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The Engining of Cargo Vessels of High Power

A Series of Papers Read and Discussed

On Tuesday and Wednesday, November 11-12th, 1947, at 2.30 p.m. at 85, Minories, E.C.3.

Chairman: THE PRESIDENT

(I) The Geared Steam Turbine

By T. W. F. BROWN, D.Sc. (Member).

Synopsis.

The general introduction gives the reason for stating that turbine machinery is particularly suitable for the propulsion of ships even up to powers of 70/80,000 s.h.p. per shaft. An explanation is furnished as to the principles controlling the choice of pressure and temperature at turbine in relation to power, and three schemes are referred to:—

(1) A 7,500 s.h.p. installation.

(2) A 13,000 s.h.p. installation with direct expansion of steam from inlet to condenser.

(3) A 13,000 s.h.p. alternative installation designed to have two stages of steam reheat between turbine inlet and condenser.

The next three parts of the paper are devoted to an examination of each of these installations in detail. In conclusion, the fuel rates, heat balances and weights for all the schemes are summarized.

General.

This paper gives particulars of geared turbine machinery suitable for the propulsion of cargo vessels as specified for this symposium by the Superintendent Engineers' Committee of the Institute. In the selection of the turbine arrangements given from the many which are possible, attention has been paid to achieving a high standard of efficiency and reliability with low weight and simplicity. The designs also lead directly to the use of still higher temperatures and pressures at inlet, with resulting further improvements in consumption.

Three schemes are given—(1) a single-screw geared turbine unit developing 7,500 s.h.p. in a single-cylinder turbine of simple construction, while the remaining schemes (2) and (3) are for units of 13,000 s.h.p., for both of which a single-screw installation has been chosen to demonstrate certain special features. Scheme (2) utilizes high-temperature steam and works with direct expansion of steam from inlet to exhaust, while Scheme (3) uses two-stage steam reheat with its accompaniment of high inlet pressure and moderate temperature.

TABLE 1.

	S.h.p. service	Description	Inlet conditions at turbine		Final conditions	
			Pressure lb./sq.in. gauge	Total temp. deg. F.	Vac. "Hg.	Wetness %
Scheme (1)	7,500	Single-cylinder geared turbine	420	790	28·5	7·5
Scheme (2)	13,000	Two-cylinder geared turbine	560	890	28·5	7·5
Scheme (3)	13,000	Three-cylinder geared turbine with two steam reheat points	1,400	790	28·5	1·75

Although the title of the symposium refers to cargo vessels of high power, turbine machinery has no limitation of size near the powers indicated. If required, geared turbine machinery to develop 70/80,000 s.h.p. per shaft can be constructed.

In modern designs of steam turbines suitable for high temperature and pressure conditions, high turbine revolutions are necessary to reduce size and consequent liability to distortion. To allow this while maintaining low propeller revolutions, double-reduction gearing is required. All the designs have been prepared to work at 120 r.p.m. of the propeller shaft. The temperatures and pressures at inlet to the turbines for the three schemes are given in table 1:—

Before detailing the separate schemes it may be permissible to point out some of the advantages which turbine machinery possesses over alternative means of propulsion, e.g.:—

(a) Turbines can be designed to have practically constant consumption between full and half power.

(b) With additional nozzles or by by-passing some stages, turbine machinery can run indefinitely at overload without detriment. Indeed if a special requirement of low consumption at part load is made, up to 250 per cent. increase in load above such power can be obtained.

(c) The lubricating oil consumption of turbine machinery is negligible over long periods.

(d) The turbine is the best instrument for the utilization of high-temperature and high-pressure steam, as there are no rapid fluctuating temperature gradients in the machinery nor are there any requirements for internal lubrication.

(e) The best propeller speed can be employed with geared turbine machinery (particularly where double-reduction gearing is employed), the reduction in speed between turbines and propeller shaft being simply obtained.

(f) Turbine machinery reliability is reflected in the very low costs of maintenance and overhaul.

(g) Few safety devices are required, the main one being an overspeed governor fitted on the L.P. turbine shaft, which operates by releasing the oil pressure holding open a combined stop and emergency valve in the steam supply line against a spring, thus allowing the valve to close. This same device shuts off steam should an interruption occur in the supply of lubricating oil to the bearings.

The selection of suitable materials for turbine parts does not present any special difficulty except possibly in the case of material for blading. The most generally useful material for H.P. turbine blading is stainless iron, which has adequate creep strength at the proposed inlet temperatures, and has the additional advantages of ease of manufacture and relatively low cost. It is known that under wet steam conditions stainless iron may be susceptible to chloride attack, and therefore if there is any likelihood of chloride contamination in the steam it may well be replaced in the L.P. turbine below the saturation point by either Monel metal or an austenitic steel of the 18/8 type. For the last few rows of L.P. blading, in which the highest stresses are obtained, K-Monel might be used with advantage, since it combines high tensile strength with excellent corrosion resistance. For temperatures between 800-900 deg. F. at inlet, molybdenum steel castings and forgings should be used for all parts subjected to the maximum steam temperature. Nozzle segments are best made of stainless iron and diaphragms should also be built up from such segments.

Attention[2] has recently been drawn to the occurrence of cylinder wastage in the wet steam region of some L.P. turbines. This phenomenon should be distinguished from the entirely different problem of blade erosion, which can occur at high blade speeds (over about 750ft./sec.) due to the impact of water droplets on the leading edges of the blades in association with wetness values up to 12 per cent. Much higher blade speeds are, of course, possible although not usual in marine turbines, and in such cases erosion shields or other similar means can be employed to overcome the effect of water in the steam. Cylinder wastage appears to be a corrosion effect and appears in the machinery of only two companies operating turbine-engined vessels.

The designs are based on the use of usual materials for the gearing. The wheel rims utilize a 0·3 per cent. carbon steel, the pinions being made from oil-quenched 3½ per cent nickel steel. The main and primary wheel centres can be fabricated, providing axial as well as radial stiffness is achieved. It is felt, however, that on balance cast iron has advantages for the main wheel centre, being a good damper of sound vibrations and has reasonable strength providing high resistance to shock is not required. In these designs the ratio of length of face to pinion diameter is small to keep the combined deflections due to torsion and bending small. In the future it is felt that harder materials and even lower ratios of face width to pinion diameter will be required in association with much higher loadings, thus leading to great reductions in the size of reduction gearing employed.

In all of the following schemes two water-tube boilers are fitted. Water-tube boilers are general in association with turbine machinery in the Merchant Navy, and two boilers have been fitted to allow one to be opened for survey at a time, leaving the other available for port use. It is not considered that many companies would install turbine machinery in association with a single boiler, an arrangement as used by one company in association with gas re-heat[2]. No auxiliary boiler is shown in the proposed arrangements, as it should not be necessary with the auxiliaries shown. If it should be required for other reasons, it can be included without making any change in the layouts by placing the boiler on a shelf adjacent to the uptakes of the main boilers.

Scheme (1).

Desirable pressures and temperatures associated with particular outputs per shaft are given in paper [5], diagram 1, referred to in the bibliography. For this power of 7,500 per shaft, the figures chosen for inlet pressure and temperature have already been tabulated.

It is considered that a single-casing impulse turbine offers the best compromise on the grounds of simplicity, reliability and cost for an installation of this power, especially if an astern turbine in the same casing can be eliminated. In addition, radial and axial blade clearances are comparatively large. The cylinder has only two glands, one pressure and one vacuum, no receiver pipes, and it can be designed to have its first critical speed of the rotor at least 30 per cent. above the highest running speed. This single-cylinder turbine should be acceptable as, without seeking to disparage the direct-coupled Diesel engine applied to single-screw drive, it has no parts of comparable vulnerability to the crankshaft of the latter engine. In addition, if spares are required to be carried aboard they are comparatively small in size and easily handled. The combined weight of turbine and gear case is considerably less than the more usual compound (H.P./L.P.) turbine drive.

The proposed arrangement is shown in Fig. 1, in which the turbine revolutions are 4,200 per minute with 120 revolutions on the propeller shafting. If all the power were transmitted through a single primary wheel and secondary pinion, the gears would be larger than the arrangement shown in Fig. 1 because the torque load would be transmitted on one contact line, while the bending in the pinion with one driving line would also have to be kept within limits, either by increase in diameter or by supporting it in a central bearing. As well as being smaller than a conventional double-reduction gear, the box has the merit of symmetry and can be used in twin-screw sets without handing. The

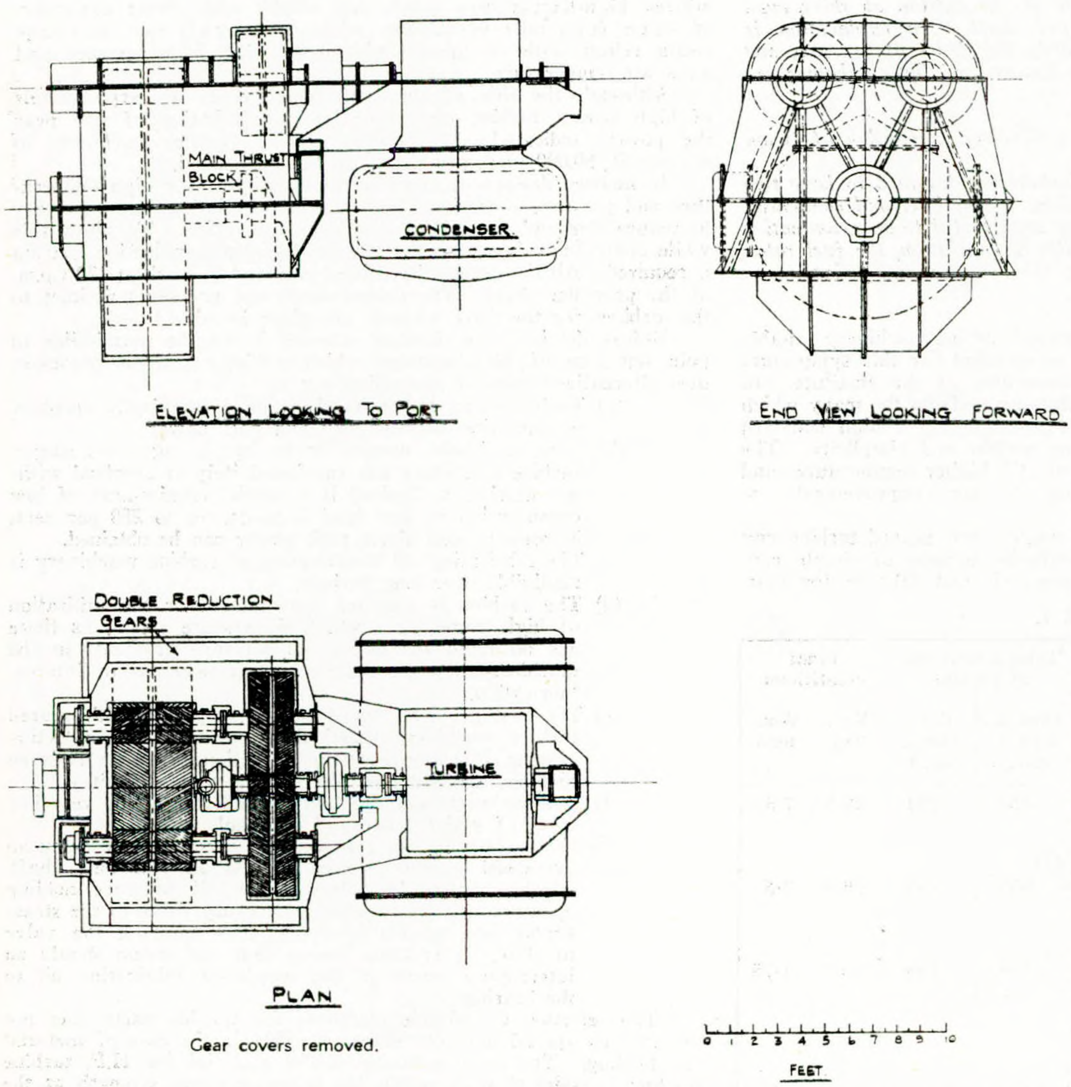


Fig. 1.—*Arrangement of single-cylinder turbine machinery of 7,500 s.h.p. and "locked-train" gearing.*

turbine shaft centre-line is at the correct height to suit an underslung condenser, and the main gear can if necessary be set to provide a low setting of the main shaft line. For these reasons a "locked-train" drive double-reduction gear has been chosen. The pinion drives two primary wheels coupled at their aft ends to quill shafts which pass through the secondary pinions and drive them through fine-tooth couplings. From these two secondary pinions the power is transmitted to the main wheel.

The special feature of the transmission is the incorporation of a hydraulic reversing and manœuvring system which is coupled to the turbine drive shaft by a fine-tooth coupling and comprises a quill shaft with impellers solidly bolted to it. Corresponding driven elements in fluid coupling casings are rigidly connected to the extremities of the pinion. If the fluid coupling nearest the turbine is filled, the coupling drives the pinion in the ahead direction; filling the astern coupling then drives the pinion astern. The ahead fluid coupling, which is of a well-known type but with special features, is designed to give 98 per cent. efficiency. The fluid coupling can have a mechanical synchronizing and solid drive incorporated in it for use when "full-away" if required, but special interlocks would then be required to ensure that the gear was out of action during manœuvring. On balance, it appears better to omit the mechanical gear as the hydraulic device alone is much simpler and more foolproof. Although there is a loss in the fluid coupling, it is largely offset by the elimination of the loss occasioned by driving astern turbines when the machinery is running ahead. In addition the fluid coupling provides a flexible element in the drive with a possibility of having considerable vibration damping between turbine and gear, and has some value as a torque limiting device. Every merit which has been claimed for electric drive is present in a cheaper, simpler and more robust form with markedly improved efficiency. The astern fluid coupling between the aft end of the quill shaft and pinion is available in several different forms which are patented, and in general terms is designed as a combination of impulse or reaction turbines and impellers, with reversal of flow between impeller and turbine elements being achieved by stationary elements all designed in the light of modern fluid dynamics. The maximum astern power with an astern fluid coupling of the size shown is about 70 per cent. of the ahead power with full steam flow.

In manœuvring, if both fluid couplings are filled the turbine will cause churning of the oil in both couplings, the one which is filled acting against the coupling which has been driving as a brake. This work on the oil results in raising its temperature which will be reduced in the oil cooler. The turbine will then be stalled and the main shaft stationary. As the turbine is stalled with the steam supply to the turbine open, at least twice the normal full load torque is available for manœuvring. This also removes the necessity of fitting a sensitive speed governor, the overspeed govern-

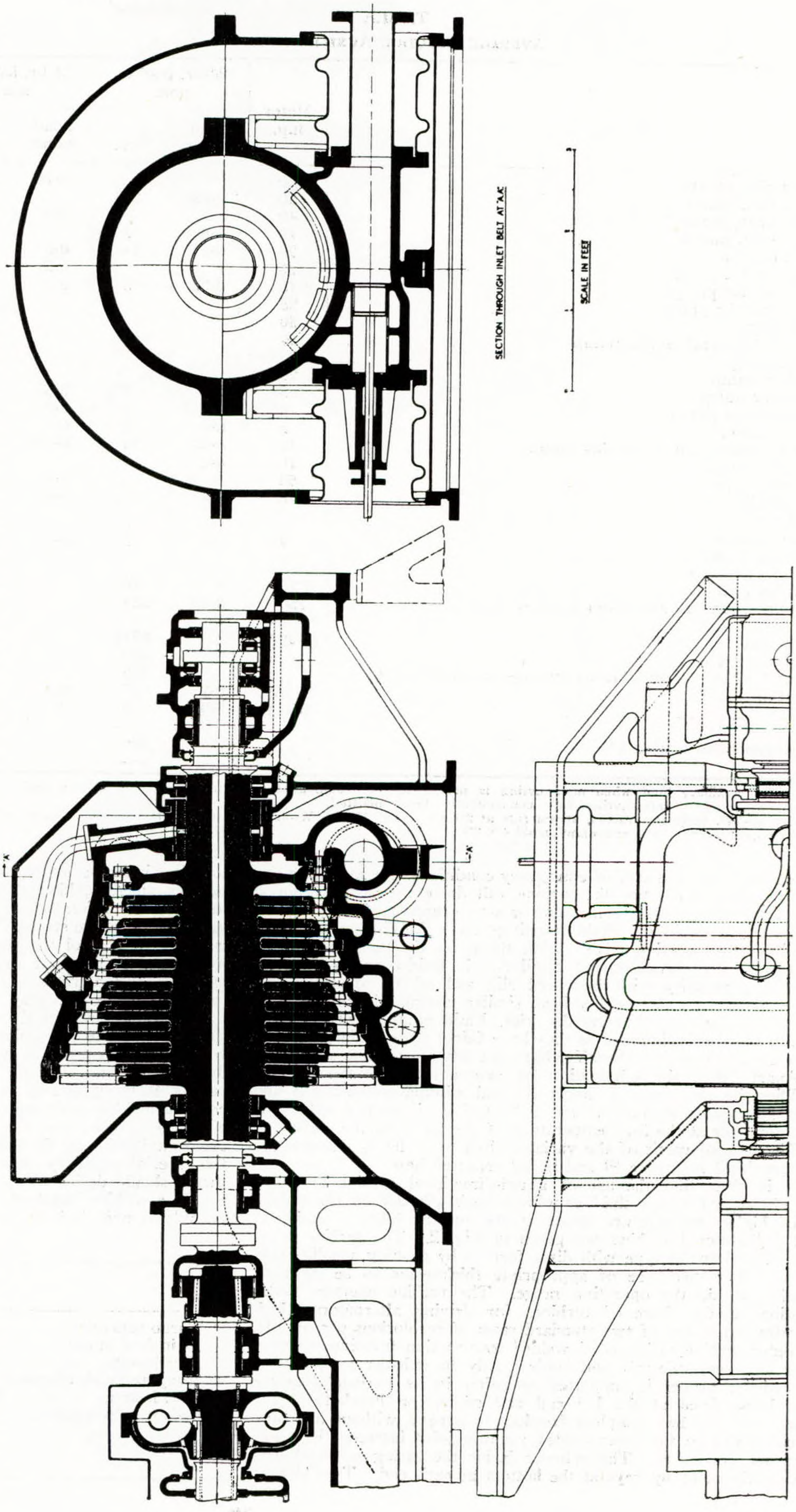


FIG. 2.—*Sectional views of single-cylinder turbine.*

TABLE 2.

AVERAGE ELECTRIC AUXILIARY LOAD.

	Motor h.p.	24-hr. load in port		24-hr. load at sea	
		Load factor	H.p.	Load factor	H.p.
Main circ. pump	120			0·75	90
Aux. circ. pump	30	0·75	22½		
Main extr. pump	40			0·5	20
Aux. extr. pump	10	0·5	5		
F.D. blower	70	0·2	14	0·5	35
L.O. service pump	25			0·5	12½
F.O. service pump	15	0·33	5	0·5	7½
F.O. transfer pump	25				
Fire pump	50				
Fire and general service pump	50		15		20
Bilge pump	25				
Ballast pump	25				
Sanitary pump	5	0·6	3	0·6	3
Fresh water pump	5	0·6	3	0·6	3
L.O. purifier	5	0·4	2	0·4	2
Refrigerator (incl. circulating pump)	15	0·67	10	0·33	5
Machine tools	10	0·2	2		
Steering gear	60			0·3	18
Ventilating fans	75	0·4	30	0·4	30
Hot water					
Water service	7½	0·4	3	0·4	3
Priming pump					
Gland exhauster					
Turning gear	10	1	10		
Windlass, capstan and cargo winches	750	0·33	250		
Total h.p.	1,432½		374½		249
Equivalent kW.			280		186
Input kW. to motors allowing average motor effy. 0·85 ...			330		219
Lighting	35 kW.	0·7	25	0·7	25
Galley	30 kW.	0·33	10	0·33	10
Total generator load in kW.			365		254

Note. The auxiliary load when manœuvring is practically the same as for continuous running at sea and the column showing load when manœuvring has consequently been omitted.
With 400-kW. turbo-generators, steam rate at 254-kW. load=24.4lb./kW.-hr. when exhausting at 20lb./sq. in. abs.
Turbo-generator steam consumption=24.4 × 254
=6,200lb./hr.

belts are shown in the drawing although only one is utilized in this scheme. This has been done deliberately to show that the casing does not become too complicated if two are required. No lagging is required in this turbine as the outer casing is subject only to exhaust temperature and the inner casing is protected in the hot portion by a simple thin steel heat shield. In addition, construction of the outer casing up to the horizontal joint can, if desired, be made integral with the condenser shell.

The fixed nozzle plate and the nozzles in the diaphragms are milled out of solid blocks and mated together to form a series of high-efficiency passages for the conversion of the energy in the steam into velocity with minimum loss. The shapes can also be controlled within fine limits and, as no welding is employed, distortion is absent and predicted performance can be achieved. Apart from cracking in cast-in nozzle plates or diaphragms, it is felt that the call for efficiency justifies the cost involved in using built-up construction for such parts. As the rotor is designed to have no critical speeds until revolutions are at least 30 per cent. above maximum allowing for temperature effects, no springs are required behind the diaphragm packing. The steam inlet is confined to the bottom half of the casing, obviating disturbance of the steam supply pipe during examination of the turbine, and is directly attached

nor remaining to take care of emergency conditions. As soon as one or other coupling is emptied the turbine will drive the primary pinion through the other, thus giving ahead or astern running of the propeller from the stop position. Fluid couplings have the correct matching conditions for a marine drive. With steam on, the turbine increases its torque as it is slowed down. Similarly, the fluid coupling increases its driving capacity with increased slip and all the operations in turbine, coupling and propeller have similar characteristics, although dealing with fluids of different densities. Fluid couplings are not sensitive to slight misalignment as they have fairly large radial and axial clearances and therefore simplify alignment between turbine and gears.

Apart from the elimination of astern turbines, this scheme simplifies the steam-pipe arrangement and eliminates the ahead and astern steam manœuvring valves, substituting oil control valves of small diameter at the low temperature of the oil from the cooler.

The lack of merit of the variable-pitch propeller is discussed in bibliographical reference (5) and is not repeated here.

It is clear that the use of a uni-directional turbine arranged with ahead and astern fluid couplings leads directly to the future use of higher temperature steam at the turbine inlet. Details of the uni-directional turbine are given in Fig. 2. The turbine is of the modern impulse type with discs formed by gashing a solid rotor forging. The discs are of appropriate thicknesses to be clear of vibration within the operating range. The turbine operates under conditions unlike those of turbines for driving alternators which generally run at one of two standard rates of revolutions per minute. The outer turbine casing is of welded construction forming a strong bridge between pedestals and subject only to exhaust temperature. The cylinder proper is supported on palms in its horizontal centre plane, being fixed at the hot end and sliding on brackets at the exhaust end. It has complete freedom to expand without imposing any constraint on the steam-sealed vacuum joint between the outer casing and condenser. The cylinder inside the casing is maintained in axial alignment by keys at the bottom at each end. Two bleeder

to the inner cylinder, vacuum tightness being maintained by bellows pipes. Although partial admission is employed, the more important consideration of nozzle height and shape is correct even with high-pressure steam. A sleeve valve with large radial clearances is fitted to control the nozzle groups in operation. The valve is maintained steam tight with reference to nozzle groups shut off by pairs of piston rings. This produces a simple casting for the casing, which is also easy to machine.

The ahead end of the turbine drive to the quill shaft and from the quill shaft to the pinion is indicated. A spur gear is also shown which could gear on to a cross shaft running at reduced speed to drive auxiliaries, such as the lubricating oil and circulating water pumps, it being pointed out that the turbine is uni-directional and consequently no reversing valves are required. Similarly a dynamo could be driven from the forward end of the turbine rotor. The usual auxiliaries are included in the loads and arrangement drawing, this special auxiliary drive leading to further improvements if an owner should decide to use this feature.

In a turbine installation more than in any other form of marine propulsion unit, it is essential to tie in the auxiliaries to match the

TABLE 3.

	24-hr. load at sea lb./hr.
Turbo generator	6,200
Main feed pump	1,000
Air ejectors	300
Evaporator air ejector	130
F.O. heaters	350
Galley, ship heating...	400
Total	8,380 lb./hr.

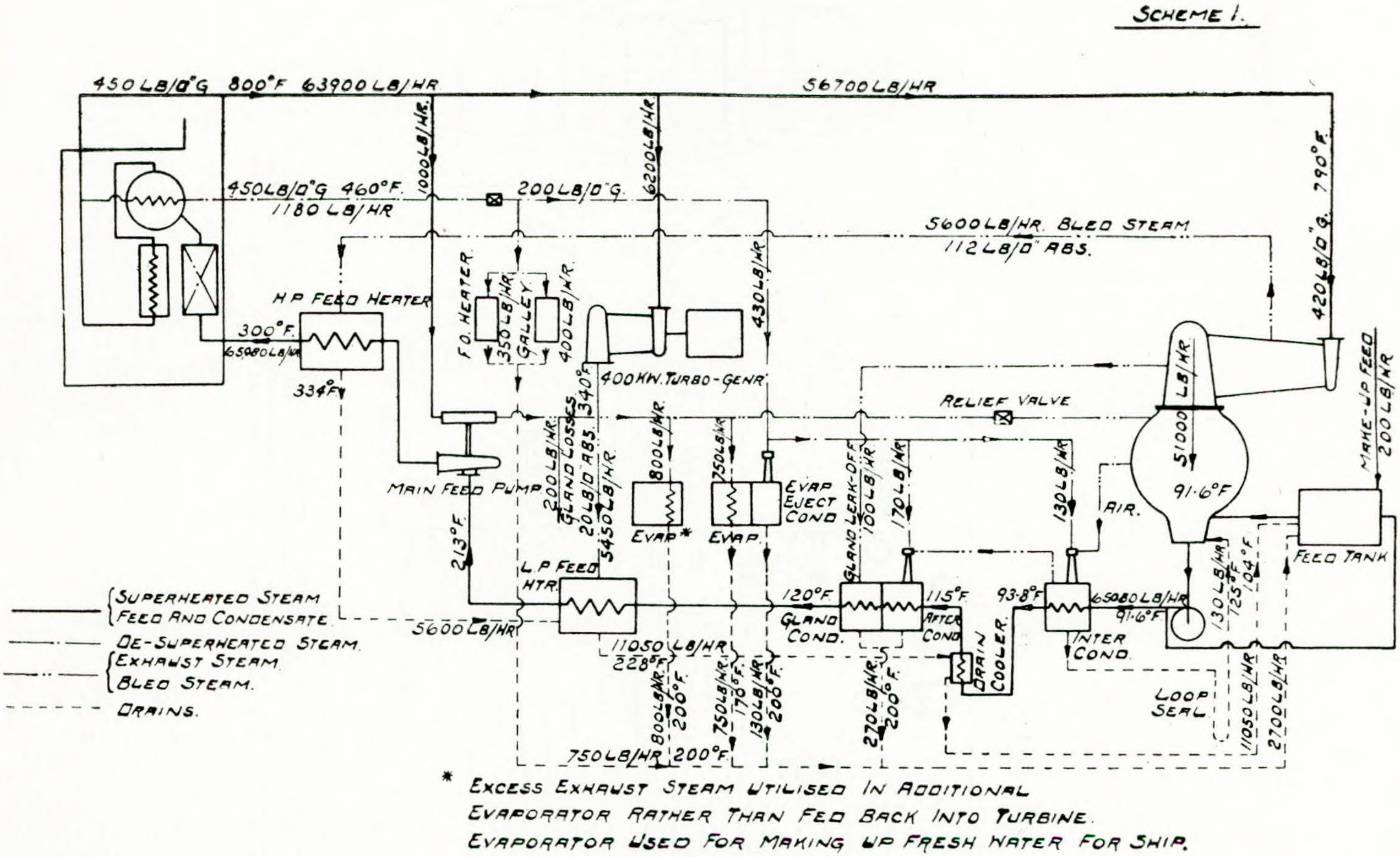


FIG. 3.—*Flow diagram for steam and feed lines, Scheme 1.*

main propulsion unit. Part steam and motor driven auxiliaries are provided as given in the preceding table, which shows the corresponding load or consumption.

It will be seen that two different outputs are required from the turbo-generators at sea and in port. This demonstrates that when the turbo-generators work non-condensing the sea load can be maintained, full output being obtained by using the condenser and auxiliary closed-feed system in port. As the turbo-generator exhaust is available, a second feed-heating supply point is not required from the main turbine at sea. Pass-out turbines for generator drive would make an even more efficient scheme, but in this paper the main turbine design has been given pride of place and the auxiliary layout has been made as simple as possible consistent with efficiency requirements, although possible improvements in auxiliaries are indicated. The complete flow diagram for steam and feed water, together with appropriate temperatures and pressures, is given in Fig. 3.

The main machinery, boilers and all auxiliaries are drawn in the proposed engine-room layout, including stand-by units for important auxiliaries—Fig. 4. The engine room length is 46ft. 6in.

The boilers shown in each of the schemes are of Foster Wheeler design, and this firm provided all the necessary data to complete the heat balance and fuel consumption figures, which are summarized in Table 10, Appendix I.

Scheme (2).

For the output of 13,000 s.h.p. a compound turbine and articulated double-reduction gear propulsion unit has been designed. The H.P. and L.P. turbines drive separate pinions at 4,000 and 2,860 r.p.m. respectively, which mesh with separate primary wheels driving the secondary pinions through quill shafts coupled to them and bolted at the aft end of the primary wheels. The power distribution from the turbines is as follows:—H.P. turbine 57 per cent., L.P. turbine 43 per cent. The secondary pinions mesh with the common main wheel driving the propeller at 120 r.p.m. This articulated construction of gearing provides a series of axial breaks in the systems and allows considerable torsional flexibility between primary and secondary gear trains. Any additional loadings due to any gearing inaccuracies present will be a minimum as the inertias are small. The main thrust block is set at the forward end of the main gear case, which provides a rigid attachment on the large area of the seating. The thrust block is accessible for examination under the arch carrying the primary gear boxes. Adequate accessibility for chocking the gearcase under the forward main bearing can be provided.

The H.P. turbine is shown in detail in Fig. 6. For the reasons set out more fully in Scheme (I), impulse construction is employed. The casing is a normal carbon-molybdenum steel casting and the rotor a carbon-molybdenum steel forging. Other features are similar to those already described. It will be seen that the pedestal at the hot end of the turbine is a separate casting. The weight of the cylinder is carried on the pedestal by palms at the plane of the horizontal centre line, and a vertical key at the bottom of the casing provides athwartship location. The forward pedestal should be fixed and the aft pedestal arranged to slide, as this will assist the design of steam piping by minimizing the movement to be taken up. As in Scheme (I), bottom admission of steam is employed.

The H.P. gland is in three portions—the inner pocket has a leak off to the H.P. turbine exhaust, the centre pocket to the gland pot and the outer pocket is connected to an ejector or gland condenser.

The L.P. turbine arrangement is illustrated in Fig. 7 and shows that the ahead portion is a single-flow reaction turbine having a solid rotor. The tapered portion is required to provide adequate blade heights at inlet, thus maintaining high efficiency, and to reduce thrust. A larger dummy would be required with a parallel rotor. Radial clearance segmental blading, side-locked in serrated grooves, is employed except for the last few rows of rotor blading where integral blading should be fitted. The dummy cylinder is integral with the casing casting. An important point is that the thrust block is adjacent to the ahead blading, protecting the blading clearances even under astern conditions when the sudden admission of astern steam may cause differential expansion in the astern portion of the rotor.

The astern turbine is designed to develop 50 per cent. of the ahead power with the normal steam flow. This power is fully adequate for all but special operating conditions in such vessels as a

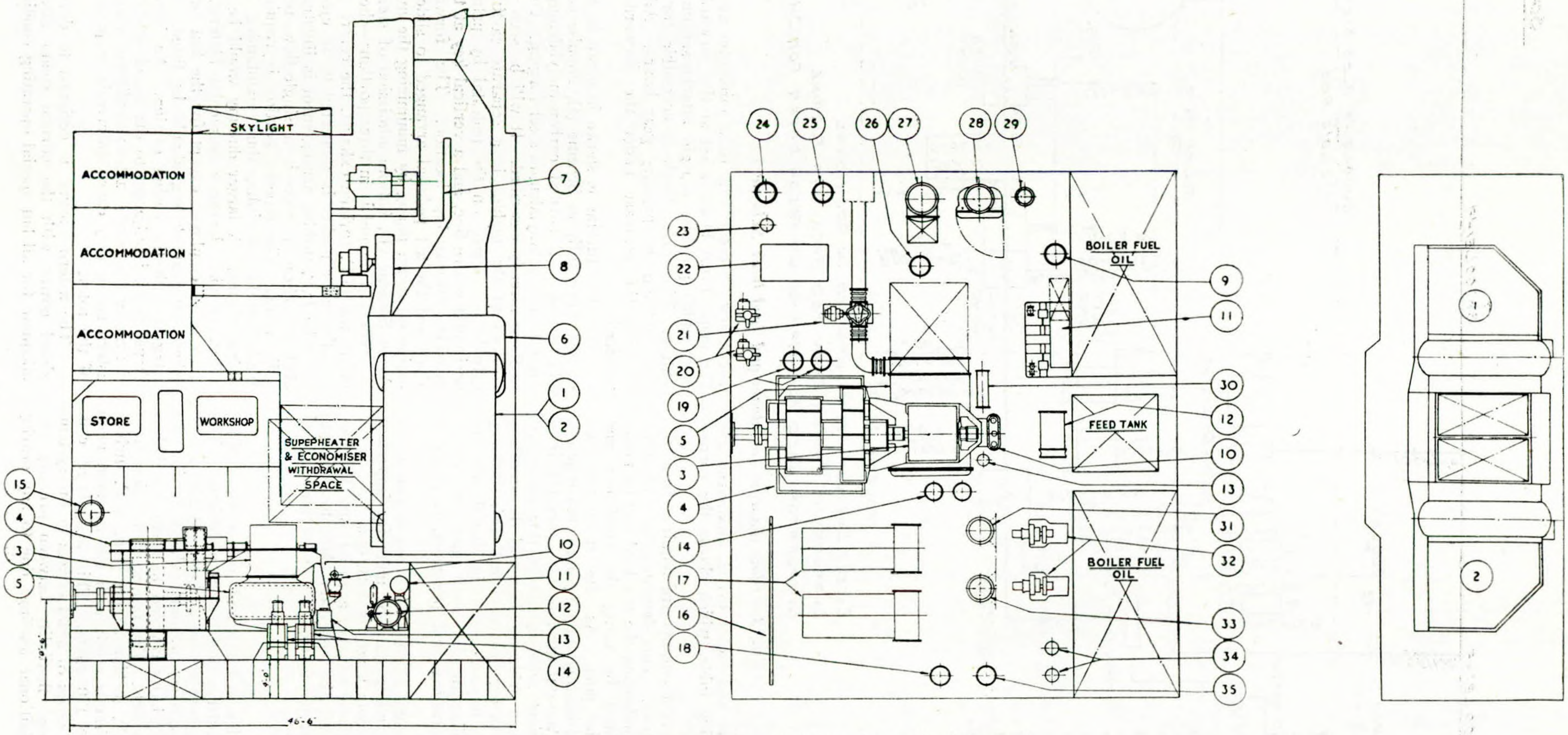
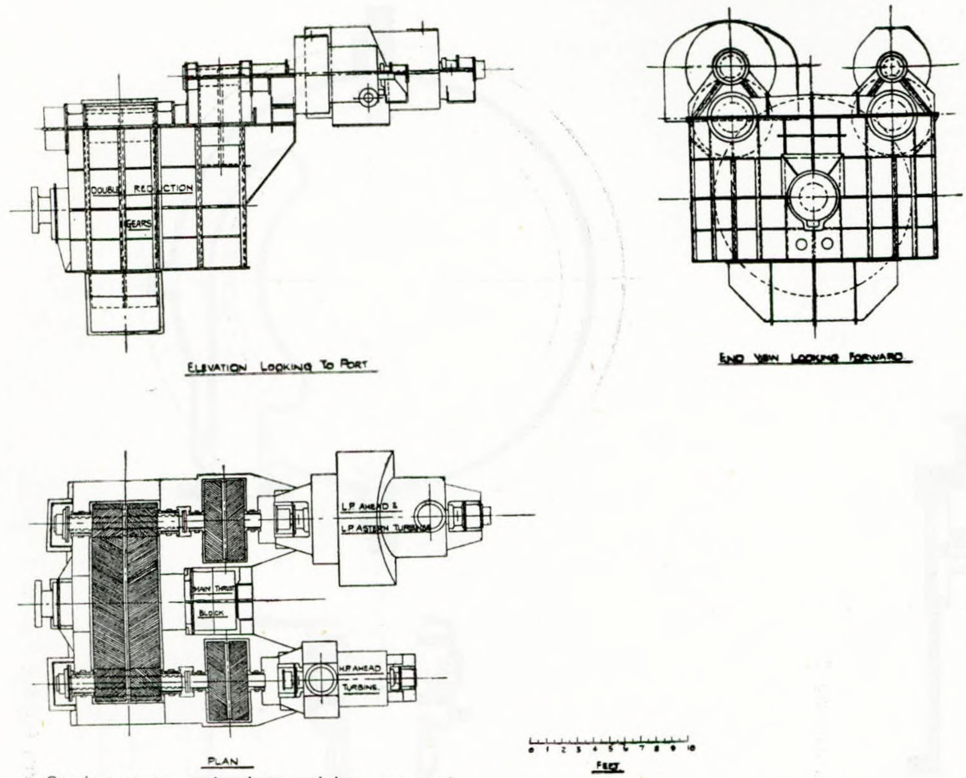


FIG. 4.—*Engine-room layout, 7,500-s.h.p. unit, Scheme 1.*

(1) Main boiler—port; (2) main boiler—starboard; (3) turbine; (4) reduction gearing; (5) main condenser; (6) air heater; (7) induced-draught fan; (8) forced-draught fan; (9) fuel-transfer pump; (10) air ejector; (11) oil-burning installation; (12) drain cooler; (13) gland steam collector; (14) extraction pumps; (15) oil cooler; (16) switchboard; (17) turbo-generators; (18) bilge pump; (19) lub. oil pumps; (20) oil purifiers; (21) main circ. pumps; (22) refrig. machine; (23) refrig. circ. pump; (24) fire and bilge pump; (25) sanitary pump; (26) fresh-water pump; (27) steam generator; (28) evaporator; (29) distiller condenser; (30) gland steam condenser; (31) L.P. feed heater; (32) main feed pumps; (33) L.P. feed heater; (34) aux. feed pumps; (35) ballast pump.



Gearing covers and primary pinions removed.

FIG. 5.—*Arrangement of compound turbine machinery of 13,000 s.h.p. with articulated D.R. gearing.*

cross-Channel steamer having long periods of running astern.

