418	419	320	375	410	390
52	72	100	115	52	68	92	118	60	75	100	120
48	70	92	110	50	67	88	120	62	70	100	118
20	32	37	50	20	30	41	54	21	26	42	52
1	1	1½	1¾	1½	1¼	2	2	1¾	2	2	1½
29·8	29·7	29	28·9	29·15	29·1	29	28·85	29·1	29·1	29·0	28·9
		267	275	212	214	266	284	240	246	261	270
460	460	470	455	465	468	463	465	460	455	460	450
550	545	635	650	655	640	653	625	620	600	600	620
13	13	14	13½	13½	13½	13½	13½	13	13	13½	13
52	52	50	51	50	51	50	51	51	50	49	49
37	36	28	27	33	32	28	26	33	30	27½	28
10	9¾	9	9	10½	10½	10	9¾	10½	10	9½	9
2,600	2,850	3,180	3,380	2,620	2,850	3,100	3,350	2,650	2,850	3,100	3,320
95	104·5	116	121·5	95·25	104	113·8	122	96·6	104	114	121·8
3,450	4,650	6,300	7,350	3,650	4,750	6,150	7,750	3,750	4,700	6,150	7,600
	42	52·1	58	36·4	43·8	53·5	63	38·5	44·7	54·5	63
13·200	14·201	15·319	15·789	12·172	13·284	14·572	15·254	13·274	14·162	15·280	16·028
13·277	14·417	15·66	16·267	12·925	13·971	15·200	15·961	12·594	13·639	14·775	15·458
Contaminated steam generator working. Port 400 kW. set running. Port excitation distiller running.				Contaminated steam generator working. Starboard 400 kW. set running. Starboard excitation distiller running.				Contaminated steam generator working. Port and starboard 400 kW. running parallel. Mains excitation.			

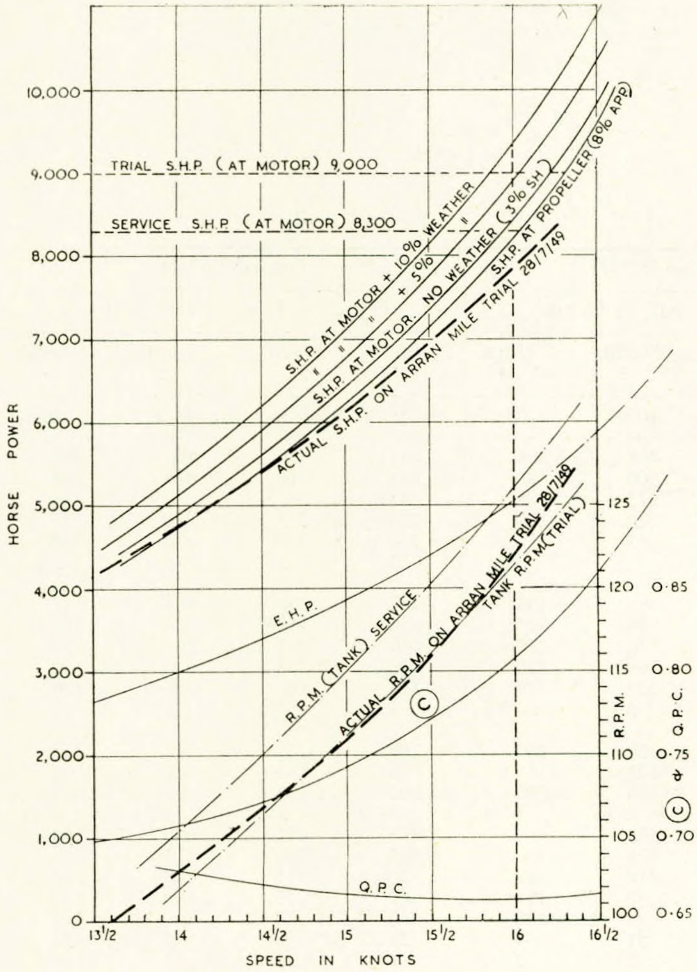


FIG. 33—*Estimated trial curves (based on tank results) winter ballast condition*

Dimensions: 510ft 0in. b.p. × 69ft. 0in. × 37ft. 0in.
Draught: 23ft. 0in. b.k. (mean)
Displacement: 16,725 tons (total)
Trim by stern: 4ft. 0in.
Level trim coefficients: Cb = 0·721; Cp = 0·747; Cm = 0·965;
L.C.B. = 6·12ft. f.

Performance

It will be appreciated that great effort has been expended in an endeavour to gain certain results and naturally the ship now being completed, exhaustive trials will be carried out.

The vessel therefore runs progressive speed trials under conditions exactly similar to those under which the model was tested, the results being shown earlier in this paper.

The scope of this paper will not permit more than an examination of the speed-power curves and compiled data (Tables 2-6) under full deadweight and various conditions of ballast (Figs. 31-34). However, it will be agreed that on plotting the actual result curves on the N.P.L. tank curves, there is a most satisfactory parity.

On Service

The tanker has now entered service but there has not been sufficient time to accumulate operating data for performance records: suffice to say the vessel and her machinery are running well with all indications of high efficiency and reliability.

A description of the vessel's last two cargoes will, the author feels, substantiate all that has been said about the ship's ability to carry composite cargoes:

Motor gasoline	7,800	tons
91 octane gasoline	668	tons
Gasoil	2,510	tons
Kerosine	915	tons
Lubricating oils (three grades)...	1,893	tons

70 octane gasoline	1,502	tons
White spirit	3,184	tons
80 octane gasoline	654	tons
Distilling benzine	3,606	tons
Lubricating oils (three grades)...	1,814	tons
High octane	2,953	tons

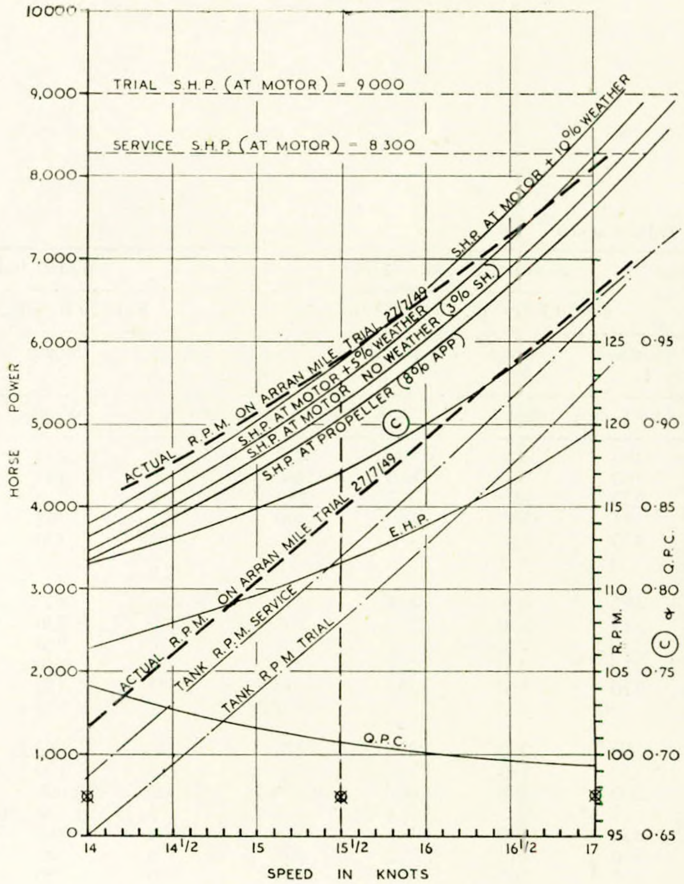


FIG. 34—*Estimated trial curves (based on tank results) Caribbean condition*

Dimensions: 510ft 0in. b.p. × 69ft. 0in. × 37ft. 0in.
Draught: 13ft. 6in. b.k. (mean)
Displacement: 9,186 tons (total)
Trim by stern: 8ft. 7in.
Level trim coefficients: Cb = 0·680; Cp = 0·721; Cm = 0·943;
L.C.B. = 7·68ft. f.

Finally, the author believes that the foregoing facts demonstrate that the design is a successful one and that the vessel will freight efficiently and safely all the varying grades of petroleum in bulk at a relatively low freighting cost and above all continue in service with the minimum of inefficiency periods.

It is also the author's firm opinion that the machinery will give little anxiety or heavy routine maintenance work to the engineers, which is what must be aimed at if one is to make life at sea at all attractive to the engineers.

The author acknowledges with grateful thanks his indebtedness to the following for information and assistance generously given: Mr. C. P. Harrison, Mr. T. H. Turner, Mr. G. A. Plummer, Mr. W. T. Butterwick, C.B.E., and also The British Shipbuilding Research Association. He also wishes to thank Miss M. C. Brown and Mr. E. J. Cuthbert for the work they have done in the preparation of the paper.

TABLE 3.

Condition

Brief statement of loading (i.e. loaded, ballast, etc.): condition No. 4 carribean ballast condition.

	Before trial	After trial
When taken	9 p.m. B.S.T.	Brodick Bay
Draught forward	9 ft. 6 in.	
Draught aft	18 ft. 5 in.	
Draught mean	13 ft. 11½ in.	
Displacement	9,595	
Block coefficient	0·6827	13 ft. 11½ in.
Prismatic coefficient	0·726	even keel
Tons per inch	64·11	
Moment to change trim 1 inch	1706	

Last undocked: 22nd June 1949
Bottom composition red lead over boiled oil.
State of bottom clean
Arrangement of plate edges i.e. riveted welded overlapped flush with fairings, etc:
Flush welded butts joggled plate edges.
Thickness of side shell amidships=0·68.

Propeller and stern arrangements

Diameter of screws: 18 ft. 6 in.
Number of blades: 4
Mean designed face pitch: 12·5 feet.
Tip clearance from hull: 1 ft. 6 in.
Mean measured face pitch: 12·69 feet.
Immersion of tips: 1·92 ft. out.
Developed area: 145 square feet.
Rotation right or left handed: right.
Centre line of shaft above keel: 11·09 feet.
Rake: 0·266 inch per foot.
Immersion of centre line of propeller / Diameter of propeller: $\frac{18{\cdot}42 - 11{\cdot}09}{18{\cdot}05} = \frac{7{\cdot}33}{18{\cdot}05} = 0{\cdot}396$
Bossings or shaft brackets: none.
Type of stern: cruiser.
Type of rudder and fin: semi-balanced double plate streamlined.

Group	Run No.	Direction	Time of day (G.M.T.) at start of run	Time on mile	Speed: Speed over (G.M.T.) Ground, knots	Speed: Mean for group, knots	r.p.m.: *	r.p.m.: r.p.m.	r.p.m.: Mean for group	Wattmeter h.p.‡	Mean of means for group	Corrected meter reading	s.h.p.	Mean of means for group	Corrected meter reading	Thrust (tons)	Total thrust corrected for trim	Mean of means for group	Apparent slip (measured pitch)	Weather: State of sea	Weather: Relative wind: Speed (knots)	Weather: Relative wind: Direction	Mean rudder angles
			h.m.	m.s.																			
I	1	N	16–01	4–58	12·081	12·932	S	90·36	90·60	2,941	2,941	38·6	3,048	3,034	29·7	31·3	32·6	32·3	–14·2%		32	10°P.	3S.
	2	S	16–29	4–21·2	13·782		S	90·83		2,940		38·0	3,020		29·0	30·6	31·9				12	85°S.	2S.
II	3	N	16–59	4–22·8	13·698	14·648	S	106·20	106·45	4,790	4,788	53·0	4,910	4,885	42·0	43·6	44·9	44·4	–10·2%	slight	36	30°P.	2S.
	4	S	17–22	3–50·8	15·597		I	106·69		4,786		52·0	4,860		41·0	42·6	43·9				16	85°S.	3S.
III	5	N	17–48	4–01	14·938	15·726	S	116·58	116·9	6,170	6,158	62·0	6,310	6,295	49·8	51·4	52·7	52·1	–7·8%	sea	34	30°P.	2S.
	6	S	18–10½	3–38	16·514		D	117·3†		6,146		61·4	6,280		48·5	50·1	51·4				16	40°S.	3S.
IV	7	N	18–33	3–43·8	16·085	16·772	D	125·82	125·81	7,726	7,789	72·0	7,900	7,979	57·5	59·1	60·4	61·2	–6·2%		37	30°P.	2S.
	8	S	18–53½	3–26·2	17·459		S	125·80		7,852		73·4	8,058		59·0	60·6	61·9				19½	90°S.	3S.

*The letters I, S or D entered in this column indicate whether the r.p.m. were increasing, steadying or diminishing during the last mile of approach.
†No autographic record available for this run. ‡Assuming a motor efficiency of 97%.

TABLE 4.

Condition

Brief statement of loading (i.e. loaded, ballast, etc.): condition No. 1 ballasted to deep load condition

	Before trial	After trial
Where taken	at Arran	
Draught forward	28 ft. 6 in.	
Draught aft	29 ft. 0 in.	
Draught mean	28 ft. 9 in.	
Displacement	21,470	
Block coefficient	0·7415	
Prismatic coefficient	0·763	28 ft. 9 in.
Tons per inch	70·55	even keel
Moment to change trim 1 inch	2,199	

Last undocked 22nd June 1949

Bottom composition red lead over boiled oil.

State of bottom clean.

Arrangement of plate edges i.e. riveted welded overlapped flush with fairings, etc.:

Flush welded butts joggled plate edges.

Thickness of side shell amidships=0·68.

Propeller and stern arrangements

Diameter of screws: 18 ft. 6 in.

Number of blades: 4.

Mean designed face pitch: 12·05 feet.

Tip clearance from hull: 1 ft. 6 in.

Mean measured face pitch: 12·69 feet.

Immersion of tips: 8·66 feet.

Developed area: 145 square feet.

Rotation right or left handed: right.

Centre line of shaft above keel: 11·09 feet.

Rake: 0·266 inch per foot.

$$\frac{\text{Immersion of centre line of propeller}}{\text{Diameter of propeller}} \qquad \frac{29-0-11\cdot09}{18\cdot05} = \frac{17\cdot91}{18\cdot05} = \cdot968$$

Bossings or shaft brackets: none.

Type of stern: cruiser.

Type of rudder and fin: semi-balanced double plate streamlined.

Group	Run No.	Direction	Time of day (G.M.T.) at start of run	Time on mile	Speed: Speed over (G.M.T.) Ground, knots	Speed: Mean for group, knots	r.p.m.:	r.p.m.: r.p.m.	r.p.m.: Mean for group	Wattmeter h.p.†	Mean of means for group	Corrected meter reading	s.h.p.	Mean of means for group	Corrected meter reading	Thrust (tons)	Total thrust corrected for trim	Mean of means for group	Apparent slip (measured pitch)	Weather: State of sea	Weather: Relative wind: Speed (knots)	Weather: Relative wind: Direction	Mean rudder angles
			h.m.	m.s.			*																
I	1	N	13–06	5–02·2	11·913	12·594	I	96·54	96·64	3,758	3,754	Torsion meter not working			39·0	40·6	40·6	40·4	–3·8%	slight	22	50°P.	5S.
	2	S	13–34	4–31·2	13·274		I	96·73		3,750					38·5	40·1	40·1			slight	16	100°S.	3S.
II	3	N	14–01	4–34½	13·115	13·639	I	103·28	103·68	4,738	4,730				45·5	47·1	47·1	46·7	–5·4%	slight	29	50°P.	4S.
	4	S	14–26	4–14·2	14·162		S	104·08		4,722					44·7	46·3	46·3			slight	18	100°S.	3S.
III	5	N	14–47	4–12·6	14·251	14·775	S	113·09	113·46	6,190	6,158				55·3	56·9	56·9	56·8	–3·8%	moderate	36	40°P.	4S.
	6	S	15–10	3–55·6	15·280		S	113·57		6,140					54·8	56·4	56·4			moderate	16	90°S.	4S.
	7	N	13–33	4–12·0	14·286			113·60		6,260					55·4	57·5	57·5			moderate	33	40°P.	4S.
IV	8	S	15–57½	3–44·6	16·028	15·458		121·44	120·585	7,610	7,525				63·0	64·6	64·6	64·7	–2·2%	moderate	15½	90°S.	3S.
	9	N	16–18	4–01·8	14·888		I	120·07		7,500					63·4	65·0	65·0			moderate	35	30°P.	3S.
	10	S	16–39	3–44·6	16·028		I	120·76		7,490					62·6	64·2	64·2			moderate	13	90°S.	3S.

*The letters I, S or D entered in this column indicate whether the r.p.m. were increasing, steadying or diminishing during the last mile of approach.

†Desk meter assuming a motor efficiency of 97%.

TABLE 5.

Condition

Brief statement of loading (i.e. loaded, ballast, etc.): condition No. 2 winter ballast condition.