The astern elements consist of two two-row wheels running in an inner casing of simple construction with large radial and axial clearances to allow for distortion without transmitting undue loads to the outer casing which forms part of the ahead casing. Steam is led into the inner casing by way of a "dumb-bell" fitted with piston rings which allow freedom to the inner casing to expand freely. The astern casing is carried in the main casing by palms set just below the horizontal joint and held by a bottom pin which locates it while allowing freedom to expand radially. The exhaust from the astern turbine is deflected away from the ahead blading, and it is not therefore compressed in the ahead turbine during astern running with consequent heating up.

In the event of astern power greater than say 60 per cent. being required, it will not be found advisable from critical speed considerations to incorporate all the necessary stages in the L.P. turbine casing. It may also happen that the power astern on one pinion may give an excessive load. A separate H.P. astern turbine should be provided overhung from the H.P. turbine rotor exhausting to an L.P. astern turbine in the L.P. casing. Such a design is shown in Fig. 8, which shows how the turbine casing is accurately positioned without transmitting heat from it to the bearing pedestal, yet with great freedom to expand.

Scheme (2) is completed by Tables 4 and 5 giving the auxiliary loads and the flow diagram and engine layout plans, Figs. 9 and 10 respectively.

TABLE 5.

AVERAGE STEAM AUXILIARY LOAD.

	24-hr. load at sea lb./hr.
Turbo generator	9,300
Main feed pump	2,300
Air ejectors	450
F.O. heaters	600
Evaporator air ejector ...	150
Galley, ship heating ...	700
	13,500 lb./hr.

In this flow diagram it will be seen that there is a slight excess of L.P. heating steam which is used to make additional distilled water, but in any particular installation this steam might be used for other purposes.

For this 13,000 s.h.p. installation the overall length of the engine room is 55ft. 0in.

TABLE 4.

AVERAGE ELECTRIC AUXILIARY LOAD.

	Motor h.p.	24-hr. load in port		24-hr. load at sea	
		Load factor	H.p.	Load factor	H.p.
Main circ. pump	200			0·75	150
Aux. circ. pump	50	0·75	37½		
Main extr. pump	70			0·5	35
Aux. extr. pump	15	0·5	7½		
F.D. blower	120	0·2	24	0·5	60
L.O. service pump	45			0·5	22½
F.O. service pump	25	0·33	8	0·5	12½
F.O. transfer pump	50				
Fire pump	90				
Fire and general service pump	90		25		35
Bilge pump	50				
Ballast pump	50				
Sanitary pump	10	0·6	6	0·6	6
Fresh water pump	10	0·6	6	0·6	6
L.O. purifier	10	0·4	4	0·4	4
Refrigerator (incl. circulating pump)	25	0·6	15	0·3	7½
Machine tools	15	0·2	3		
Steering gear	100			0·3	30
Ventilating fans	130	0·4	52	0·4	52
Hot water					
Water service	10	0·4	4	0·4	4
Priming pump					
Gland exhauster					
Turning gear	15	1	15		
Windlass, capstan and cargo winches	1,300	0·33	430		
Total h.p.	2,480		637		424½
Equivalent kW.			475		316
Input kW. to motors allowing average motor effy. 0·85 ...			559		372
Lighting...	60	0·67	40	0·67	40
Galley	50	0·4	20	0·4	20
Total generator load in kW.			619		432

Note: Auxiliary load when manœuvring is approximately the same as for continuous running at sea.
With 650-kW. turbo-generators, steam rate at 432-kW.=21.5lb./kW.-hr. when exhausting at 20lb./sq. in. abs.
Turbo generator steam consumption=21.5 × 432
=9,300lb./hr.

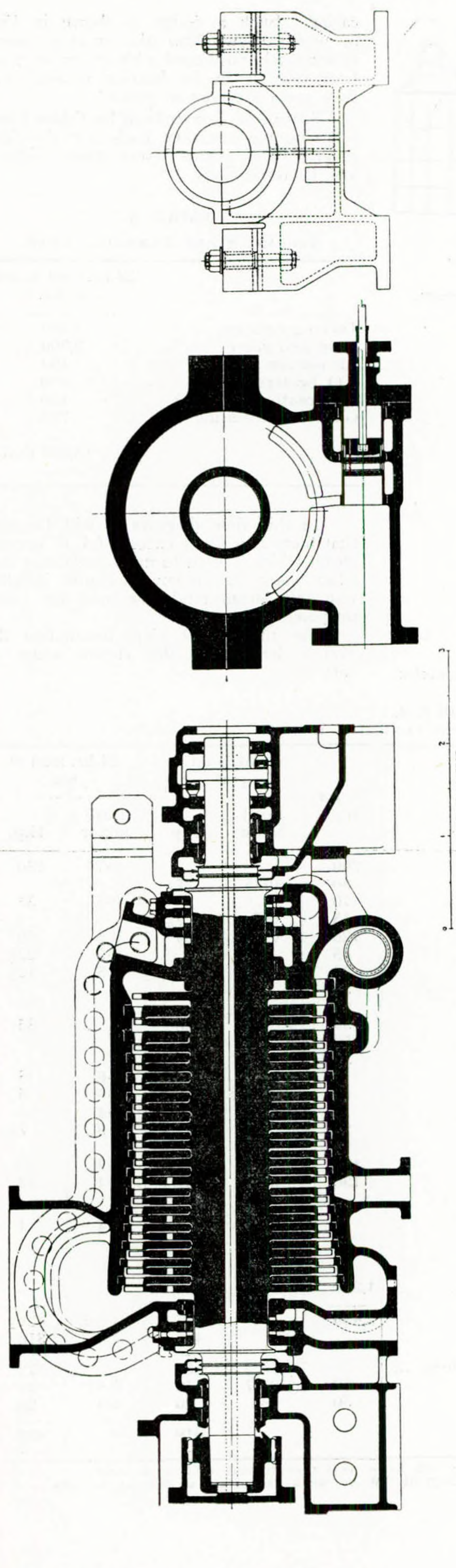


FIG. 6.—*Sectional elevation of H.P. Turbine, Scheme 2.*

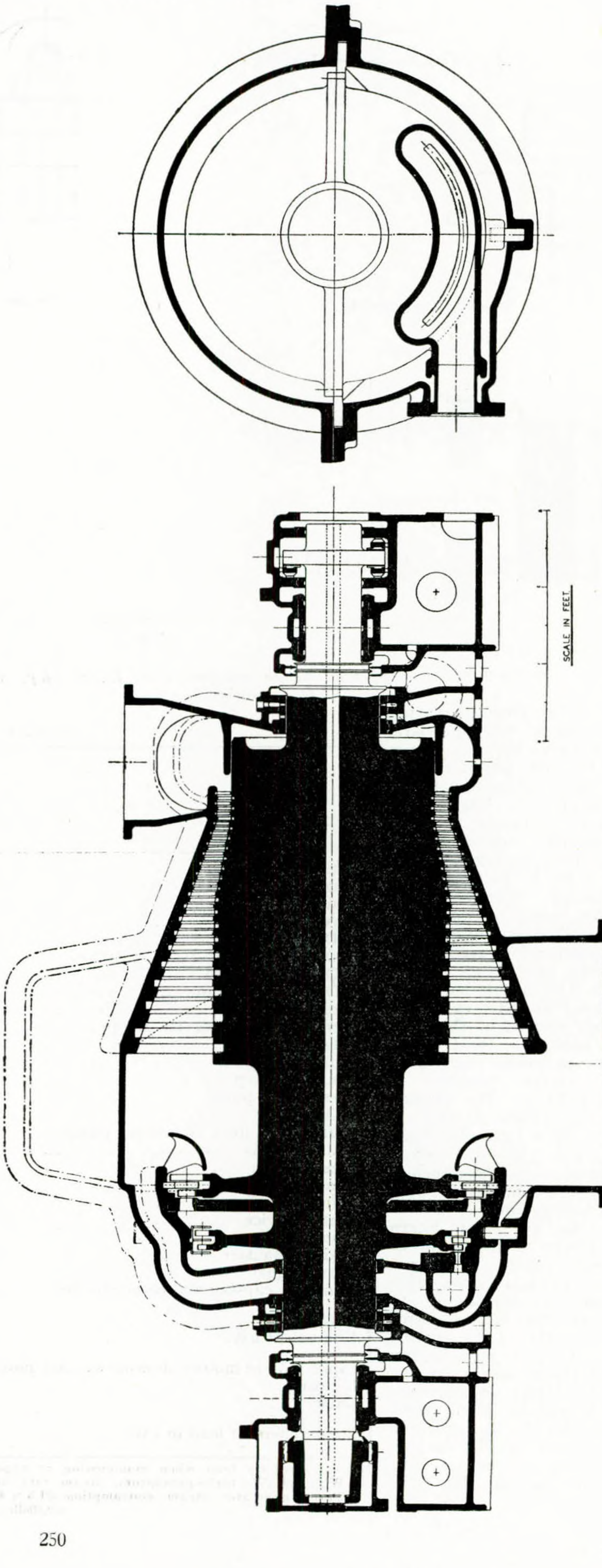


FIG. 7.—*Sectional elevation of L.P.* **ahead and astern turbine,** *Scheme 2.*

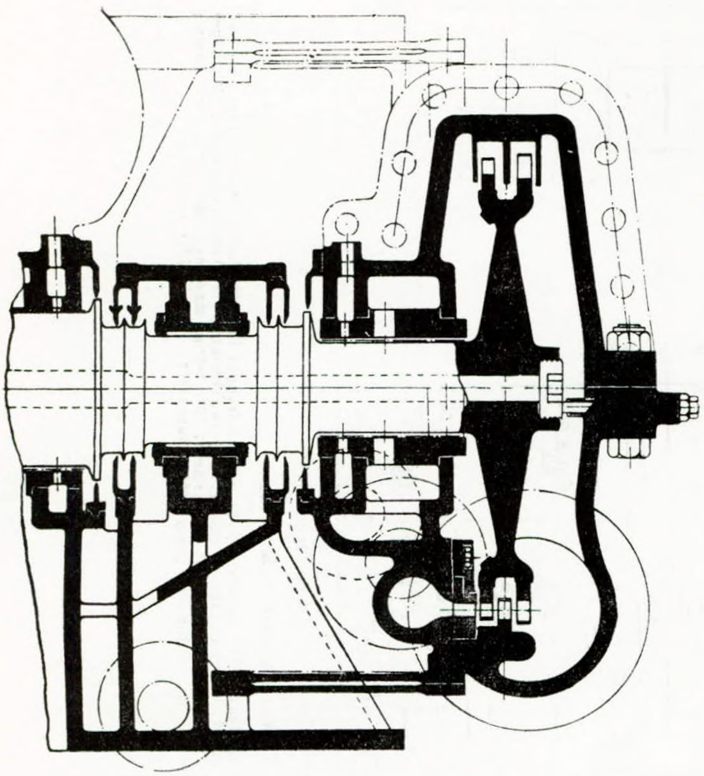


FIG. 8.—*Sectional elevation of H.P. astern turbine.*

Scheme (3).

The design in this case is an alternative to the direct expansion of steam from inlet to condenser described in Scheme (2) by providing two steam reheat points in the expansion. For simplicity the steam is reheated where it leaves one turbine and before entry to the next. The use of this cycle also illustrates the provision of a three-casing turbine design in association with an articulated design of double-reduction gear.

The reason for considering the reheat cycle is to obtain an increase in efficiency as compared with the direct cycle. Two alternative systems are in use: the gas reheat system, in which the steam after expansion in the H.P. turbine has its temperature raised to the original value by returning it to a reheater incorporated in the boiler, and the steam reheat system adopted in this scheme in which the heating is carried out by boiler steam in reheating chambers similar to feed heaters. In the latter system the reheat temperature is limited to a value rather below the saturation temperature corresponding to boiler pressure.

Both systems have been successfully applied within the last few years, the gas reheat system in the S.S. "Examiner"[3] and the C.P.R. "Beaver" Class vessels[2], the steam reheat system in the ore carrier S.S. "Venore"[4].

Although these schemes are discussed in these papers, a summary may be useful.

Gas reheat results in a theoretically greater efficiency than steam reheat, since it makes possible a higher reheat temperature, but the complication caused by the presence of the reheater in the boiler and by the large steam pipes to and from the reheater makes it suitable only for the largest powers. With steam reheat the reheaters may be placed close to the turbines, and this gives a simplified arrangement since the large diameter interconnecting pipes are short, and the heating steam connections are small since this steam is at boiler pressure. The net result is that pressure losses are small with live steam reheat. A high boiler pressure is necessary with steam reheat in order to obtain an adequate reheat temperature, but on the other hand the maximum temperature can be relatively low. In this connection it may be pointed out that high pressure unaccompanied by high temperature does not cause any serious

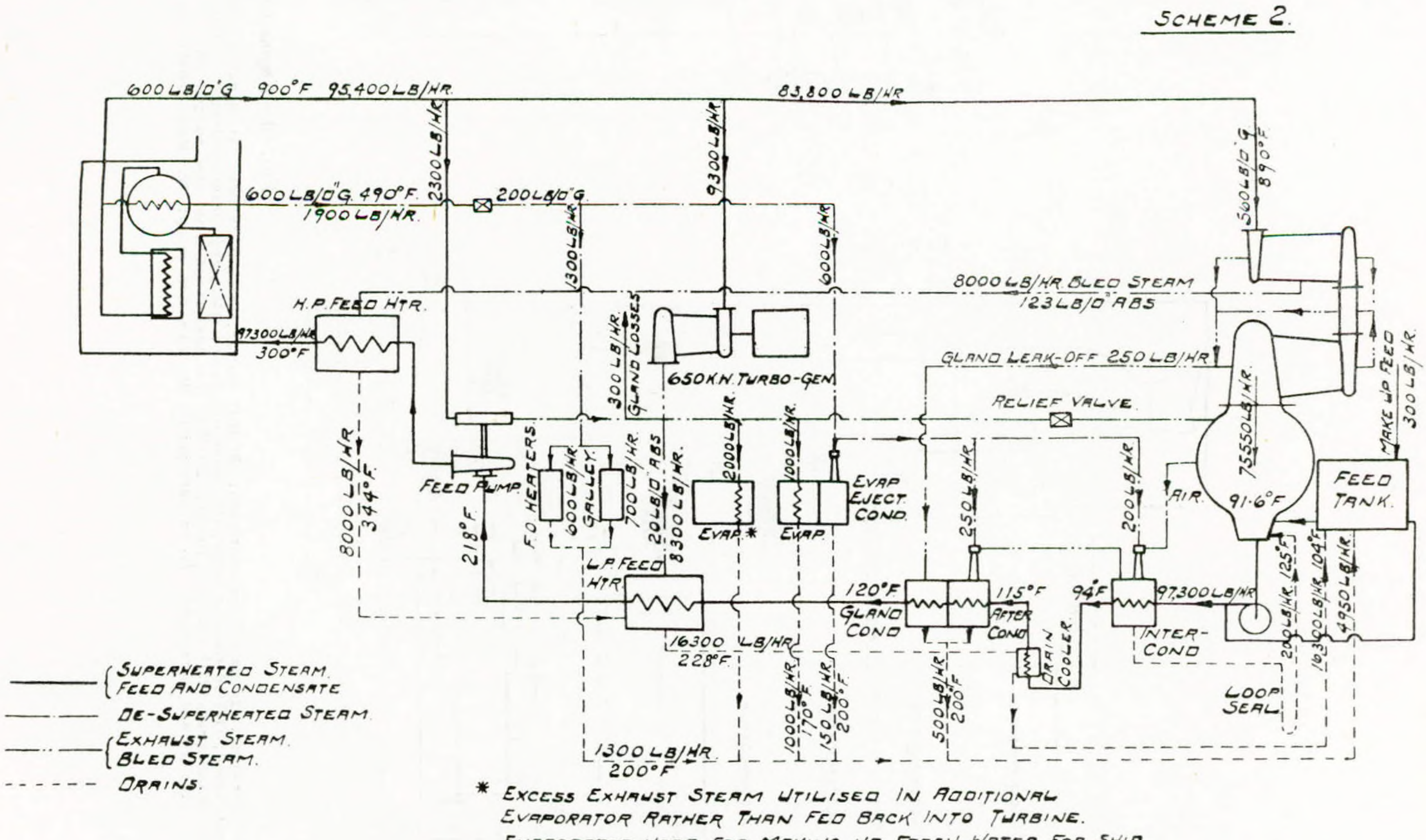


FIG. 9.—*Flow diagram for steam and feed lines, Scheme 2.*

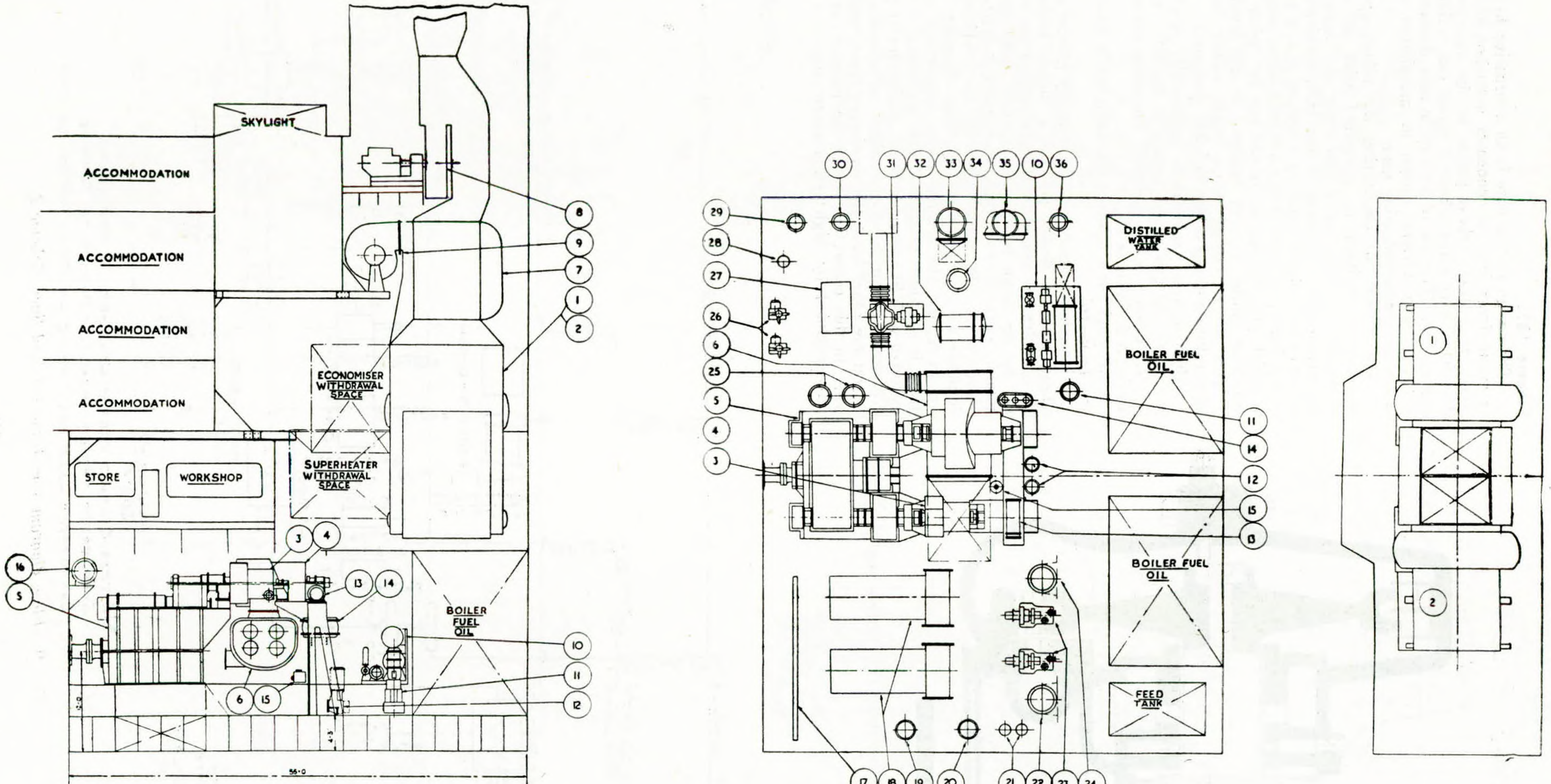


FIG. 10.—*Engine-room layout, 13,000-s.h.p. unit, Scheme 2.*

(1) Main boiler—port; (2) main boiler—starboard; (3) H.P. turbine; (4) L.P. turbine; (5) reduction gearing; (6) main condenser; (7) air heater; **(8) induced-draught fan;** (9) forced-draught fan; (10) oil-burning installation; (11) oil transfer pump; (12) extraction pump; (13) gland steam condenser; (14) air ejector; (15) gland collector; (16) oil cooler; **(17) switchboard;** (18) turbo-generators; **(19) bilge pump;** (20) ballast pump; (21) aux. feed pumps; (22) H.P. feed heater; **(23) main feed pumps;** (24) L.P. feed heater; (25) lub. oil pump; **(26) oil purifiers; (27) refrig. machine;** (28) refrig. circ. pump; (29) fire and bilge pump; (30) sanitary pump; (31) main circ. pump; **(32) drain cooler;** (33) steam generator; (34) fresh-water pump; **(35) evaporator;** (36) distiller condenser.

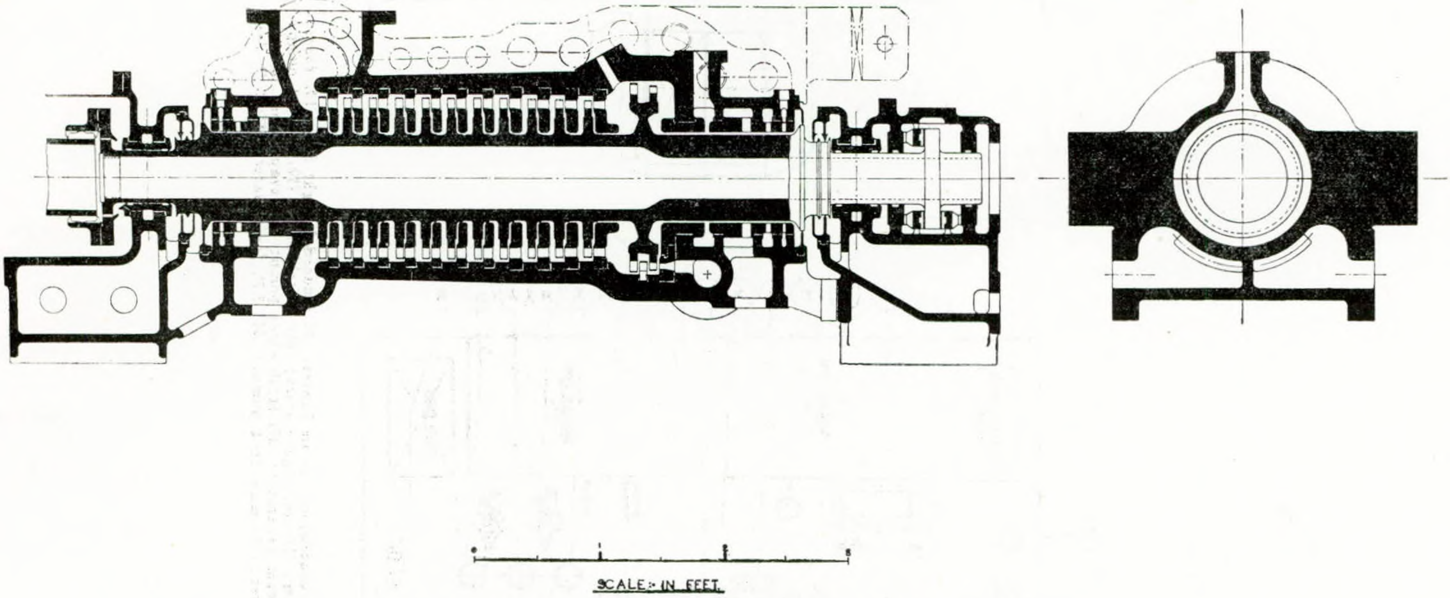


FIG. 11.—*Sectional elevations of H.P. turbine, Scheme 3.*

design difficulties. Increases in wall thicknesses are required but overall sizes are reduced due to the greater steam density, and the weight summary at the end of this paper shows that the total increase in weight is very small. The reduced complication of the steam reheat system as compared with the gas reheat system makes it suitable for the intermediate power range—say from 8,000 h.p. to 16,000 h.p. Neither system can be recommended for installations of low power.

Steam flow through the gas reheaters ceases during astern running, whilst the boilers are still required to generate a considerable quantity of steam, although with astern turbines it is very difficult to develop a power of 50 per cent. of the ahead. In the steam reheat system no adverse effects can result from this cause, and indeed the steam normally supplied to the reheaters becomes available for use in the astern turbines. In the gas reheat system, however, special precautions must be taken to prevent the reheater from burning out under these conditions, and this problem may present considerable practical difficulties. In fact, it is evident from the paper(3) that it was chiefly consideration of this difficulty which led to the adoption of uni-directional turbines with electrical

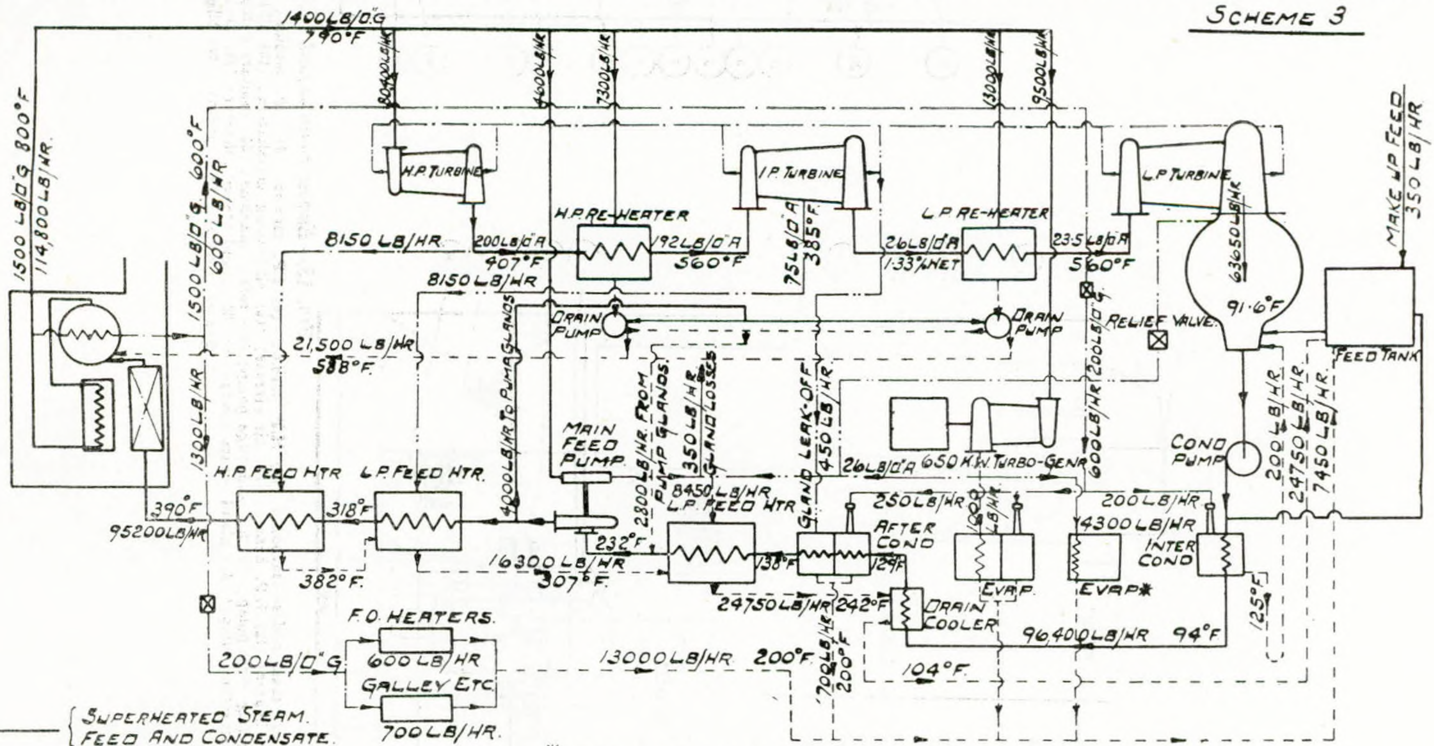


FIG. 12.—*Flow diagram for steam and feed lines, Scheme 3.*

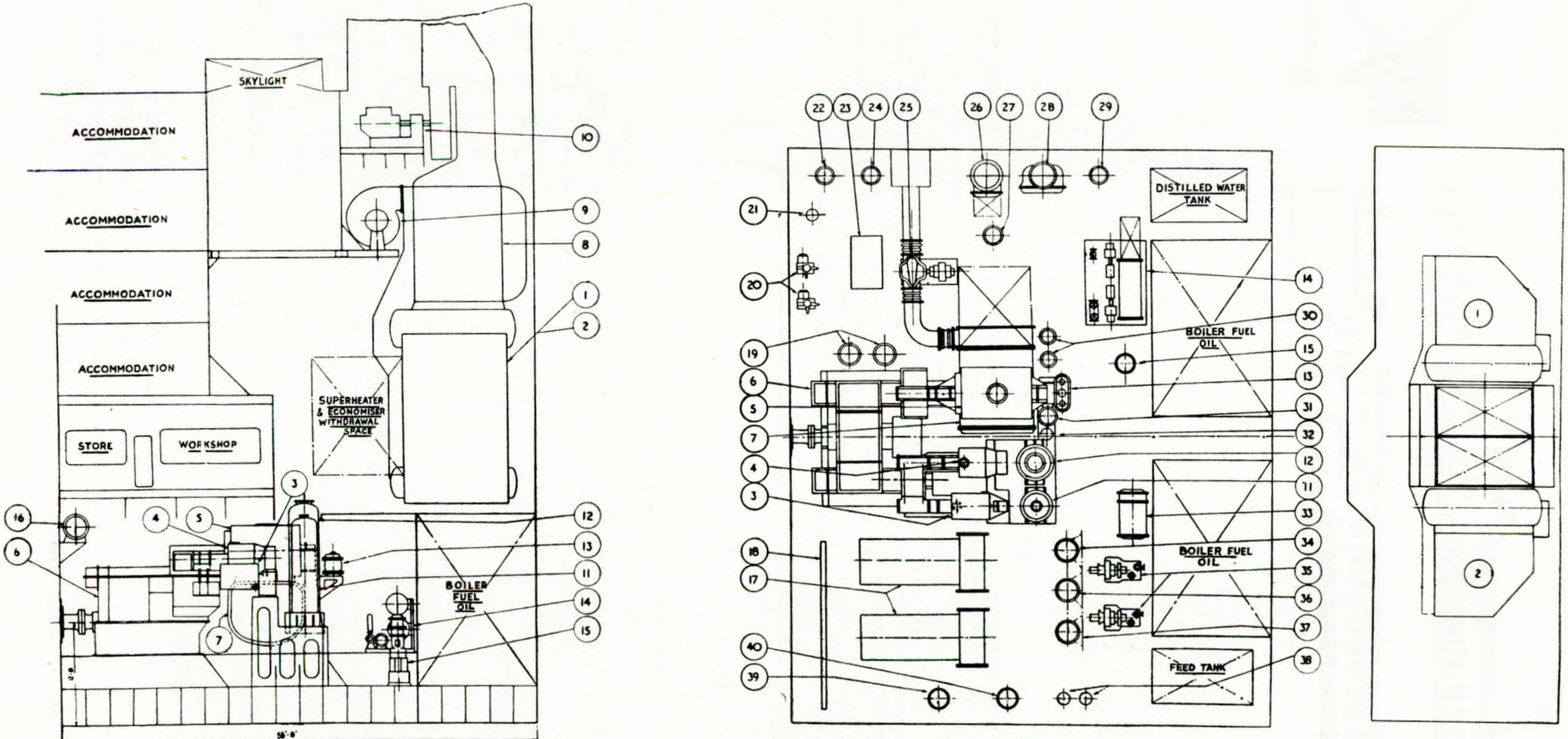


FIG. 13.—*Engine-room layout, 13,000-s.h.p. unit, Scheme 3.*

(1) Main boiler—port; (2) main boiler—starboard; (3) H.P. turbine; (4) I.P. turbine; (5) L.P. turbine; (6) reduction gearing; (7) main condenser; (8) air heater; (9) induced-draught fan; (10) forced-draught fan; (11) H.P. heater; (12) L.P. heater; (13) air ejector; (14) oil-burning installation; (15) oil-fuel transfer pump; (16) oil cooler; (17) turbo-generators; (18) switchboard; (19) lub. oil pumps; (20) oil purifiers; (21) refrig. circ. pump; (22) fire and bilge pump; (23) refrig. machine; (24) sanitary pump; (25) main circ. pump; (26) steam generator; (27) fresh-water pump; (28) evaporator; (29) distiller condenser; (30) extraction pumps; (31) gland steam condenser; (32) gland steam collector; (33) drain collector; (34) H.P. feed heater; (35) main feed pumps; (36) I.P. feed heater; (37) L.P. feed heater; (38) aux. feed pumps; (39) bilge pump; (40) ballast pump.

transmission, in spite of the fact that the resulting greater transmission loss cancelled out most of the gain in efficiency due to the use of reheat.

The special part of this design, that of the H.P. turbine, is shown in Fig. 11. The H.P. turbine runs at 5,350 r.p.m., the corresponding figures being 5,350 for the I.P. turbine and 4,100 for the L.P. turbine. The power distribution over the pinions would be H.P. 33 per cent., I.P. 29 per cent. and L.P. 38 per cent. In this case the solid rotor forging is chambered out to give reduced weight and consequently an adequate margin from critical speed considerations, while having a large number of stages to handle the heat drop and adequate gland length to seal the turbine cylinder against leakage of high-pressure steam. Features described in the earlier schemes are again employed, but the drawing gives accurately the scantlings required to operate with an initial pressure of 1,400lb./sq. in.

TABLE 6.

SCHEDULE OF MACHINERY WEIGHTS.

	Scheme (1)	Tons	Scheme (2)	Tons	Scheme (3)	Tons
	Turbine cylinder	11	H.P. turbine	8	H.P. turbine	3·5
	Gearcase and couplings	50	L.P. turbine	20	I.P. ahead and H.P. astern	4·5
					L.P. ahead and astern	19·5
			Gearing	77	Gearing	79
	Condenser	25	Condenser	38	Condenser	32
Turbines ... ... / Gearing ... ... / Condenser		86		143		138·5
Boilers (dry)		150		229		238
Total weight main machinery and boilers		236		372		376·5
Stern gear, shafting and propeller		75		152		152
Auxiliaries incl. turbo-generators		57		140		145·5
Cocks, valves, piping and fittings		103		139		144
Funnel, ladders, gratings, floor-plates, tanks and spare-gear		89		174		174
Total dry weight		560		977		992
Water in boilers, oil and water in system, etc.		32		50		45
Total weight "Steam up" ...		592		1,027		1,037

Total weight is complete in every respect unless pumps for sanitary water, calorifiers and refrigeration load are required to suit a special service vessel or abnormal hotel load.

The auxiliaries allowed for meet the outline specification and are detailed in the tables giving the auxiliary loads appropriate to each of the schemes.

The essential differences between Schemes (2) and (3) are summarized as follows:—

	Pressure	Temperature	Materials
Scheme (2)	Medium	High	Special
Scheme (3)	High	Medium	Normal but increased in scantlings.

AVERAGE ELECTRIC AUXILIARY LOAD.

The auxiliaries are identical with those for Scheme (2), but with the addition of two reheater drain pumps. Total h.p. = 15.

	In Port	At Sea
Average generator load in kW.	619	445

The generator load when manœuvring is approximately as for continuous running at sea.

With 650-kW. turbo-generators, the steam rate at 445-kW. load $=21{\cdot}3$lb./kW.-hr. when exhausting at 26lb./sq. in. abs.

Turbo generator steam consumption $= 21{\cdot}3 \times 445$
$= 9{,}500$lb./hr.

AVERAGE STEAM AUXILIARY LOAD.

	24-hr. Load at Sea
Turbo generator	9,500
Main feed pump	4,600
Air ejectors	450
Evaporator air ejector	150
F.O. heaters	600
Galley, ship heating	700
	16,000lb./hr.

The flow diagram of steam and feed water is given in Fig. 12.

No drawings of turbine and gearing for this scheme have been given as it is felt that the layout of machinery in Fig. 13 is sufficient. The engine room length for this alternative to Scheme (2) is again 55ft. 0in.

Summary.

In this section the three schemes are compared on the basis of fuel consumption, heat balances and weight. Apart from pressure and temperature considerations which have a secondary effect on general layout, the three schemes enable the arrangements for single-cylinder, compound and three-cylinder turbines to be demonstrated, all driving through double-reduction gearing of the articulated type. In addition the drawings of the H.P. turbines for Scheme (2) and Scheme (3) show how an increase in pressure from 560lb./sq. in. to 1,400lb./sq. in. at the turbine inlet affects scantlings such as the joint flanges in the turbine casings, all drawings being made to scale. It will be seen that the practical difficulties of high pressure are largely those of increased thicknesses.

Machinery weights are given in Table 6. For the turbines and gearing the weights of the separate turbines, condenser and gear cases complete are given individually, and then the boiler weights are added. This gives the relative weights due to the changed inlet conditions and the number of cylinders employed.

The other engine-room weights are aggregated under convenient headings, and it will be seen that the weights for the supply of cocks, valves, piping, floorplates, ladders, gratings, etc. are on a generous scale. They can be relied on as there is no necessity to give low weights in order to demonstrate the outstanding characteristics of steam-turbine drive for vessels requiring the powers mentioned in this symposium.

Consumptions and Heat Balances.

The specific steam rates (non-bleed) for the main turbines at full and half load are given in Table 7. The turbine consumptions given in this paper are calculated on the basis of designed clearances being increased in service by 25 per cent. at glands and other constrictions, which will give a reasonable estimate of consumption after the turbine has operated for some time. A warning is given against direct comparisons of the steam rates in this paper with those which may not have such allowances. The half-load consumptions prove that the change in efficiency over this large range of power output is not large.

TABLE 7.

Scheme	Steam rate (non-bleed) lb./s.h.p./hr. Full load	Half load
(1)	7·01	7·71
(2)	6·03	6·51
(3)	5·37	5·80

The fuel burned under boilers can be of a very low-grade residual type—in fact anything which can be burnt—and this is recognised by the specification of the Superintendent Engineers'

Committee which states that the gross calorific value of boiler oil should be taken as 18,500 B.Th.U.'s/lb. The lower value compared with that allowed for Diesel fuel increases the specific consumption figure for turbine machinery in comparison with Diesel machinery, although such low-grade boiler oil is bought at a great deal lower cost per B.Th.U.

Heat balances are given in Table 8.

In turbine-engined ships the total fuel is not the addition of the fuel consumption for main propelling machinery and that for auxiliaries, but is obtained from a consideration of the installation as a whole. In the auxiliary consumption figures, average figures have been used and are higher than those given in papers in bibliographical references [(2)] and [(3)]. With the calorific value given above and the boiler efficiencies given in Table 10, the "all purposes" and "propulsion only" fuel rates per s.h.p./hr. shown in Table 9 are arrived at. If the change in consumption with a different boiler efficiency is required, it can be arrived at by direct proportion. The turbo generator consumption is given in Table 11 to complete the data used in this paper.

TABLE 8.
SUMMARY OF HEAT BALANCES.

	SCHEME (1)		SCHEME (2)		SCHEME (3)	
	B.T.U./ s.h.p./hr.	Per cent.	B.T.U./ s.h.p./hr.	Per cent.	B.T.U./ s.h.p./hr.	Per cent.
Heat supplied in boiler fuel ...	11,300	100	10,130	100	9,680	100
Loss in boiler (η=88%)	1,360	12	1,220	12	1,160	12
Heat to steam	9,940	88	8,910	88	8,520	88
Steam losses	90	0·8	65	0·65	40	0·4
Heat absorbed by feed pump and turbo-generator	195	1·7	200	2·00	225	2·3
Heat to evaps., F.O. heaters, galley, etc.	375	3·3	400	3·95	570	5·9
	— 660	— 5·8	— 665	— 6·6	— 835	— 8·6
Heat to main turbine and feed heating	9,280	82·2	8,245	81·4	7,685	79·4
Loss to condenser	6,600	58·5	5,620	55·5	5,060	52·3
Heat corresponding to turbine work	2,680	23·7	2,625	25·9	2,625	27·1
Transmission loss	135	1·2	80	0·8	80	0·8
Heat corresponding to shaft work...	2,545	22·5	2,545	25·1	2,545	26·3

TABLE 9.

Scheme	Fuel Rate			
	All purposes		Propulsion only	
	lb./s.h.p./hr.	tons/24 hrs.	lb./s.h.p./hr.	tons/24 hrs.
(1)	0·61	49·0	0·58	46·6
(2)	0·54	75·2	0·52	72·3
(3)	0·52	72·3	0·50	69·6

TABLE 10.
BOILER PARTICULARS (FOSTER WHEELER).

	SCHEME (1)	SCHEME (2)	SCHEME (3)
No. of boilers per ship	2	2	2
Working press. sup't'r. outlet lb./sq. in. gauge	450	600	1,500
Total temp. sup't'r. outlet deg. F.	800	900	800
Feed temp. to boilers deg. F. ...	300	300	390
Evaporation per boiler lb./hour — service ...	32,500	48,600	58,300
includes desupt. steam — trial ...	40,000	54,000	70,000
H.S./boiler. Boiler sq. ft.	3,790	4,625	3,770
Waterwall ,,	484	280	525
Economiser ,,	3,760	4,770	6,580
Superheater ,,	1,150	1,700	1,470
Airheater ,,	2,815	5,250	8,890
Efficiency (G.C.V.) %	88	88	88
Oil, calorific value, B.T.U./lb. ...	18,500	18,500	18,500
Oil burnt/boiler lb./hr., normal...	2,280	3,550	3,400
,, ,, ,, trial ...	2,840	4,000	4,330
Furnace volume, cu. ft. ...	970	1,025 (combined)	1,000

The figures given above, which have been supplied by the courtesy of Messrs. Foster Wheeler, Ltd., enable any comparisons to be made on specific rates of oil burned or evaporations in relation to heating surfaces of the boiler.

These fuel rates are not figures for new machinery under trial trip conditions, but are such as should be obtained after years of service providing the machinery has been reasonably maintained.

The use of Diesel generators to provide power for the M.D. auxiliaries has been examined, but apart from the necessity of carrying and handling two grades of fuel oil in the engine room, etc., it was found that the "all purposes" consumption was little changed from that obtained using turbo-generators for electric power. The expense and space occupied would also be greater. *The use* of coal has not been considered in this paper because of lack of availability at present.

None of the designs shown requires a foreign licence, and they are available to all British marine turbine builders. After the 1914-18 war there was a rush to take out foreign licences for Diesel engines which had the unfortunate tendency of retarding the production of engines wholly British in origin. It is hoped that the like will not occur in the marine field as regards gas-turbine machinery after the recent war.

In conclusion, thanks are due to the Council of Pametrada for agreeing to this paper being given by the author, who also thanks Mr. W. Sampson of Messrs. Foster Wheeler, Ltd. for preparing the special designs of boilers shown in this paper and actually producing a further one for Scheme (3) when calculations had shown that a change in consumption rate could be achieved. The author also thanks his colleagues at Pametrada who assisted in the preparation of this paper, particularly Mr. Yates, the senior designer, who prepared the turbine particulars, and Mr. Wilkinson who assisted in the calculations.

TABLE 11.
SCHEDULE OF STEAM CONSUMPTIONS OF AUXILIARY TURBO-GENERATORS.

	SCHEME (1)	SCHEME (2)	SCHEME (3)
Output in kW. condensing ...	450	700	700
,, ,, non-condensing	290	475	500
Initial conditions, pressure, lb./sq. in. gauge	420	560	1,400
Temperature, deg. F. ...	790	890	790
Consumption— lb./kW./hr. 27 in. vacuum — 100% load	15·7	14·5	14·4
75% ,,	16·2	14·9	14·8
50% ,,	17·5	16·1	15·9
Consumption— lb./kW./hr. 20 lb./sq. in. abs. back pressure — 100%	24·3	21·3	21·1
75%	24·9	21·9	21·7
50%	26·9	23·7	23·5

The generators fit particularly well the two conditions of output for sea load and port requirements. For use of exhaust steam at sea the flow diagrams of steam and feed water should be consulted.

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(3) "A 1,200-pound Reheat Marine Installation". B. Fox and R. H.

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(II) The Direct-coupled Diesel Engine

By C. C. Pounder (Vice-President).

Synopsis.

Mention is made of the propelling engine types which form the basis of the comparisons. These include some of the latest Harland and Wolff designs. The main machinery, auxiliaries, shafting, pipe systems and all other matters which a complete marine installation comprises, receive attention. Weights, spaces, overhauling heights and other comparative particulars are given for the 7,500-s.h.p. single-screw and the 13,000-s.h.p. twin-screw machinery sets. The paper closes with some general remarks regarding engine details.

(A) PRELIMINARY CONSIDERATIONS.

Alternative Propelling Engines.

There are four engine designs within the Harland and Wolff range of types and sizes which fulfil the power requirements of this symposium. Two are double-acting engines; two are single-acting; all are two-stroke cycle.

The engine shown in Fig. 1 is a double-acting two-stroke engine of established design, 620 mm. (24·41in.) bore, 1,400 mm. (55·12in.)

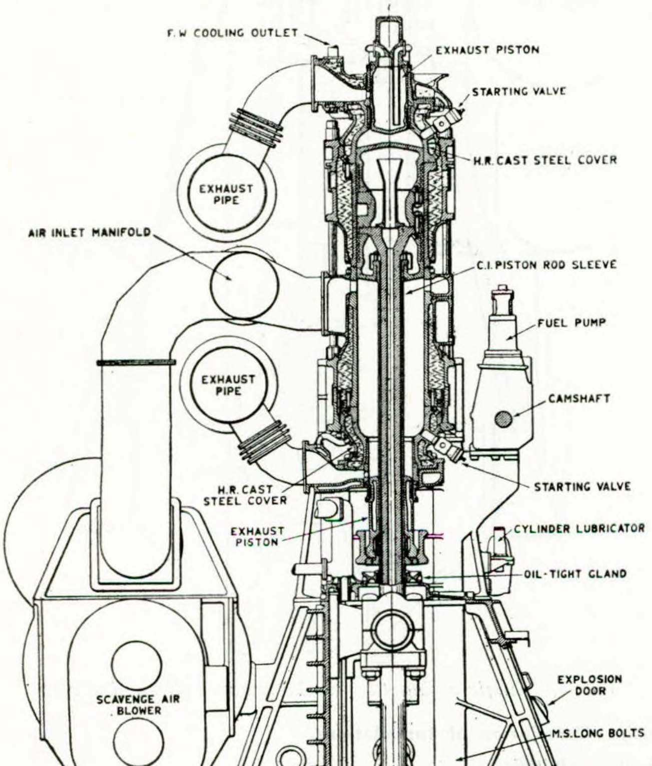


FIG. 1.—*Double-acting two-stroke engine. Scale 1:70.*

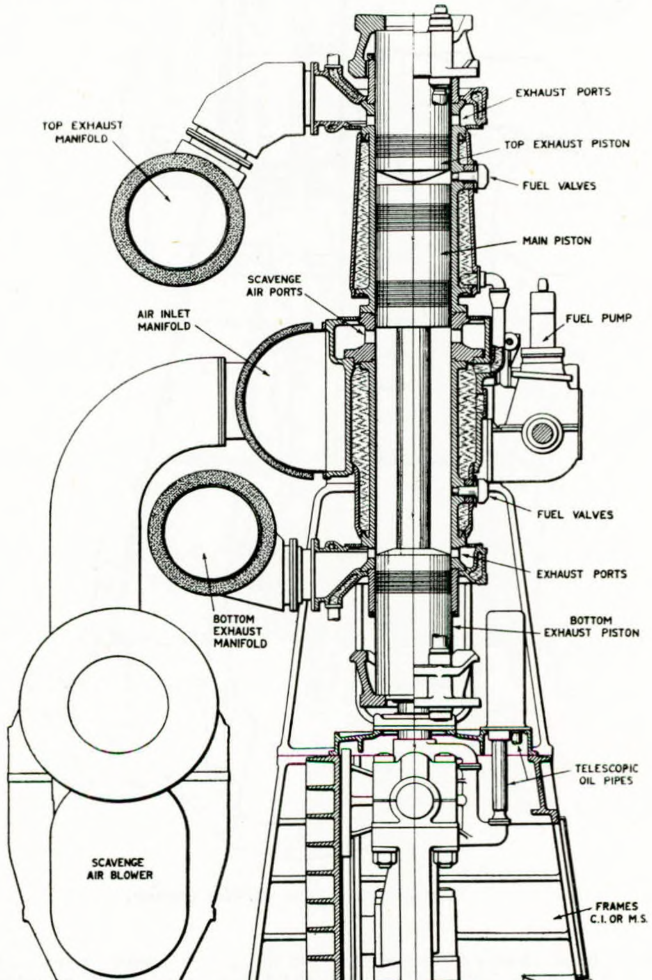


FIG. 2.—*Double-acting two-stroke engine; alternative type. Scale 1:60.*

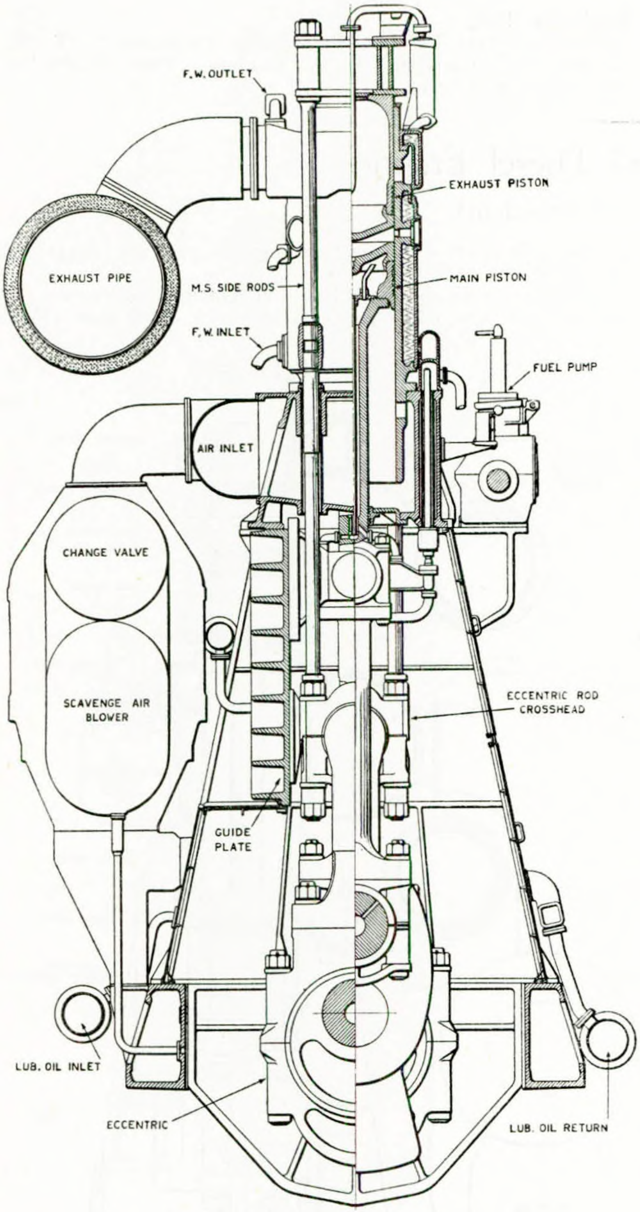


FIG. 3.—*Single-acting two-stroke engine.*

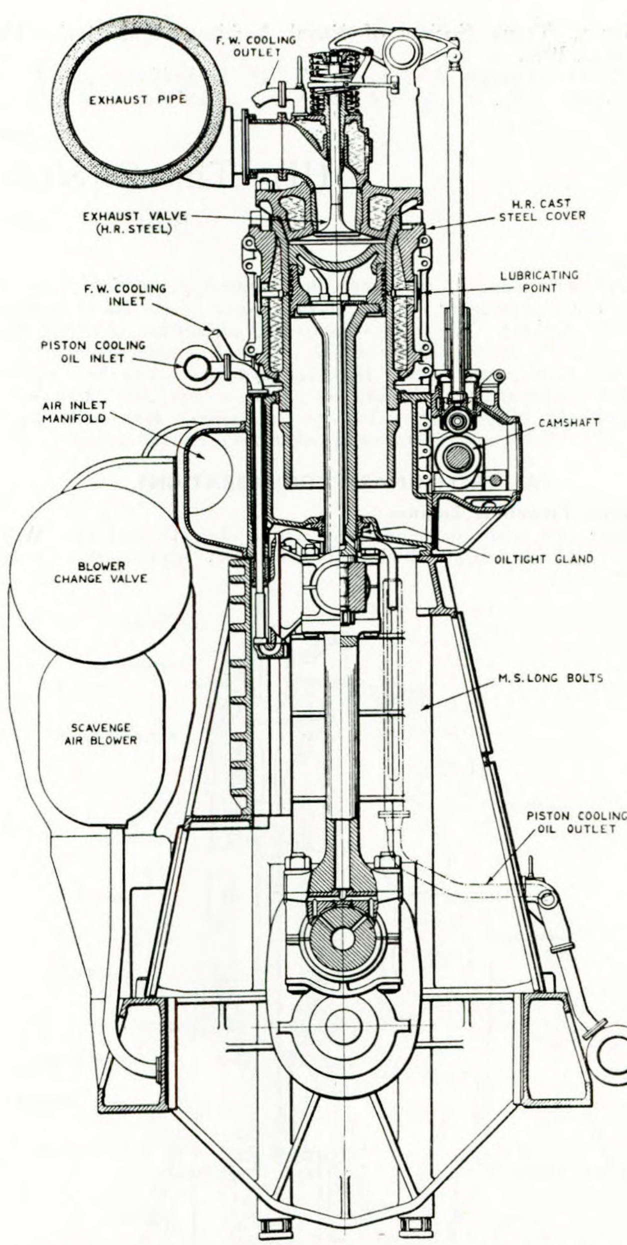


FIG. 4.—*Single-acting two-stroke engine; alternative type.*

stroke. It is chiefly characterized by the piston valves which are arranged in the cylinder covers and which are actuated by an eccentric drive.

The engine illustrated by Fig. 2 is also a double-acting two-stroke design; it is 590 mm. (23·23in.) bore, with 1,270 mm. (50·00in.) main stroke plus 430 mm. (16·93in.) exhaust stroke. Cylinder covers are dispensed with in this engine, and the exhaust pistons—which are operated by eccentrics—are made the same diameter as the main pistons.

In Fig. 3 a single-acting two-stroke engine is shown. It is a very simple engine. As in Fig. 2, the exhaust pistons are made the full diameter of the working cylinders. The cylinders are 750 mm. (29·53in.) bore, with 1,500 mm. (59·06in.) main stroke plus 500 mm. (19·68in.) exhaust stroke.

Fig. 4 shows a single-acting two-stroke engine in which an exhaust poppet valve is centrally arranged in the cylinder cover. This design may be regarded as the two-stroke edition of the well-known four-stroke engine, of which a countless number have been made during the last thirty-five years. The cylinders are 740 mm. (29·13in.) bore, 1,400 mm. (55·12in.) stroke.

All the propelling engines offered herein are rated well below their maximum continuous output.

Outline Description of Installations.

(*a*) *Main engines.*

The construction of the respective engines will be fairly clear from Figs. 1 to 4. The following notes are intended to supplement the information provided by the diagrams.

The bedplates and frames are substantially constructed in cast iron or fabricated steel, as may be desired, the allowable stresses being low. All joints and bearing surfaces are accurately faced and fitted metal-to-metal. The bedplate is bolted to strong seatings which are incorporated in the ship's double-bottom and an oiltight sump is arranged in the well of the seatings under the bedplate; the underside of the bedplate flanges is machined and a sealing bar welded to the tank top around the bedplate. The holding-down bolts and cast-iron chocks are of standard form. The main bearing keeps and whitemetal-lined bushes are of cast steel, the bushes being circular and so arranged that the bottom halves can be removed while the crankshaft is in place. To facilitate removal the bushes are made with an eccentricity of one millimetre. The length of all the main bearings is the same and mild steel studs hold down the keeps.

In the engine types illustrated in Figs. 2 and 3 the main and exhaust piston loads are equalised at each crank, leaving only the forces caused by the differential obliquity effect, the torque-reaction, and the seaway racking forces to be absorbed by the frames and bedplate. In the engines having cylinder covers, i.e. Figs. 1 and 4, the piston load reaction is taken by long mild steel bolts which extend from the cylinder covers to the bedplate bottom, the bending forces thereby engendered being carried within the strong cross-section of the girders. Accordingly, the frames and bedplates of the engines having full-sized exhaust pistons are considerably lighter than those of engines with cylinder covers, even allowing for the scantlings of the former being, perhaps, over-generous.

There are no unbalanced primary or secondary forces in the crankshaft systems of any of the engines; unbalanced couples, if any, are negligibly small.

The thrust blocks are integral with the bedplates. They are normally of the double-collar design with a whitemetal-lined stationary disc between the collars. But they may also be of Michell type. The thrust loading is of the order of 15 kg. per sq. cm. (say 200lb. per sq. in.), when calculated on the basis of brake thrust multiplied by an experiential co-efficient.

The crankshafts are fully-built, made in two sections coupled together by solid flanges and mild steel fitted bolts; the journal-pieces and crankpins are of forged mild steel, the crankwebs cast steel. In strength the crankshafts are about 10 per cent. in excess of Lloyd's rules. The eccentric sheaves—in engines having eccentric drives—are integrally cast with the webs, as are also the balance weights, if there are any. The crank webs are of special cross-section.

The position of the propeller blades is marked on the aft crankshaft coupling, for docking purposes; the usual gauges are supplied for showing when the aftermost crank is on top dead-centre. At the final coupling-up in the ship all cranks are adjusted to clock gauge, a chart being provided to show the readings.

The turning wheel is made in halves, of cast iron, and the teeth are cast on the rim. The turning gear, which is motor-driven, can rotate the crankshaft through one revolution in five to six minutes.

In the double-acting engines of Fig. 1 when looking forward upon the aft end for a single-screw installation, the cranks for a six-cylinder engine are arranged 1 and 6, 3 and 4, 2 and 5, with 120 deg. between the pairs. Two cylinders fire simultaneously, the sequence being: top, 1 and 6; bottom, 2 and 5; top, 3 and 4; bottom, 1 and 6; top, 2 and 5: bottom, 3 and 4. In the double-acting engines of Fig. 2, the cranks for a single-screw six-cylinder engine are arranged: 1.6.3.5.2.4; the angles between 1 and 6, 3 and 5, 2 and 4, are 30 deg.: the angles between 6 and 3, 5 and 2, 4 and 1, are 90 deg. The firing order is: 1T, 6T, 2B, 4B, 3T, 5T, 1B, 6B, 2T, 4T, 3B, 5B. For all seven-cylinder engines the crank sequence is: 1.7.2.5.4.3.6, with 51·43 deg. between cranks. For the double-actors the firing order is: 1T, 4B, 7T, 3B, 2T, 6B, 5T, 1B, 4T, 7B, 3T, 2B, 6T, 5B; for the single actors: 1.7.2.5.4.3.6. In the single-acting engine of Fig. 4, the crank sequence, also the firing order, for an eight-cylinder engine is 1.8.3.4.7.2.5.6, with 45 deg. between cranks; for a ten-cylinder engine the arrangement is 1.10.3.6.7.2.9.4.5.8, with 36 deg. between cranks. The differences in firing order and crank sequence are responsible for some apparent anomalies in crankshaft and especially in shafting weights.

The loading of crankshaft and connecting rod bearings is well within the limits which experience has shown to be reliable. The quotation of simple figures, to establish this point, would be misleading.

The forged mild-steel connecting rods are of normal design, with top- and bottom-end bearing bushes of whitemetal-lined cast steel, and bolts of mild steel. Carbon steel of 38-42 tons per sq. in. ultimate strength is used for the crossheads; the guide shoes—which may be articulated for double-acting engines—are steel castings, with whitemetal on ahead and astern faces. The guide plates and bars are cast iron. Crossheads and guides are marked to show when the pistons are in top and bottom positions. When running ahead, the crosshead shoes of single-screw sets slide on the guide plates; those of twin-screw sets on the guide bars. This has been standard practice for many years and is determined by the necessity for ready access to the crankcase from the front of the engine.

The piston rods are of forged, heat-treated, mild steel, 28-32 tons per sq. in. ultimate tensile strength, one piston rod per forging. Each rod is provided with a stout flange at its upper end for bolting to the piston; at its lower end—in a double-acting engine—it is attached to the crosshead by a wrought-iron top nut and a mild-steel bottom nut, or by a reduced tapered end and wrought-iron nut if a single-acting engine. The threads are of special form, cut with the greatest accuracy and given an exceedingly fine finish. The threads in the nuts are equally well made, with the correct mating fit. In double-actors, e.g. Figs. 1 and 2, each rod is enclosed in a cast-iron sleeve which is screwed into the piston at the top end and slides freely on the rod at the bottom. The annular space between the sleeve and the rod is utilized for conveying oil to the piston, thereby cooling the sleeve. The sleeve passes through a stuffing box located in the lower exhaust piston and through a gland in the diaphragm plate on the top of the crankcase.

The construction of the pistons will be clear from the diagrams. The piston ends are castings of chrome-molybdenum steel. In the double-acting engines, cast-iron distance-pieces are arranged between the piston ends; the piston-rod flange is sandwiched between the lower end and the distance-piece. The form and proportions of the pistons are well-adapted for withstanding the temperature variations. In the single-acting engines, Figs. 3 and 4, the heat-resisting cast-steel piston end is screwed-on to the cast-iron body. The piston rings are of Ramsbottom type, with diagonal slot, pinned.

The exhaust pistons, Figs. 1, 2 and 3, are constructed on the same basis as the main pistons. In some engines the exhaust pistons are single castings of chrome-molybdenum steel. Both the main and exhaust pistons are oil-cooled through a telescopic pipe system. The exhaust pistons are tap-bolted to cast steel yokes; the latter are connected by mild-steel side rods to the eccentric drive. This drive comprises whitemetal-lined mild-steel straps, which ride on the sheaves cast with the crankwebs, and eccentric rods fitted at their upper ends with either simple guides or with crossheads—depending upon the size and type of engine. The eccentric bolts and tap bolts are mild steel; all bearings are of ample dimensions. Gauges are supplied for registering the position of the piston valves relative to their cranks when on top dead-centre.

In engines having cylinder covers, viz. Figs. 1 and 4, these are sturdily constructed of chrome-molybdenum cast steel. The cylinder liners are turned all over, outside as well as inside; the material is vanadium cast iron. There are heavy flanges on the liners; these are securely held to the covers by tap bolts, or bolts, of 3½ per cent. nickel steel. The cylinders are fitted with mechanical lubricators. All cylinder liners and covers are fresh-water cooled. Longitudinal scavenging is a characteristic of all the engines illustrated.

In single-acting engines of the poppet valve type, the central exhaust valve is a single forging of the highest class of heat-resisting steel; the valve seat is pearlitic cast iron. The poppet valve actuating gear is shown in Fig. 4. The cams and rollers are case-hardened carbon steel; the camshaft is chain-driven from the crankshaft. During the reversal of the engine a lost motion clutch comes into operation—unlike the four-stroke engine wherein the rollers are drawn clear of the cams and the camshaft is moved fore-and-aft to bring a second set of cams into line with the rollers.

Each cylinder is provided with its own fuel injection pump—or pair of pumps if the engine is double-acting. The delivery of fuel is regulated by varying the effective stroke-volume of the pump. All the pumps are connected to the manœuvring lever—enabling the amount of fuel injected into the cylinders to be controlled—and to the emergency governor, for stopping the engines. The governor is of the inertia type, so arranged that the fuel cut-out speed can be adjusted when the engine is running. The fuel pumps are operated by hardened-steel cams, mounted on a camshaft, acting upon hardened-steel rollers. The plungers are of case-hardened steel; the solid steel bodies are fitted with sleeves of special cast iron. A pressure gauge, with device for damping-out pressure fluctuations, is connected to each pump. Fuel oil is prevented from reaching the lubricating oil.

The fuel surcharging pump is located near the manœuvring platform and is lever-driven from the engine. The fuel oil flows from the service tanks to the surcharging pump and is delivered through effective filters to the fuel pumps, thence to the fuel valves on the cylinders. The fuel valves are of automatic, spring-loaded type, operated by the fuel pump pressure. The bodies are of mild steel and the spindles case-hardened carbon steel. In the single-acting engines there are two fuel valves per cylinder; and in the double-acters, two fuel valves at the top end and two at the bottom end.

A starting valve of automatic type is arranged on each cylinder—at each cylinder end for the double-acting engines. The valves are of ample dimensions for ensuring quick-starting; they are operated by compressed air, the supply of which is controlled by an air distributor rotating at engine revolutions. The valve bodies are of cast and mild steel, with spindles, valves and seats of stainless steel. A safety valve of standard design is mounted on each single-acting cylinder and on the top and bottom ends of each double-acting cylinder.

The scavenge blowers—of which there are two per engine—are of rotary type. The rotors, which are turned by gear wheels on the rotor spindles, are hollow iron castings and the casings cast iron. Spring couplings, on the driving shafts, relieve the chains of load when manœuvring; and a change-over valve, operated by the engine reversing gear, interchanges the suction and delivery ports when reversing, thus ensuring an ample air supply to the cylinders when running astern. Each scavenge blower draws its air from the engine room through an intake silencer and discharges to the air manifold on the back of the engine. The blower sleeve bearings and the gears are served with oil from the forced-lubrication system.

The fuel-pump camshaft and the scavenge blowers are chain-driven from the crankshaft. The drive for each half of the blower comprises a duplex chain, 2·5in. pitch, except for the engine of Fig. 3, in which the chain is 2in. pitch. For the fuel-pump camshaft there is a single chain, 4in. pitch for the double-acting engines, 3in. and 3·5in. duplex for the single-acting engines. On each chain-drive there is an adjustable, spring-loaded jockey wheel. This may be placed either on the slack side or on the driving side of the chain. Only in short chains is it important to have the jockey wheel on the slack side. The crankshaft chain wheels are 0·45 per cent. carbon-steel castings, the jockey wheel forgings being of the same material, machined all over.

The crankcase is completely enclosed. Large portable oil-tight inspection doors, of light construction, are clipped to the frames. All running gear is thus quickly exposed and rendered easily accessible.

The manœuvring gear is arranged at the front of the engine, at mid-length, and is of simple form. There are only two levers, i.e. one for air and fuel, one for reversing.

Oil vapour is extracted from the crankcase, which is completely sealed, by means of a fan and suitable ducting; an oil saveall is arranged under the fan. The vapour is discharged at the top of the funnel.

The exhaust manifolds, which are made of welded or riveted steel plates, are provided with flexible expansion pieces at and between the cylinders; the bends are cast iron or steel. All exhaust pipes are well-lagged and cleaded with galvanised steel plates, as may be required; and they are led to large, effective silencers, constructed of steel plates, which are housed in the funnel. By-pass valves are arranged in the exhaust pipes, whereby the exhaust gases can be deflected, wholly or fractionally, to the waste-heat boilers. Indicating gear, pyrometers, thermometers, pressure gauges and all necessary instruments are provided.

(b) Shafting, sterntubes, propellers.

The tunnel shafting is made approximately 5 per cent. in excess of Lloyd's rules for strength and, in way of the plummer blocks, is smooth-turned to a diameter ⅛in. larger than the shaft body. The solid-flange couplings are bolted together by mild-steel tapered bolts, all bolt-holes being bored from the same template. On such bolts the taper is ¾in. on the diameter per foot of length; a so-called parallel bolt is given a taper of $\frac{1}{16}$in. per foot. Guard plates, with sight holes, are fitted over the couplings. There are two bearings to each shaft and all shafts are made the same length, if practicable, for convenience of manufacture and assembly. One length per line is ordered with extra-thick coupling flanges—usually 1¼in. additional material on face and on back, for each flange—to allow for discrepancies at the ship.

The thrust and tail shafts are respectively made about 10 per cent. and 10 to 15 per cent. above Lloyd's requirements for strength. A continuous gunmetal liner is shrunk on to the tail shaft in way of the sterntube. The aft end of the liner is reduced ¼in. on the diameter, outside the sterntube bushes; a water groove is turned on the liner, just forward of the gland face in the shaft tunnel. A spare propeller shaft, securely strapped, is carried in the tunnel, at the aft end. Shaft withdrawing gear is provided and housed on board.

The plummer blocks, which are of the self-lubricating type, are constructed with lower halves of whitemetal-lined cast iron: the hinged upper halves, of $\frac{3}{16}$-in. galvanised steel, are stiffened at the rims. The aftermost plummer block, which is of robust design, strongly seated, is made of cast iron throughout. Top and bottom halves are whitemetal-lined and there is a circumferential whitemetal strip at the forward end face to take the thrust of a towing collar which is forged on the aftermost intermediate shaft length. In normal circumstances the collar is ¼in. to ⅜in. clear of the block. Each tunnel bearing is fitted with a branch pipe on the underside of the block, with a control cock for water circulation. The outlet pipes are led to a drain pipe which terminates in the engine-room bilge, or as may be approved. The water service pipe serves as handrailing in the tunnel. Cast-iron bulkhead stuffing boxes, in halves, are provided.

The taper of the propeller shaft cone is ⅞in. on the diameter per foot of length. The thimble-point on the shaft is equal in length to about one half of the nut depth, to facilitate the entering of the nut upon the thread. The keyway is cut with a special milling tool in such a way as to avoid the stress-raising influence of sharp corners. In twin-screw ships the screwing is right-handed on both shafts, the old-time practice of screwing the nut left-handed for a right-hand propeller, and vice-versa, having been discarded.

The heavy cast-iron sterntubes are of standard pattern, flanged at the forward end and secured by a wrought-iron nut at the aft end, with a lignum-vitæ-lined gunmetal bush at each end; the wood in the lower halves having end grain. A water-circulating cock, fitted to the after peak bulkhead, cools the tail shaft gland. The gland is operated by toothed gearing on the nuts.

The propellers are of manganese bronze, solid form, with blades of aerofoil section and bosses streamlined; the streamlined cones are of bronze or cast iron, as may be agreed; the rope guards are cast iron. A recess is cut in the forward face of the boss, into which the end of the tail shaft liner projects, the space between being properly filled by a rubber ring. This is important, as many tail shafts have failed by corrosion fatigue arising out of an ill-fitting rubber ring. Strong eye-plates, with shackles, are riveted under the counter of the vessel for lifting the propeller. For single-screw ships a spare propeller in manganese bronze is carried on board; in twin-screw ships a pair of spare bronze propellers are provided.

(c) Pipe systems.

In all pipe arrangement work the highest standards of mercantile practice are adhered to. Classification society requirements are fully met. The bilge, ballast, fresh-water circulating, deck, sanitary, fuel-oil transfer, and kindred systems, require no comment. The pipes are, in general, of lap-welded steel, with welded flanges; valve chests are of cast iron and fittings of welded steel or cast iron as may be most convenient. The auxiliary steam and exhaust pipes are made of copper.

The main-engine cylinder jackets and covers are fresh-water circulated from a closed circuit, the fresh-water circulating pump discharging through the cooler to the lowest part of the cylinders and the water returning to the pump suction from the highest part of the engine system. A small make-up tank is arranged at the most elevated point of the circuit. The main salt-water circulating pump, drawing from the sea, discharges to the fresh-water cooler and thence overboard. The auxiliary Diesel engines are most usually fresh-water circulated.

The starting air system is simple. Compressed air at 25 atmospheres (356lb./sq. in.) is stored in the air reservoirs, sufficient in quantity for many more than the twelve engine starts required by Lloyd's rules. The air reservoirs are charged by a motor-driven two-stage air compressor; there is also available a small emergency steam-driven compressor. The compressed air is piped from the reservoirs to the engines through a master stop valve, which is arranged to be within reach of the engineer at the manœuvring station. The compressed air pipes are made of hot-rolled solid-drawn steel, with screwed-on flanges. For the auxiliary Diesel engines, starting air is also obtainable from the air bottles provided. An engine-room air service system is included.

The forced-lubrication oil system comprises a drain tank, arranged in the double bottom, capacious enough to contain all the oil in circuit. The forced-lubrication pump draws from this tank through a strainer and discharges through a filter and a cooler to the engine oil connections, whence the oil drains back to the drain tank. Sight glasses are provided in the piston cooling and bearing oil return pipes. The oil system is a common one; it serves the piston cooling as well as all bearings. Centrifugal purifiers are introduced into the scheme, for continuous use at sea if required, the oil being heated in a head tank on its way from the pressure line to the purifiers. The lubricating oil pumps, also the filters and coolers, are duplicated as working and stand-by units. The quantity of oil circulated through the engine must be great enough to cool all the parts, as well as to lubricate them. A spare charge of oil is carried in one or more storage tanks, with connections to the system. Pipe lines for deck filling and discharge are included. The oil coolers may be circulated by a branch line from the main circulating system. The drain tank suction is as far removed as practicable from the oil return, to avoid frothing. All lubricating oil pipes are lapwelded steel, thoroughly scoured after bending to ensure freedom from scale.

The fuel-oil transfer pump—which can draw from and discharge to any bunker tank—delivers the bunker fuel to an elevated unpurified oil tank, from which it flows into the centrifugal purifiers, to be again raised by means of the purifier pump to the purified oil tank, whence it gravitates into the main engine surcharging pump. The system includes fuel oil filling and discharges to deck. The fuel-oil pipes are made of lapwelded steel, with welded-on flanges.

All necessary cross-connections, high and low injections, weedclearing pipes, suction and discharge lines for the refrigerator circulating pump, also the pump itself, are included. Flanges for low-pressure services, e.g. bilge and ballast, are made to the B.S.I. marine flange table for 50lb./sq in.; for fuel-oil transfer suctions and discharges to that for 100lb./sq. in.; and to other tables according to the pressures.

Fresh- and salt-water coolers are of the multitubular type, with tubes of aluminium brass $\frac{3}{4}$in. outside diameter 18 l.s.g. thick, tube-plates being of rolled naval brass, bodies and doors of cast iron. Lubricating oil coolers are similar in construction, with tubes $\frac{1}{2}$in. outside diameter 20 l.s.g. thick, but the bodies may be fabricated steel. The auxiliary condensers are of non-vacuum type, constructed of the same materials as the coolers, with tubes $\frac{3}{4}$in. outside diameter 18 l.s.g. thick. The condensate flows into a float-controlled tank. To obviate the rising of vapour into the engine room, a salt-water circulated coil is arranged in the tank.

There are no main engine-driven pumps. All engine-room and deck auxiliaries are motor-driven.

The engine-room pumps are, in general, of vertical centrifugal type, non-self-priming, except where one or more of the duties involves a suction lift, either direct or frictional; the motors are arranged to have, approximately, 25 per cent. speed reduction.

Air for combustion and ventilation is supplied to the machinery spaces by motor-driven axial fans located in the ventilating trunks. Platforms and ladders, extended valve spindles, lagging, service and storage tanks—all are in accordance with high-class practice. The engine-room floor plates are $\frac{5}{16}$in. thick below the chequers and $\frac{1}{4}$in. thick in the shaft tunnels. Handrails and toe-plates are provided where necessary. The platforms around the engines are generally of chequered plating, with open-spilled gratings elsewhere, for access to all valve chests and gear. A generous amount of engine-room outfit, as distinct from spare gear, is included. Store rooms for engineers and electricians, also workshop, are provided. The workshop contains: a $6\frac{1}{2}$in. × 6ft. lathe, driven by a 2-b.h.p. motor running at 1,450 r.p.m.; a 21-in. drilling machine capable of drilling holes up to 2in. diameter, driven by a 2-b.h.p. motor, 1,400 r.p.m.: a double-wheel grinder, driven by a 2-b.h.p. motor, 1,600 r.p.m. This is for the single-screw ships. For the twin-screw vessels, a $9\frac{3}{8}$in. × 9ft. lathe is added, driven by a 5-b.h.p. motor, 950 r.p.m. The funnel is included in the machinery weight; fire extinguishers and oily-water separators are excluded. The demarcation between engineers' pipes and shipbuilders' pipes is that customarily observed in the Belfast shipyards.

(*d*) *Miscellaneous.*

The steam requirements vary widely between ship and ship. One company may ask for a relatively small oil-fired boiler of the Cochran type, arranged at engine-room floor level, and it will fully meet steam requirements, with all-round economy of supervision, upkeep, and so on. Another may specify an exhaust-heat boiler, but cannot absorb all the steam and so the excess amount of gas is by-passed. Yet another may require an exhaust-gas boiler and utilizes to advantage every pound of steam provided. Taking the mean position, one alternatively-fired gas and oil composite-type thimble-tube boiler is arranged on each main-engine exhaust pipe and is complete with fuel-oil gravity system, electrically-driven blower, and automatic feed-pump control. Each boiler evaporates 1,000lb. of steam per hour on exhaust gas and 2,000lb. when using oil fuel; the working pressure is 100lb. per sq. in. by gauge. An adjustable gate-valve—as mentioned earlier—is arranged in the exhaust pipe, to by-pass the gases to the silencer to suit the varying demands for steam. The boiler is lagged with magnesium slabs 4in. thick and cleaded with planished steel plates. Steam connections are made to the emergency air compressor, evaporator, feed pumps, oil-fuel gravity and rectified tanks, whistle, deck and engine-room steam hoses, steaming-out tanks, deck connections for Suez Canal portable electric light plant, also galleys, baths, and so on.