	Before trial	After trial
Where taken	Brodick Bay	
Draught forward	20 ft. 6 in.	
Draught aft	25 ft. 7 in.	
Draught mean	23 ft. 0½ in.	
Displacement	16,746	
Block coefficient	0·7220	even keel
Prismatic coefficient	0·748	23 ft. 0½ in.
Tons per inch	67·47	
Moment to change trim 1 inch	1980	

Last undocked: 22nd June 1949
Bottom composition red lead over boiled oil.
State of bottom clean
Arrangement of plate edges i.e. riveted welded overlapped flush with fairings, etc:
Flush welded butts joggled plate edges.
Thickness of side shell amidships=0·68.

Propeller and stern arrangements

Diameter of screws: 18 ft. 6 in. — Number of blades: 4
Mean designed face pitch: 12·5 feet. — Tip clearance from hull: 1 ft. 6 in.
Mean measured face pitch: 12·69 feet. — Immersion of tips: 5·24 feet.
Developed area: 145 square feet.
Rotation right or left handed: right.
Centre line of shaft above keel: 11·09 feet. — Rake: 0·266 inch per foot.

$$\frac{\text{Immersion of centre line of propeller}}{\text{Diameter of propeller}} \qquad \frac{18{\cdot}42 - 11{\cdot}09}{18{\cdot}05} = \frac{14{\cdot}49}{18{\cdot}05} = 0{\cdot}783$$

Bossings or shaft brackets: none.
Type of stern: cruiser.
Type of rudder and fin: semi-balanced double plate streamlined.

Group	Run No.	Direction	Time of day (G.M.T.) at start of run	Time on mile	Speed		r.p.m.			Wattmeter h.p.‡	Mean of means for group	Corrected meter reading	s.h.p.	Mean of means for group	Corrected meter reading	Thrust (tons)	Total thrust corrected for trim	Mean of means for group	Apparent slip (measured pitch)	Weather			Mean rudder angles
					Speed over (G.M.T.) Ground, knots	Mean for group, knots		r.p.m.	Mean for group											State of sea	Relative wind Speed (knots)	Relative wind Direction	
			h.m.	m.s.			*																
I	1	N	04–56	4–55·8	12·171	12·925	S	95·31	95·66	3,642	3,651	44·4	3,683	3,684	36·4	38·0	38·7	38·4	–7·8%	slight	30	40°P.	4S.
	2	S	05–22	4–23·2	13·678		I	96·00		3,660		44·0	3,685		35·7	37·3	38·0			slight	23	90°S.	2S.
II	3	N	05–49	4–31	13·284	13·971	D	104·01	104·07	4,750	4,715	53·0	4,810	4,760	43·8	45·4	46·1	45·5	–7·0%	slight	33	40°P.	3S.
	4	S	06–13	4–5·6	14·658		S	104·12		4,680		51·8	4,710		42·5	44·1	44·8			slight	17	90°S.	1S.
III	5	N	06–36	4–7·8	14·527	15·200	S	114·19	114·15	6,150	6,165	64·0	6,390	6,340	53·5	55·1	55·8	55·5	–6·2%	slight	33	40°P.	3S.
	6	S	06–58	3–46·8	15·873		I	114·11		6,180		63·2	6,290		52·8	54·4	55·1			slight	16	90°S.	2S.
IV	7	N	07–22	3–56	15·254	15·961	I	122·15	122·6	7,748	7,752	74·4	7,893	7,954	63·0	64·6	65·3	65·0	–3·8%	moderate	42	40°P.	2S.
	8	S	07–44	3–36	16·667		I	123·0†		7,756		74·3	7,970		62·4	64·0	64·7			moderate	16	90°S.	2S.

*The letters I, S or D entered in this column indicate whether the r.p.m. were increasing, steadying or diminishing during the last mile of approach.
†No autographic r.p.m. record available for this run. ‡Desk meter assuming a motor efficiency of 97%.

TABLE 6.

Condition

Brief statement of loading (i.e. loaded, ballast, etc.): condition No. 3 summer ballast condition.

	Before trial	After trial
Where taken	at Arran	
Draught forward	13 ft. 4½ in.	
Draught aft	23 ft. 2¾ in.	
Draught mean	18 ft. 3⅝ in.	
Displacement	12,969	
Block coefficient	0·704	18 ft. 3⅝ in.
Prismatic coefficient	0·736	even keel
Tons per inch	65·46	
Moment to change trim 1 inch	1820	

Last undocked 22nd June 1949

Bottom composition red lead over boiled oil.

State of bottom clean

Arrangement of plate edges i.e. riveted welded overlapped flush with fairings, etc:—

Flush welded butts joggled plate edges.

Thickness of side shell amidships = 0·68.

Propeller and stern arrangements

Diameter of screws: 18 ft. 6 in.

Number of blades: 4

Mean designed face pitch: 12·5 feet.

Tip clearance from hull: 1 ft. 6 in.

Mean measured face pitch: 12·69 feet.

Immersion of tips: 2·90 feet.

Developed area: 145 square feet.

Rotation right or left handed: right.

Centre line of shaft above keel: 11·09 feet.

Rake: 0·266 inch per foot.

Immersion of centre line of propeller / Diameter of propeller $\frac{23·23 - 11·09}{18·5} = \frac{12·14}{18·05} = 0·656$

Bossings or shaft brackets: none.

Type of stern: cruiser.

Type of rudder and fin: semi-balanced double plate streamlined.

					Speed		r.p.m.													Weather			
																					Relative wind		
Group	Run No.	Direction	Time of day (G.M.T.) at start of run h.m.	Time on mile m.s.	Speed over (G.M.T.) Ground, knots	Mean for group, knots	*	r.p.m.	Mean for group	Wattmeter h.p.‡	Mean of means for group	Corrected meter reading	s.h.p.	Mean of means for group	Corrected meter reading	Thrust (tons)	Total thrust corrected for trim	Mean of means for group	Apparent slip (measured pitch)	State of sea	Speed (knots)	Direction	Mean rudder angles
I	1	N	04–34	4–36½	13·020	13·177	S	95·34	95·70	3,420	3,475	42	3,520	3,540	—	—			–10·2%	slight	16	0	1S.
	2	S	05–10½	4–30	13·333		S	96·06		3,530		42·5	3,560		—	—				slight	14	0	—
II	3	N	05–44	4–13½	14·201		S	104·69	104·18	4,630	4,631	51·4	4,707	4,684						slight	20	0	—
	4	S	06–14	posts obscured by fog			S	103·67		4,631		51·5	4,660		41·0	42·6	44·0			slight	12	10°P.	—
	5	N	06–36	4–06	14·634		I	105·5†		4,684		52	4,780		41·5	43·1	44·5			slight	14	0	3S.
	6	S	posts obscured by fog																	slight	9	0	3S.
III	7	N	09–20	4–07·2	14·563	14·482		105·27	105·38	4,702	4,693	51·5	4,740	4,720	42·0	43·6	45·0	44·8	–9·4%	slight	18	10°P.	2S.
	8	S	09–48	4–10	14·400		I	105·48		4,683		51	4,700		41·5	43·1	44·5			slight	14	80°S.	2S.
IV	9	N	10–31	3–55	15·319	15·66	I	116†	116	6,324	6,314	64·3	6,500	6,459	52·1	53·7	55·1	54·6	–7·8%	slight	25	30°P.	3S.
	10	S	10–53	3–45	16·000		S	115·98		6,304		63·3	6,418		51·1	52·7	54·1			slight	14	80°S.	—
V	11	N	11–19	3–48	15·789	16·267	D	121·5†	121·6	7,376	7,358	70·9	7,510	7,490	58·0	59·6	61·0	60·9	–7·0%	slight	32	30°P.	3S.
	12	S	11–44	3–35	16·744		S	121·75		7,340		70·5	7,470		57·8	59·4	60·8			slight	24	80°S.	4S.

*The letters I, S or D entered in this column indicate whether the r.p.m. were increasing, steadying or diminishing during the last mile of approach.

†No autographic revs. records available for these runs.

‡Assuming a motor efficiency of 97%.

Discussion and Author's Reply

MR. T. HALLIDAY TURNER (Member) said all the details were, he thought, fully and ably described in the paper. There were one or two novel features in the machinery of this vessel, one being a mixed pressure governing gear to the 400 kW. auxiliary generating sets. It had been evident that under certain conditions a considerable surplus of exhaust steam had to be utilized and this was done by having the 400 kW. set take this surplus and so do useful work. The drawings gave a full indication of the main turbine and its governor gear, and these turbines were similar to those in a number of ships, the *Bermuda* and the *Bel* ships for instance, and they had given no anxiety whatsoever.