Compressed air is stored in cylindrical air reservoirs, constructed of welded steel plates with dished ends and securely supported from the deck. The reservoirs are of sufficient strength for a working pressure of 25 kg. per sq. cm. (356lb. per sq. in.). For the single-screw ship there are two reservoirs each of 550 cu. ft. capacity; for the twin-screw vessel two reservoirs each of 800 cu. ft. capacity. The reservoirs are fitted with the necessary manholes and doors, relief and stop valves, drain and pressure gauge valves, fusible plugs with guards—three plugs per reservoir arranged along the bottom—and so on; the insides of the reservoirs are thoroughly scaled and varnished.

Air bottles, of 180 litres (6·35 cu. ft.) each, are supplied for the emergency starting of the auxiliary Diesel engines. Each bottle consists of a steel plate rolled and welded, with dished ends welded to body; it is fitted with a combined inlet and outlet valve, inspection door and drain valve.

The number and capacity of the auxiliary Diesel engines necessarily depend upon the type of ship, upon whether it is completely or partly insulated, or not insulated at all; also upon whether the refrigerating plant is driven by its own horizontal Diesel engines or is motor-driven by current obtained from the engine room. The auxiliary engines assumed herein are of the four-stroke single-acting Harland and Wolff trunk type, with six cylinders per engine, 24 hours per day continuous rating. The engines offered are completely balanced for primary and secondary forces and couples. The point regarding rating is noteworthy, as it is customary for many makers of marine auxiliary Diesel engines to offer engines with only a 12-hour rating. Each engine is self-contained with a lubricating oil pump, cooler and filter, also a priming pump. The covers and jackets are fresh-water cooled from the auxiliary system for twin-screw sets; in single-screw installations the auxiliary engines are occasionally salt-water cooled, but fresh-water cooling is much to be preferred, the extra cost notwithstanding. The engines are all made to the same handing; all working parts are interchangeable; each engine is mounted on a common bedplate with its dynamo. Hand turning gear is fitted to each engine. Provision is made for circulating the jackets, covers and coolers when in drydock. An ample amount of spare gear is provided for the auxiliary engines.

Regarding the electrical load: the maximum continuous sea load is the one which chiefly interests the engine builder. This is estimated for new ships by taking the total connected load and multiplying it by a diversity factor obtained from analysis of many voyage reports from comparable ships. Sometimes the total connected load is split-up and each section is multiplied by its own diversity factor. Thus the auxiliaries which continuously carry a full load at sea, e.g. the refrigerator, the main circulating pumps, the forced-lubrication pumps, and so on, are grouped together, and the total motor brake horse-power obtained. This figure is multiplied by 0·9, because the motors supplied by reputable makers always have power in hand. The remainder of the sea connected load, excluding stand-by units, is then grouped and the aggregate power is multiplied by a lower factor; this may be of the order of, say, 0·45. The mixed diversity factor thus obtained may vary from, say, 0·60 to 0·75. By way of example: one twin-screw vessel of approximately the power with which this symposium is concerned shows, when homeward bound, a maximum sea load of 1,600 amperes, a minimum of 1,300 amperes and an average of 1,400 amperes. In this particular ship there are four 330-kW. generating sets. In another vessel, having four 250-kW. sets, the average amperage on the outward voyage is 1,350, with 1,700 on the homeward run. The maximum is 2,000 amperes. In yet another comparable twin-screw vessel, which has four 350-kW. auxiliary Diesel generators, the maximum sea load is 650 kW., the average 550 kW. and the minimum —at night—460 kW. In this vessel the maximum engine-room load is 200 kW., with an average of 140 kW. to 150 kW. In all the foregoing examples the voltage is 220. In passenger-carrying vessels the two extreme conditions are the load at midnight—which is a minimum—and the load when manœuvring into port, which is a maximum.

(B) 7,500-S.H.P. SINGLE-SCREW INSTALLATION.

(a) Weights of Machinery.

Main engines and shafting.

Assuming (i), a double-acting two-stroke engine of the type shown in Fig. 1: with 7 cylinders 620 mm. (24·41in) bore, 1,400mm. (55·12in.) stroke; 108 r.p.m.; 5·2 kg./sq. cm. (74·0lb./sq. in.) brake mean effective pressure; 6·5 kg./sq. cm. (92·5lb./sq. in.) mean indicated pressure; 5·04 metres/sec. (992ft./min.) piston speed: bedplate and frames fabricated: the weight of engine, including all customary pipes, gratings, also thrust block = 530 tons. Spare gear = 14 tons. Shafting—say 180 feet from thrust block coupling to tail-shaft thimble-end—together with sterntube, propeller, plummer blocks, bulkhead stuffing boxes, and so on, complete: including spare bronze propeller, tail shaft, and customary details = 129 tons.

Alternatively assuming (ii), a double-acting two-stroke engine of the type shown in Fig. 2: with 7 cylinders, 590 mm. (23·23in.) bore, 1,700 mm. (66·93in.) combined stroke; 108 r.p.m.; 5.2 kg./sq. cm. (74·0lb./sq. in.) brake mean effective pressure; 6·4 kg./sq. cm. (91/0lb./sq. in.) mean indicated pressure; 4.57 metres/sec. (900ft./min.) piston speed; bedplate, frames and entablature fabricated; the weight of engine, including all customary pipes, gratings, also thrust block = 510 tons. Spare gear = 15 tons. Shafting, etc., including spares, all as above-described = 120 tons.

Alternatively assuming (iii), a single-acting two-stroke engine of the type shown in Fig. 3: with 7 cylinders, 750 mm. (29·53in.) bore, 2,000 mm. (78·74in.) combined stroke; 105 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·4 kg. per sq. cm. (91·0lb. per sq. in.) mean indicated pressure; 5·25 metres per sec. (1,033ft. per min.) piston speed; bedplate, frames and entablature fabricated; the weight of engine, including all customary pipes, gratings, also thrust block = 560 tons. Spare gear = 16 tons. Shafting, etc., including spares, as above-described = 131 tons.

Alternatively assuming (iv), a single-acting two-stroke engine of the type shown in Fig. 4; with 10 cylinders 740 mm. (29·13in.) bore, 1,400 mm. (55·12in.) stroke; 108 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·5 kg. per sq. cm. (92·5lb. per sq. in.) mean indicated pressure; 5·04 metres per sec. (992ft. per min.) piston speed; bedplate and frames fabricated: the weight of engine including all customary pipes, gratings, also thrust block = 570 tons. Spare gear = 10 tons. Shafting, etc., including spares, as above-described = 127 tons.

In the engine described at (i) two cylinders fire simultaneously: in the other engines no two cylinders fire together.

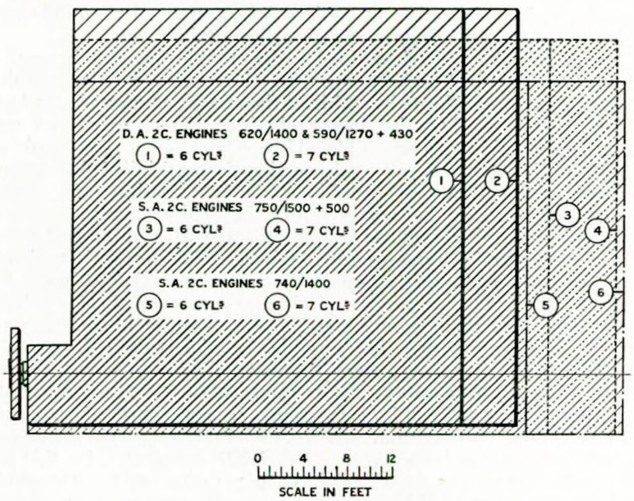


FIG. 5.—*Comparison of engine sizes.*

Fig. 5 shows an outline comparison of the four engine types and Fig. 6 shows an arrangement of the single-acting two-stroke engine described at (iii) above and illustrated in section at Fig. 3.

Diesel generators:	3 off, 300 kW.
Emergency generator:	1 off, 15 kW., 1,000 r.p.m.
Air compressors:	2 off, 100 cu. ft. per min. free air, 450 r.p.m., 40 b.h.p.
Emergency air compressor, steam driven:	1 off, 15 cu. ft. per min. free air.
Air reservoirs and bottles:	2 off, 550 cu. ft. 1 off, 180 litres (6·35 cu. ft.).
Salt-water circulating pumps, main:	2 off, 300 tons per hr., 60ft., 31 b.h.p., 1,250/1,500 r.p.m.
Salt-water circulating pumps, auxy:	2 off, 40 tons per hr., 60ft., 7 b.h.p., 1,600 r.p.m.
Fresh-water cooling pump, main:	1 off, 300 tons per hr., 50 ft., 28 b.h.p., 1,400 r.p.m.
Lubricating-oil pumps:	2 off, 350 tons per hr., 130ft., 110 b.h.p., 1,150/1,400 r.p.m.
Fuel-oil transfer pumps:	2 off, 25 tons per hr., 100ft., 8 b.h.p., 700 r.p.m.
Purified fuel-oil pump:	1 off, 5 tons per hr., 100ft., 3 b.h.p., 1,150 r.p.m.
Domestic fresh-water pump:	1 off, 10 tons per hr., 100ft., 7 b.h.p., 1,250/1,500 r.p.m.
Fresh-water cooler:	1 off, 3,400 sq. ft.
Lubricating-oil coolers:	2 off, 2,000 sq. ft.
Lubricating-oil purifiers:	2 off, 300 galls. per hr., 2 b.h.p., 1,450 r.p.m.
Lubricating-oil filter:	1 off, 350 tons per hr.
Lubricating-oil heater:	1 off, 20 kW.
Fuel-oil purifiers:	2 off, 300 galls. per hr., 2 b.h.p., 1,450 r.p.m.
Fuel-oil filter, main:	1 off, twin, 3 tons per hr.
Evaporator:	1 off, 1,500 galls. per 24 hours.
Distiller:	1 off, 1,500 galls. per 24 hours.
Auxiliary boiler:	1 off, exhaust gas and oil-fired.
Auxiliary boiler feed pump:	2 off, 2,600lb. per hr., 4½in × 3in. × 6in. vertical simplex.
Auxiliary condenser:	1 off, 150 sq. ft. non-vacuum.
Bilge pump:	1 off, 110 tons per hr., 60ft., 15 b.h.p., 1,250/1,500 r.p.m.
Ballast pump:	1 off, 150 tons/40 tons per hr., 60/140ft., 24 b.h.p., 1,300/1,600 r.p.m.
Sanitary pump:	1 off, 40 tons per hr., 100/140ft., 13 b.h.p., 1,400/1,700 r.p.m.
Ventilating fans:	4 off, 32·5in. dia., 15,000 cu. ft. per min., 1in. w.g., 5 b.h.p., 1,130 r.p.m.
Vapour extraction fan:	1 off, 12·5in. dia., 3,500 cu. ft. per min., 3in. w.g., 5 b.h.p., 1,750 r.p.m.

Total weight (including spares) = 202 tons.

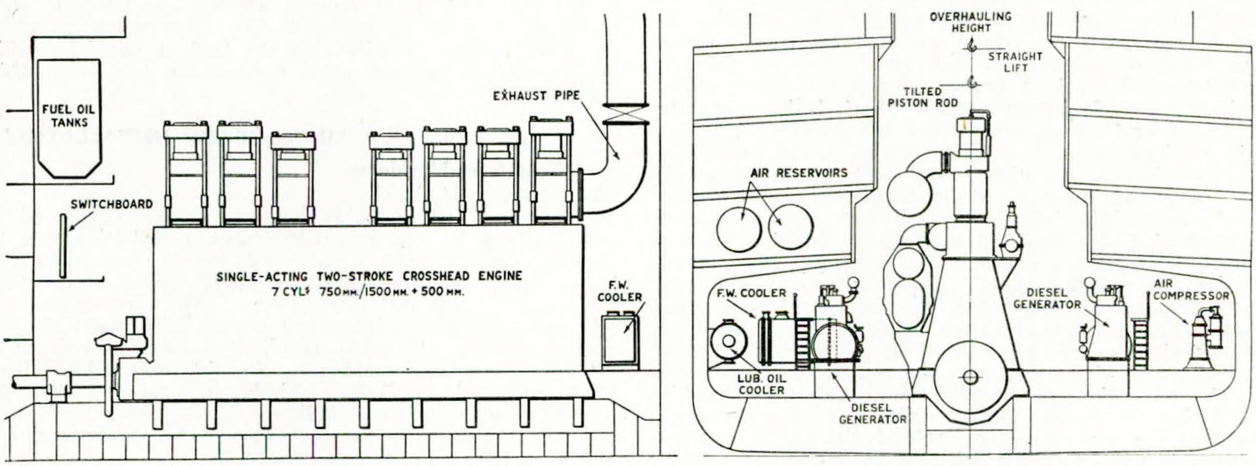


FIG. 6.—*Single-screw ship; single-acting engine.*

Auxiliary machinery.

The following auxiliaries and pumps are assumed—ignoring the effect of slight variations between the engine types. The capacities of pumps and horse-powers of motors which are quoted refer, in each instance, to a single unit:—

Pipe systems, etc.

Aggregate weight (empty) of all piping, fittings, floorings, ladders, gratings, workshop, store-rooms, ventilation system, tanks, funnel, silencers, hangers, clips, spanners, cranes, overhauling gear, etc. = 154 tons.

Water and oil.

Aggregate weight of water and oil in engine, systems, tanks, etc., for the four engine schemes = 36 tons, 40 tons, 40 tons, 35 tons, respectively.

(b) Summary of Weights.

Total weight of machinery, in running condition:—

(i) assuming main engines as in Fig. 1 = 1,065 tons.
(ii) „ „ „ „ „ Fig. 2 = 1,041 tons.
(iii) „ „ „ „ „ Fig. 3 = 1,103 tons.
(iv) „ „ „ „ „ Fig. 4 = 1,098 tons.

(c) Space Occupied.

Length of engine over end frames, double-acting engines of Figs. 1 and 2 = 40ft.

Length of engine over end frames, single-acting engine of Fig. 3 = 49ft.

Length of engine over end frames, single-acting engine of Fig. 4 = 50ft.

Length of engine room, for double-acting engines of Figs. 1 and 2 = 61ft.

Length of engine room, for single-acting engines of Figs. 3 and 4 = 70ft.

The widths of casings for all the engine types = 22ft., 22ft. and 18ft., at main, upper and bridge decks respectively.

The overhauling heights from tanktop to crane hook are summarized below, viz:—

(i) double-acting engine of Fig. 1 = 39ft. assuming piston and rod tilted; = 43ft. for straight lift of piston and rod. In this engine the cylinder cover and top part of liner are removed.

(ii) double-acting engine of Fig. 2 = 45ft. assuming piston and rod tilted; = 49ft. for straight lift of piston and rod. If top part of liner removed, these dimensions become reduced respectively to 39ft. and 43ft.

(iii) single-acting engine of Fig. 3 = 35ft. assuming piston and rod tilted; = 38ft. for straight lift of piston and rod.

(iv) single-acting engine of Fig. 4 = 32ft. for withdrawing liner, or lifting piston and rod, tilted; = 34ft. for straight lift of piston and rod.

(C) 13,000-S.H.P. TWIN-SCREW INSTALLATION.

(a) Weights of Machinery.

Main engines and shafting.

Assuming (i), double-acting two-stroke engines of the type shown in Fig. 1: each with 6 cylinders 620 mm. (24·41in.) bore, 1,400 mm. (55·12in.) stroke; 110 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·5 kg. per sq. cm. (92·5lb. per sq. in.) mean indicated pressure; 5·13 metres per sec. (1,010ft. per min.) piston speed; bedplates and frames fabricated: the weight of two engines, including all customary pipes, gratings, also thrust blocks = 926 tons. Spare gear = 14 tons. Shafting—say 190 feet from thrust block coupling to tail shaft thimble-end—together with sterntubes, propellers, plummer blocks, bulkhead stuffing boxes and so on, complete; including two spare bronze propellers, one tail shaft and customary details = 226 tons.

Alternatively assuming (ii), double-acting two-stroke engines of the type shown in Fig. 2: each with 6 cylinders, 590 mm. (23·23in.) bore, 1,700 mm. (66·93in.) combined stroke; 110 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·4 kg. per sq. cm. (91·0lb. per sq. in.) mean indicated pressure; 4·66 metres per sec. (917ft. per min.) piston speed; bedplates, frames and entablatures fabricated: the weight of two engines including all customary pipes, gratings, also thrust blocks = 894 tons. Spare gear = 15 tons. Shafting, etc., including spares, all as above-described = 211 tons.

Alternatively assuming (iii), single-acting two-stroke engines of the type shown in Fig. 3: each with 6 cylinders, 750 mm. (29·53in.) bore, 2,000 mm. (78·74in.) combined stroke; 106 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·4 kg. per sq. cm. (91·0lb. per sq. in.) mean indicated pressure; 5·30 metres per sec. (1,043ft. per min.) piston speed; bedplates, frames and entablatures fabricated: the weight of two engines, including all customary pipes, gratings, also thrust blocks = 980 tons. Spare gear = 16 tons. Shafting, etc., including spares as above-described = 219 tons.

Alternatively assuming (iv), single-acting two-stroke engines of the type shown in Fig. 4: each with 8 cylinders 740 mm. (29·13in.) bore, 1,400 mm. (55·12in.) stroke; 116 r.p.m.; 5·2 kg. per sq. cm. (74·0lb. per sq. in.) brake mean effective pressure; 6·5 kg. per sq. cm. (92·5lb. per sq. in.) mean indicated pressure; 5·41 metres per sec. (1,066ft. per min.) piston speed; bedplates and frames fabricated: the weight of two engines, including all customary pipes, gratings, also thrust blocks = 932 tons. Spare gear = 10 tons. Shafting, etc., including spares as above-described = 208 tons.

In the engine described at (i) two cylinders fire simultaneously; in the other engines no two cylinders fire together.

Fig. 7 shows an arrangement of the single-acting two-stroke engine described at (iii) above and illustrated in section at Fig. 3.

Auxiliary machinery.

The following auxiliaries and pumps are assumed—ignoring the effect of slight variations between the engine types. The capacities of pumps and horse-powers of motors which are quoted refer, in each instance, to a single unit.

Diesel generators:	4 off, 330 kW.
Emergency generator:	1 off, 25 kW.
Air compressors:	2 off, 250 cu. ft. per min. free air; 100 b.h.p.
Emergency air compressor, steam driven:	1 off, 15 cu. ft. per min. free air.
Air reservoirs and bottles:	2 off, 800 cu. ft. 2 off, 180 litres (6·35 cu. ft.).
Salt-water circulating pumps, main:	3 off, 300 tons per hr., 60ft., 36 b.h.p., 1,200/1,500 r.p.m.
Salt-water circulating pump, auxy:	1 off, 60 tons per hr., 60ft., 8 b.h.p., 1,450/1,800 r.p.m.
Fresh-water cooling pumps, main:	2 off, 280 tons per hr., 50ft., 26 b.h.p., 1,200/1,600 r.p.m.
Fresh-water cooling pump, auxy:	1 off, 50 tons per hr., 50ft., 6 b.h.p., 1,200/1,600 r.p.m.
Lubricating-oil pumps:	3 off, 350 tons per hr., 50lb. per sq. in. head, 110 b.h.p., 1,000/1,400 r.p.m.
Lubricating-oil drain pump:	600 galls. per hr., 80ft. head, 1 b.h.p., 880 r.p.m.
Fuel-oil transfer pumps:	2 off, 40 tons per hr., 100ft., 12 b.h.p., 700 r.p.m.

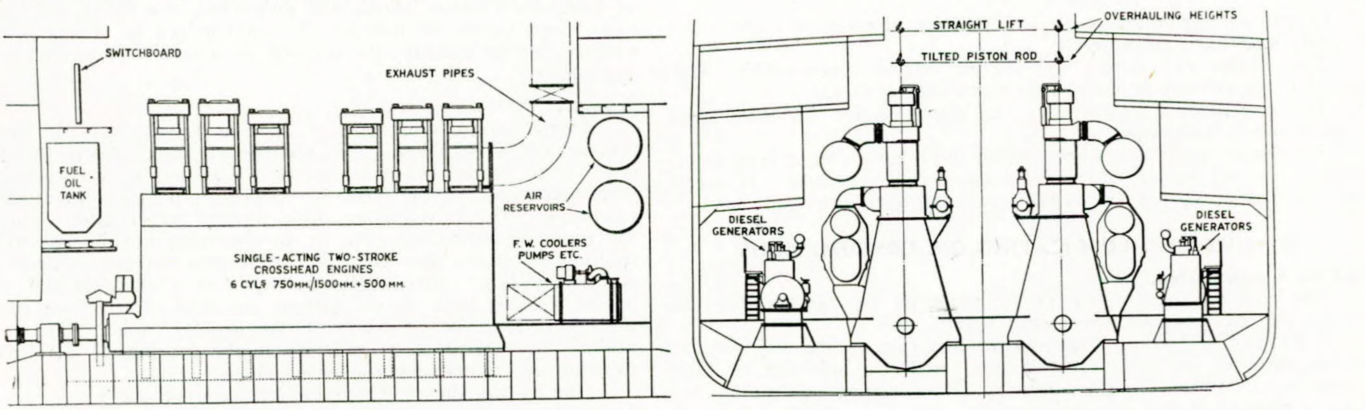


FIG. 7.—*Twin-screw ship; single-acting engines.*

Purified fuel-oil pump:	1 off, 5 tons per hr., 100ft., 3 b.h.p., 1,150 r.p.m.
Domestic fresh-water pump:	1 off, 20 tons per hr., 100ft., 8 b.h.p., 1,450/1,800 r.p.m.
Fresh-water coolers, main:	2 off, 3,000 sq. ft.
Fresh-water cooler, auxy:	1 off, 500 sq. ft.
Lubricating-oil coolers:	2 off, 2,500 sq. ft.
Lubricating-oil filters:	2 off, twin, 280 tons per hr.
Lubricating-oil purifiers:	2 off, 500 galls. per hr., 3 b.h.p., 1,450 r.p.m.
Fuel-oil purifiers:	3 off, 500 galls. per hr., 3 b.h.p., 1,450 r.p.m.
Fuel-oil filters, main:	2 off, twin, 3 tons per hr.
Fuel-oil filters, auxy:	2 off, twin, 700lb. per hr.
Evaporator:	1 off, 25 tons per 24 hours.
Distiller:	1 off, 25 tons per 24 hours.
Auxiliary boiler:	2 off, waste-heat and oil-fired.
Auxiliary boiler feed pumps:	1 off, 6,000lb. per hr. steam, 6in. × 4in. × 7in. simplex.
	1 off, 10,000lb. per hr., 3 b.h.p., 750/1,500 r.p.m.
Auxiliary condenser:	1 off, 150 sq. ft. non-vacuum.
Bilge pumps:	2 off, 120 tons per hr., 60ft., 14 b.h.p., 1,250/1,600 r.p.m.
Ballast pump:	1 off, 200 tons per hr., 60ft., 26 b.h.p., 1,200/1,500 r.p.m.
Sanitary and general service pumps:	2 off, 80 tons per hr., 140ft., 26 b.h.p. 1,200/1,600 r.p.m.
Ventilating fans:	6 off, 15,000 cu. ft. per min., 1in. w.g., 5 b.h.p., 1,000/1,300 r.p.m.
Vapour extraction fans:	2 off, 3,500 cu. ft. per min., 3in. w.g., 5 b.h.p., 1,750 r.p.m.

Total weight (including spares) = 281 tons.

Pipe systems, etc.

Aggregate weight (empty) of all piping, fittings, floorings, ladders, gratings, workshop, store-rooms, ventilation system, tanks, funnel, silencers, hangers, clips, spanners, cranes, overhauling gear, etc. = 244 tons.

Water and oil.

Aggregate weight of water and oil in engine, systems, tanks, etc., for the four engine schemes = 55 tons, 60 tons, 55 tons, 50 tons respectively.

(b) Summary of Weights.

Total weight of machinery, in running condition:—

(i) assuming main engines as in Fig. 1 = 1,746 tons.
(ii) " " " " " Fig. 2 = 1,705 tons.
(iii) " " " " " Fig. 3 = 1,795 tons.
(iv) " " " " " Fig. 4 = 1,725 tons.

(c) Space Occupied.

Length of engine over end frames, double-acting engines of Figs. 1 and 2 = 35ft.

Length of engine over end frames, single-acting engines of Fig. 3 = 43ft.

Length of engine over end frames, single-acting engines of Fig. 4 = 41ft.

Length of engine room for double-acting engines of Figs. 1 and 2 = 65ft.

Length of engine room for single-acting engines of Figs. 3 and 4 = 73ft. and 71ft. respectively.

The widths of casings for all the engine types = 42ft., 32ft. and 23ft. respectively at main, upper and shelter decks.

The overhauling heights are the same as for the 7,500-s.h.p. single-screw installations.

The engine room dimensions, in all the schemes, have been determined with full regard to the desiderata of the engineer. If necessary the engine spaces are capable of compression.

(D) FUEL AND LUBRICATING OIL CONSUMPTIONS.

Fuel Oil Consumption.

The fuel oil consumption is the same, as regards specific rate, for all the main engines considered herein.

For the 7,500-s.h.p. single-screw installation, the main-engine consumption is 30 tons per 24 hours, with the addition of, say, 3 tons per 24 hours for the auxiliary Diesel engines, assuming two engines running at less than full load.

For the 13,000-s.h.p. twin-screw set the main engine consumption is 52 tons per 24 hours with approximately 4·5 tons per 24 hours additional for the auxiliary Diesel engines assuming three engines running at less than full load.

The calorific value assumed is 19,300 B.T.U. per lb. gross, and the figures given are average service figures, not test-bed results.

Lubricating Oil Consumption.

The service lubricating oil consumption, in Diesel machinery of the same type and power, varies more widely from ship to ship than might be expected. Accordingly the figures given are, perhaps, over-generous.

For the 7,500-s.h.p. single-screw set, the main-engine crankcase oil consumption is likely to be about 18 to 24 gallons per 24 hours, with 14 to 18 gallons for the cylinders, i.e. 32 to 42 gallons for the main engine; in addition 8 gallons per 24 hours can be expected to be used at the auxiliary Diesel engines; that is, the total daily consumption = 40 to 50 gallons.

For the 13,000-s.h.p. twin-screw installation, the crankcase oil consumption for the main engines can be expected to be 26 to 32 gallons per 24 hours, with about 24 to 28 gallons for the cylinders, i.e. 50 to 60 gallons for the main engines; and the auxiliary Diesel engines will consume about 12 gallons per day; that is, the total daily consumption = 62 to 72 gallons.

(E) SOME MISCELLANEOUS COMMENTS.

Engine overhaul.

In the years before the war the author used to wrangle with shipowners about the essentiality of ample cranage facilities in the engine room and the avoidance of heavy manual labour; also the need for space, above and about the machinery, for direct lifts. In those days, when negotiating an order, the power crane with its traversing gear was too often jettisoned in favour of something cheaper. Then, there were available a reasonable number of skilled men who possessed a satisfactory degree of knowledge of the various Diesel engine types. Now, the position is not good. At some of the repair ports it is bad. Thus: routine overhauling which took 3 days before the war now occupies 14 days. To quote but one example at random: the recent price of a complete cylinder liner for a direct-coupled engine, of another firm's make, was £240. This was reasonable. The cost of fitting the liner was nearly £800 and the ship was held at the crane berth for three days. This was not reasonable. A well-known shipowner recently said that it was nothing less than a nightmare to have to send a vessel to a repair yard. Although costs were almost unbelievably high the loss of time was an even greater cause of concern. His remarks were general, applying to hulls as well as engines. Most of the company's vessels are steamers.

In planning a machinery room the first essential is to ensure that space is allowed for straight-lifting the engine parts with as little preliminary stripping as possible; the second, that power-operated gear is provided for all the major manipulations.

The Use of Boiler Oil in Diesel Engines.

If present efforts to burn boiler grade fuel in Diesel engines are really successful, trends in marine practice may be deflected.

Having regard to the widespread and increasing preference over a long period for Diesel engines and the inevitable stimulative effect of this upon steam machinery, it is not surprising that there is, at the moment, an appreciable swing towards the use of high-pressure-temperature turbines. Rather is the swing overdue. With the passing of time, for reasons which will duly arise, the swing will again be away from steam machinery. It would indeed be unhealthy if, in a great maritime country, there could be only one form of propelling machinery.

Fatigue.

Every progressive type of marine engine has its distinctive denominational problem. In high-temperature steam installations it is creep; in the Diesel engine it is fatigue. It is surprising how disproportionate is the effect, in service, of insignificant modification to design. The introduction of a slightly larger fillet, or a taper; the rounding-off of a corner, or an alteration in thickness of a few millimetres; these may mean the difference between fatigue failure and non-failure. Just as the strength of a chain is the strength of its weakest link, so the fatigue strength of a Diesel engine is, metaphorically, the strength of its smallest fillet. These simple notions need iteration *ad nauseam* in the hearing of all workers in drawing offices and manufacturing shops.

One of the things which experience over the years has taught the author is the uselessness of relying upon such specimens for relative tests. High-class steels which show markedly superior behaviour in tension, bending, twisting, notch-effect, fatigue strength, and so on, as test pieces, will fail long before mild steel, in an actual engine. Moreover, the fatigue strength of mild steel, as exemplified by failures in the component parts of heavy engines, is much lower than is commonly supposed. It can be only 20 or 30 per cent. of ordinarily accepted values.

Another puzzling problem which is so often bound-up with stress fatigue is corrosion fatigue. When minute relative movement occurs between two steel blocks under high surface pressure there is "fretting", caused by the mechanical tearing of particles from the steel surfaces. These particles rapidly oxidise, under the local heat which is generated, forming a brownish corrosion product; and small corrosion pits are formed on the surfaces. This fretting corrosion action results in reduction of fatigue resistance. Thus, in a screwed member, small corrosion pits form on the threads, and the notch-effect of the pitting is to increase the tendency to fatigue. Another example is a bolted connection which is insufficiently tightened to ensure a frictional load great enough to counteract the tendency for the nut face alternately to dilate and contract under pulsating load. Then the minute relative movements occur.

Cylinder Covers.

In double-acting two-stroke engines of the type shown in Fig. 1, as the result of prolonged sea experience, the bottom exhaust cylinder liner is now cast integrally with the bottom cylinder cover. To provide an effective working surface—the material being chrome-molybdenum cast steel—a cast-iron liner is pressed into the exhaust cylinder. The difficulties which arose in earlier days, when cylinder covers were made of high-alloy chrome cast steel, passed away with the substitution of chrome-molybdenum steel.

The Tightening-up of Piston-Rod Nuts.

Fig. 8 shows the ratchet gear used for tightening-up piston-rod nuts to the prescribed amount. Many engineers have the idea that the flogging-up of heavy nuts by spanner and tup is fully satisfactory, if supervised by an experienced engine fitter. Careful measurements have shown, however, that the tightening-up by one experienced man can be twice as severe as that of another equally skilled man.

Insufficient tightening is a greater fault than overtightening. In new engines, piston-rod nuts are tightened and re-tightened four times—twice on the test bed, twice during installation. This brings stability.

Crankshaft dampers.

In the earliest double-acting and single-acting two-stroke engines, a crankshaft damper was incorporated; it was coupled to the forward end of the crankshaft and its function was to minimise the cumulative effect of torsional oscillations. Essentially it consisted of inertia masses so connected to the shaft that the stiffness or elasticity of the connection, and thereby the contribution of the masses to the natural frequency of the shaft vibration, varied continuously and periodically—between two pre-determined limits—during the rotation of the shaft. By changing the natural frequency, the torsional vibrations never reached their maximum. Repeated tests showed, however, that there were no critical oscillations to be dealt with in any of the engines and accordingly for the last twelve or more years no damping device has been fitted to any two-stroke engine crankshaft. For four-stroke engines the question never arose.

Whitemetal.

Under war-time regulations whitemetal thicknesses were much reduced; 3 mm. was the official figure. This could, and should, have been a good thing. But reduction of thickness was sometimes accompanied by an increase in hardness and the combination was not satisfactory. Bonding was difficult. In service the hard whitemetal cracked and lubricating oil penetrated the bonding. Bearings not a year old could be quickly reduced to flattened pieces the size of a penny.

With the re-establishment of peace-time technique these legacies will come to an end. Thin metal of normal quality, without dovetails, may be satisfactory when bushes are filled by a first-class maker but, as machinery may be repaired anywhere in the world, it is advisable that pre-war metal thicknesses be provided and use made of dovetails to a reasonable extent.

Scavenge Fires.

Scavenge belt fires occur, from time to time, in two-stroke propelling engines of all makes. Normally, they are associated with deterioration of cylinder liners and piston rings. The points to consider are: the piston rings, their type and condition; the cylinder liner wear; the frequency and thoroughness of scavenge belt cleaning; the amount of lubricating oil used; and so on. The contents of the scavenge belt must be kept fluid and regularly drained away, and all ports, ledges and passages cleaned at every opportunity. The essential requirement is a tight piston. Accordingly the proper functioning of piston rings is of high importance.

Conscientious attention to water-jacket temperatures and cylinder lubrication is also necessary. Analysis of available records seems to show that fires tend to be more prevalent in temperate zones where there is high humidity, and when leaving harbour with relatively cold engines; that is, in circumstances where condensed moisture is liable to form on the cylinder walls. All this points to the advisability of maintaining cylinder wall temperatures as high as possible. Thus the fresh-water circulating inlet temperature at the cylinder jackets should be not lower than 120 deg. F.

Excessive back pressure in the exhaust pipe system can be a factor in the causation and frequency of scavenge fires. The cutting of holes through the silencer diaphragms and passages for reducing the pressure-drop, until the exhaust line pressure is below the scavenge air pressure, can be helpful. It is doubtful if, in slow-running marine engines, there is any appreciable pressure-wave phenomenon. A leaking blower change valve might cause higher cylinder temperature, and therefore an increased risk of fire. One is reluctant definitely to link scavenge fires with mean indicated pressures.

As an example of normal experience: a twin-screw installation, comprising 16 cylinders total, of the size and type illustrated in Fig. 1, had its first scavenge fire in one cylinder two years after entering service, after 74,000 sea miles. There was a second fire in the same cylinder a week later, after 2,600 more miles. At the end of the voyage, no broken piston rings were found, but several were gummed-up in their grooves. Six rings were worn and these were renewed, together with two scraper rings. From the time of the second fire until the time of making this record, 131,000 miles were covered without any further fire. The amount of lubricating oil used was 11 per cent. above the builders' recommended quantity.

Piston-Ring Gaps.

The disadvantages of having too great a piston-ring gap are common knowledge; the likely consequences of too small a gap are not so well realized. If, under working temperatures, the piston ring ends butt together there may be: scuffing and heavy initial wear of cylinder liners; increased risk of piston seizures during shop tests and trial runs; piston-ring sticking and fracturing, especially in exhaust pistons; breakage of piston-ring stops; reduction in mechanical efficiency on shop tests; and overloading of turning gear motors.

It is only the rings adjacent to the combustion space which require increased end clearance for circumferential expansion, but all rings must be the same, as it is not practicable to supply rings with varying dimensions of gap.

The extra friction against liner walls, caused by butting rings, will increase the piston-ring temperature and so further aggravate the conditions; the torn-off particles abrade the rings and the liners. It can be assumed that butting rings exert the greatest pressure against the coolest part of the liner wall, i.e. at the bottom of the stroke where the wear is negligible. Engine efficiency on the test bed has been known to suffer when the load has reached 75 per cent. of the designed power.

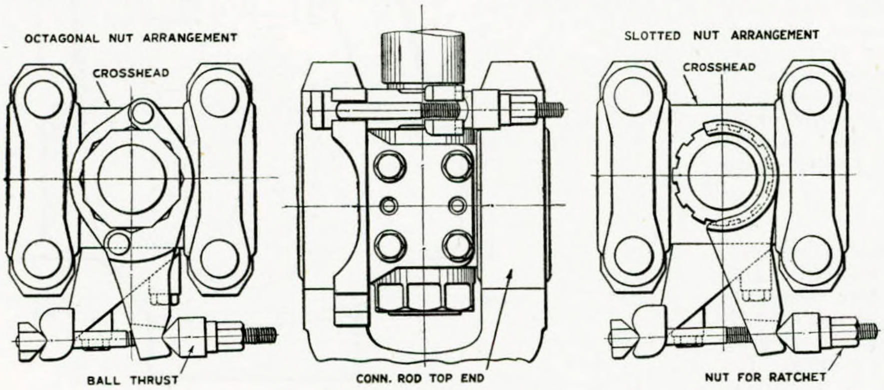


FIG. 8.—*Ratchet for tightening crosshead nuts.*

The effect of end-butting of piston rings can sometimes be seen at the turning gear, where the hot-engine load can be three and four times that of the cold engine. To meet contingencies the gap should allow for a temperature difference of, say, 500 deg. F.

Fuel Consumption.

It is characteristic of the Diesel engine that, over its lifetime, its fuel economy remains constant. In this it is unlike steam machinery. By way of example: in one well-known firm, whose fleet of high-class turbine steamers receives every care in maintenance, whose superintendent is a really able, clear-thinking engineer, there is a steady deterioration in fuel economy of one per cent. per annum. That is, after ten years' running the fuel consumption is 10 per cent. more than when the machinery was new.

The author must, in conclusion, enter a demurrer against the terms of reference of the symposium. While the information given by the various contributors will be very useful to members of the Institute, it would be incorrect to assume that the results as they stand will provide the shipowner with a true weight-and-space comparison of prime movers.

In the author's opinion, a machinery comparison should take into account machinery weights plus weights of seatings plus bunker steel weights plus weight of bunkers for any prescribed length of voyage. This is what the shipowner is compelled to buy, and what he has to carry along in his ship, to ensure its propulsion. These factors operate in favour of the Diesel engine as against steam machinery and in favour of the direct-coupled engine as against the various forms of indirect drive.

In making this demurrer the author realizes that the Committee would, without doubt, consider these various factors in their preliminary deliberations.

(III) The Geared Diesel Engine

By C. C. Pounder (Vice-President).

Synopsis.

After some introductory remarks regarding geared drives, the author describes the characteristics of the magnetic coupling in some detail for the benefit of those engineers who may not be conversant with it. Machinery descriptions, for the single-screw and twin-screw installations, follow. Fuel and lubricating oil consumptions are given; then there are further remarks about magnetic couplings, and a detailed reference is made to a four-engined single-screw installation. A summarized comparison of weight and space particulars for geared and direct-coupled engines ends the paper. Engines designed and built by Messrs. Harland and Wolff, Limited, form the basis of the installations.

There is more scope for variation with geared Diesel engines than there is with direct-coupled engines. Accordingly, for the same horsepower, different men can be expected to choose different engine sizes and different gearing ratios.

For ready comparison, the author has assumed propeller revolutions not dissimilar from those which would be used for direct-coupled engines, namely 100 r.p.m., and has also assumed two different levels of engine revolution, viz. 220 r.p.m. and 300 r.p.m. These revolution rates are sufficiently far apart to enable the effect of other speeds to be assessed fairly clearly.

The two cylinder sizes chosen from the Harland and Wolff range of available engines are: 530 mm. (20·87in.) bore, 820 mm. (32·28in.) main stroke, 360 mm. (14·17in.) exhaust stroke; 370 mm. (14·57in.) bore, 580 mm. (22·83in.) main stroke, 245 mm. (9·65in.) exhaust stroke. Fig. 1 shows a sectional view of an engine having the smaller cylinder dimensions.

The author has assumed that the terms of reference of the symposium require the described installations to be equal to the well-established standards of high-class mercantile practice.

The geared Diesel engine is probably the most promising form of indirect drive, reckoned as an alternative to the direct-coupled engine and considered in terms of weight, space occupied, first cost, and so on. In general terms, the claims made for the geared drive are: (i) increased reliability, by having more than one engine per screw; (ii) some of the engines can be shut-down when the ship is running light or in partly-loaded condition, the others then being operated at their most efficient rating; (iii) maintenance is easier, because the engines are of more manageable size; (iv) engines can be overhauled, seriatim, at sea; (v) by modifying the number of engines per ship and cylinders per engine, the propelling machinery of a fleet of vessels of different sizes and powers can be standardized upon a single cylinder size, with advantage to initial cost, delivery time and replacements. How many of these factors really become operative must depend upon circumstances.

(A)—THE ELASTIC COUPLING.

While geared Diesel engines have been made, on the Continent, with a heavy flywheel as the only damping agent between engine and gearing and while reports received before the war seemed to show satisfactory results therefrom, it is nevertheless the author's opinion that—in the present state of knowledge—an elastic coupling of some kind should be interposed between engine and gearing. The functions of a flexible coupling are: (a) to serve as a cushion, preventing the transmission of detrimental torque pulsations from engine to gearing; (b) to act as a quick-disconnecting clutch—for use when manœuvring; (c) to limit the transmissible torque to say 1·5 to 2·5 times its normal value and thus obtain a measure of protection if there should be a seizure in one of the engines. The forms which such a coupling can take are: (i) mechanical; (ii) hydraulic and (iii) electric. The first-named is the least attractive, and it cannot easily be made to serve as a clutch. Cost, weight, space and reliability are the factors to be considered as between

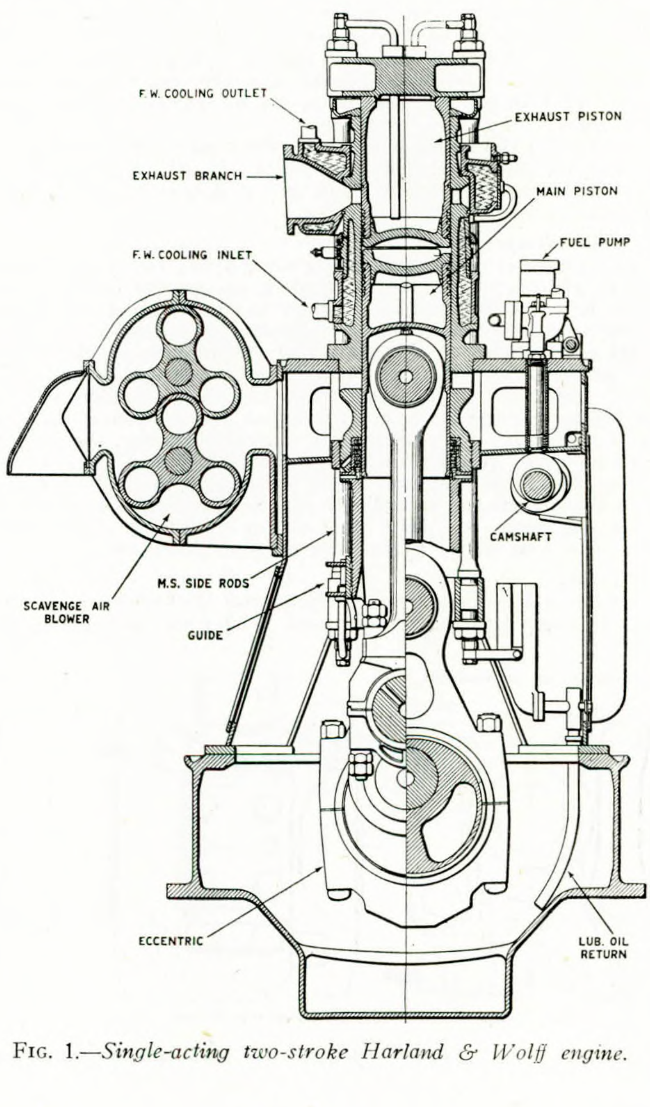


Fig. 1.—*Single-acting two-stroke Harland & Wolff engine.*

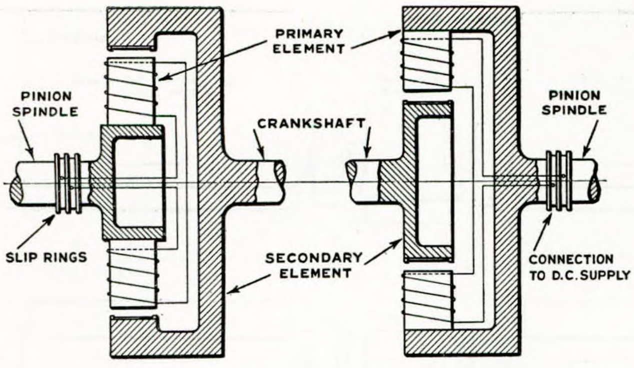


FIG. 2.—*Slip coupling elements; alternative arrangements.*

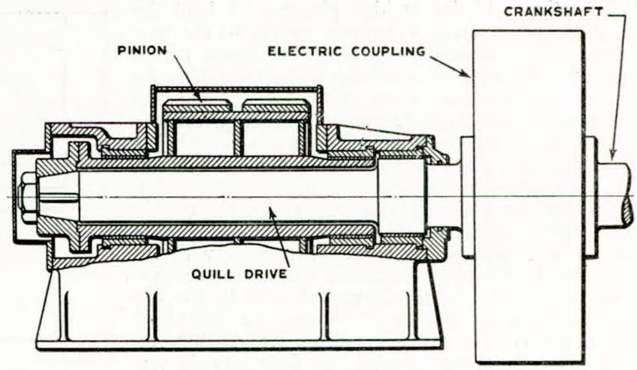


FIG. 4.—*Quill drive for pinions.*

hydraulic and electric air-gap couplings. In this paper the electric coupling is the preferred type.

Although it was first applied to a geared marine Diesel engine set forty years ago—there being two 300-s.h.p. engines coupled to a single shaft, the reduction ratio being 7 to 1—only within the last fifteen years has the electro-magnetic coupling found significant application.

The mechanical power supplied by a prime mover to an electrical generator is converted through the air gap between stator and rotor into electrical power. If the stator, as well as the rotor, is arranged to rotate about its own centre, mechanical instead of electrical power is given out. This is the basic principle of the electric coupling.

The coupling consists of two electromagnets. One, the primary element, is excited from the ship's direct current supply; the other, the secondary, is excited inductively. Fig. 2 shows, diagrammatically, the elements of the coupling, and Fig. 3 a typical assembly. The primary element is a multipolar magnet ring. It has the greater moment of inertia and is attached to the pinion. The secondary element, which is provided with squirrel-cage windings having short-circuiting end rings, is electro-mechanically more robust and can better endure variable torque than can the salient pole windings; it is therefore connected to the Diesel engine. In one form of magnetic coupling the primary element is on the inside and the secondary member on the outside; in another design the primary part is on the outside and the secondary part on the inside. Both forms are indicated in Fig. 2. These variations do not affect the coupling characteristics. The two members, which may be cast steel or fabricated steel, are overhung on their respective shafts and are separated by a radial air gap which may be anything from 5 mm. (0·2in.) to 10 mm. (0·4in.). The shafts, also the slip coupling flanges and the mating flanges are of ample dimensions; the coupling bolts are numerous and sturdy; the flanges are spigoted. British designers, especially, always strive to avoid constructions in which there are heavy, overhanging masses. With the electric coupling it is difficult to avoid overhang, the alternative being very cumbersome indeed. Two-engine arrangements have been made with a quill drive on the pinion shaft, as shown diagrammatically in Fig. 4. The idea is to correct any tendency to distortion of the gearing caused by the overhanging weight of the primary element. Sometimes a flywheel has been fitted to each pinion shaft on the end remote from the coupling.

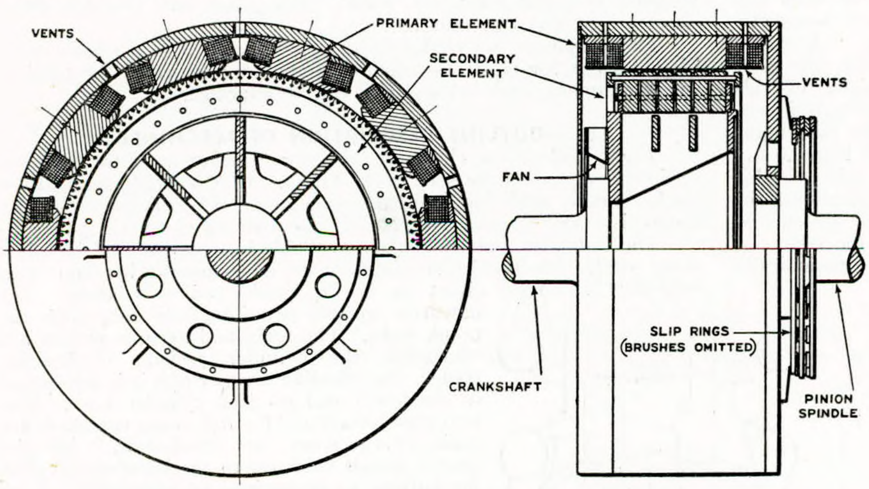


FIG. 3—*Electro magnetic coupling; a typical assembly.*

In Fig. 5, shown later, the overhanging weights of the primary and secondary elements are respectively 8 tons and 6 tons; in Fig. 6, the corresponding weights are 4 tons and 3 tons. The respective outside diameters of the couplings in Figs. 5 and 6 are 8ft. and 6ft.; the lengths 3ft. and 2ft.

The working of the coupling depends upon the principle that, when the primary element is excited, a magnetic field is created; the lines of force pass from the poles across the air gap, cutting the conductors of the secondary element and inducing in them an electromotive force, or voltage, sufficient to drive through the conductor bars and end-rings a current of the amount needed. The interaction of this current and the magnetic field produces a torque, causing rotation of the secondary element. Under the influence of this torque the speed of revolution of the secondary element increases until it reaches a value just sufficiently below the speed of revolution of the primary member to circulate the secondary current required to transmit the torque. The small difference in rotational speed between the two elements—usually about one or two per cent.—is termed the slip; hence the name magnetic slip coupling. The torque transmitted by the coupling is not dependent upon the engine speed, but—apart from the excitation—is determined only by the actual amount of the slip. This is illustrated by curves Y, Y_1, Y_2, Y_3 in Fig. 16. The engine speed changes, but the relationship between torque and actual value of the slip remains the same. Reduction of excitation current is followed by a relative increase in slip and enables the propeller torque and revolutions to be reduced, the engine speed remaining constant. In Fig. 2 the secondary element is driven by the Diesel engine; the primary element therefore has a rotational speed less than the secondary by the amount of the slip. Excitation current is continuously applied to the couplings when the engines are in use and the latter are started, stopped, reversed and controlled exactly as for direct-coupled engines. Expressed another way: during the transmission of the torque a small fraction of the power is lost, as is always so in power transmission. Excitation and induction power losses appear as heat in both elements; there are also iron and windage losses, as well as resistance losses in the conductors. As the torque across the air gap must of necessity be the same as the engine torque the power losses do not appear as a difference in torque; they become apparent as a difference in rotational speed between the primary and secondary elements. The excitation current, which is necessary for creating and maintaining the magnetic field, can be expected to be about one or two per cent. of the transmitted power. In Fig. 16, the curves X and Y are characteristic for two different coupling types.

There is no doubt that the magnetic slip coupling—sometimes called the electric air-gap coupling—while transmitting the mean torque, reduces to negligible proportions the shocks which have their origin in torsional irregularities of the engine; the oscillations on the pinion side correspond in frequency with those on the engine side but have greatly reduced amplitude. Resonance cannot occur at the coupling. Under sudden changes of load the coupling tends to change relatively slowly to the new conditions, thus smoothing-out the acceleration forces engendered. Pitching in a seaway is an example. As the propeller leaves the water the engines accelerate, as in a direct-coupled arrangement, until the governors cut off the fuel. When the propeller descends again the suddenly imposed loads, as the blades meet the water, are not transmitted in their intensity to the crankshafts, thus relieving the shafting system. When the engine is stopped the coupling is still capable of transmitting torque. The transmissible torque increases with increased speed difference between the elements—i.e. with the slip—up to the maximum allowed by the design. This may be 1·5 to 2·5 times the full load torque, at say 6 or 7 per cent. slip. At greater slip the transmitted torque begins to fall. With one engine out of service the excitation current may be reduced and the slip increased, reducing the ship speed until the propeller torque lies within the capacity of the remaining engine or engines. Full torque at low speeds, especially with a coupling member which has a small wr^2 value, may cause the firing impulses to accentuate the turning moment irregularities.

More specifically; in the magnetic slip couplings which are embodied in the schemes which follow, the air gap is 8 mm. (0·32in.); the lowest revolutions per minute at full excitation are 50 per cent. of the normal revolutions. The approximate maximum static torque at full excitation is 1·7 times the full-load torque, thus enabling the engines to be reversed, when running at full speed, with fully-excited couplings, without the coupling halves being stalled. The torque at 100 per cent. slip—i.e. when one element is rotating at full speed while the other is at rest—is 50 per cent. approximately; the slip at normal torque is approximately one per cent. The excitation current at 220 volts is about one per cent. of the main-engine output—which must be allowed for when computing the electrical load on the Diesel generators. The runaway speed is 25 per cent. above the normal designed revolutions. By reason of wear and slight inaccuracies in initial setting, the two elements may not be exactly concentric and thus the air gap will not be uniform. A radial magnetic attractive pull between the elements at the place of minimum air gap is thus created. The maximum value of this force is reached when the excitation current is much below normal and falls well below maximum at full excitation. For the two-engine arrangement, the magnetic unbalanced radial force at 1 mm.—say 0·04in.—is about 4 tons; and rather less than one-half of this amount for the four-engined scheme. Normally the stiffness of the shafts, the adequacy of the bolting and the insignificant amount of wear which occurs at the bearings render negligible the effect of the magnetic pull. The width of the air gap is checked periodically by the insertion of gauges.

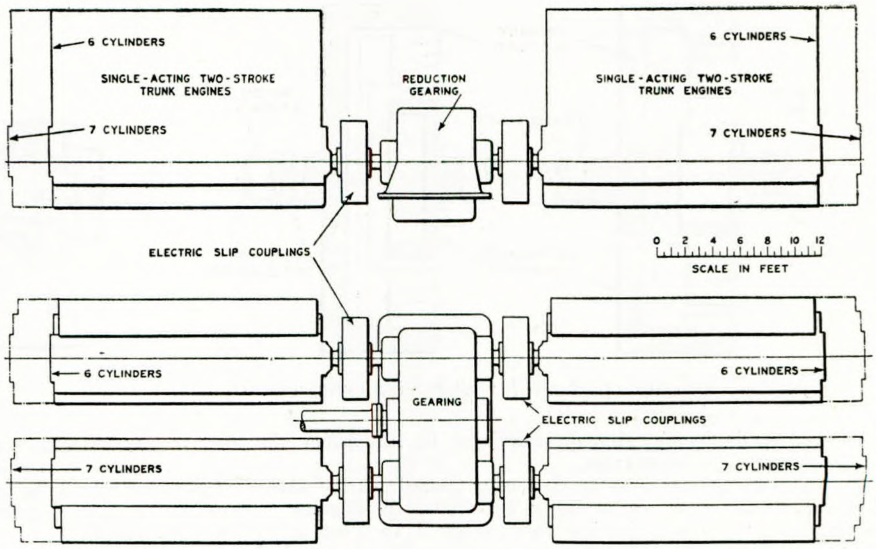


FIG. 6.—*Geared engine set; four reversible engines.*

Teeth, for the Diesel engine turning gear, can be cut on the circumference of innerpole type slip couplings if so desired.

The circuit breakers, resistances, switches, fuses, and other devices are mounted in a suitable cubicle.

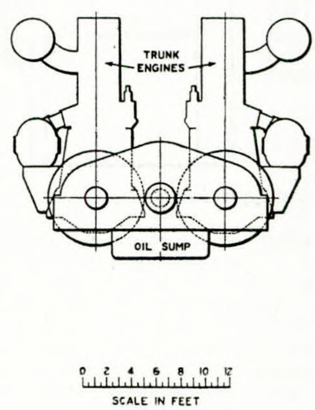
The couplings are air-cooled by a self-acting fan arrangement incorporated in the design. Ducts from the engine-room ventilation system are led to the neighbourhood of the couplings to minimize the drawing-in of oil mist. A wire-meshed guard surrounds the couplings. As about one-half of each coupling is normally below the flooring, splash guards are desirable, although the couplings are designed to expel entrained water. Toe-plates and portable floor plates are provided as required.

The couplings are designed and constructed in accordance with Lloyd's rules for electrical machinery and for service in the tropics, where dampness and condensation can be expected.

(B)—OUTLINE DESCRIPTION OF MACHINERY.

The Diesel engine chosen as the prime mover is of the simplest construction, as will be seen from Fig. 1. It is a single-acting two-stroke engine, with longitudinal scavenging. The place of cylinder covers is taken by exhaust pistons, operated by eccentric gear.

The bedplates and frames are available in cast iron and in fabricated steel. Each exhaust piston yoke is connected by four long mild-steel side-rods direct to the eccentric rod crossheads. The eccentric sheaves are integrally cast with the crank webs. The main and exhaust pistons are oil-cooled, the cylinder jackets fresh-water cooled. An effective scraper box is incorporated in the lower end of each cylinder liner. The scavenge blower and the fuel pump camshaft are chain-driven from the crankshaft. All the engine details conform to the latest practice of the builder, as determined by experience. Each engine has an overspeed governor.



One of the characteristics of this engine type, as demonstrated by experience, is its quietness when running. Accordingly the engine-room noise level will be no greater than for direct-coupled engines. For six-cylinder engines the crank sequence is 1.5.3.6.2.4, with 60 deg. between cranks; for seven-cylinder engines the crank sequence is 1.7.2.5.4.3.6, with 51·43 deg. between cranks. The primary and secondary forces and couples are negligible.

There are two schemes of engine arrangement. The first, for 220 r.p.m., is shown in Fig. 5, where two engines are coupled together. The second, for 300 r.p.m., is outlined in Fig. 6; there are four engines, two for each pinion. With the numbers of cylinders shown, the required propulsion powers are developed at easy ratings.

The gearing, which is single-reduction

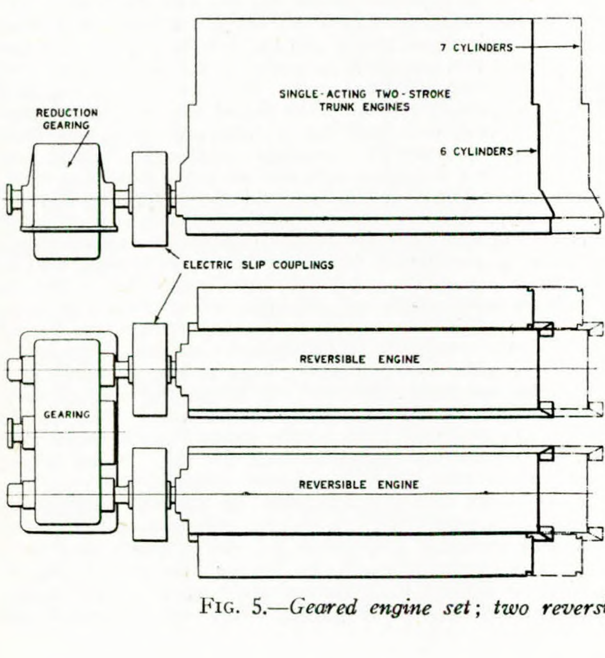


FIG. 5.—*Geared engine set; two reversible engines.*

double-helical, is constructed in accordance with present-day standards for marine work. The proportions and details differ somewhat from those of turbine gearing. The gear case can be made of cast iron or of fabricated steel; it is stoutly ribbed to withstand the propeller thrust, the thrust block being incorporated in the gear case. The main wheel is cast iron or fabricated steel, with a shrunk-on rim of special steel, in which the teeth are cut; the pinions are made of special steel, shrunk-on to the shafts. Plain, or sleeve, bearings are used both for main wheel and for pinions. The gear case joint is arranged horizontally through the centres of main wheel and pinions. Independent lubricating-oil pumps are assumed with oil cooler, strainers, sump tank and so on, but a preferred arrangement is a cogwheel pump driven from the main shaft, with an independent standby pump. Good-class makers agree that the gearing losses are about 1·5 per cent.; that the efficiency of slip couplings, of the sizes being considered, is 98·5 per cent.; that the excitation current to be supplied from auxiliary sources is one per cent. For the purposes of this paper, a combined efficiency of 95 per cent. is assumed. Between engine crankshaft flange and tunnel shafting there is a loss in revolutions and in torque. The slip in revolutions between engine and pinion is taken into account when determining the gear ratio and the loss of torque at the gearing in computing the power. The gain accruing from propeller revolutions at 100 per minute, against 105 to 116 for the direct-coupled engines, is ignored.

The crankshafts and tail shafts are approximately 10 per cent. in excess of Lloyd's rules for strength, the tunnel shafts 5 per cent. The plummer blocks are of self-lubricating type; the stern tube and details are of standard design; the propeller is of manganese bronze, solid form. A spare propeller is carried on deck and a spare tail shaft in the tunnel, at the aft end. Composite type oil-fired and waste-heat boilers—one per engine—are taken into account in compiling the weights. Compressed air, for manœuvring purposes, is stored in welded-steel air reservoirs, and there are air bottles for the emergency starting of the auxiliary engines.

The auxiliary Diesel engines are of the four-stroke single-acting Harland and Wolff type, six cylinders per engine, rated for continuous night and day service. All forces and couples, of primary and secondary orders, are completely balanced within the engine.

In extensiveness and in quality the pipe arrangements, with their associated valves, fittings and other details, conform to all accepted canons of mercantile practice. Flanges are made to B.S.I. marine standards, or their equivalent. The main engine cylinder jackets are fresh-water-circulated from a closed circuit. The main engine forced-lubrication and cooling oil system is common to all the engines; each auxiliary engine has its own oil pump. The machinery spaces are ventilated by air trunks in which mechanical fans are arranged, delivering an air weight considerably in excess of the combustion requirements of the machinery. Tanks, lagging, floor plates, toe-plates, handrails, ladders, platforms, cranes, overhauling gear, spanners, spanner racks, and all other things necessary for the completion of the engine room are taken into account. The funnel and silencers are included in the machinery weight. All items are in general accordance with the practice more fully described in the paper on direct-coupled Diesel engines.

(C)—7,500-S.H.P. SINGLE-SCREW INSTALLATION.

(a) Weight of Machinery.

Main engines and shafting.

Assuming (i), two seven-cylinder single-acting two-stroke trunk engines, of the type shown in Fig. 1, arranged as in Fig. 5, delivering 8,000 b.h.p., total, through magnetic couplings and gearing to the tunnel shafting, the cylinder size being 530 mm. (20·87in.) bore, 1,180 mm. (46·46in.) total stroke; 220 r.p.m.; 4·50 kg. per sq. cm. (64·0lb. per sq. in.) brake mean effective pressure: 6·0 metres per sec. (1,180ft. per min.) piston speed; bedplates, framing and gearcase fabricated: the weight of engines plus electric couplings plus gearing complete, including thrust block, also engine spare gear and one pinion complete with shaft = 377 tons. Shafting at 100 r.p.m.—say 180ft. from gear-shaft coupling to tail-shaft thimble-end—together with stern tube, propeller, plummer blocks, bulkhead stuffing boxes, and so on, complete; including spare bronze propeller, tail-shaft and customary details = 133 tons.

Alternatively, assuming (ii), four seven-cylinder engines, of the type shown in Fig. 1, arranged as in Fig. 6, delivering 8,000 b.h.p. total, through magnetic couplings and gearing to the tunnel shafting, the cylinder size being 370 mm. (14·57in.) bore, 825 mm. (32·48in.) total stroke: 300 r.p.m.; 4·80 kg. per sq. cm. (68·31b. per sq. in.) brake mean effective pressure; 5·80 metres per sec. (1,140ft. per min.) piston speed; bedplates, framing and gearcase fabricated: the weight of engines plus electric couplings plus gearing complete, including thrust block, also engine spare gear and one pinion complete with shaft = 353 tons. Shafting at 100 r.p.m., including spares, all as previously described = 133 tons.

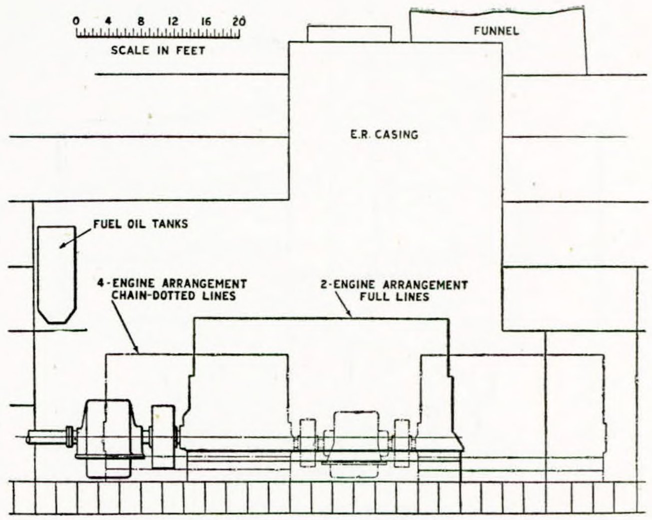


FIG. 7.—*Single-screw ship; two- and four-engine drive.*

The above-mentioned engines have trunk pistons. Many engineers would prefer crosshead-type engines. The effect of such a change would be to increase the total machinery weight of (i) by 25 tons, and (ii) by 24 tons.

Auxiliary machinery.

Assuming equivalent practice to that detailed in the paper on direct-coupled engines: the total weight of auxiliary machinery, including spares = 195 tons.

Pipe systems, etc.

Aggregate weight (empty) of all piping, fittings, floorings, ladders, gratings, workshop, store-rooms, ventilation system, tanks, funnel, silencers, hangers, clips, spanners, cranes, overhauling gear, etc. = 177 tons.

Water and oil.

Aggregate weight of water and oil in engines, systems, tanks, etc. = 40 tons.

(b) Summary of Weights.

Total weight of machinery in running condition:

(i)	two-engine arrangement, trunk type	= 922 tons
(ii)	four- „ „ , „ „	= 898 tons
(iii)	two- „ „ , crosshead type	= 947 tons
(iv)	four- „ „ , „ „	= 922 tons

(c) Space Occupied.

Length of engine, forward end frame to aft face of gearcase for two-engine arrangement = 46ft.

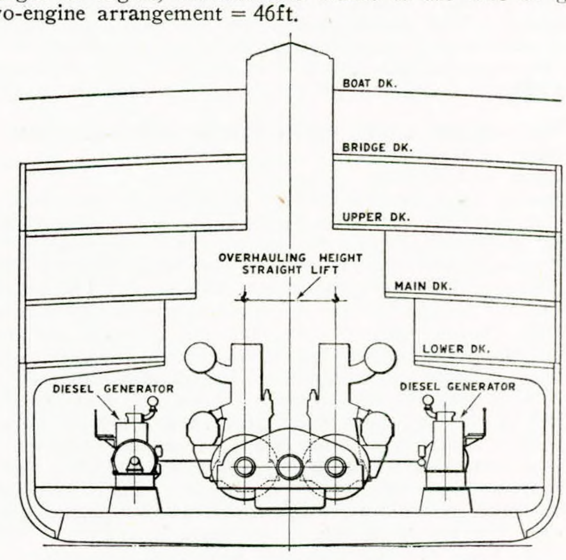


FIG. 8.—*Single-screw ship; two-engine drive.*

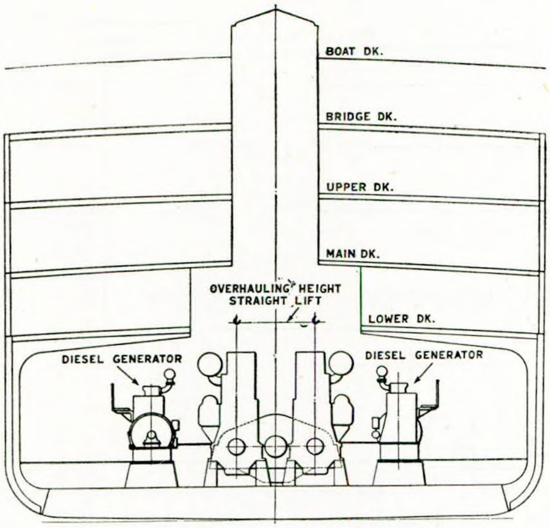


FIG. 9.—*Single-screw ship; four-engine drive.*

Length of engines over end frames, four-engine arrangement = 61ft.

Length of engine room, two-engine arrangement = 63ft.

Length of engine room, four-engine arrangement = 74ft.

Width of casings, two-engine arrangement = 29ft., 22ft., and 10ft. at lower, main and upper decks.

Width of casings, four-engine arrangement = 20ft. and 10ft. at lower and main decks.

Overhauling heights, seating top to crane hook are:

(i) two-engine arrangement, trunk type = 22ft.
(ii) four- „ „ , „ „ = 17ft.
(iii) two- „ „ , crosshead type = 24ft.
(iv) four- „ „ , „ „ = 19ft.

In (iii) and (iv), the overhauling dimensions assume that the pistons and rods are given a straight lift.

Figs. 7, 8 and 9 show typical engine-room arrangements.

As an alternative to the foregoing, it is useful to consider the effect of lowering the propeller revolutions to, say, 85 per minute. The propulsive efficiency at, say, 85 r.p.m. can be greater than that at 108 r.p.m.—the speed of the direct-coupled engines—by 5 per cent. That is: the fuel consumption rates become equal for geared and for direct-coupled machinery, when reckoned in terms of effective horse-power at the propeller. This equality extends from fuel consumption to steaming radius, bunker weight, deadweight carrying capacity, and so on.

The additions to the machinery weight are confined to: (i) the altered gearing ratio, i.e. a larger main wheel and smaller pinions; (ii) heavier tunnel shafting, plummer blocks, tail shaft, stern tube and propeller. The pinion centres are unaltered; the engines are the same. The engine room length, also the casings, remain as before.

The increased weight arising from the described alterations = 19 tons, an addition of about 2 per cent. to the installation weight.

(D)—13,000-S.H.P. TWIN-SCREW INSTALLATION.

(a) Weight of Machinery.

Main engines and shafting.

Assuming, (i): two six-cylinder single-acting two-stroke trunk engines per shaft line, of the type shown in Fig. 1, arranged as in Fig. 5, delivering 14,000 b.h.p. total, through magnetic couplings and gearing to the tunnel shafting, the cylinder size being 530 mm. (20·87in.) bore, 1,180 mm. (46·46in.) total stroke; 220 r.p.m.; 4·60 kg. per sq. cm. (65·4lb. per sq. in.) brake mean effective pressure; 6·0 metres per sec. (1,180ft. per min.) piston speed; bedplates, framing and gearcases fabricated: the weight of two sets of engines plus magnetic couplings plus gearing complete, including thrust blocks, also engine spare gear and one pinion complete with shaft = 672 tons. Two lines of shafting at 100 r.p.m.—say 190ft. from gear shaft coupling to tailshaft thimble-end—together with stern tubes, propellers, plummer blocks, bulkhead stuffing boxes and so on, complete; including two spare bronze propellers, one tailshaft and customary details = 217 tons.

Alternatively, assuming (ii): four six-cylinder engines per shaft line, of the type shown in Fig. 1, arranged as in Fig. 6, delivering 14,000 b.h.p. total, through magnetic couplings and gearing to the tunnel shafting, the cylinder size being 370 mm. (14·57in.) bore,

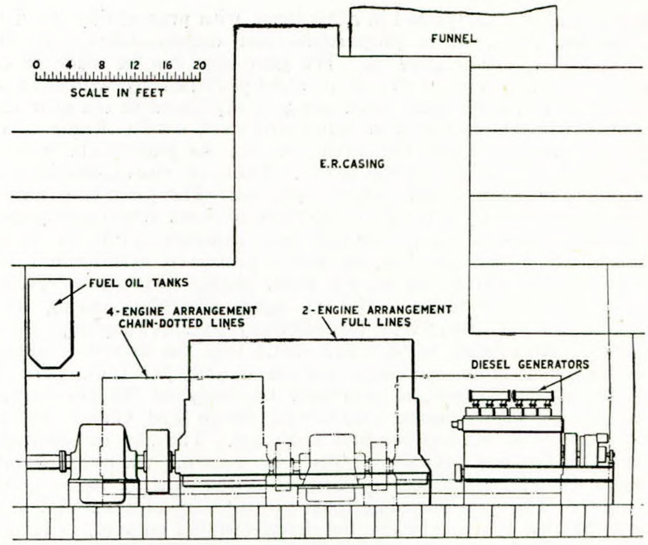


FIG. 10.—*Twin-screw ship; two- and four-engine drive.*

825 mm. (32·48in.) total stroke; 300 r.p.m.; 4·90 kg. per sq. cm. (70lb. per sq. in.) brake mean effective pressure; 5·80 metres per sec. (1,140ft. per min.) piston speed; bedplates, framing and gearcases fabricated: the weight of two sets of engines plus magnetic couplings plus gearing complete, including thrust blocks, also engine spare gear and one pinion complete with shaft = 632 tons. Two lines of shafting at 100 r.p.m., including spares, all as previously described = 217 tons.

The above-mentioned engines have trunk pistons. The effect of a change to the crosshead type of engine would be to increase the total machinery weight of (i) by 43 tons, and (ii) by 41 tons.

Auxiliary machinery.

The total weight of auxiliary machinery, including spares = 240 tons.

Pipe systems, etc.

The aggregate weight (empty) of all items comprised in the pipe arrangements, as previously enumerated = 292 tons.

Water and oil.

Aggregate weight of water and oil in engines, systems, tanks, etc. = 65 tons.

(b) Summary of Weights.

Total weight of machinery in running condition:

(i) two-engine arrangement, trunk type = 1,486 tons
(ii) four- „ „ , „ „ = 1,446 tons
(iii) two- „ „ , crosshead type = 1,529 tons
(iv) four- „ „ , „ „ = 1,487 tons

(c) Space Occupied.

Length of engine, forward end frame to aft face of gearcase,

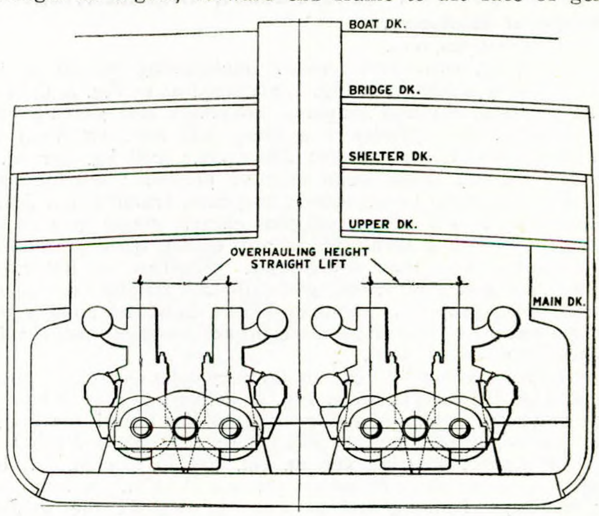


FIG. 11.—*Twin-screw ship; two-engine drive.*

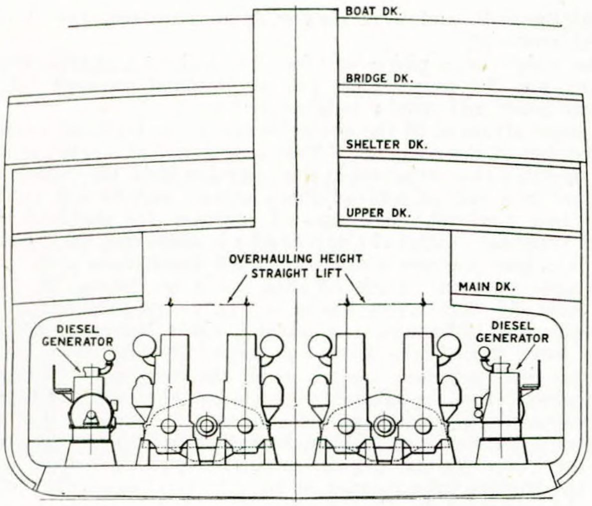


FIG. 12.—*Twin-screw ship; four-engine drive.*

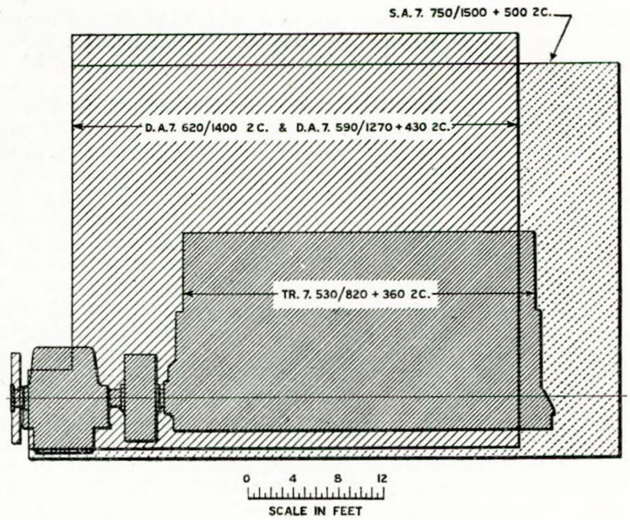


FIG. 13.—*Geared and direct-coupled engines compared.*

for two-engine arrangement = 42ft.

Length of engines over end frames, four-engine arrangement = 56ft.

Length of engine room, two-engine arrangement = 73ft.

Length of engine room, four-engine arrangement = 70ft.

Width of casings, two-engine arrangement = 55ft. and 10ft. at main and upper decks.

Width of casings, four-engine arrangement = 37ft. and 10ft. at main and upper decks.