MR. NELSON said that as mentioned in the paper the usage compass of steam in a tanker was very wide and under certain conditions the steam in the exhaust, or should one say the low pressure range, reached high quantities. In order to avoid passing the excessive exhaust steam into the main condenser when main propulsion machinery was shut down, which constituted a danger unless the main turbines were rotated, the mixed pressure control took this surplus into a later stage of the auxiliary turbine, simultaneously shutting down high pressure steam to the high pressure belt and *vice-versa*.

The gear had required certain adjustments and minor modifications but now was working perfectly.

MR. G. A. PLUMMER (Member) said that some of the deepest consideration, from his own experience, had been in connexion with the feed system, which was really an excellent one. It compared favourably with most central power stations, and there were very few ships at the present time—at any rate in this country—with such a complete system.

The boiler plant, as would be seen, he thought, from Fig. 2 and Figs. 12(a) and 12(b), appeared, on the face of it, to take up a fairly small amount of space compared with boilers of the natural circulation design, but here they had really been spread out somewhat. In future, there could be a very considerable reduction in the size of such boilers, particularly if advantage were taken of the more recent advances that had been made in the combustion of fuel oil.

Great care had been devoted to preventing the possibility of serious corrosion in the air heater. One was working, here, at a fairly high efficiency—something of the order of 86 per cent on the gross calorific value which was about 91 per cent on the net calorific value. To achieve that it was necessary to get down to fairly low exit gas temperatures. A limit had been fixed at what was thought at the moment to be the danger point, and precautions had been taken whereby the temperatures at low loads could be maintained above this critical point. That made for a rather complicated multiplicity of dampers, and there again in future designs, there could be still further improvement and simplification.

His own reactions after the experience he had had of this ship led to one very valuable lesson. One was advancing a little too quickly. The time had come, he thought, to get down to a certain amount of rationalization and simplification in engineering, particularly in the Mercantile Marine.

A study of the various documents and drawings right through the paper would show that the author had been a pioneer on this ship. He had not been satisfied to sit down to old-established ideas and conventional features. In respect of the boilers, again, he had been a pioneer. It was true that in Germany there had been some eighty-four Lamont boilers in German naval vessels up to 1,200lb. per sq. in. pressure. In France in the French Mercantile Marine there were Lamont boilers up to 900lb. per sq. in. 900 deg. F., and even in this country the British Navy had employed a few very small very light-weight Lamont boilers for steam gunboats.* But this was the first time in the British Mercantile Marine that the Lamont boiler had been employed.

He would like to ask whether the author did not consider, particularly with turbo-electric drive, that a considerable increase in the steam pressure and temperature could be tolerated and might even be attractive in future designs? Could not more use be made, also, of alternating current for auxiliaries, even cargo pumping? In other words, could not the main engines be considered as a central power station to supply power for all purposes on the ship? Could they not be used for propulsion, auxiliary power and lighting, pass-out steam being mainly used for all heating purposes?

In the last few years he had had something to do with central power stations. Each could, he thought, learn something from the other. He seriously put forward these proposals, and thought something still further could be done if steam was going to compete, as it should do, with Diesel and other forms of drives.

MR. NELSON said that he would refer firstly to the question of higher pressures and higher temperatures in modern installations. He had thought himself rather venturesome in going up to 500lb. per sq. in. and 800 deg. F. because there was one factor which must be remembered in considering this question. One must not be carried away too much by what was done in land practice. On land an installation was set down to a solid foundation and generally worked at a fixed, steady rate. From what he had seen of high pressure installations, he really thought there was a lot to learn and there were big advances to be made in steam supply, particularly the piping and the jointing. He had seen a certain amount of what he would call not ignorance but lack of knowledge or lack of appreciation of the stresses which were set up on these high pressure pipes when they were varying from full power at the highest temperature to lowest temperature on the plant being shut down. One must remember that on the marine job the plant was very often being shut down and opened up and therefore these stresses were set up. Before venturing into the higher pressure and higher temperature spheres, as far as he was concerned, he must be convinced that a careful study had been made of the stresses that were being put on to the pipe when it was jointed up cold as it could not be jointed up when it was hot. He had seen engineers installing high pressure pipes where they had made the trouble even worse by introducing fancy bends. There must be the right draw-up clearances between flanges when the pipe was cold so that the stresses on the pipe and flange would be halved when they were hot. However, he quite appreciated the point about high efficiencies, although one must bear in mind these factors, added to the fact that the marine plant was subjected to vibration in a heavy sea and to different influences peculiar to marine service which would probably give trouble,

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* LAY, H. A. K. and BAKER, L. 1948-9. Trans.I.Mar.E., Vol. 60, p. 190, "Steam Gunboat Machinery—A Light-weight Steam Plant".

and it was in this direction that attention should be directed.

With regard to a.c. supply, this had been near to his heart for many years, and he had wondered why Lloyd's Register were coming down on a.c. As had been seen, in order to satisfy Lloyd's he had had to put in two 30 kW. motor generators because they would not allow 200 volts on the deck. He was therefore taking 220 volts d.c. and converting them at a loss into 100 volts. He did not think that was right and in his opinion—and he was thinking in terms of safety—they were being unduly penalized.

With special regard to a.c. current, he thought many people forgot that most of the motors driving auxiliaries on a ship, whether ancillary or purely auxiliary, were required to run at varying speeds to obtain the full efficiency of these units. Unfortunately with a.c. one had to run the motor at constant speed, and, as an instance, an a.c. motor driven centrifugal main circulating pump where the output must be regulated according to main engine power and sea water temperatures by restricting the suction of the pump, the consequential turbulence and agitation caused endless trouble on the pump and condenser.

One must put the advantages and disadvantages of d.c. against a.c., and although the installations costs of a.c. would probably be cheaper, he had grave doubts whether whole-hearted adoption of a.c. would ultimately show the dividends claimed by some people. It was mainly a question of variable speeds.

Mr. B. Rogers (Member) asked whether Mr. Nelson could give any indication of the fuel cost per shaft horse-power of the vessel and the fuel cost per kW. hour from the main alternator.

Mr. Nelson pointed out that they were not in a true position yet to find out what they were getting out of the ship in the way of fuel consumption for propulsion only. He defied anyone, with a plant of this sort, to be able offhand to differentiate what was the amount of steam actually being used for the propelling unit. In modern vessels, in this case steam driven, auxiliary plant had increased enormously to meet the demands of all the mechanical means which had been introduced into the working of the tanker together with all the added amenities for the officers' and crew's comfort.

It all took steam initially from the main steam generating plant and unless the Chief Engineer meticulously logged times, checked bunkers, etc., it was difficult to obtain an actual figure as to what power was being developed at the propeller shaft per lb. of fuel consumed. Therefore any figure must be an estimate.

He had been able to work out a figure of 0·702lb. fuel per b.h.p. developed at propeller shaft or 0·935 per kW. hour. However, there was every indication that this figure would be very much improved upon in future.

Captain J. P. Thomson (Associate) observed that very little had been said about the hull and hull design. The question of the design of a tanker to maintain a speed on any particular schedule was one requiring the closest attention. For instance, a tanker on the Eastern run would probably average a knot per hour more speed than would be possible when on the North Atlantic run. The form of the hull had to receive the most careful consideration in order to give the best service speed for the trade intended. The vessel carried her parallel body well forward above the water line and still maintained a good underwater form. He recalled the earlier vessel with the sharp bow which used to be spoken of as a big ship with a small bow.

Construction was a subject about which they did not know much, but they were particularly well pleased with the corrugated bulkheads and corrugated designs for cleaning the tankers. The Butterworth system, to which the author had already referred, contributed real efficiency in this respect. The design was not final and improvements would no doubt be possible.

Mr. Nelson said that Capt. Thomson observed that very little had been said about the hull and hull design: the author however would remind him of the title of the paper. The scope of the paper would not allow for embracing detailed design of the hull and it was thought sufficient had been given to illustrate clearly the major points of the vessel to dovetail in with the general theme of the paper.

He endorsed the remarks with regard to the meteorological conditions of the trade factors as shown early in the paper and naturally the comment that the design was not final.