Overhauling heights, seating top to crane hook are:

(i) two-engine arrangement, trunk type = 22ft.
(ii) four- „ „ , „ „ = 17ft.
(iii) two- „ „ , crosshead type = 24ft.
(iv) four- „ „ , „ „ = 19ft.

In (iii) and (iv) the overhauling dimensions assume that the pistons, with piston rods, are lifted out vertically.

Typical machinery-room arrangements are illustrated in Figs. 10, 11 and 12.

In Fig. 13 the relative sizes of a geared and a direct-coupled engine are shown.

(E)—FUEL AND LUBRICATING OIL CONSUMPTIONS.

Fuel Oil Consumption.

With relatively small, fast-running engines the fuel rate per brake horse-power may be 5 or 6 per cent. more than for large engines. For the engines mentioned in this paper, however, the fuel rate is no higher than for direct-coupled engines. The losses at magnetic couplings and gearing are assessed at 5 per cent.

Assuming a gross calorific value of 19,300 B.T.U. per lb., the all-purposes fuel consumption for the 7,500-s.h.p. installation is 35 tons per 24 hours, with 60 tons for the 13,000-s.h.p. installation. These figures include 3 tons and 4·5 tons respectively for the auxiliary Diesel engines.

Lubricating Oil Consumption.

With trunk-type engines, the total lubricating oil consumption for main engines, gearing, and auxiliary generators can be expected to be 53 to 58 gallons per 24 hours for the 7,500-s.h.p. installation, and 87 to 92 gallons for the 13,000-s.h.p. installation. For crosshead-type engines, the respective amounts will be lower, becoming, say, 45 to 50, and 70 to 75 gallons.

All the amounts are capable of reduction.

(F)—GENERAL.

In the ordinary magnetic coupling the slip serves no purpose other than to generate the low-frequency current which excites the secondary element. Hence, if secondary excitation can be provided otherwise, a magnetic coupling is obtained in which there is no slip. This occurs when exciting current, for both elements, is supplied from an external source. See Fig. 14.

In this diagram the squirrel-cage secondary element is replaced by a second salient-pole magnet system similar to the primary element. Either element may rotate inside the other; or the pole-shoes may face each other. Each is provided with slip-rings for receiving the excitation current. There is the customary air-gap between the elements.

With this arrangement the driven element rotates at the same speed as the driving element; the two members tend to remain in constant, definite angular relationship, according to the torque transmitted. Increase or decrease in torque produces a corresponding angular displacement of the elements. A quill pinion shaft would probably be desirable, in this scheme. At starting and stopping, the pinions would probably be badly hammered.

There is another design of electric coupling in which reversal takes place at the coupling—by reversing the rotating field—the engine itself being non-reversible. This is achieved by using three-phase current, instead of direct current, to excite the primary element. The three-phase frequency is twice that corresponding to the engine speed. The excitation winding of the primary element is suitable both for direct current, when running ahead, and alternating current, when running astern.

This type of coupling is considerably heavier than the normal design and the space occupied is greater. These facts must be set against the savings which accrue from the elimination of engine reversing gear.

Comparisons between hydraulic and electric couplings show hydraulic couplings, with lubricating oil pumps, coolers, seats, tanks, pipes, and fittings, in working condition, to be appreciably heavier than electric couplings complete with their equipment; the cost also is substantially higher. The space occupied is greater to the extent of pumps and other gear, but this is negligible. The hydraulic slip losses, at 3 per cent., can be balanced against the electrical losses plus excitation. The power absorbed by lubricating-oil pumps and cooler circulating can be disregarded.

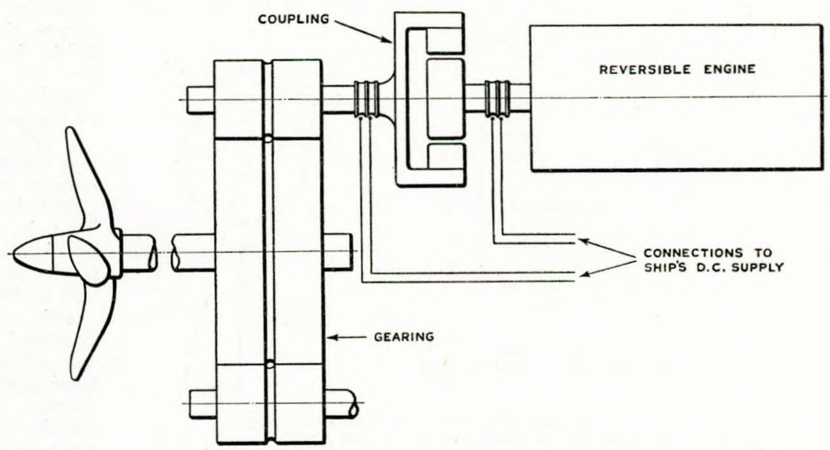


FIG. 14.—*Non-slip magnetic coupling; diagrammatic sketch.*

The largest geared engine set with which, to date, the author has had contact, is briefly described below. It was built in 1939 and the single-screw vessel which it propelled was under the British flag during the war.

There were four 7-cylinder, 520 mm. (20·47in.) bore, 700 mm. (27·56in.) stroke, single-acting two-stroke trunk engines, of cross-scavenge type, running at 237 r.p.m., driving single-reduction gearing through electro-magnetic couplings; and they were designed collectively to deliver 8,900 b.h.p. to the couplings and 8,500 s.h.p. at 85 r.p.m. to the propeller. This was the normal rating, but the engines were to be capable of 10 per cent. continuous overload and 25 per cent. overload for two hours. The engine arrangement was similar to that in Fig. 6, i.e. two engines were coupled to each pinion. The overall dimensions of the assembled engines and gearing were: 61ft. long, 22ft. wide, 13ft. high. The lower half of the gear-case was fabricated steel, with welded-in cast-steel main bearing and thrust-block housings. The fabricated steel gear-case cover was water-jacketed. The vent pipe carried a basket of granulated anhydrous calcium sulphate. This air drier was taken out periodically and roasted at about 500 deg. F. for driving out the moisture.

The gearing was double-helical, the particulars being: (i) main gear wheel: pitch circle dia. 119·60in.; face width 38·25in. total, 36in. nett; diametral pitch 2·5; spiral angle 23 deg.; pressure angle 25 deg. normal; tooth pressure 1,490lb. per inch of face; pitch line speed 2,650ft. per min.: number of teeth 299. (ii) pinions: pitch circle dia. 42·80in.; number of teeth 107. The teeth were of full involute form, hobbed and lapped. The bearing loads are indicated in Fig. 15.

The main wheel was carried by two whitemetal-lined cast-steel bearings, each 21in. dia., 14in. long; and each pinion by two double-row, spherical roller bearings. The gear wheel and pinions, also the bearings and thrust block, were lubricated by controlled splash, the principle being that the gear wheel functioned as an element in a geared pump, lifting oil from the oil pan and splashing it on to the inside of the fresh-water circulated cover, where it was cooled, then caught by troughs and led to reservoirs above the bearings. Gear rim scrapers were provided for use at low speeds.

The electro-magnetic couplings, of outer pole type, were each designed for 2,230 b.h.p. at 240 r.p.m., 120 field volts, 150 amperes normal field current. The steady maximum torque was 1·5 times the normal, with a transient maximum—for suddenly applied loads—of about 2·0. The guarantee for full-load slip was 1·2 per cent. and for full-load efficiency, including excitation, 97·7 per cent. On trials the slip was less than 1·2 per cent. and the efficiency more than 97·7. The outer, or field, elements were secured to the pinion shafts. The spiders were made of fabricated steel plates, heavily ribbed, and the salient poles were steel laminations riveted together and bolted to the inside of the spider rims; the rims were welded to the spiders. The field coils were copper straps wound on edge and insulated with asbestos and mica. The exciting current was brought-in through brushes mounted on the gear case and bronze collector rings mounted on the fields. The inner, or secondary, elements were fastened to the crankshafts and consisted of rigidly-braced fabricated-steel-plate spiders, with cores of one-piece circular punchings shrunk on to the spider rims. The windings were squirrel cages of heavy copper bars driven into slots in the punchings and brazed to short-circuiting rings at the ends, and were generously proportioned for absorbing the heat produced.

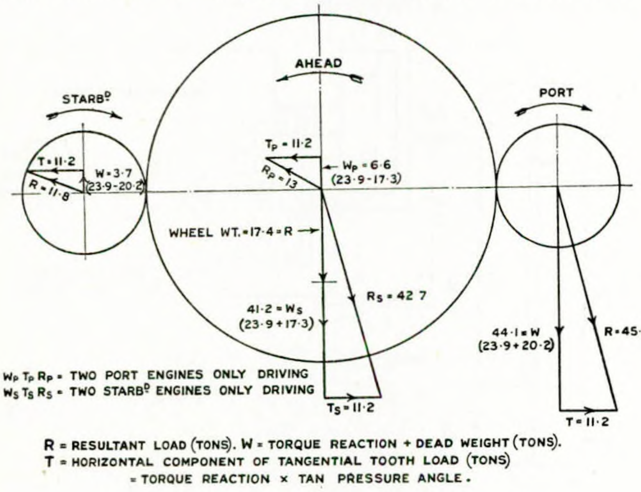


FIG. 15.—*Bearing load diagram.*

The weight of a pinion as lifted, i.e. without magnetic coupling elements, was 9·5 tons. Each primary element weighed 5·2 tons. The gear wheel and spindle weighed 17·5 tons.

A lever arranged at the engine manœuvring platform controlled the operation of the couplings. This lever actuated a selector switch which operated the contactors in the control cubicle, the cubicle being located at one end of the auxiliary power switchboard. At the manœuvring platform, there was an ammeter for the field circuit of each coupling. Interlocks prevented the energising of a coupling when its engine was not running or when the turning gear was in the engaged position. Each coupling could be cut-out by switch. There were six positions of the lever, viz. astern, stop, ahead, stop, half-power and full-power. In starting, either before or after the engines were started, the lever was moved to full-power position. A normal field was thus created in all the couplings. If the field were applied before the engines were started, the propeller began to turn as soon as the engines rotated, the electric couplings thus functioning as solid couplings. When it was desired to try the engines alone, they were started and run with the couplings out of action. While the engines were running at reduced revolutions the propeller could pick-up speed by the coupling control being moved to full power. When the coupling fields were cool they would draw a current somewhat higher than normal but, after a few hours running, the field current approached normal value.

The engines could be reversed in two ways. By one method, the engines were dealt with exactly as if direct-coupled. That is, the magnetic coupling control was left at full-power position, the slip-couplings behaved like solid couplings, and the propeller followed the engine movements. The alternative method was to reduce the engine speed; to disconnect the engines by moving the slip-coupling control to the stop position; to reverse the engines and bring them to not more than 140 r.p.m. in the astern direction; finally to energise the coupling fields by moving the controls to full power, and thus to reverse the propeller.

The second method gave quicker reversal because the starting air had to reverse only the engines and the secondary coupling elements. As soon as the engines were running on fuel again, full torque was available to reverse the propeller. When reversing at full ship speed by this method, the engines were never brought to more than 140 r.p.m. astern before the couplings were energised and given time to reverse the propeller and bring it to approximately 50 r.p.m. The reason was that the propeller, under these conditions, required so much torque at engine full speed that the couplings could not possibly bring it up to this speed, and therefore the propeller would hold the pinion elements to a lower speed, the high slip causing excessive heating of the coupling secondary elements. In making reversals at full ship speed the slip-couplings were carefully watched and at the first signs of distress—such as smoke arising from the paint on the armature bars—the engine speed was reduced to a minimum and held there for a few seconds, to make sure that the couplings reached normal slip.

With the second method, the required maximum torque need not be as high as with the first method, but the starting torque is higher. With full engine speed the starting torque is the same as at 100 per cent. slip; with lower speeds it is greater. With direct reversing, a maximum torque of say 175 per cent. of full-load torque should be assumed, with say 50 per cent. starting torque. With the second method, the maximum torque may be, say, 150 per cent. and the starting torque 75 per cent. See Fig. 16—the slip couplings were designed to curve X. The reversing characteristics of the propeller are the deciding factors.

When manœuvring at low speeds in, say, narrow waters, the two forward engines were run continuously ahead and the two aft engines continuously astern. The propeller was then coupled, as required, to either the ahead-running or the astern-running engines. That is, the selector switch lever was moved first to the ahead position—applying excitation to the forward couplings only—then to the astern position, exciting only the aft couplings. Manœuvring was entirely controlled by the use of the coupling control lever, in ahead, stop, and astern positions. The engine revolutions were arranged to suit coupling control. The energising of couplings in opposition to engine direction was prevented, as was also the overheating of secondary elements. In the half-power position the field current was reduced to about 70 per cent. of the normal value, thereby reducing the heat in the field when the ship was operated for long periods at low speed.

After being two or three years in service, the gearing became so worn that it had to be completely replaced. The engines were sub-

APPENDIX. DIRECT COUPLED AND GEARED DIESEL ENGINE PROPOSALS. SUMMARY OF DATA.

Installation Size	7500 S.H.P. Single Screw						13000 S.H.P. Twin Screw					
Type of Drive	Direct Coupled				Geared		Direct Coupled				Geared	
Number of screws	1	1	1	1	1	1	2	2	2	2	2	2
Engine type	D.A.2C.	D.A.2C.	S.A.2C.	S.A.2C.	S.A.2C.	S.A.2C.	D.A.2C.	D.A.2C.	S.A.2C.	S.A.2C.	S.A.2C.	S.A.2C.
Engines per screw	1	1	1	1	2	4	1	1	1	1	2	4
Cylinders per engine	7	7	7	10	7	7	6	6	6	8	6	6
Bore, m/m. (inches)	620(24·41)	590(23·23)	750(29·53)	740(29·13)	530(20·87)	370(14·57)	620(24·41)	590(23·23)	750(29·53)	740(29·13)	530(20·87)	370(14·57)
Stroke, m/m. (inches)	1400(55·12)	1270(50·00) 430(16·93)	1500(59·06) 500(19·68)	1400(55·12)	820(32·28) 360(14·17)	580(22·83) 245(9·65)	1400(55·12)	1270(50·00) 430(16·93)	1500(59·06) 500(19·68)	1400(55·12)	820(32·28) 360(14·17)	580(22·83) 245(9·65)
Engine, r.p.m.	108	108	105	108	220	300	110	110	106	116	220	300
Propeller, r.p.m.	108	108	105	108	100	100	110	110	106	116	100	100
Piston speed, mean. Metres per sec. (ft. per min.)	5·04(992)	4·57(900)	5·25(1033)	5·04(992)	6·0(1180)	5·8(1140)	5·13(1010)	4·66(917)	5·30(1043)	5·41(1066)	6·0(1180)	5·8(1140)
Brake, mean effective pressure. Kg./sq. cm. (lb./sq. in.)	5·2(74·0)	5·2(74·0)	5·2(74·0)	5·2(74·0)	4·5(64·0)	4·8(68·3)	5·2(74·0)	5·2(74·0)	5·2(74·0)	5·2(74·0)	4·6(65·4)	4·9(70)
Auxiliary machinery, engine room and deck	Electric	Electric	Electric	Electric	Electric	Electric	Electric	Electric	Electric	Electric	Electric	Electric
Fuel consumption, main engine, tons per day	30	30	30	30	32	32	52	52	52	52	55·5	55·5
Fuel consumption, auxiliary engines, tons per day	3	3	3	3	3	3	4·5	4·5	4·5	4·5	4·5	4·5
Total fuel consumption at sea, tons per day (basis 19300 B.T.U. per lb. gross)	33	33	33	33	35	35	56·5	56·5	56·5	56·5	60·0	60·0
Lubricating oil, main engines, galls. per day	38	42	38	32	45	50	55	60	55	50	75	80
Lubricating oil, auxiliary engines, galls. per day	8	8	8	8	8	8	12	12	12	12	12	12
Total lubricating oil at sea, galls. per day	46	50	46	40	53	58	67	72	67	62	87	92
Weight of main engines, gearing and magnetic couplings, thrust blocks and gratings, etc., mounted thereon; including spare gear to Lloyd's schedule, tons	544	525	576	580	377	353	940	909	996	942	672	632
Weight of all shafting, sterntubes, propellers, plummer blocks, etc.; also spares, including bronze propellers, and tail shaft, tons (180 ft. thrust block coupling to tail shaft thimble for 7500 s.h.p.; 190 ft. for 13000 s.h.p.)	129	120	131	127	133	133	226	211	219	208	217	217
Weight of auxiliary diesel engines, pumps, boilers, air reservoirs, etc.; also all spare gear, tons	183	183	183	183	195	195	253	253	253	253	240	240
Weight of all pipe arrangement work, including floors, gratings, ladders, tanks, silencers, funnel, workshop, storerooms, ventilation, etc., tons	173	173	173	173	177	177	272	272	272	272	292	292
Weight of water and oil in engines and circuits, machinery in running condition, tons	36	40	40	35	40	40	55	60	55	50	65	65
Total weight of machinery in running condition, tons	1065	1041	1103	1098	922	898	1746	1705	1795	1725	1486	1446
Length of engine room, feet... ...	61	61	70	70	63	74	65	65	73	71	73	70
Width of casing at successive decks,ft.	22/22/18	22/22/18	22/22/18	22/22/18	29/22/10	20/10/	42/32/23	42/32/23	42/32/23	42/32/23	55/10/	37/10/
Overhauling height, tank top to crane-hook, feet (assuming straight lift for piston plus rod)	43	49	38	34	22	17	43	49	38	34	22	17
Overhauling height, tank top to crane-hook, feet (assuming tilted lift for piston plus rod) ...	39	45	35	32	—	—	39	45	35	32	—	—

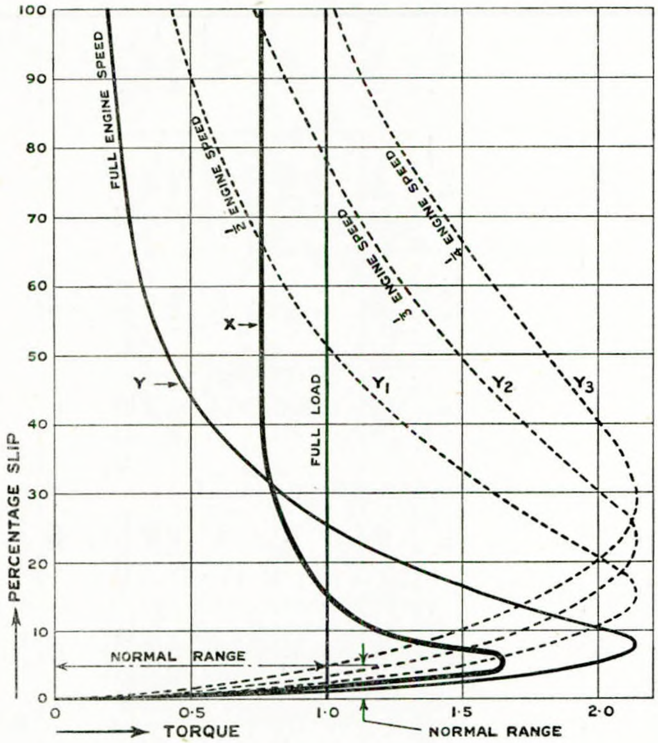


FIG. 16.—*Torque variation diagram.*

sequently run at not more than 7,500 s.h.p. Soon afterwards the author lost touch with the ship, but, so far as he knows, the machinery continued to run well.

(G)—COMPARISON OF GEARED AND DIRECT-COUPLED ENGINES.

For equal powers delivered to the tunnel shafting abaft the engine room, and with the same propeller revolutions, geared Diesel engines can be expected to consume about 5 per cent. more fuel than direct-coupled engines.

The machinery weight is about 18 per cent. in favour of geared engines, taking the geared machinery weight as the basis of reference. Similarly, with the gross volume of the geared engine room as basis, the space saved is 20 per cent. If, in the ships chosen for comparison, the vertical pitch of the decks were slightly different, the saving in space could be even more favourable to the geared engine. Whether or not the 13 per cent. tonnage rule allows full advantage to be taken of this saving must depend upon hull considerations. The relative usefulness of the various spaces saved will differ from ship to ship.

In single-screw ships, the 5 per cent. fuel differential can be offset by an equivalent improvement in the propulsive coefficient, obtainable by lowering the designed propeller revolutions. The increase in machinery weight thus entailed is small, being about 2 per cent.; the engine-room length and size of casings are not affected. The weight of oil bunkers, the bunker steel weight, and so on, then become the same as for a direct-coupled installation, to the direct advantage of the geared machinery. For twin-screw ships this procedure cannot normally be followed, as the available stern-end space does not permit of the propellers being made sufficiently large in diameter. Contrary to what is so frequently stated, the revolutions are seldom a compromise between a slow-running propeller and a faster running Diesel engine; the aft end arrangement is usually the determinant.

As regards costs; comparing a geared engine set with a direct-coupled installation, the pipe systems, auxiliary machinery and shafting are approximately the same; that is, notable differences are confined to the main engines. An analysis of the weights shows that if a geared set is 100 the corresponding direct-coupled engine weight is 134 to 164, with an average of 149. Superficially, it would seem obvious that the geared arrangement is the less costly. Actually, however, the result is not so obvious. Reckoned in pounds sterling per ton of weight, the direct-coupled engine is a cheap machine to produce, and experience shows that, in manufacturing costs, the number of the components can be a more important factor than the weight of the parts. In addition, there is the gear-cutting, the costly electrical work, and so on. On balance, for the powers mentioned and with the schemes described in this paper, the geared engines should cost not more than the direct-coupled engines.

To conclude: for small powers, the first cost of a geared drive is definitely higher than that of a direct-coupled engine and the fuel consumption may be as much as 10 per cent. greater, sometimes even more. For the engine powers with which this symposium deals the position is probably one of equality. For larger powers the first cost could be made favourable to the geared drive, granted suitably chosen engine units.

(IV) Diesel-Electric Propulsion

By J. G. BELSEY, O.B.E., M.A. and J. G. ROBINSON (Member).

Synopsis.

This paper expresses the personal views of the authors in regard to Diesel-electric propulsion.

It outlines the advantages apparent under present conditions, touches generally upon the staff question, operation and maintenance; gives tables of machinery data, weights, dimensions and fuel/lubricating oil consumptions for prime movers of British and Continental design; discusses the main electrical A.C. units, zero voltage switching with synchronized alternators, overall efficiencies, use of A.C. and D.C. driven auxiliaries; and puts forward engine-room layouts with different machinery in two existing hull forms requiring 7,500 and 13,000 s.h.p. respectively.

Since several comprehensive papers dealing with the principles and merits of Diesel-electric propulsion for ships have already been read before the Institute, the authors, who enjoy an unbiased position, have endeavoured to avoid so far as possible a repetition of the main facts well known to marine engineers, and have attempted to put forward other reasons why serious consideration should be given to this type of machinery for the powers stipulated.

The tables included have been prepared for comparison with data applicable to other forms of propulsion and also to give particulars of certain Diesel engines and electrical equipment suitable for the project.

Before dealing with individual points, a brief resumé of the ideas leading up to this proposal may not be out of place.

The direct-coupled slow-speed marine Diesel engine has to date, by its high thermal efficiency, been generally adopted for powers up to about 7,500 b.h.p. per shaft. For powers of the order of those under review, such engines lose some of their appeal on the score that they must be very large and all serviceable parts correspondingly heavy for ships' staff to handle. Where, therefore, fuel economy ranks high in determining the type of machinery required for a ship, the Diesel-electric method of propulsion presents a solution and gives an overall efficiency which, when considered in conjunction with the benefits that can be obtained, makes it an attractive alternative to the large direct-drive unit.

The well-known Ricardo paper covering this subject put forward large numbers of small 1,500-r.p.m. Diesel engines as prime movers, whilst later authors suggested units having rotational speeds of the order of 200/300 r.p.m.

Development of the 300/450 r.p.m. range of engines during the war years just past makes this latest class of prime mover the most attractive for the project. The revolutions are such that the principal mechanical and electrical machines can meet the need economically and, at the same time, no special fuel requirements are necessary; in fact the authors venture to suggest that the time is not far distant when engines of this speed will operate on fuel oils having a viscosity greatly in excess of that now considered the limit for engines of this type and having characteristics very much inferior to those at present found in Diesel oil, and yet will not suffer from the excessive wear and heavy maintenance which has hitherto been associated with the use of heavy fuel oils.

On the electrical side, A.C. three-phase drive is necessary in order to deal with the powers under consideration, and experience has already established that any number of A.C. electrical alternators can be operated in parallel without the need of complex synchronizing gear for manœuvring purposes, whilst the propelling motor itself can be accommodated in any reasonable space and from the maintenance angle almost ignored. The most efficient propeller speed can readily be obtained by determining the number of poles required, and for these two particular ships it is suggested to use 50-cycle equipment at 2,300 volts.

It is the authors' opinion that although an overall s.h.p. loss of up to 10 per cent. is experienced with electric drive, as compared with a loss of 2-4 per cent. in the case of direct-drive machinery located amidships, the advantages of indirect electric propulsion are overriding, and for the larger power where twin propellers would be needed for direct-Diesel drive a gain in propulsive efficiency is assured.

Personnel.

One point which it is felt has not been sufficiently emphasized in the past and which at the present time commands more attention than ever, is the position regarding engine-room staff. Much has been done in recent years to make an engineer officer's life more congenial, but although machinery has been made more reliable, there is, speaking generally, still a wide disparity between the duties to be performed by marine engine-room staffs and their colleagues working ashore as operating and maintenance engineers.

Marine engineers realize that space is a crucial factor aboard ship, and that the physical limitations are such that they cannot expect to have conditions equal to those prevailing in shore establishments, but much could be done to minimize the difference if those concerned with the design and operating of the ships kept this point prominently in mind when deliberating the type and arrangement of machinery most suitable for a particular ship.

The request is for efficient machinery arranged in such a manner that the operators can get to grips with any defective part, without having to disturb other parts having nothing whatever to do with the work on hand.

Proper facilities and tools should be provided which will enable the engine-room personnel to do the major overhauling and servicing of all machinery during the sea passage, thereby affording an opportunity for the majority of the engineers and assistants to go ashore in port, instead of having to turn to and work lengthy periods preparing the ship for the next voyage.

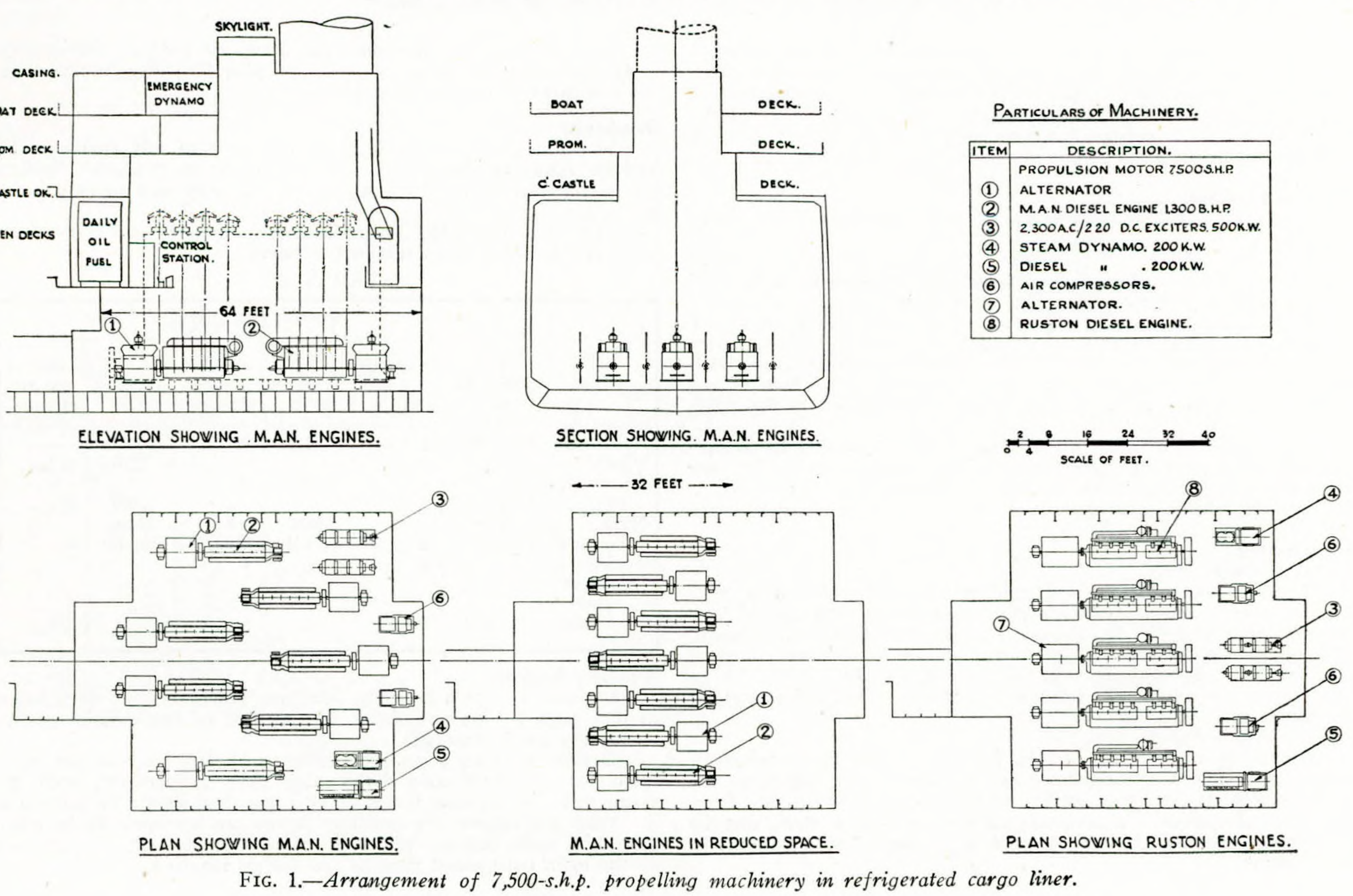
Watchkeepers on ships where more than one per watch is carried, look for interesting work which they can undertake concurrently with their routine duties and if, as does happen on some ships, there is nothing suitable to be found, the hobby of model-making or suchlike is resorted to. If the opportunity and facilities were made available for stripping down and re-erecting one or more of the main units during the sea passage, the same personnel would whole-heartedly set about the task, providing the work involved did not demand a large expenditure of physical energy and the proper tools were available to execute the job in a prompt workmanlike manner.

The shipowner would benefit considerably by the introduction of such routine maintenance, especially if a spare engine was provided, not only from the direct saving on repair costs but from the fact that the work was done by men familiar with the operation of the machinery and, therefore, in the most advantageous position to undertake the overhaul with assurance and in "freight-earning time".

It may be found even more profitable to carry one or more engine-room ratings or senior apprentices who could give full-time assistance to the regular maintenance of the machinery, and so reduce to the minimum the work to be dealt with during the periodical drydockings or refits.

Doubts possibly exist as to whether the marine engineer has sufficient electrical knowledge to enable him to operate this type of machinery with confidence. On this point, the manufacturers of electrical equipment advise that the average engineer is quite capable of effecting any regular electrical repair on the present-day equipment, and experience during recent years is proving this statement to be correct.

When considering the bodily comforts which it is hoped multi-unit propulsion will confer on the sea-going engineer, the time is appropriate to draw the Diesel engine manufacturers' attention to the



ITEM	DESCRIPTION.
	PROPULSION MOTOR 7500S.H.P.
1	ALTERNATOR
2	M.A.N. DIESEL ENGINE 1,300 B.H.P.
3	2,300 A.C./220 D.C. EXCITERS. 500 K.W.
4	STEAM DYNAMO. 200 K.W.
5	DIESEL " . 200 K.W.
6	AIR COMPRESSORS.
7	ALTERNATOR.
8	RUSTON DIESEL ENGINE.

FIG. 1.—*Arrangement of 7,500-s.h.p. propelling machinery in refrigerated cargo liner.*

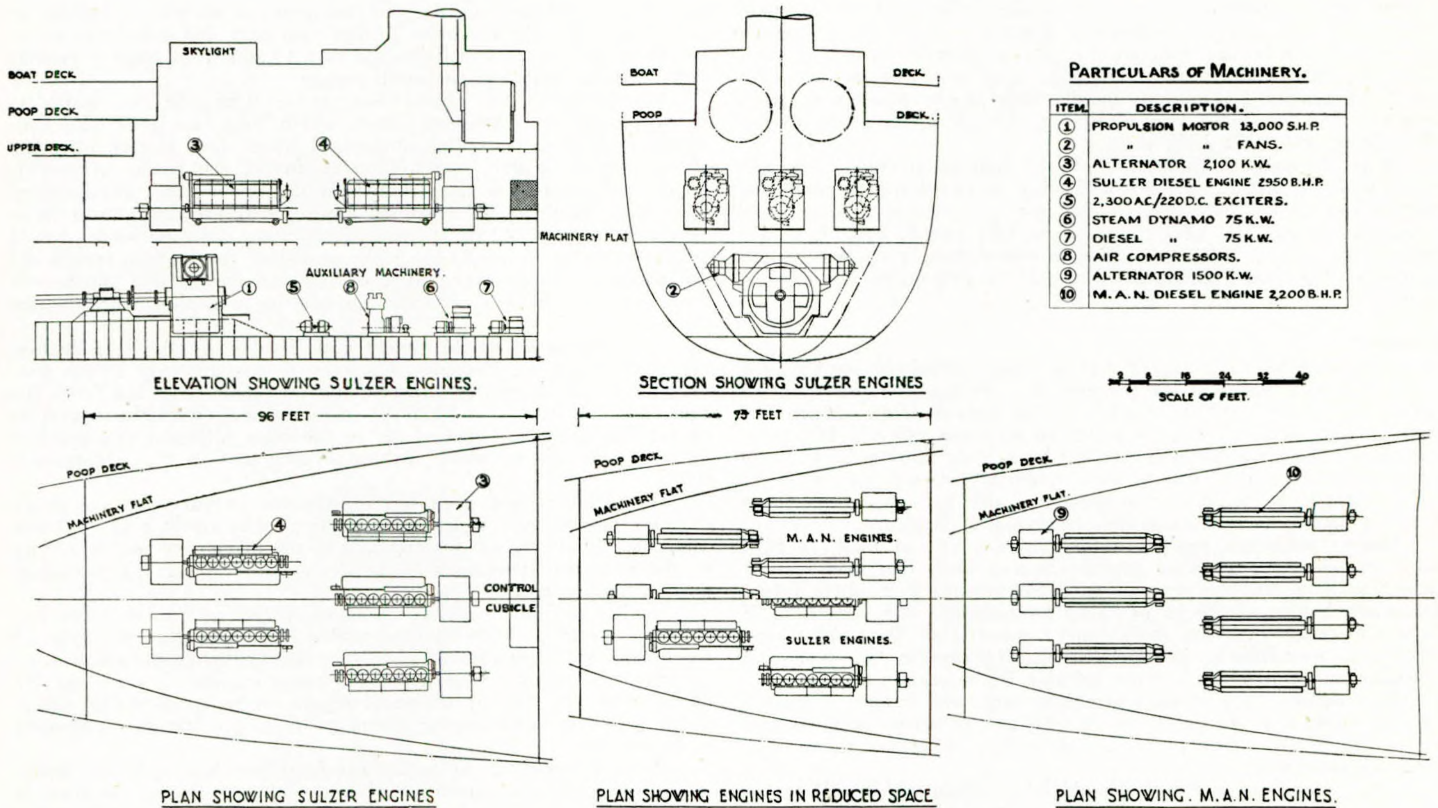


FIG. 2.—*Arrangement of 13,000-s.h.p. propelling machinery in oil tanker.*

question of reduction of noise. The employment of silent camshaft drive, pressure-lubricated well-designed valve gear, enclosed in noise eliminating covers, and where superchargers or scavengers are employed, attention to silencing the air intake and blower itself, would do much to reduce the objectionable noise level at present associated with the class of engines under consideration.

Space and Layout.

Connected with the foregoing is the question of machinery space and layout.

It is the authors' opinion that with Diesel-electric propulsion there are few ships where the machinery spaces have to be reduced to the absolute minimum size, and it is suggested that advantage should be taken to provide maximum space between the units, and adequate lighting and ventilation to allow all work to be done under the most congenial conditions. The benefits resulting from such apparent luxury would be continuous throughout the whole of the ship's life, whether work be done by shore labour or by the ship's personnel.

As a point of interest, the machinery illustrated can be accommodated in a space half the length of that shown, as can be seen from the middle illustration in Fig. 1, although the authors would not recommend this practice.

Maintenance.

Piston Cooling.

Another gain available with medium-size Diesel-electric units is (in the case of certain trunk-piston engines, such as the M.A.N. engine put forward) the absence of piston-cooling arrangements. Efficient as some piston-cooling systems are, they still have attendant troubles, and the majority of engineers would be pleased to be relieved of the continual attention demanded to keep this part of the engine in good order.

Engine Driven Pumps.

Service pumps directly driven from an engine are generally a source of trouble, and for the schemes put forward in this paper the lubricating-oil and cylinder-cooling-water services proposed are from common rail systems, the necessary care being taken in designing the pipe-lines to ensure that the flow is not impeded at any point in the systems.

This arrangement for the lubricating oil permits the quality to be kept the same in all engines, and also simplifies the handling for treatment in the cleansing plant.

Overhauls.

The dimensions and weights of the parts of any engine to be handled influence the quality and amount of work done. Within limits, the smaller the parts the more efficiently and quickly a set job is done.

The following table illustrates the characteristics of the parts applicable to the engines referred to herein.

TABLE 1.

Part	Sulzer.		Ruston Hornsby.		M.A.N.	
	Weight in lb.	Time at sea to change.	Weight in lb.	Time at sea to change.	Weight in lb.	Time at sea to change.
Cyl. cover	990	2 hrs.	1,120	2 hrs.	585 }	—
Piston	1,210	6 ,,	} 1,120	6 ,,	270 }	6 hrs.
Connecting rod ...	1,100	9 ,,			396	3 ,,
Main bearing	Man-handle	2 ,,	Man-handle	2 ,,	Man-handle	3 ,,
Fuel pump	,,	$\frac{3}{4}$,,	,,	1 ,,	,,	1 ,,
Inlet/exh. valve ...	,,		,,	1 ,,	,,	1 ,,
Fuel valve	,,	$\frac{1}{2}$,,	,,	$\frac{1}{2}$,,	,,	1/3 ,,

Auxiliary Engines.

In special ships where the auxiliary power is high in relation to that required for propulsion, this method of transmission shows up to very good advantage.

Usually on such ships the multiple auxiliary engines are over-rated for the continuous duties they have to perform, with the result that they demand far more attention than should be necessary.

With this scheme the auxiliary power can economically be taken from the main busbars whilst the ship is at sea, and in port one of the main units could provide the energy required.