Mr. S. Hunn mentioned that it was not always appreciated that the minutest trace of one petroleum product mixed with another might put the contaminated parcel completely off specification and make it unsaleable. Cases had occurred when a quantity of contaminant too small to be detected had found its way into a particular product, and upon arrival at the port of discharge it was impossible to use the parcel as such; the only course had been to ship it back to the refinery where it had to be re-treated. He could not stress too strongly the need for the designer of composite cargo carrying tankers to give special consideration and care to complete isolation of the different types of products carried, and he felt that the vessel described by the author was a very fine effort in this direction.

Correspondence

Mr. Carlton Garratt (Member) wrote that the author mentioned, in discussion following the paper, the difficulty of separating main engine fuel consumption from the numerous auxiliary services. But information as to all purposes consumption would be very interesting, as it could be compared with a motor ship performance—all purposes being taken in its broadest sense of one round voyage, including discharge of cargo.

It was surprising to note the statement on p. 182 that inconsistency in the distribution of strength of hull had contributed to rapid internal deterioration. He had not experienced such a marked effect, though too much stiffening in a cofferdam was agreed to be undesirable. It was noted that a cofferdam was introduced between Nos. 4 and 5 tanks. Would the author agree that cofferdams were much less essential for segregation now that bulkheads were of all welded construction?

Finally, he ventured to disagree with the author regarding corrugated bulkheads. Cracking troubles had not yet been completely cured and, in addition, the extra depth of stringer increased the amount of horizontal area, which was an objectionable feature when cleaning tanks, even if a mechanical system of tank cleaning was used.

Replying to Mr. Carlton Garratt, **Mr. Nelson** wrote that he would venture to suggest that the conditions varied so much on each particular voyage that to attempt a comparison for a voyage between the vessel in question and a similar motor ship would be purely hypothetical. However he gave hereunder some figures and hoped they would prove useful.

Consumptions

	Duration, days	hrs.	mins.	Speed, knots	Main engines and ship's services	Heating coils, cleaning tanks, etc.	Total consumption
Load passage	10	18	—	14·20	50·8 tons	52·0 tons	539 tons
Light passage	10	09	42	15·53	44·8 tons	57·0 tons	523 tons
Discharging 13,800 tons (consisting of 7 different grades at Rio de Janeiro and Santos)							69 tons
Total consumption round voyage							1,131 tons

With regard to the question of flexing being a contributory factor in connexion with internal deterioration, this was now an accepted fact that internal wastage was most severe at positions on shell, stringers or bulkheads where the effects of flexing were most evident. Tanker designers and owners' technical staffs were fully alive to this and were concentrating on bringing this flexing down to a minimum.

The undesirability of stiffness in way of cofferdams was accepted but was truly related to his previous remarks and was purely a question of length of span.

He agreed that cofferdams might not be so essential in light of the fact that bulkheads were now all-welded, but surely the question of the lead of pipelines was a salient feature when considering segregation of grades.

He would venture to assure Mr. Garratt that the present design of corrugated bulkhead as developed and fitted in this country was not revealing the troubles mentioned and the objectionable feature of the extra depth of stringer when cleaning tanks did not really exist.

MR. C. P. HARRISON (Member) wrote that having explained the factors involved in the design of the ship herself, the author had marshalled his reasons for using turbo-electric machinery. This machinery was far from experimental and it took its place on its own merits for equal consideration with other forms of propulsive machinery.

There seemed still to be an impression that it was complicated—it was not. The electrical side itself could almost be ignored; it had fewer vulnerable parts than a motor driven circulating pump; the steam end was a simple single casing job, and as the author had shown, was easily adaptable to the particular steam conditions and auxiliary steam demands for which he had to cater in his tanker.

Service experience would no doubt prove the first choice to be right and the advantages would be more apparent in the later period of the ship's life.

MR. E. G. WARNE (Member) wrote that he had found some difficulty in checking the actual length of the engine room in which it was originally proposed to install either Diesel or steam machinery. The author remarked "given a machinery space of 97 feet in length" and included a plan (Fig. 3) on which there was no scale. Turning, therefore, to the principal plans (Fig. 12) it was seen that the engine room length was 88 feet, extending from frame Nos. 10-47. One had to accept this and transfer a length representing 88 feet to Fig. 3, frame Nos. 10-47, which, however, showed the engine room extending from frame Nos. 10-49.

On this basis, the outline for a six-cylinder opposed-piston engine gave a length of 47 feet and that for a single-acting two-stroke engine, 60 feet. The drawing indicated a 6,000 b.h.p. opposed-piston engine, and the text gave it as 6,600 b.h.p. One was further perplexed to note that the single-acting, two-stroke engine outline drawing for comparison was of 9,000 b.h.p.

It would be ascertained from the paper that a speed of 15·45 knots was attainable with the ship loaded, the power being 7,600 s.h.p. and the revolutions 121·8 r.p.m. Table 4 more or less confirmed this. The service speed of the ship was given as 14¾ knots, and from the curves shown in Fig. 31 it was seen that the corresponding horse-power on the Arran mile trial was 6,200 s.h.p. and the revolutions 113·5 r.p.m. A six-cylinder opposed-piston engine, 670 mm. bore and 2,320 mm. stroke, working at a mean effective pressure of 5 kg. per sq. cm. (71lb. per sq. in.) should, according to his calculations, give about the same result. The length of this engine, including the thrust block and flywheel, was approximately 56 feet, leaving 32 feet clear, fore and aft, in an 88 feet engine room.

However much the provision of steam in a tanker might be a dominating factor in the choice of machinery, it was clearly not a question of the space taken up by the Diesel engine that decided the issue. A series of eight 16,000-ton tankers had just been built in this country, in which oil engines of the size and power apparently needed by the author had been installed, and there were other tankers with even higher power fitted with double-acting, two-stroke Diesel machinery built abroad.

In order to ascertain the tendency to install double-acting two-stroke machinery in large tankers, he had examined a list of orders placed between 1935 and 1940, omitting all tankers under 13,500 tons deadweight. In those five years there were sixty-five of these large tankers built and fitted with double-acting two-stroke machinery, although the speeds in most cases did not exceed 13 knots. The aggregate output of these engines was very probably more than 300,000 b.h.p. Not one of the engines, however, was built in this country.

In the first two months of this year, seven large motor tankers totalling about 120,000 tons d.w.c. had been ordered, launched or completed, with machinery for these vessels (the smallest 12,600 tons d.w.c. and the largest 20,000 tons d.w.c.) specified to be of the double-acting two-stroke-type, having an aggregate output of approximately 45,000 b.h.p.

MR. NELSON replied to Mr. Warne that the length of engine room as shown on Fig. 12 was later increased and extended to frame No. 49.

The reason for not setting down the dimensions of a 9,000 s.h.p. opposed-piston engine was the difficulty in procuring the actual dimensions of an engine of that power.

Mr. Warne's calculations to the effect that the six cylinder opposed-piston engine of the dimensions given would give the vessel 14¾ knots at the same power as the existing machinery were beside the point.

What he wanted was 9,000 s.h.p. which would easily give 15¾ knots in smooth water on trials under full designed deadweight, which power was calculated to give the vessel a steady speed of 14¾ knots in actual service and he would say that Mr. Warne's engine had not the power to meet this requirement. This meant that the Diesel engine must be 9,000 s.h.p. and he still maintained that to get a direct-drive slow running Diesel engine of this power, together with having to provide a flat and space for two relatively large Scotch type boilers, also the many varied auxiliaries in the larger space as shown would constitute a problem.

He must disagree with Mr. Warne in saying that the provision of steam might be the dominating factor. Space was one of the factors.

He had noted Mr. Warne's remarks on the recent installing of double-acting two-stroke machinery in large tankers being built abroad, but he could only enlarge on the remarks on p. 180 of the paper.

It had been proved again and again that in order to get satisfactory results from double-acting two-stroke machinery, a regular system of maintenance and overhaul was essential. With liners or cargo-liner-type of vessels with constant trade and regular ports this was apparently achieved. With a tanker it was an entirely different matter: trade and loading and discharging ports might not be regular and the stay in each port was only a matter of hours. Further the oil installations, where the tanker lay, were invariably many miles from a source where facilities for repairs together with holding of vital spare parts were available. A tanker should be fitted with machinery which required the least regular short period maintenance of the main machinery also the minimum amount of replacement of spare parts between the inefficiency periods.

Mr. Nelson wrote that in the course of the discussion it had been pointed out to him that there were certain errors in the Tables at the end of the paper. These corrections had been made in the Tables as now published.

INSTITUTE ACTIVITIES

MINUTES OF PROCEEDINGS OF THE ORDINARY MEETING HELD AT THE INSTITUTE ON TUESDAY 14TH MARCH 1950

An ordinary meeting was held at the Institute on Tuesday, 14th March 1950 at 5.30 p.m. Mr. A. F. C. Timpson (Member of Council) was in the Chair. A paper entitled "Factors Governing the Design of a Modern Tanker with Special Reference to Machinery", by Mr. W. L. Nelson, O.B.E. (Member of Council), was read and discussed. Eighty-seven members and visitors were present and five speakers took part in the discussion.

A vote of thanks to the author, proposed by Mr. S. Eady Laxton, was carried with acclamation.

The meeting terminated at 7.10 p.m.

JUNIOR SECTION

Lecture at Bristol

Mr. Hogg's lecture given at the Merchant Venturers' College, Bristol, on the 17th February 1950, was followed with keen interest by all present.

The lecture was demonstrated with slides which enabled those present to follow in detail the points made by the lecturer.

At the conclusion Mr. Hogg asked for questions and for quite a long while he was kept busy answering many of a varied nature, showing that the questioners had missed little, if anything, of the points brought out during the lecture.

Mr. E. Poole, Vice Principal of the College of Technology, took the Chair, and introduced the lecturer in a few well-chosen words.

At the termination of the proceedings a vote of thanks to Mr. Hogg was proposed by Mr. Gilling and seconded by Mr. Peck; Councillor Walker proposed a vote of thanks to the Chairman.

MEMBERSHIP ELECTIONS

Elected 3rd April 1950

MEMBERS

Walter Johnston Andrews
Leslie Birts, B.Sc., A.C.G.I.
Arthur Graham Bridgman, Lt.-Com'r(E), R.C.N.
Thomas Forrest Browell, Lieut.(E), R.N.
James William Burns
George Cottington, O.B.E.
Christopher Rostron Cragie
Bernard Cowling Fleuret
George Florance Fife
George Alexander Jack
George Edward Kerr
Adriaan Knape
Robert Linton
Charles McLachlan
James Barker McLaren
George McMullan
James Moffat
Hilton Conmor Newson
Noel Richardson Nickalls
George Frederick Osborne, Lieut.(E), R.N.
Norman Percival Rice
George Felix Souza
Alexander Spence
Frederick Horace Tickell
Louis Hector Ivanhoe Todd

ASSOCIATE MEMBERS

Gordon McKay
F. C. A. Ward, Lieut.(E), R.N.

ASSOCIATES

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Donald Abraham Mathew
William Purdie McNab
Clarence Frank Nossiter
William Charles Richards
Herbert Edwin George Streater
John William Waller
Ronald Thomas Woods

TRANSFER FROM ASSOCIATE TO MEMBER

Ralph Bartlett
William Frederick Hardstone
James Lawrence Kinmond
Norman Lofthouse
Walter John George Wilson

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Oliver Addison Glass
John Higgins

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER

William Hutton Peterson, Lieut.(E), R.N.Z.N.

TRANSFER FROM STUDENT TO ASSOCIATE

William Francis John Dunlop Ewart

Elected 1st May 1950

MEMBERS

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Charles Ernest Burke
Eric George Douglas, Lieut.(E), R.N.
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Pestonji Byramji Javat
Paul Kennedy
Carl Wilhelm Prohaska
Leslie William Reavill, Lieut.(E), D.S.C., R.N.
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ASSOCIATES

John Robert Armstrong
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Ralph Melville Richards

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
Frederick Aitchison Davidson

TRANSFER FROM ASSOCIATE TO MEMBER
Bruce Anderson
Ronald Campbell Brown, B.Eng.
Laurence Edward Massey
Alan Herbert Morton
William Roberts
John Rodgers
Thomas Norman Taylor

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Cecil Thomas Tippin

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
William George Clark, B.Eng.

TRANSFER FROM GRADUATE TO ASSOCIATE
Leslie Alan Lee

LOCAL SECTIONS

Sydney

The Annual General Meeting of the Sydney Local Section was held on Thursday, 9th March 1950 at 7.45 p.m. at Science House, Sydney. Mr. H. A. Garnett (Local Vice-President) was in the Chair and the assembly consisted of seventy-two members and guests. The Minutes of the preceding meeting held on the 24th August 1949 were read and confirmed.

The Committee presented the first Annual Report reviewing the proceedings of the first year of the local section, following an inaugural meeting of members of the Institute resident in New South Wales, held at Science House, Sydney, on 13th December 1948, when it was resolved to seek the approval of the Council of the Institute to form a Local Section in Sydney. The approval was granted and the Local Section was formed with a membership which reached sixty-nine by February 1949 and gradually increased to eighty-one by the end of the year.

The Council laid down a set of rules for the guidance of Local Sections Overseas and approved of the Local Rules which were drawn up for the working of the Section and these are now in operation.

There was no doubt that the establishment of this Section had filled a long-felt want and, judging by the enthusiasm of the members, its success was assured. Partly as the result of the inauguration of the Local Section, several new members had joined the Institute in the past few months.

The first function was a dinner at Petty's Hotel on 18th January, to the President of the Institute, Sir Robert Micklem, C.B.E., who was visiting Australia at the time and this was attended by sixty-two members and guests and proved a most successful evening.

Following this, an ordinary Meeting was held at Science House on 29th April, when Mr. H. P. Weymouth delivered a paper, which had since been published in the TRANSACTIONS, on "Some Interesting Examples of Geared Diesel Drives"; the attendance was fifty-eight.

The next Meeting on 24th August was attended by seventy-seven members, when Mr. C. R. Bickford, the Chief Engineer of the Maritime Services Board of N.S.W., gave a paper on "Modern Port Developments with Particular Reference to Sydney".

The final function for the year was the annual dinner held at the Carlton Hotel on 16th November and attended by eighty members and guests, including Mr. Sydney A. Smith, a former Member of the Council, who was on a visit to Sydney at the time. Again, this was a most successful evening and the good fellowship which exists between the members of the Section was very much in evidence.

The audited statement of the accounts showed that the year's operations had resulted in a credit balance of £29 10s. 11d.

It was regretted that the deaths of two members, Mr. D. W. Wansey and Mr. A. G. Burn occurred during the year.

The adoption of the Annual Report and Balance Sheet for the year 1949 was proposed by the Chairman, seconded by Mr. H. P. Weymouth and carried unanimously.

The following members were elected office bearers for 1950:—

Chairman: H. A. Garnett
Honorary Secretary: Eng'r Capt. G. I. D. Hutcheson, R.A.N.(ret)
Honorary Treasurer: N. A. Grieves
Committee: W. G. C. Butcher, D. W. Findlay, F. W. Harper, G. T. Marriner, F. W. Ward, and H. P. Weymouth.

A lecture was then delivered by Mr. J. Ross on "Closed Feed Systems". This was illustrated by lantern slides and the whole subject was dealt with in a most interesting and comprehensive manner. A discussion followed with several members taking part.

A vote of thanks to Mr. Ross was proposed by Mr. D. Findlay and carried by acclamation.

Supper was served at the conclusion of the meeting and a further opportunity afforded to members and their guests for informal discussion.

The first meeting for Students and Apprentices of the Sydney Local Section of the Institute was held at Science House, Sydney, on Friday, 14th April 1950, and was an unqualified success. The Local Vice-President was in the Chair and the attendance numbering 120 completely filled the hall.

Careful attention was paid to Mr. Butcher's paper which was followed by a talk by Mr. C. McLachlan, the Superintendent Engineer of the Adelaide Steamship Company, on "Training and Experience at Sea". This dealt with many of the immediate problems of the young men and aroused a great deal of interest. After the talk the Honorary Secretary of Sydney Local Section, Eng'r Capt G. I. D. Hutcheson, R.A.N.(ret), spoke for a few minutes on the advantages of those who might be qualified seeking membership of the Institute. A discussion then ensued.