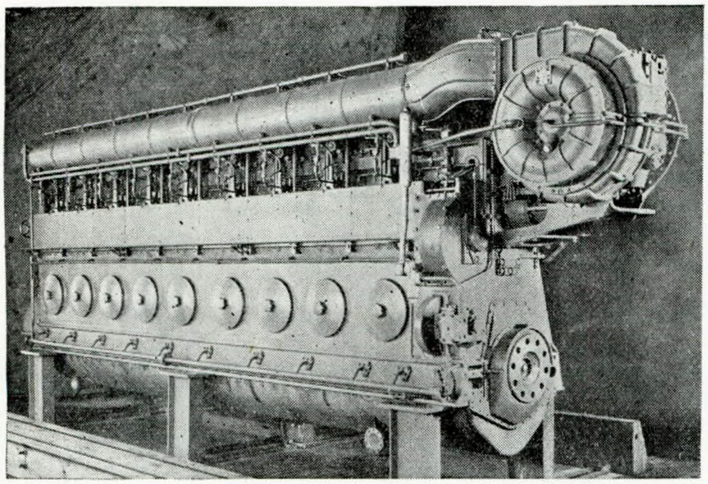


Fig. 3.—*M.A.N. engine.*

The very slight increase in fuel consumption by running one main set at say half load, is more than compensated by the saving on wear and tear of the auxiliary engines.

Further, the number and size of auxiliary engines can be reduced to the absolute minimum, resulting in a saving of capital cost.

Cost.

Under present conditions it has not been possible to obtain accurate costs for comparable installations, but the engine makers indicate that for the smaller power the total cost of the indirect prime movers will be about 70 per cent. of that for a direct-driven Diesel engine of equal power, the electric machinery and propeller shafting excluded. When comparing the costs of direct drive and indirect electric drive, the authors would like to point out that it is apparent that although the equipment cost is higher for indirect electric drive for powers up to about 7,500 s.h.p., above 7,500 s.h.p. the cost becomes progressively lower than for direct drive.

Choice of Prime Movers.

The particulars recorded here by no means cover the whole range of excellent makes of engines at present in production and suitable for this purpose. The particular makes were chosen because they represented three entirely different types:

(a) The 4-S.C.S.A. M.A.N. engine which was designed in Germany in 1937, ostensibly for use in yachts and other peaceful vessels; this engine was developed considerably and used during the recent war for submarine service.

Messrs. Wilton-Fijenoord, the Dutch licensees have, however, fixed the continuous engine rating for Merchant Navy use at only 73 per cent. of the power used for warship purposes.

Incidentally, the submarine engines produced in this country, whilst somewhat heavier, are comparable with this particular engine.

(b) The well-known Sulzer 2-S.C.S.A. engine needs no introduction, and although it has not been shown on the 7,500-s.h.p. installation drawings, three of the seven-cylinder units would provide a satisfactory solution to the project.

(c) The 4-S.C.S.A. Ruston Hornsby engine has gone through much development in recent years, and although it is heavier than the M.A.N. engine the principal dimensions and characteristics are approximately the same.

The decision to put forward two different types of ships as examples, was to give as broad and unbiased a picture of the transmission system as possible, it being considered more appropriate to show the proposed machinery in existing proved hull forms rather than produce hulls around the machinery.

Electrical Propulsion Equipment.

The main alternators chosen would be of the double unit type arranged to comprise two half alternators with independent field and A.C. windings normally operating in series, but the polarity of one half arranged for reversal during the switching for manœuvring, thereby allowing this to be done at substantially no voltage. This scheme for "no voltage" switching is a B.T.-H. Co. patent and allows the alternators to be held in synchronism during switching, since the intermediate busbars or synchronizing busbars are held at normal voltage (see Fig. 4) when the polarity of one half alternator is reversed. This system permits the drawing of the auxiliary power supply from the main propulsion power system and enables it to be maintained even when manœuvring.

The propeller motor is arranged to operate as a synchronous unity power factor motor between half and full speeds, the alternator frequency varying over approximately a 2:1 range. Below half speed the motor runs as an induction motor, a squirrel-cage winding being incorporated for this purpose, with variable slip depending on voltage. Large motors of this type have been in marine operation for several years and have been proved to be absolutely reliable and trouble-free.

The control gear for each alternator would be arranged for lining up as a single panel on the control platform. Each panel would carry an A.C. ammeter, field ammeters, engine tachometer and an overload relay.

The propeller motor control cubicle and lever gear would comprise H.T. reversing switches, field switches and propeller motor protective gear with a set of indicating and alarm instruments forming a meter panel. The authors suggest that the inclusion of an integrating type Wattmeter would be an added luxury in assessing power and enabling the engineers readily to detect engine efficiencies, etc.

A.C. or D.C. Drive for Auxiliaries.

The question of whether to use A.C. tapped from the main alternator or D.C. supply for driving the auxiliaries, both engine room and deck, has been closely considered by the authors, and it has been decided to adopt a 220-volt D.C. auxiliary supply for both the 7,500-s.h.p. refrigerated cargo ship and the 13,000-s.h.p. oil tanker. The main reason for this choice is that the A.C. supply frequency would vary with the engine speed and this, although satisfactory for auxiliaries such as engine circulating pumps, is not ideal for most other auxiliaries and in particular for the refrigeration equipment in the 7,500-s.h.p. ship and the cargo pumps in the

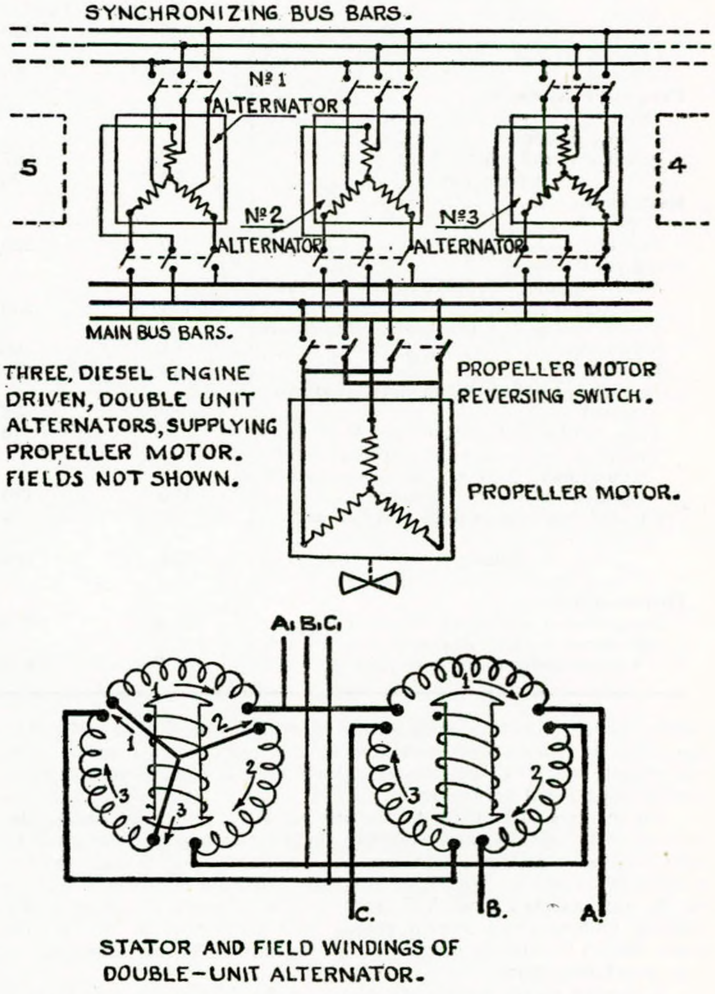


Fig. 4.—*B.T.H. system of A.C. Diesel-electric ship propulsion.*

TABLE 2.

	7,500-s.h.p. Cargo Liner			13,000-s.h.p. Oil Tanker		
Prime Movers:						
Main engines, make and type	M.A.N. 4-S.C.S.A.	Ruston Hornsby 4-S.C.S.A.	Sulzer 2-S.C.S.A.	M.A.N. 4-S.C.S.A.	Ruston Hornsby 4-S.C.S.A.	Sulzer 2-S.C.S.A.
Number of engines	7	5	3	7	7	5
,, cylinders each engine ...	6	7	7	9	8	7
Bore	400 mm.	432 mm.	510 mm.	400 mm.	432 mm.	510 mm.
Stroke	460 mm.	457 mm.	550 mm.	460 mm.	457 mm.	460 mm.
Engine speed, r.p.m.	450	435	340	450	435	340
M.E.P. lb./sq. in.	122	118	74	122	118	74
Piston speed, ft./min.	1,350	1,305	1,225	1,350	1,305	1,225
B.H.P./engine	1,300	1,860	2,975	2,200	2,125	2,975
B.H.P., total	9,100	9,300	8,925	15,400	14,875	14,875
Supercharge/scavenge type	Buchi blower	Buchi blower	Recip. scavenge pump	Buchi blower	Buchi blower	Recip. scavenge pump
Fuel consumption based on 19,300 B.T.U.'s gross/lb.:						
Full load, lb./b.h.p./hour	0·360	0·37	0·370	0·360	0·37	0·370
¾ ,, ,, ,, ,, ...	0·375	0·37	0·375	0·375	0·37	0·375
½ ,, ,, ,, ,, ...	0·395	0·40	0·385	0·395	0·40	0·385
Tons/day for 7,500 s.h.p.	31·8	32·5	32·6	54·8	56·5	56·3
,, ,, ,, auxiliaries at sea ...	Nil	Nil	Nil	Nil	Nil	Nil
Estimated port consumption working cargo	2·0	2·0	2·0	4·5	4·5	4·5
Estimated M.E. lub. oil consumption, gals/day	85	88	145	148	150	240
Estimated aux. lub. oil consumption at sea	Nil	Nil	Nil	Nil	Nil	Nil
Type of engine-room aux. machinery ...	Electric	Electric	Electric	Electric	Electric	Electric
,, ,, deck ,, ,, ...	Electric	Electric	Electric	Electric	Electric	Electric
Type of propulsion	Diesel-electric	Diesel-electric	Diesel-electric	Diesel-electric	Diesel-electric	Diesel-electric
Propeller speed, r.p.m.	110/120	110/120	110/120	110/120	110/120	110/120
Alternators (Direct Coupled):						
Type	Double unit	Double unit	Double unit	Double unit	Double unit	Double unit
Volts	2,300	2,300	2,300	2,300	2,300	2,300
Cooling system	Enclosed circuit with air coolers					
Efficiency	95·5%	96·0%	96·0%	96·0%	96·0%	96·0%
Propeller Motor:						
Type	Silent pole synchronous					
Cooling system	Motor fans with water-tube air coolers					
Efficiency, full load	97·0	97·0	97·0	97·5	97·5	97·5
,, at 90 r.p.m.	96·2	96·2	96·2	97·0	97·0	97·0
Exciters:						
Type	2 Electric motor driven					
Voltage	220	220	220	220	220	220
Weights (in tons):						
Main engines, bedplates, chocks and attached parts	134	233	162	194	339	271
Lloyds spares, C.I. propeller and tailshaft	34	36	35	36	38	37
Alternators	98	90	105	126	128	145
Exciters (2)	14	14	14	16	16	16
Propelling motor intermediate shaft and thrust	87	87	87	124	124	124
Fans, cables and control cubicle ...	15	15	14	22	22	20
Engine-room coolers, pumps, auxiliary generators, boilers, air compressors, uptakes, silencers, funnel	170	170	170	210	210	210
Oil and water in engines and boilers ...	22	24	24	40	42	42
Totals	574	669	611	768	919	865
Dimensions:						
Length of engine room required ...	32′ 0″	32′ 0″	32′ 0″	75′ 0″	75′ 0″	75′ 0″
Maximum height required above tank top for overhauling any part ...	10′ 6″	18′ 0″	13′ 0″	10′ 6″	18′ 0″	13′ 0″

tanker, which require a wide range of speed control that only D.C. can give. It is also considered that marine engineers are much more conversant with D.C. appliances, which give the simplest and most readily understood arrangement.

An existing drawback to the use of A.C. power at sea is the lack of A.C. control and starting equipment specially designed to meet marine conditions, whereas D.C. apparatus of proven quality is already available. In time, no doubt, suitable equipment will be on the market, but land A.C. gear which is perfectly suitable for starting motors from a grid supply may not prove at all suitable when the A.C. supply comes from the more limited capacity of ship generating plant.

Referring to the machinery layout in the 7,500-s.h.p. refrigerated cargo liner, the size of the existing engine room allows the various units to be arranged in a very satisfactory manner, and the object of making this space resemble a shore power station has been substantially realised.

Suitable overhead lifting gear and the most satisfactory type of workshop tools would be supplied to facilitate such work as changing cylinders, pistons or even crankshafts without having to lift the prime movers from their seatings, whilst re-wiring and renewal of parts in the alternators and the propelling motor can be completed in place without difficulty.

In the case of the M.A.N. submarine engines, the entire prime movers, each weighing 17½ tons, could be lifted up bodily through the skylight, if an occasion made this necessary.

With the arrangements shown, various schemes could be devised to legislate for several different conditions. As examples, the main units could without congesting the layout be relocated to permit the propelling motor to be installed in the engine room, two or more of the alternator units could be provided with self-driven pumps and exciters, the latter sufficiently large to produce enough direct current for driving the refrigerating plant or other D.C. equipment when the ship is at sea. Alternatively, the motor driven A.C./D.C. exciters could be arranged with sufficient capacity to take care of all the D.C. loads when at sea, leaving the auxiliary generating sets to do this work when the ship is in port. The emergency dynamo could be used for initial excitation purposes in the event of both auxiliary sets being out of commission.

With the engines referred to, there would be sufficient heat remaining in the exhaust gases to justify the installation of waste-heat/oil-fired boiler equipment.

As a point of interest, the dotted outline of a large direct-drive Diesel engine of similar power has been shown, the weight of which is more than twice the combined weights of the seven M.A.N. engines, alternators, exciters, and propelling motor. This statement does not take into account the saving there would be if the line shafting and bearings were eliminated due to placing the propeller motor aft, nor has allowance been made for the reduced number of auxiliary generating sets. In the case of the 13,000-s.h.p. oil tanker, the five Sulzer 2-S.C. engines give a very spacious layout in the existing engine room, whilst the space between the Ruston and M.A.N. units still allows adequate room for easy maintenance.

As with the cargo liner, the machinery requirements can be varied to suit several conditions. The choice here was to have all ship and engine services operated from 220-volt direct current and to install two motor-driven exciters, the output of each being capable of driving cargo pumps of a capacity of 1,000 tons per hour in addition to essential services when the ship is in port. At sea, each could provide excitation, auxiliary loads and yet have surplus energy to run the cargo pumps at 500 tons per hour without interfering with the speed of the ship.

The auxiliary boilers would be of the waste-heat/fuel type and when at sea running at full power, the heat from the exhaust gases could be used to drive the steam dynamo and thereby improve the overall fuel consumption.

The foregoing is written mainly to show how flexible this form of propulsion is and how for these powers the weight ratio shows up to good advantage.

Acknowledgments.

In conclusion the authors would emphasize that the views expressed herein are personal and in no way reflect the policy of their employers who have been kind enough to grant the facilities for preparing this short paper. It is also desired to thank Mr. John Lamb and the engine and electrical manufacturers for the guidance and data which they have given.

(V) Turbo-Electric Propulsion

By C. Wallace Saunders, D.C.M., A.C.G.I. (Member) and T. Halliday Turner (Member).

Synopsis.

The paper stresses that the information in it is derived from existing ships with this type of machinery, and therefore its contents are based on practice rather than theory; it deals primarily with the steam end rather than the electrical end, because the latter is already in print in the several papers that have been read before this Institute and the Institution of Naval Architects. It deals with the difficulty of utilizing to the best advantage the exhaust steam from boiler feed pumps during "STANDBY" and "SLOW RUNNING"; the efficient use of full steam from the boilers during reversals without detriment to the turbine, there being no astern turbine; the relation of the first critical speed to the maximum running speed, emphasizing very strongly that it should be well above the emergency tripping speed; turbine efficiency ratio; method of stating steam consumptions such that authentic data are obtained for both main turbine and auxiliary power, that is, "PROPELLING ONLY" and "ALL PURPOSES", bringing in evaporators, etc.; trial readings and curves to arrive at correct information in relation to guarantees.

Introduction.

The authors have, in this paper, primarily confined themselves to recounting the performance of machinery of this type in existing vessels; further, they have treated the "TURBO" end rather than the "ELECTRICAL" end of the title for the following reasons:—

(1) The electrical end has been quite elaborately dealt with in theory, detail, and practice in previous papers by one of the present authors and by Mr. Belsey of The British Thomson-Houston Co., Ltd., and its design has practically not altered, except the control gear, which has, in some cases, been simplified to mechanical operation only. This simplified control gear has been fitted to four "Bel" type ships, and has proved itself in the very important feature of reliability.

(2) It was felt that by shelving the electrical end, with its well-known quick manœuvring attributes without detriment to itself, the re-action of this manœuvring on the steam end could be dealt with in much more detail within the limited length allowed for the papers of this symposium.

(3) That all those concerned with propelling machinery would derive from this paper vital information that is the result of actual experience on trials and at sea—information it is so necessary, when planning new tonnage, that the engineers should be able to give to their managements, and which can be verified on British-built ships actually in service.

(4) By means of the details given later in this paper, it is hoped that a very clear, authentic picture has been given on the following points:—

(a) Reliability.
(b) Flexibility.
(c) Weight and space.
(d) Lb. steam per s.h.p./hour, and so, with agreed boiler efficiency, lb. of fuel per s.h.p./hour, and tons of fuel per day.

The authors would like to have gone into greater detail in connection with the most important of all problems for any steam-driven machinery, viz. the supply of pure distilled, or double-distilled, water feed to the boilers, which is so necessary for their low upkeep and life, and which also, in turn, is so necessary for clean steam of the highest quality to be admitted to the turbine; and further, the absolute isolation of any contaminated steam or condensate from the boiler feed system. This would be of too great a length for that permissible for the papers of this symposium, and the authors are of the opinion that it might be the subject of a further paper in the future. In short, it is essential and is quite possible of achievement, that any contaminated steam whatever should have its own pipeline entirely separate from the clean steam line, and the use of a "de-aerator" to get rid of the oxygen and CO_2 is considered as essential because, in this way, boilers can be given the only thing they should ever be given, viz. pure feed water free of oxygen and CO_2.

The authors regret they have not given some of the information asked for in item B of point 6 in the specification issued for guidance to authors, and would comment as follows:—

As the horse powers are fixed, the shafting, stern tubes and bearings, propellers, ladders and gratings, etc., should be common to all papers, and as they are not conversant with these weights, rather than give approximate figures, which might be misleading, they felt they should be left out.

Again, they considered it would not be their problem to decide which type of boilers should be used; therefore they have not dealt with them, or their funnels and uptakes. They hope they have shown, with their consumption figures, that boilers for this type of machinery will be no bigger than those required for any other type of steam-driven machinery, and as engine seatings are not to be included, they feel they will not be at a disadvantage on the point "steaming weight of machinery".

Steam Conditions.

It is unusual for a turbine designer to be offered a free choice of the working conditions for his design, and many of the consequences of the choice are not matters which concern him alone, because the steam comes from boilers which, in turn, have their own problems.

Turbines can be designed with a high efficiency for any steam conditions without impairing the reliability of the turbines, and the design pressure and temperature can be raised to any figures which the boiler makers are able to offer.

The relative reliability of different makes and types of boilers is not the province of the turbine designer to discuss, and nothing in the demands of the turbine for "ahead" propulsion steam influences the reliability of the boilers, though it is a different matter if "astern" running requires the steam to be de-superheated.

The boiler plant, therefore, must have had expert and very careful consideration from the point of view of its reliability, cost of maintenance, and of its efficiency, apart from any question of the type of turbine it is to supply.

If the materials used and the design are both good, turbines, even for the highest steam conditions, have a long life and maintain their efficiency and rate of output with a very little diminution from their original performance over the whole life of the ship. Provided the turbines are given clean steam at the designed pressure and temperature, past performances prove that they will cause little maintenance expense to the shipowner whatever level of pressure and temperature has been chosen.

Variation from the designed pressure, of course, causes a variation in the maximum power obtainable, but it does not impair the turbine efficiency ratio seriously, nor does it influence the reliability of the turbine.

Variation from the designed steam temperature does not vary the maximum power obtainable, and only slightly increases the turbine efficiency ratio with rise of temperature, but unless the design and materials of the turbine are such that sudden changes of temperature (especially excessive rises) can be accepted by the turbine, its reliability will be impaired.

This variation in steam temperature is induced during manœuvring, but it is essentially a boiler problem which will have to be considered in determining the choice of the boiler and on this the turbine designer should not have to adjudicate.

He can only proceed on the basis of being given an ideal boiler, which will always give him clean steam at constant pressure, with only a moderate variation in steam temperature.

He will expect the best boiler efficiency that can be offered, because he will generally be blamed for an excess fuel consumption and, if 87 per cent. boiler efficiency is asked for, he will have to declare at what feed temperature he expects the boiler maker to obtain this efficiency. Raising the feed temperature to increase the thermal efficiency of the turbine will either reduce the thermal efficiency of the boiler by raising the funnel temperature or greatly increase its air heater dimensions, weight and cost.

The choice of feed temperature is therefore another matter over which the turbine designer has not got complete freedom. He is only free to vary the number of stages into which he divides the feed heating, and this has not a large influence on the overall economy.

The relation between inlet steam pressure and inlet steam temperature is, of course, a matter which directly concerns the turbine designer, because a high turbine efficiency and a low steam temperature taken together produce a high wetness in the low-pressure blading of the turbine. Though its erosive effects can be resisted by proper choice of blade materials and by trapping and removing some of the moisture from the steam in the last stages of the turbine, nevertheless, having fixed the initial steam temperature at the highest which is satisfactory for the superheater, there is a limit to the initial steam pressure. Any pressure higher than this requires reheating the steam after partial expansion in the turbine, and this is a further complication in the boiler design. It is clear that it can be done successfully, and the cost of doing it is for the shipowner to examine and reason out with the boiler designers. The turbine designer can listen to this discussion with complete complacency, provided that he is not invited to complicate his problems by adding "astern" turbines to his design.

Although land power stations are running successfully on high steam pressures and temperatures, this does not necessarily indicate that the same steam conditions are desirable at sea. A land power station has the assistance of other power stations to give full power standby in moments of emergency. It does no manœuvring at the end of its full power run, and it can shed load at its own convenience. It certainly never has to put full power on "astern" to save the power station from wreck, and there is little difficulty in giving the boilers perfectly pure feed water.

Yet the majority of power stations, having very much larger generating units than these proposed cargo ships, do not, in general, use steam conditions higher than 600 P.S.I.G., 825 deg. F. total temperature.

Auxiliary Steam and Power.

Power stations do not have such awkward problems with the drive of auxiliary machines as ships, and the engine room of a ship, besides being producer of power, is a very large domestic consumer. The amounts of auxiliary power and of auxiliary exhaust steam are out of all proportion to the amounts required in land power stations, and definitely do affect the choice of steam conditions.

One most important reason for limiting the boiler pressure is the size of the main feed pump. With 500 P.S.I.G. boiler pressure, the steam consumption of a turbo-feed pump is 7 per cent. of the boiler output when the ship is on full power, but the proportion rises to 40 per cent. of the boiler output when the main engine is at standby. At full power it is beginning to interfere with the best thermal efficiency for, although the feed pump exhaust steam can be absorbed in the feed heaters, that prevents bled steam being used for feed heating. The reduction in the thermal efficiency is, of course, due to the steam being used at only 50 per cent. efficiency in the relatively small feed pump turbine, whereas it could have done work in the main turbine at over 75 per cent. efficiency before being used for feed heating, so that it is better to use bled steam for feed heating than to use auxiliary exhaust steam.

Also, an auxiliary steam consumption of 7 per cent. of the feed flow represents a temperature rise in the feed of 75 deg. F., so that the temperature rise due to the feed pump exhaust is more than would be normal in the middle stage of a three-stage feed-heating arrangement. This indicates that the rise in temperature is beginning to raise the feed-pump exhaust pressure excessively to the detriment of the feed-pump turbine consumption.

At "standby" very little of the feed-pump exhaust can be used in the feed heater, and a large quantity of exhaust steam with a high temperature (because of the relatively low efficiency of the small feed-pump turbine) has to be thrown away. It is usually spilled over to the main condenser through a pressure regulating device, so as to limit the excessive rise in feed-pump turbine exhaust pressure. Admitting this high temperature steam to the main condenser, however, besides being a dead loss of heat, causes a gradual rise in the temperature of the L.P. turbine. This can result in leaking casing joints from distortions or, what is much worse, in a bent turbine rotor.

It must be remembered also that unless the glands of a main turbine are sealed, a vacuum cannot be maintained in the condenser, and the atmosphere of hot steam and air in the turbine accelerates corrosion, which can cause a rapid deterioration of the turbine. Unfortunately, the turbine glands cannot be steam sealed unless the turbine is continuously rotated to equalize its expansion. A turbo-electric arrangement allows the turbine to be kept running at a slow speed throughout the standby period and, because of this, any trouble with bent rotors has so far been avoided. Trouble with casing joints has been experienced, and this was inevitable with the temperature reached in the L.P. part of the turbine during long standby periods, but this was minimized by making better arrangements for de-superheating the feed pump exhaust and by improving the bolting and lagging of the casing joints.

There is a great need, therefore, to do something to eliminate the surplus exhaust steam by developing electric drive for the main feed pumps, and this will do much to accelerate progress towards higher overall thermal efficiencies.

This preamble may seem to be far from the purpose of this paper, but it is necessary to be definite that the choice of steam conditions does not rest wholly with the designer of turbo-electric machinery. The same problems, of course, have to be solved for the designer of geared-turbine machinery, and some of the difficulties are much more pressing upon him.

The *paper read in November, 1946 by G. B. Warren before the American Society of Naval Architects and Marine Engineers on the "Development of Steam Turbines for Main Propulsion of High Power Combatant Ships" gives a detailed statement of the considerations governing the application of increased pressure and temperature to geared turbines, and should be studied by those interested in turbo-electric propulsion, as some of the conclusions reached apply equally to geared and turbo-electric machinery.

Astern Power.

Detailed attention is given in Mr. Warren's paper to the high temperatures reached in the ahead blading and in the astern exhaust steam while reversing, and it would not be out of place to state what happens in this respect in a turbo-electric propulsion turbine.

On receiving a reversal order, the engineer at the controls moves

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his speed control to the minimum speed position, and this shuts off all steam to the turbine. He then moves the electrical control to "Stop", cutting off the excitation from the alternator and motor, reducing the power to zero. The turbo-alternator continues to run idle at a high speed, while the direction lever is being put into the "Astern" position, closing the reversing switch. Then, when the excitation is switched on, power is generated by the momentum of the turbo-alternator, but no steam is admitted until the momentum has been absorbed by the turbo-alternator speed having been pulled down to the minimum speed setting on the governor. It will then be held at the minimum governing speed by the governor. In the meantime, the motor will have stopped and reversed, and when the motor has synchronized, he can proceed to bring up its revolutions in the reverse direction on the speed control. All this can be done as quickly as the controls can be manipulated, without any harm to the turbo-alternator.

It must be noted, however, that through all this the turbine is rotating in the same direction, and even if it were stalled by the reversing torque, and full steam flow were continuously admitted by the speed governor, the exhaust steam temperature could not rise above the inlet temperature because no work is being done on the steam, and the turbine is acting in the same way as a reducing valve.

Actually, any stalling period is only measured in seconds because the stalled torque is double the full-load full-speed torque, and the turbine very quickly passes out of the stalled condition. This can be appreciated because the steam is passing through the same highly-efficient ahead turbine.

With an astern turbine, the ahead momentum of the turbine gears, etc. is not absorbed in the propeller, because the propeller persists in trying to run ahead, and the momentum has to be destroyed by admitting steam to blading which is already running in the wrong direction. The ahead momentum therefore does work on the astern steam which, if admitted quickly while the turbine is still running at high speed ahead, will cause a colossal rise in exhaust temperature. The authors have not heard of this temperature being measured accurately, but they have seen various devices calculated to evade its effects, such as providing reduced power astern, arrangements for supplying only de-superheated steam to astern turbines, water sprayers to de-superheat the exhaust steam, and water jackets on the astern turbine casings.

It has been found that seagoing engineers are not much impressed with all this, but achieve safety by taking their time in admitting astern steam, so allowing hull friction to absorb some of the ahead momentum.

Once the astern turbine has been stopped from running ahead, steam can be admitted at any rate of flow that the boilers can give, and the conditions in the astern turbine exhaust are just the same as in the turbo-electric turbine, but the turbine parts have been subjected to a greater range of temperature and will have correspondingly greater difficulty with expansions and distortions due to temperature stresses. Even if the astern turbine is in a separate casing, and is not normally running, it still has to cope with the temperature rise caused by destroying the ahead momentum and has to suffer the consequences of this temperature rise.

The turbo-electric turbine imposes no limitations on the speed of operation of the controls.

Turbine Efficiency Ratio.

On the question of overall turbine efficiency ratio, it is claimed that the rotation losses of an astern turbine running idle in the ahead direction are small. No doubt the astern turbine can be designed to keep the ahead losses small, but there is an unsuspected loss which is of greater magnitude. Having chosen a turbine speed and last stage blade area to get a low exhaust leaving loss, the span of the turbine between bearing centres fixes itself because the critical speed of the turbine must be a safe amount above the maximum running speed, and a 20 per cent. margin is strongly recommended. Into this space between the bearings, both the ahead and astern blading, and an adequate gap between them, has to be arranged. Examination of sectional drawings will show that length sufficient for 50 per cent. more L.P. ahead stages has been used for the astern turbine and exhaust gap. This length could be used for more ahead stages to improve the ahead efficiency. It may be argued that already the ahead turbine has been designed for optimum steam speed/blade speed ratio, and more stages would be beyond the optimum. Perhaps this is so, but the stage diameters at the inlet end of the turbine could have been reduced with a larger number of stages and still made to suit the optimum velocity ratio. This would increase the blade heights of the shorter H.P. blades, and would give a notable improvement in their stage efficiencies, partly due to reduced inter-stage leakage but mostly to the improved steam flow conditions in longer blades.

The electrical losses with turbo-electric propulsion have mostly been exaggerated. They can definitely be taken as :—

(a) Alternator losses 3·0 per cent.
(b) Motor losses 3·0 per cent.
(c) Excitation and fan losses 1·3 per cent.

making in all 7·5 per cent. transmission loss. These losses can be determined precisely in the Works' tests.

Propulsion gear losses are often quoted as being only 2 per cent., but adding up the bearing losses of the gears, which can be calculated from the drawings, makes one doubt this figure, particularly for double-reduction arrangements, and the authors have never seen any tests published of the losses having been measured. They can easily be determined by metering the lubricating oil flow and measuring its temperature rise, but allowance has to be made for the large amount of heat usually radiated from the gear case by lagging it during the test. The radiation losses, of course, would not otherwise appear in the temperature rise of the oil and, if not accounted for, would credit the gears with too high an efficiency.

Chosen Steam Conditions.

From the foregoing conclusions, and because it seems to be agreed that at the present time boiler steam conditions of 500lb. per sq. in. pressure and 800 deg. F. total temperature will ensure perfectly reliable boilers, and as these conditions with powers so low as 7,500 s.h.p. represent nearly the optimum overall thermal efficiency of boilers, turbines and auxiliaries, with little to be gained unless re-heating is adopted, the authors have chosen to give an example of the steam consumption which may be expected with these boiler conditions.

The feed temperature at normal full power, which seems to be acceptable to the boiler designer for these steam conditions, is about 300 deg. F. Any higher feed temperature with some boiler designs would require air heaters as big and as heavy as the boilers if a boiler efficiency of 87 per cent. is to be achieved.

Turbine Efficiency Ratio.

The choice of turbine stages depends upon the heat drop and vacuum, but for boiler steam conditions of 500lb. P.S.I.G. 800 deg. F. there is no difficulty in arranging the turbine stages in a single casing without excessive leaving losses and with reasonably high efficiency, which has been proved in practice to be comparable with geared-turbine installations.

For steam conditions higher than these, it might be profitable to divide the turbine into two casings, and it is preferred to direct-couple them in tandem.

The electrical reduction ratio with 120 r.p.m. on the propeller shaft can easily be made 26-1 without having an excessively large diameter propulsion motor, and this gives a turbine speed of 3,120 r.p.m., which is not too low to get good overall turbine efficiency for these powers of 7,500 s.h.p./13,000 s.h.p.

Method of Stating Steam Consumptions.

The method of stating consumption figures has never been rationalized, and it is suggested that this is a matter to which the Institute should give some attention, perhaps on the lines of the British Standard Test Code 752, which was prepared for land turbines.

When examining reports and log books of existing ships, it is found that the results are given as fuel consumptions, generally stated for all purposes and for propulsion only. It is impossible to make any useful analysis of the turbine performance from these figures, as the efficiency of the turbine can only be examined in terms of its own steam consumption. Usually there are no data from which to determine the boiler efficiency, and until this is known it is impossible to deduce from the fuel consumption the steam/oil ratio and arrive at the steam consumption.

In a well-designed ship great efforts are made to use all surplus auxiliary steam to the best advantage. If not contaminated, it is mixed up with the main flow of steam in the main turbine and its feed heaters.

It is quite useless, therefore, to refer the steam consumption to the "Not feed heating" condition, because this convention becomes worthless, just as nowadays no one talks of boiler evaporation as "from and at 212 deg. F."

Some of the auxiliary steam cannot be separated out from the main turbine consumption, but this is of no real consequence if the consumption for propulsion only is defined as consisting only of the steam used in the main turbine, including its bled steam, plus the steam used in the air ejector, plus the steam used by the feed pump in pumping this propulsion condensate back to the boilers.

Auxiliary Power.

To find the all-purposes consumption, when we have Diesel-driven generators, it is only necessary to add their fuel consumption directly to the main propulsion fuel consumption. If they are steam driven, however, it is not so simple, because the condensate from the auxiliary turbo-generators will go back to the main boilers and be heated in the main feed heaters by extra steam bled from the main turbine, and it will also require additional power for the main feed pump.

It is possible, however, to construct a graph from which to read the increase of high-pressure steam consumption against the auxiliary turbo-generator load for different main propulsion powers, making accurate allowance for the extra steam then required by the main turbine and by the main feed pump on account of the heating and pumping of the auxiliary condensate.

Evaporated Make-up.

It is necessary also to make allowance for the make-up feed. The make-up rate is an unpredictable amount, much of it outside the control of the engineer. For instance, the authors have known the captain to blow the whistle so vigorously and continously that he blew away 12 tons of steam in one watch.

The engineer can measure the amount of make-up and should "log" it, but besides the steam used for distilling it, there needs to be added the extra bled steam for heating the distillate, and the extra feed pump steam for pumping it back into the boilers.

Again, a graph can be constructed to make the proper allowance in terms of high-pressure steam taken from the boilers, plotted against shaft horse power load.

Auxiliary Steam.

In some ships so much low-pressure steam is needed that special boilers have to be installed. If these are oil-fired, then a simple addition of their oil fuel consumption can be made for the "all purposes" fuel consumption if the condensate returns direct to the auxiliary boiler and does not enter the main feed system.

If the auxiliary boilers are operated by high-pressure steam taken from the main boilers, and the condensed heating steam is taken back to the main boilers with the main feed, then allowance has to be made for the extra steam for heating it, or may be for the reduction of H.P. steam, because of the heat taken from these hot drains in the feed heater. To this has to be added the extra feed pump steam for pumping it back to the boilers.

This is, of course, on exactly the same lines as for the evaporator, and it can be plotted similarly.

It must not be thought that these corrections are of little consequence. They may be large and on an oil tanker the demand for reduced-pressure steam when heating and pumping cargo can be as much as one-third of the maximum duty of the main boilers,

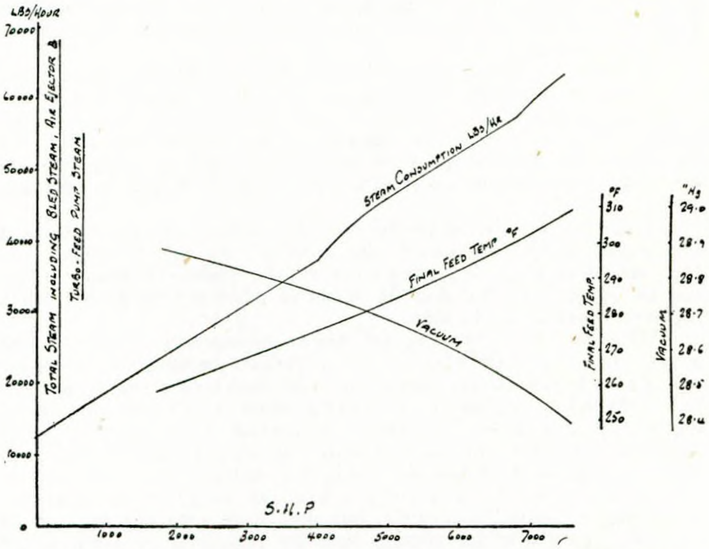


FIG. 1.—*7,500-s.h.p. turbo-alternator. Normal steam conditions:—415lb. per sq. in. g., 725 deg. F. t.t., 73 deg. F. water, 30in. barometer. Guarantee steam consumption curves.*

TABLE 1. TRIAL READINGS.

MAIN TURBINE GAUGE BOARD.

Time	Main steam press., P.S.I.G.	Main steam temp., °F.	1st throttle press., P.S.I.G.	2nd throttle press., P.S.I.G.	Abs. vac. inches, Hg.	H.P. B.S. press., P.S.I.G.	I.P. B.S. press., P.S.I.G.	L.P. B.S. vac., inches	Feed pump exhaust press., P.S.I.G.	Final feed temp., °F.
6·00	400	709	340	370	0·90	56	16·5	7	17·25	296
7·00	405	703	340	370	0·99	55	15·5	6·5	16	302
8·00	405	696	340	375	0·99	57	16·75	6	17·25	301
9·00	405	702	340	375	0·95	57	16·75	6	17·25	297
Mean	404	702·5	340	372·5	0·958	56·2	16·4	6·4	17·1	299
10·00	395	690	320	355	0·92	53	18	8·25	18·25	305
11·00	410	696	355	380	1·00	59	20·5	7	21·25	305
12·00	425	708	370	399	0·98	62	22	6·5	22·5	303
1·00	425	701	370	399	1·01	62	22·5	6	23	307
Mean	413·75	699	354	383	0·98	59	20·75	6·9	21·25	305
Speed Trials at Arran: Displacement 10,855 tons. Propeller mean pitch 14·07 ft. Full power 121·1 r.p.m. 16·691 knots + 1·54% slip. 7,646 s.h.p.										
Mean	424·2	717·5	365	345	1·05	73	22·25	1·56	23·0	311·25
Low power 110·18 r.p.m. 15·46 knots — 0·7% slip. 5,565 s.h.p.										
Mean	414·2	728	355	226·2	1·045	46·5	13·5	7·9	13·2	294

which accounts for the large difference between propulsion oil and all-purposes oil on tankers.