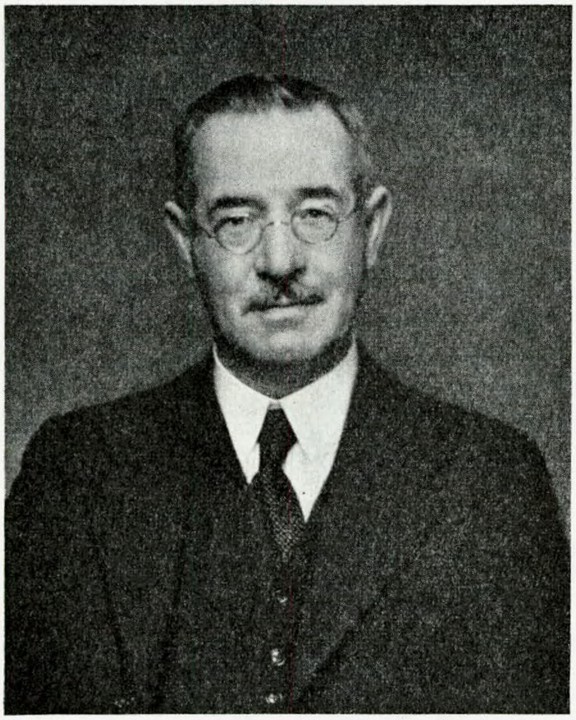
At the conclusion of the discussion a vote of thanks was proposed to Messrs. Butcher and McLachlan by Mr. J. Munro and he voiced the feeling of all concerned that the meeting had been a great success. It was gratifying that every student and apprentice who accepted an invitation was present and many expressions of appreciation were received.

Cardiff

On the 11th and 25th March 1950 a party of twenty Members, Students and apprentices visited the steel works of Messrs. Guest, Keen and Baldwins. The visits were most interesting and all the various stages in the manufacture of iron and steel were shown and explained to the parties by experienced guides.

On the 20th March 1950 Mr. S. B. Jackson (Member) delivered his lecture entitled "Diesel Driven Heat Pump Evaporators" at the South Wales Institute of Engineers. The lecture was very interesting but the attendance was much lower than usual owing to the weather conditions.

OBITUARY



Arthur Huntley Parker (Member 6741) was born in 1869 and educated at the Coopers School. At the age of sixteen he joined the staff of Messrs. Fraser and Fraser of Bromley-by-Bow as a junior in the drawing office. His aptitude as a draughtsman was soon recognized and he obtained rapid promotion. After a short period in the workshops he joined (in 1892) Messrs. Holmans, London, sailing as third engineer. In 1896 he transferred to the New Zealand Shipping Co., rising to Chief Engineer of the *Rakaia* in 1904. Mr. Parker sailed in various vessels of the New Zealand Shipping Co. until he was appointed Assistant Superintendent Engineer in 1920, being promoted to Superintendent Engineer in 1925, which position he held until his retirement in 1938. In his official capacity he showed consideration for juniors and seniors alike and his efforts were largely responsible for the great improvements in accommodation for engineers afloat. Mr. Parker was elected a Member in 1931 and was a Member of Council from 1937 to 1940. He was also one of the earliest Life Members of the Guild of Benevolence. He died on 17th January 1950 aged eighty-one years. He leaves a widow and two daughters.

William Elder (Member 6826) was born in 1874 at Garlieston, Scotland. He served his apprenticeship with Messrs. Cochran and Co., of Birkenhead. His sea service was with Messrs. W. Johnston and Co. and Pacific Steam Navigation Co. of Liverpool. In 1902 he went to Australia where he was employed as an engineer at Hoskins Iron and Steel Works, later going to Messrs. Buyacott and Co., Engineers of Sydney, New South Wales, as Works Manager. In 1912 he commenced business on his own account in an engineering shop at Balmain, Sydney, and in 1924 transferred his activities to private practice as a consulting engineer in which he was actively engaged up to the time of his death on the 3rd October 1949. He was elected a Member in 1931.

Edward William Green (Member 1395) was born in 1873 and educated at Harrow. He served his apprenticeship with Clydebank Engineering and Shipbuilding Co., Ltd., followed by a few years at sea with the Orient Line. In 1899 he was appointed Managing Engineer of Messrs. R. and H. Green, Ltd., Blackwall, and shortly after was elected a director. He became a Member of the Institute in 1899. Mr. Green was awarded the O.B.E. for his services during the 1914-18 war. In 1939 he was appointed a director of Messrs. R. and H. Green and Silley Weir. He died on 31st January 1950 at the age of seventy-seven and leaves a widow and son.

Thomas Henry Hunstone (Member 7570) was born in 1871. He served his apprenticeship with Bailey, Pegg and Co., Brierley Hill, and his following sea-service was with Burdick and Cook. He then spent a short time with Lysaght's Mills, Newport, and the Monmouthshire Steel and Tin Plate Co. In 1906 he was appointed Chief Engineer of the Staffordshire County Council Institution, Wordsley, and served there until he retired about twelve years ago. He was elected a Member in 1934. He died on 1st February 1950 and leaves a widow.

Neil Lang (Member 3553) died on board the Douglas Steamship Company's S.S. *Haiyang* in Colombo on Christmas Day 1940 after a lifetime spent as an engineer in the Far East. He served his apprenticeship with Messrs. Denny and Co., at Dumbarton, and thereafter worked with this firm and also with Messrs. John Brown and Co., of Clydebank. Then the engineers' strike of 1897-98 made him decide to join other Clydeside engineers of his day in seeking a career in the Far East. 1898-1904 were spent with Messrs. Butterfield and Swire at Shanghai until an attack of cholera caused him to return home in 1904 when he took his Chief Engineer's Certificate. This, incidentally, was his only leave from the China coast spent at home in forty-two years. In 1905 he returned to China in the S.S. *Fengtien* from Scott's of Greenock, and in 1908 he joined the Douglas Steamship Company of Hongkong. Through the years of the 1914-1918 war, and up to the time of his death he sailed with this company who ultimately erected a memorial tablet on his grave in Colombo as a mark of their esteem. He was elected a Member in 1919.

Norman McCrum (Member 9598) was born in 1902 at Sunderland, but at an early age went with his family to Ireland and was educated at Dundalk Grammar School. He served his apprenticeship with Messrs. Richardson Westgarth and Co., Sunderland. His sea-going career was with Messrs. Runciman and Co., Newcastle-on-Tyne, Elder Dempster and Co., and the Belfast Steam Ship Co. He joined the City of Leicester Waterworks in 1940, being appointed Superintendent in 1943. Mr. McCrum was elected a Member in 1943. He was also a Member of the Leicester Engineering Society. He died on 27th February 1950 after a brief illness and leaves a widow.

John Bernard Norris (Member 8189) was born in 1895. He served his apprenticeship with Messrs. Napier and Co., Acton Vale, and his sea service was with Messrs. Flannery, Baggallay and Johnson. He was awarded Lloyd's Medal in 1918. In 1925 he formed his own company, The Thale Engineering Co. He was appointed Engineer to the Huileries du Congo Belge (Levers) in 1928 and rose by successive stages until he was nominated Chief Engineer in 1945. Mr. Norris was elected an Associate in 1936 and a Member in 1942. He was also an Associate Member of the Institution of Mechanical Engineers. He died very suddenly at Alberta, Belgian Congo, on 18th January and leaves a widow and two sons.

GEORGE WILLIAM ROGER (Member 2404) was born at Banchory, Aberdeenshire, in 1872 and educated at Glasgow Academy and Albany Academy. He served his apprenticeship with Messrs. Mackie and Thomson's Shipyard, Glasgow. He attended Glasgow University in 1894 and obtained his Diploma for Naval Architecture. Returning to Messrs. Mackie and Thomson's Shipyard he was appointed assistant shipyard manager supervising the construction of many steam trawlers, sailing ships and steamers. With the late W. Smeddon he formed the Irvine Shipbuilding Co., Ltd., and as a director and manager built twenty-seven vessels in the Irvine Shipyard. In 1902 he sold his interest in the Irvine Yard and started consulting work in Glasgow and later in London, being appointed assistant manager to Fletcher's Dry Docks, London, in 1906. He was later manager and director on the Board until the business was sold in 1925. He was with this company nineteen years. During the 1914-18 war most of the work at Fletcher's was for the Admiralty and included the repair in dry dock and afloat of many destroyers, minelayers and minesweepers. Whilst with this company he also had experience in lifting sunken vessels in the Thames. After the war he helped to found, with the directors of Fletcher's, the Union Electric Welding Co., of which he was Managing Director. He was then appointed special surveyor to the Salvage Association for Lloyd's and was sent abroad as special officer to supervise claims and repairs on damaged vessels. In 1932 at the request of Mr. H. Devitt and the late Admiral Sir Charles Madden, he supervised and, with unemployed dockyard workers from Chatham and Rochester, carried out the conversion of the German sailing ship *Peking* into the training ship *Arethusa*. For this voluntary work he was presented with a handsome honorarium. Since 1937 Mr. Roger had retired and settled down at Gourock but still did odd surveying work on yachts on the Clyde. He was elected a Member in 1920 and was a member of the Institution of Naval Architects, The Society of Consulting Marine Engineers and Ship Surveyors. He died on the 15th December 1949, at the age of seventy-seven and leaves a widow.

RAYMOND FREELAND SCARFFE (Associate 11093) was born in 1916 and was educated at Tiffins School, Kingston-on-Thames, and University College, Southampton. He served his time as a premium apprentice with the British Power Boat Company of Hythe, Southampton, and on receiving his articles entered the design office, where he became the designer and firm's representative on official Admiralty, R.A.F. and War Office trials of high-speed craft. With the liquidation of the company consequent on the conclusion of the war he took a temporary post as designer with Messrs. Aldous Successors of Brightlingsea. He set up in private practice in April 1947 as consultant naval architect and surveyor. He was elected an Associate in 1946, and was also an Associate Member of the Institution of Naval Architects, Member of the Institute of Navigation, and Associated Member of the Institute of Welding, and a Member of the American Institute of Welding. He was also an accredited Surveyor for Navigators and the Marine Departments of several other Insurance Companies. He died on 12th February 1950.

GEORGE ALEXANDER SCULLARD (Member 1553) was born in 1861. He served his apprenticeship with Messrs. Thomas Daynes, Calpe Foundry, Gibraltar. He joined the Bland Line (H. M. Bland and Co., Ltd., Gibraltar) in 1884 and was appointed Chief Engineer of the S.S.*Gibel Tarik,* when this newly-built vessel went out to Gibraltar to start her Moroccan service. In 1890 he was appointed Marine Superintendent of the company, a post which involved the supervision and maintenance of the company's vessels, tugs, tenders and lighters, and also management of the company's shipyard ashore, where contructional and repair work was undertaken. Mr. Scullard's long and distinguished active career with Blands came to an end in 1928 when he retired but to mark the esteem in which he was held he was given the title of Technical Adviser to the company, which he held until his death on the 31st March 1950, at the age of eighty-nine years. He was elected a Member in 1901.

THOMAS JAMES SMITH (Member 7884) was born in 1885. He served his apprenticeship with Maudsleys Sons, and Field, Westminster. His sea-going service was with the P. and O. S.N. Co., and in 1917 he became Admiralty Inspector of Mines and Torpedoes. After 1920 he served with the Soudan Cotton and Development Co. and the Thames Valley Rubber Co. In 1924 he took up an appointment as District Surveyor with Insurance Engineers Ltd., with whom he was connected at the time of his death. He was elected a Member in 1935. He died suddenly at his home in Plymouth on the 28th June 1949, and leaves a widow and married daughter.

LEONARD SUMNER (Companion 2193) died on the 14th May 1942. He was elected a Companion in 1931.

EDWIN C. TALBOT (Member 4197) was born in 1889. He served his apprenticeship with the Dynevor Engineering Co., Ltd., Neath, and received technical training at Swansea Technical College. He served at sea from 1909 to 1919, being chiefly engaged on minesweeping during the war. In 1921 he was appointed Assistant Works Manager with the Port Talbot Dry Dock Co., Ltd., Port Talbot. He joined the staff of the Prince of Wales Dry Dock Co., Ltd., Swansea, in 1923 and was appointed Works Manager in 1926 and Manager in 1937. During the 1939-45 war the Company's workshops were destroyed by enemy action and for his work in connexion with their rebuilding, Mr. Talbot was awarded the M.B.E. He resigned from the Prince of Wales Dry Dock Co., Ltd. in 1947 and set up as a consulting engineer and marine surveyor at Swansea. He was elected a Member of the Institute in 1921. He was also Vice-Chairman of the Bristol Channel Dry Docks and Ship Repairers Association. He died on the 20th October 1949 and leaves a widow.

THOMAS AUGUSTUS WELLUM (Companion 8805) was born in 1888 and after a period with United Water Softeners he became sales and commercial director of Super Centrifugal Engineers, Ltd. He left this organization to become sales manager of British Separators, Ltd., and later transferred to Messrs. Henry Hughes and Son, Ltd., as sales manager. He was elected a Companion in 1939 and was an Associate Member of the Institution of Chemical Engineers. He died after a major operation on the 22nd October 1949, at St. Charles' Hospital, London, and leaves a widow. He was sixty-one.

GEORGE EDWARD WINDELER (Member 5408) died on 24th February. Mr. Windeler, who was seventy-five years of age, was the senior partner in the firm of Rhodes and Windeler, consulting engineers, and for over twenty-five years he was associated with the Mirrlees Watson Co., Ltd., of Glasgow, and Mirrlees Bickerton and Day, Ltd., of Hazel Grove, Stockport, in the development of the Mirrlees Diesel oil engine. He was born in Cork, Ireland, in 1874 and went with his father, who was in military service, to India, and the early part of his general education was obtained in European schools in India. In 1883 he returned to England and continued his education at the Workington Secondary School and at evening classes. He was first employed by the West Cumberland Iron and Steel Co., Ltd., of Workington, where for nearly three years he worked in the pattern shop, and on iron and steel production. For about eighteen months he was with Mr. Thomas Johnson, of Wigan, where he designed and repaired small high-speed steam engines, and for a further two and a half years or more he was with Walker Brothers, of Wigan, where he was employed in the drawing office and on testing work. In 1890 Mr. Windeler came to London, to join the firm of Ferranti and Co., Ltd., which then had its London office at Charterhouse Square, and he was employed there on the design and construction of high power steam engines and alternators for electric power generation. He was sent up to the firm's Hollingwood Works in Lancashire, where he continued to do designing work, and later on gained experience in inspection work and outside erection. Returning to the drawing office, he was made chief designer on the engine side and finally was appointed to be in charge of the works plant and the design and preparation of special tools for the engine, switchgear and meter departments of the Ferranti firm. About 1902 the late Mr. Charles Day, who had left Ferranti's to become general manager of the Mirrlees Watson Co., Ltd., of Glasgow, invited Mr. Windeler to join him in Glasgow and he appointed him manager of the Diesel engine department. Mr. Day had then gained the sanction of his board to build two 40 b.h.p. Diesel engines for works' power house use, and the 20 b.h.p. prototype engine was also put into service. On these single-cylinder engines Mr. Windeler carried out his pioneer experiments on cylinder lubrication and eventually evolved the idea of the floating gudgeon-pin. The air compressor was re-designed as a two- and later as a three-stage machine, and other improvements were introduced. In the Scotland Street works at Glasgow Mr. Windeler designed and built the four-cylinder 160 b.h.p. Diesel engines directly coupled to dynamos running at 400 r.p.m., two sets of which were first installed in H.M.S. *Dreadnought*, and similar sets later in H.M.S. *Indomitable*, H.M.S. *Shannon*, and H.M.S. *Minotaur*. For naval pinnace propulsion a four-cylinder engine of lighter construction, developing 120 b.h.p. at 400 r.p.m., was also produced. In the summer of 1908 the Diesel engine side of Mirrlees Watson was transferred to the new works of Mirrlees Bickerton and Day, Ltd., at Hazel Grove, near Stockport. Mr. Windeler was appointed works manager at Hazel Grove, and slow-speed, high-speed, and Admiralty engines were designed and built. In 1916 Mr. Windeler was made chief engineer to the company. As the 1914-18 war progressed it was decided to build Ricardo high-speed engines for tanks at Hazel Grove and the excellent way in which Mr. Windeler organized and carried out that work earned for him the warm thanks of Sir Harry Ricardo. Mr. Windeler also built the two-stroke Mirrlees-Nobel engine and today some examples of that engine are still giving good service. In 1928 Mr. Windeler severed his connexion with Mirrlees Bickerton and Day, Ltd., and founded with Mr. W. F. Rhodes, the firm of Rhodes and Windeler. From the beginning the firm specialized in a wide range of mechanical equipment and plant, which embraced not only oil engines, but gas engines, and high-speed steam engines. Mr. Windeler interested himself particularly in the analysis of vibration problems and the tracking down of trouble caused by vibration in mechanical plant, and also the investigation of industrial accidents. He also had a wide knowledge of smelting and the heat treatment of metals. For many years Mr. Windeler had acted as consultant for the Ashanti Goldfields Corporation, Ltd. Mr. Rhodes retired from the business about twelve years ago and last year Mr. Windeler took into partnership Mr. John Liller Faris, A.M.I.Mech.E., and Mr. William Henry Cowlin, A.M.I.Mech.E. He was elected a Member in 1925. He was also a Member of the Institution of Mechanical Engineers and of the Diesel Engine Users' Association.