Trials of 7,500-S.H.P. Turbo-Electric Ship.

Having now defined the method of stating the steam consumption, we can proceed to give some figures obtained on the trials of a turbo-electric ship of 7,500 s.h.p. maximum power, 6,800 s.h.p. normal full power.

It is always to be realized that the maximum power at the given r.p.m. is continuously available year in and year out throughout the life of the ship without detriment to the machinery and still complying with Lloyd's Regulations. In effect, this means that the maximum power stated can be taken as the service power, the turbine being designed for it and the temperature rises on the electrical end being still within Lloyd's Rules and the guarantees.

These trial figures can be analysed with reference to the steam consumption curves previously described, and which are:—

Fig. 1 (T.2724) which gives the total steam consumption

TABLE 2.

CONTROL INSTRUMENTS. TORSION METER CONSTANT 1·45.

Time	Propeller r.p.m.	S.h.p. indicator	S.h.p. integrator	Torsion meter		Propulsion meters				Mean S.h.p.	Alternator excitation		Motor excitation	
				Reading	S.h.p.	Volts	Amps.	KW.	S.h.p.		Volts	Amps.	Volts	Amps.
6·00	113·3	6097	6550	80·2	6260	2857	987	4878	6327	—	77	171	219	270
7·00	114·7	6347	6570	81·7	6460	2853	1003	4956	6428	—	76·3	175	219	270
8·00	115·1	6343	6590	80·5	6400	2863	990	4909	6367	—	78·1	180	220	270
9·00	114·6	6290	6260	80·0	6320	2863	973	4825	6258	—	78	181	220	270
Mean	114·4	6269	6492	80·6	6380	2859	988	4892	6345	6357	77·2	177	219·5	270
10·00	113·8	6300	6960	85·5	6700	2950	970	4956	6428	—	78	181	220	270
11·00	115·4	6820	6820	83	6610	2920	1050	5310	6888	—	79	180	220	270
12·00	117·5	6800	6890	84·5	6850	2930	1030	5227	6780	—	79	182	222	270
1·00	116·5	7000	7180	85·7	6900	2920	1100	5563	7215	—	78	180	221	270
Mean	115·8	6718	6962	84·7	6765	2942	1037·5	5263	6827	6818	78·5	180·8	220·8	270
Speed Trials:														
Full Power	121·1	7703	—	91·4	7708	2988	1121	5802	7526	7646	83·4	187	219	270
Low Power	110·25	5659	—	73·1	5558	2800	870	4219	5478	5565	79·8	183	219	270

TABLE 3.

AUXILIARY POWERS. (AMPS. AT MOTOR STARTERS.)

Time	Circ. pump	Extraction pumps		Lubn. pump		Motor vent. fan		Forced draught		180-kW. Diesel engines			400 kW. T.G.	Total load	Volts	KW.
		For'd	Aft	Port	St'rb'd	For'd	Aft	Port	St'rb'd	Port	Centre	St'rb'd				
6·00	250	—	40	20	—	56	57	49	45	—	—	—	1125	1125	220	—
7·00	250	—	39	20	—	56	57	39	32	—	—	—	1095	1095	220	—
8·00	260	—	38	21	—	58	59	39	33	—	—	—	1105	1105	220	—
9·00	260	—	38	20	—	56	57	53	32	—	—	—	1090	1090	220	—
Mean	255	—	38·75	20·25	—	56·5	57·5	45	35·5	—	—	—	1104	1104	220	243
10·00	260	—	39	20	—	58	58	40	29	—	—	—	1010	1010	220	—
11·00	260	—	39	20	—	58	56	43	30	—	—	—	1100	1100	222	—
12·00	260	—	39	20	—	58	58	53	35	—	—	—	1100	1100	224	—
1·00	260	—	39	20	—	58	57	52	35	—	—	—	1200	1200	223	—
Mean	260	—	39	20	—	58	57·2	47	32·2	—	—	—	1102	1102	222·8	245
Speed Trials:																
Full Power	250	—	39	19·5	—	56·75	58	59·25	58	—	—	—	1242·5	1242·5	220	273
Low Power	250	—	38	20	—	57	58	53	58	—	—	—	1246	1246	220	274

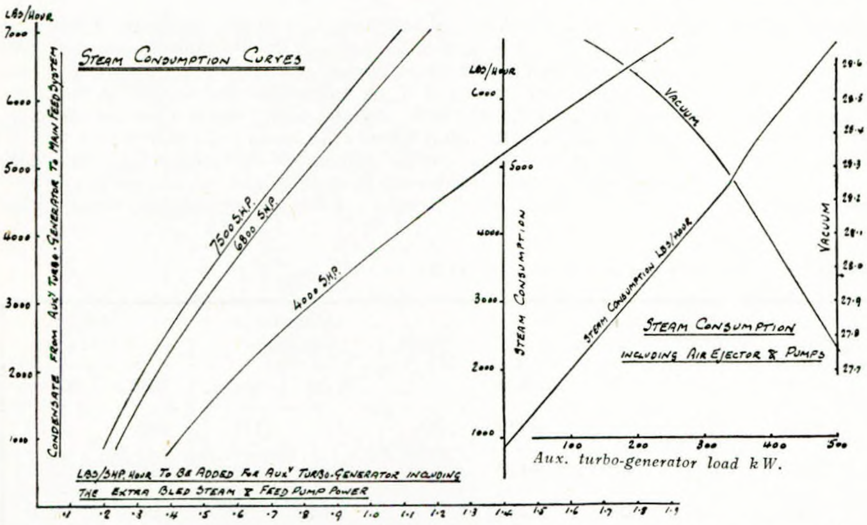


FIG. 2.—*400-kW. turbo-generator. Normal steam conditions:—415lb. per sq. in. g., 725 deg. F. t.t., 73 deg. F. water, 30in. barometer.*

for the main turbine, and bled steam for feed heating its own condensate, together with air ejector steam, and feed pump steam for pumping this quantity of feed back to the boilers. The curves also indicate the design vacuum and feed temperatures.

Fig. 2 (T.2725) the steam consumption of the 400-kW. turbo-generator, plus extra high-pressure steam to the main turbine for feed heating it, plus extra feed pump steam to pump the extra feed back to the boilers, and these curves also indicate the design vacuum.

Fig. 3 (T.2726) the steam consumption of L.P. evaporator, which used bled steam from the I.P. branch of the main turbine, all stated as extra high-pressure steam for the main turbine to provide extra bled steam for feed heating the make-up, together with the extra feed-pump steam for pumping the make-up back to the boilers.

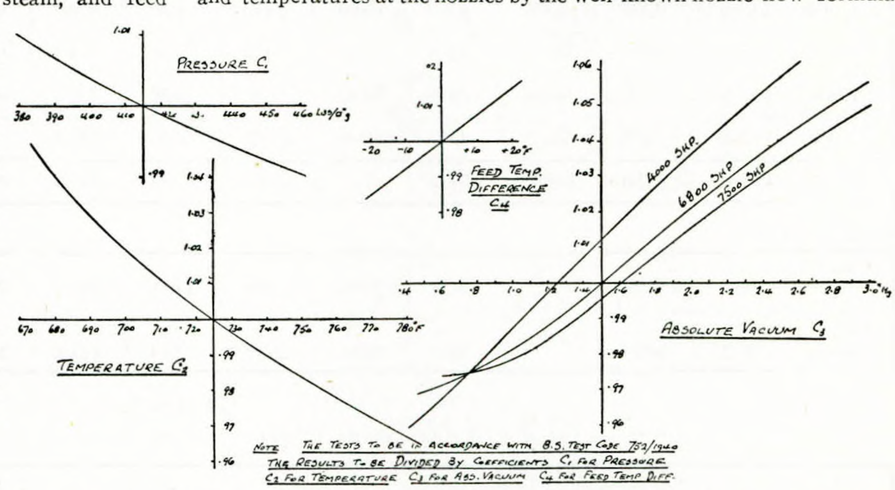


FIG. 4.—*7,500-s.h.p. turbo-alternator. Normal steam conditions:—415lb. per sq. in. g., 725 deg. F. t.t., 73 deg. F. water, 30in. barometer. Steam consumption correction curves.*

Notes on Trial Readings.

Certain other points must be made clear. During the first four hours, the evaporator was running at a high rating to produce extra distilled water to fill up the feed tank.

During the second four hours, the evaporator was shut down and the water level in the feed tank observed to determine the rate of make-up. The measurements made showed that in the first four hours the evaporator was working at the rate of 26·1 tons per day, and that the mean rate of make-up going to the boilers was 10·9 tons per day. This is a high rate, and after the trials it was much reduced after attending to various leakages.

The table of powers states the shaft horse powers as measured by four different methods, namely:—

(a) by kilo-Watt indicating meter calibrated in shaft horse power from the workshop tests of the main propulsion motor efficiency.

(b) by kilo-Watt integrating meter, also calibrated in shaft horse power from the workshop tests.

(c) by Siemens torsion meter loaned for the trials with its zero error checked before and after the trials, but without the shaft modulus having been calibrated.

(d) by volts and amps. in the propulsion circuit with the excitation adjusted to give Unity Power Factor throughout, using the two wattmeter method and sub-standard instruments specially fitted for the trials.

To avoid any suspicion of favouring any one method of measurement, the mean shaft horse power of all four sets of readings was accepted as the true mean shaft horse power.

There was no meter for measuring the feed flow, but the turbine nozzles had been carefully calibrated, and the steam flow to the turbine was calculated from the pressures and temperatures at the nozzles by the well-known nozzle-flow formula.

The fuel consumption was at the rate of 2·125 tons per hour during the first four hours, but only the specific gravity of the oil 0·982 was known, and not its real calorific value, nor its analysis, but it was assumed to be 18,000 B.T.Us. per lb.

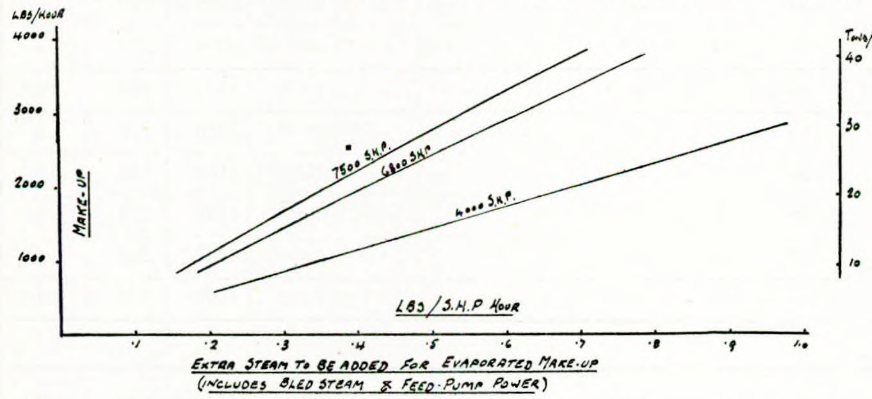


FIG. 3.—*7,500-s.h.p. turbo-electric ship. Normal steam conditions:—415lb. per sq. in. g., 725 deg. F. t.t., 73 deg. F. water, 30in. barometer. Steam consumption of bled steam evaporator.*

There was no funnel-gas analysis and the CO_2 recorder was inoperative, but the funnel-gas temperature was recorded as 495 deg. F. during the first four hours, which seems to confirm the boiler efficiency calculated from the steam/oil ratio.

Correction to Designed Conditions.

A proper comparison of the results with the guarantees requires the use of correction factors to bring the results back to the designed conditions, and these are given as curves of divisors in Fig. 4 (T.2722) giving corrections for pressure, temperature, absolute vacuum, and feed temperature difference for the main turbine, and Fig. 5 (T.2723) giving correction for pressure, temperature, and absolute vacuum for the auxiliary turbine.

Summary of Trial Results.

First four hours with evaporator working and 6,357 s.h.p.

Mean steam conditions 404 P.S.I.G., 702·5 deg. F., 0·958in. abs. vac., 299 deg. F. feed.

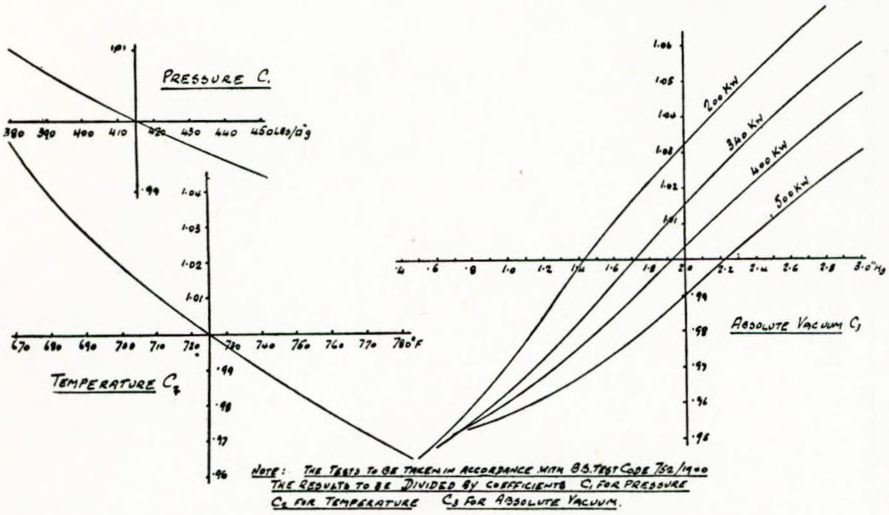


FIG. 5.—*400-kW. turbo-generator. Normal steam conditions:—415lb. per sq. in. g., 725 deg. F. t.t., 73 deg. F. water, 30in. barometer. Steam consumption correction curves.*

Designed steam conditions 415 P.S.I.G., 725 deg. F., 1·45in. abs. vac., 296 deg. F. feed.
Correction factors 1·003, 1·017, 0·980, 1·002
Total correction—Divide trial results by 1·0016

Calculated Steam Flow.
1st throttle valve 340 P.S.I.G. Vol. = 1·89ft.^{3}/lb. = 31,700lb./hour.
2nd " " 372·5 " " = 1·71 " " = 19,300 " "

Total steam to main turbine (including bled steam)	= 51,000
" " " air ejector	300
" " " feed pump	3,000
" " " aux. turbo-generator (243 kW.)	3,950
" " " evaporator (26·1 tons/day)	3,450
Total steam consumption	61,700lb./hour.

Correction to designed conditions/1·0016 = 61,600lb./hour.
Total all purposes = 9·69lb./s.h.p. hour.

Guarantees as Figs. 1, 2 and 3.

Main turbine ejector and feed pump	55,000
Aux. turbine (243 kW.)	3,950
Evaporator (26·1 tons/day)	3,450
Guaranteed total steam	62,400lb./hour.

Oil fuel consumption 2·125 tons/hr. = 4,760lb./hr. = 0·7487lb./s.h.p.
Make-up used 10·9 tons/day.
Boiler evaporation 61,700 + 1,020 = 62,720lb./hour.
Steam/oil ratio = 62,720/4,760 = 13·2
Boiler steam conditions 441lb./P.S.I.G., 705 deg. F., 299 deg. F. feed = 1,096 B.T.U./lb.
Boiler efficiency 13·2 × 1,096/18,000 = 80·3 per cent.
Funnel temperature, 495 deg. F.

Second four hours without evaporator and 6,818 s.h.p.
Mean steam conditions 413·75 P.S.I.G., 699 deg. F., 0·98 abs. vac., 305 deg. F. feed.
Designed steam conditions 415 P.S.I.G., 725 deg. F., 1·50 abs. vac., 300 deg. F. feed.
Correction factors 1·001, 1·021, 0·981, 1·0035.
Total correction 1·006.

Calculated Steam Flow.
1st throttle valve 354 P.S.I.G. Vol. = 1·79ft.^{3}/lb. = 33,200lb./hour.
2nd " " 383 " " = 1·66 " " = 19,800 " "

Total steam to main turbine	53,000
" " " air ejector	300
" " " feed pump	3,000
" " " aux. turbine (245 kW.)	4,080
" " " evaporator 0	0
Total steam consumption	60,380lb./hour.

Correction to designed conditions/1·006 = 60,020lb./hour.

Guarantee steam from Fig. 1

Main turbine ejector and feed pump	57,500
Aux. turbine (245 kW.)	4,080
Guaranteed total steam	61,580lb./hour.

The boiler conditions for the run were 460 P.S.I.G., 707 deg. F., 305 deg. F. feed = 1,090·5 B.T.U./lb. and if the boiler efficiency had been 87 per cent. and the calorific value of the fuel had been 18,500 B.T.U., the steam fuel ratio would have been

$$\frac{18{,}500 \times 0{\cdot}87}{1{,}090{\cdot}5} = 14{\cdot}75.$$

The boiler evaporation with 10·9 tons/day make-up from feed tank would be 60,380 + 1,020 = 61,400lb./hour and the oil consumption 61,400/14·75 = 4,160lb./hr./6,818 = **0·611lb.**/s.h.p./hr.

If the evaporator were making 10·9 tons/day, the steam consumption would be increased by 1,450lb./hour, the boiler evaporation all purposes would be :—

60,380 + 1,450 + 1,020 = 62,850lb./hour.

and the oil fuel = 62,850/14·75 = 4,260lb./hr./6,818 = **0·625lb.**/s.h.p./hour.

The oil fuel for main turbine air ejector and feed pump = 56,300/14·75 = 3,820lb./hr./6,818 = **0·559lb.**/s.h.p./hour.
and for the main turbine only
= 53,000/14·75 = 3,600lb./hr./6,818 = **0·527lb.**/s.h.p./hr.

If the auxiliary load of 245 kW. were taken on Diesel engines with a fuel consumption of 0·55lb./kW. hr., then the figures become :—

Main turbine ejector and feed pump	56,300lb./hour.
Evaporator 10·9 tons/day	1,450
Make-up 10·9 " "	1,020
Total evaporation	58,770lb./hour.

Oil fuel consumption 58,770/14·75 = 3,980lb./hr. = 0·583lb./s.h.p./hour.
Diesel oil fuel = 0·55 × 245/6,818 = 0·02

Fuel consumption all purposes = **0·603lb.**/s.h.p./hour.

New Design for 7,500-S.H.P. Normal Full Power.

The consumption for the new design can be obtained by proportioning it from the above figures for the steam conditions which we have chosen.

With the boiler conditions at normal full power of 500 P.S.I.G., 800 deg. F. and feed temp. 300 deg. F. the "heat at boilers" will be 1,140 B.T.Us. per lb., so that with oil at 18,500 B.T.Us. per lb. and 87 per cent. boiler efficiency, the steam/oil ratio is 14·1lb. of steam per lb. of oil.

The steam conditions at turbine stop valve would then be 475 P.S.I.G., 775 deg. F., and with 1·50in. abs. vac., 73 deg. F. sea temperature, the heat drop from the turbine stop valve would be 493 B.T.Us. per lb.

With a single-screw arrangement and a single turbo-alternator, the rated load and steam pressure having both been increased by the same proportion, it is reasonable to take the turbine efficiency ratio the same as for the 6,800-s.h.p. design, and basing the figures on the previous guarantees, the steam consumption for 7,500 s.h.p. load will now be :—

Turbine, air ejector and feed pump	= 61,000lb./hour.
Auxiliary turbo-generator, say 260 kW.	= 4,300
Evaporator, say 10 tons/ day	= 1,400
Make-up, say 10 tons/day	= 935
Total boiler evaporation	= 67,635lb./hour.

Oil consumption = 67,635/14·1 = 4,800lb./hour = 51·5 tons/day.
All purposes = **0·640lb.**/s.h.p./hr

Main turbine air ejector and feed pump = **0·576lb.**/s.h.p./hr.
With Diesel-driven auxiliary generators at 0·55lb./kW.-hour :—
Boiler evaporation = 67,635 − 4,300 = 63,335lb./hour
Oil for boilers 63,335/14·1 = 4,500lb./hour = 0·599lb./s.h.p./hr.
Oil for Diesels 260 × 0·55/7,500 = 0·019 " " "

Oil consumption, all purposes = **0·618** " " "

With two turbo-alternators, they will be each of only half the rating and will therefore have a lower efficiency, so that with this

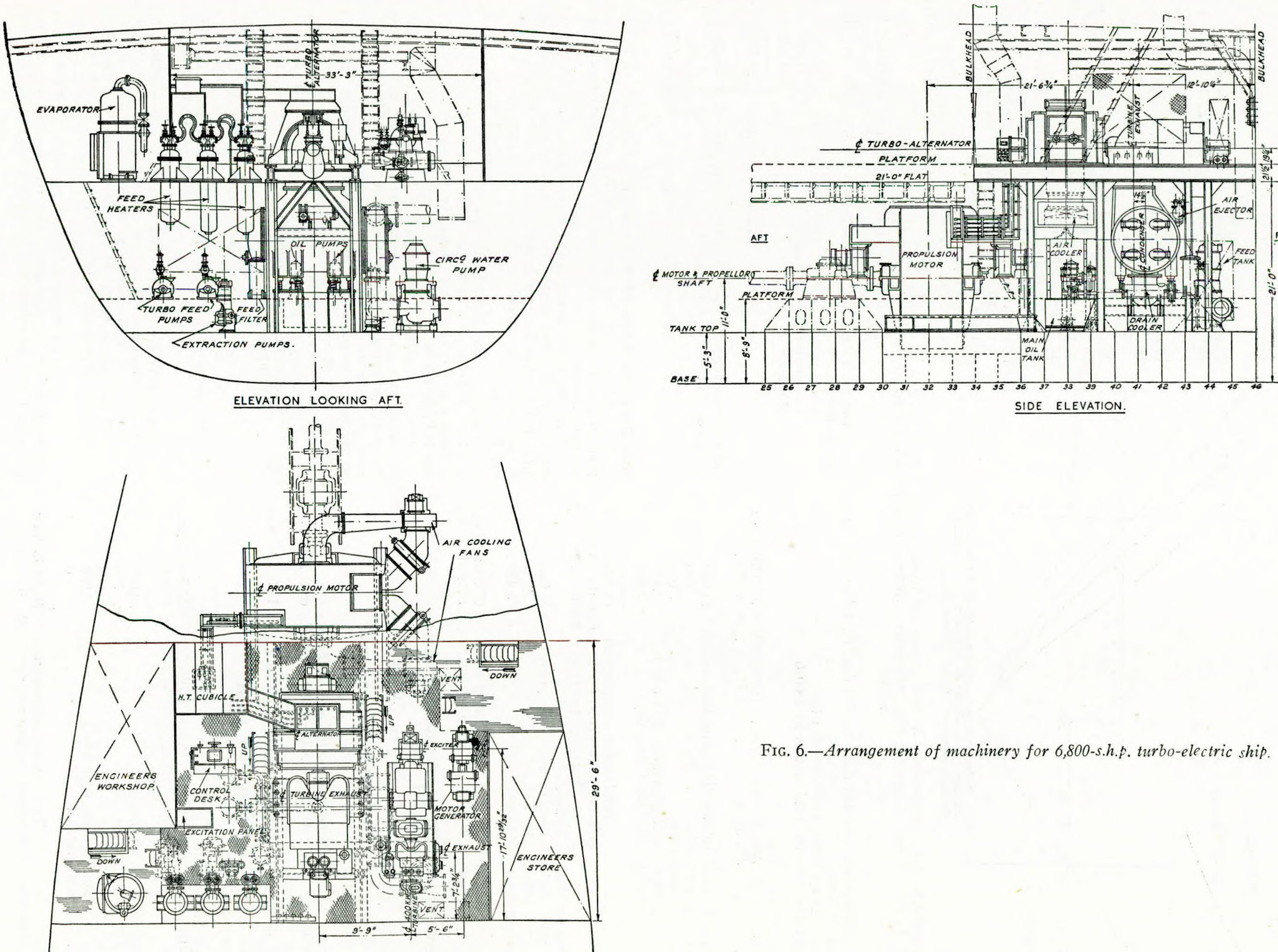


FIG. 6.—*Arrangement of machinery for 6,800-s.h.p. turbo-electric ship.*

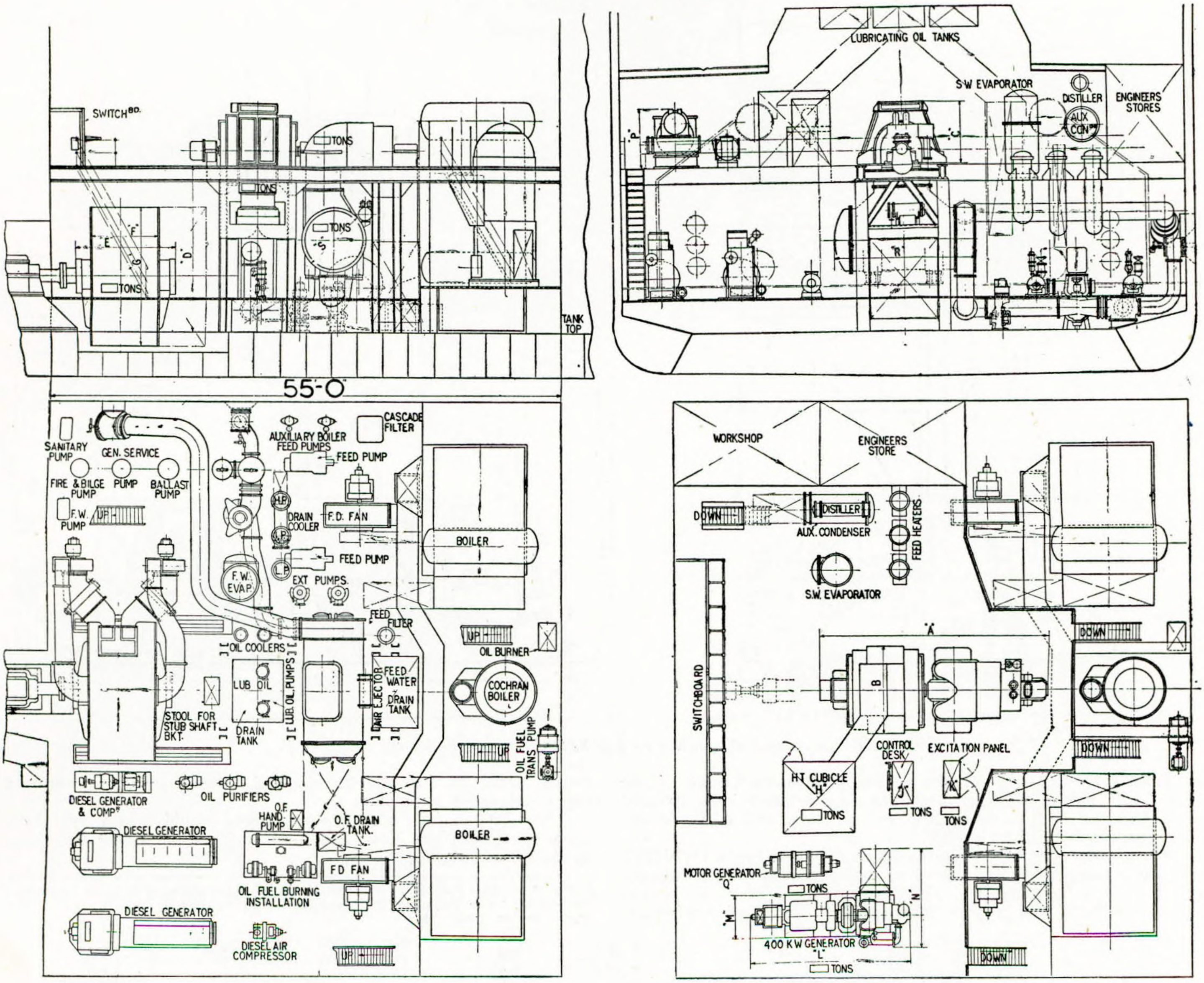


FIG. 6A.—*Turbo-electric A.C. machinery 7,500-s.h.p. single screw proposed for general cargo ships.*

arrangement 4 per cent. higher fuel consumption than with one turbo-alternator would be expected.

13,000-S.H.P. Proposal.

If this is a single-screw arrangement with one turbo-alternator, the turbo-alternator efficiency will be 2 per cent. better than the 7,500-s.h.p. single-screw arrangement.

If, however, it is a twin-screw arrangement with two turbo-alternators, there will again be a 4 per cent. drop of efficiency, and the fuel consumption would be 4 per cent. worse than the 13,000-s.h.p. single-screw arrangement.

For the twin-screw arrangement the power will need to be reconsidered, because it will have to be increased for twin screws to maintain the same speed as with a single-screw arrangement, but this is a matter on which the shipbuilder would have to be consulted before comparative fuel consumptions could be stated with certainty.

Weights of Machinery.

For the 6,800-s.h.p. machinery described above, the weights of the propulsion machinery were :—

Main engine (turbo-alternator and its air cooler, motor and its fans and air cooler)	137 tons.
Control gear	6·5 „
Condensing and feed-heating plant with circulating and extraction pump	48·5 „
400-kW. turbo-generator complete	12 tons.
Total	204 „

As boiler weights vary so much between different makers and as these and the shafting, thrust block, ladders and gratings, together with the general service auxiliaries, are all the same as for geared turbines, it is thought that no attempt should be made to estimate their weight, for unless the equipment were designed to a detailed shipowner's specification, great variations might appear which have nothing whatever to do with turbo-electric propulsion.

Recently, the authors have seen weights of boilers of 50 per cent. bigger output, higher efficiency, and higher pressure and temperature, which were much lighter than the weights which might have been quoted in the list given above, and the designer of turbo-electric machinery cannot involve himself in such variations.

For 7,500 normal shaft horse power single-screw with one turbo-alternator, the above total weight will be increased to 235 tons.

For 7,500 normal shaft horse power single-screw with two turbo-alternators, the total weight will be 300 tons.

For 13,000 normal shaft horse power single-screw with one turbo-alternator, the total weight will be 300 tons.

For 13,000 normal shaft horse power twin-screw with two turbo-alternators, the total weight will be 425 tons.

Space Required.

The 6,800-s.h.p. machinery given above was installed in exactly

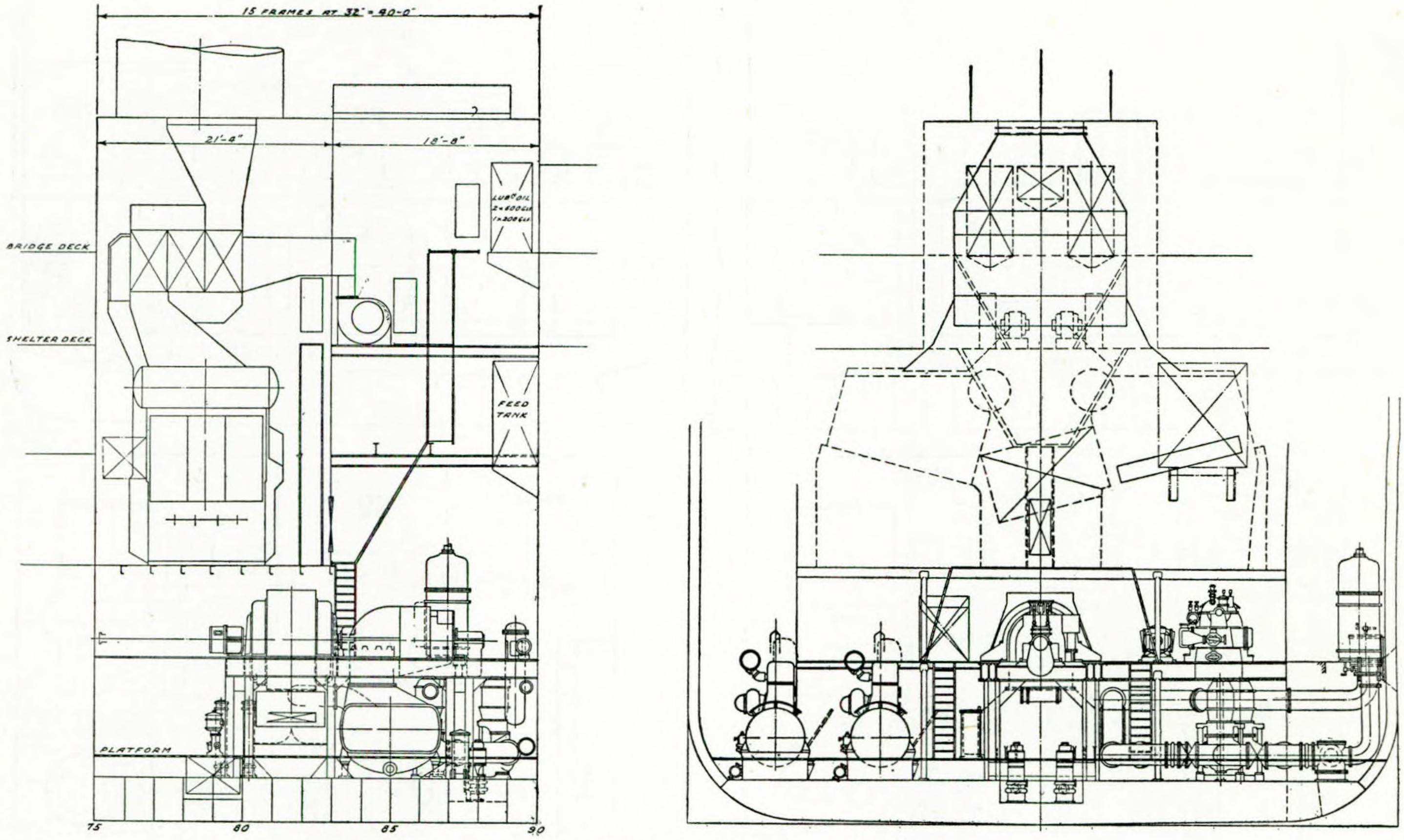


FIG. 7.—*Arrangement of machinery for 8,000-s.h.p. turbo-electric cargo ship.*

the same space as that for the geared turbine machinery of the same power and, in addition, there was enough room for a 400-kW. turbo-generator with direct-coupled exciter and complete condensing plant, and a motor-driven exciter.

The arrangement of this machinery is shown in Fig. 6 (5/05/244) and, when completed, was found to be convenient in all respects. The watch-keeping engineers, after 16 months' service in the Far East, all declared that they could not wish for a more comfortable engine room, or for less anxiety and general personal wear and tear in maintaining the plant.

For 7,500 normal s.h.p. the machinery would occupy very little more space, and the 55-ft. length of engine room would be ample, as shown in *Fig. 6a.

* Fig. 4 of Mr Saunders' contribution to the Institute of Marine Engineer's Papers on "The Engining of Post-War Cargo Vessels of Low Power", which were read in June, 1944.

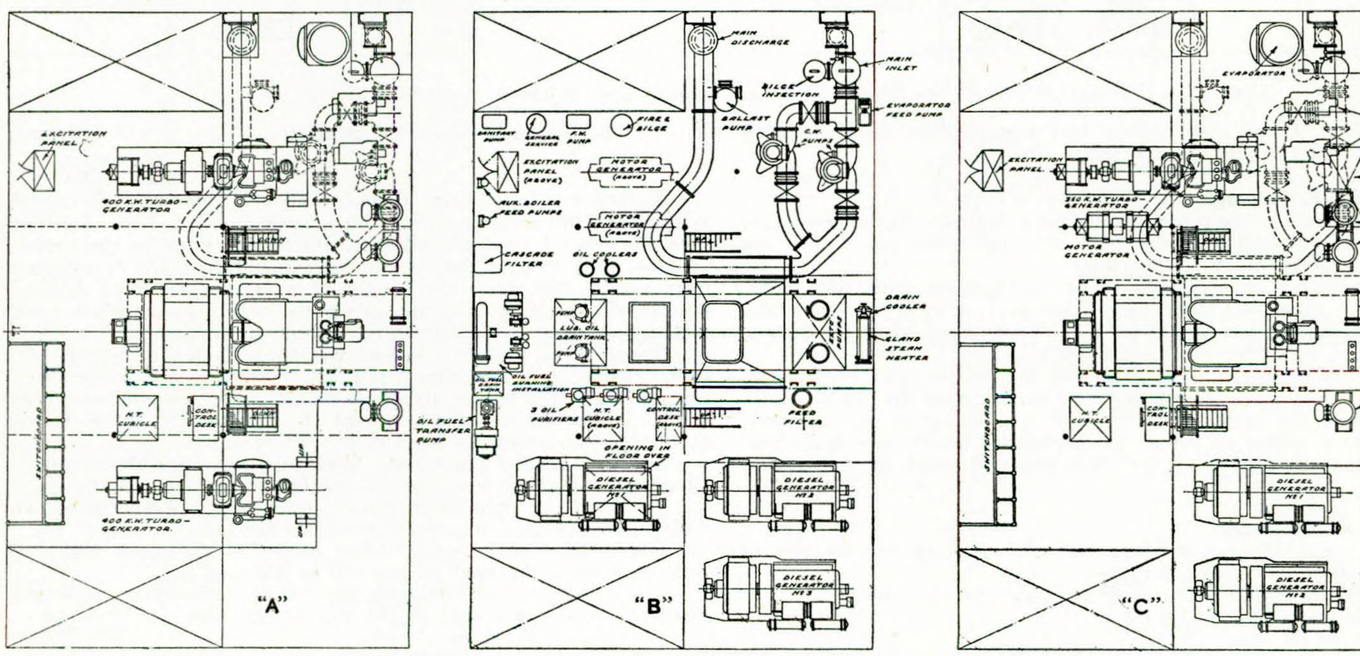


FIG. 8.—*Arrangement of machinery for 8,000-s.h.p. turbo-electric cargo ship.*

An alternative arrangement of the main engine room is shown in Figs. 7 and 8 (P5-2013/4), where the propulsion motor is in a separate compartment at the aft end of the ship. The engine room length in this case is 40ft.

Maintenance.

Reference was previously made to the comfort of the engineers, but from the superintendent engineer's point of view, the defects list is of even greater importance. When that ship returned to this country after its long service in the Far East, during which time no news whatever of her was available, the chief engineer's defect list had only three pressure gauges to repair and a meter to recalibrate, with no work whatever to be done on the main propulsion machinery. It had been hoped to get a report on the behaviour of a new throttle valve material, but there had been no necessity to open the steam chest for valve maintenance, and the ship sailed again without any opportunity of inspecting the valves and seats, and no deterioration had been noticed.

A general external inspection showed that the maintenance work amounted only to a few minor adjustments on the brush gear.

One of the authors of this paper has previously stated the record of other ships in this respect.

Prices.

Firm prices cannot be given at the time of writing with any likelihood that they will be correct at the time of publication and, furthermore, unless they were based upon a detailed shipowner's specification, they would not be so closely related to the prices of other types of machinery that a proper comparison is ensured.

There is no need to accept statements that turbo-electric machinery must be much more expensive than geared turbo machinery, and any gross excess must arise from circumstances which are not related to design or manufacture.

The U.S.A. have published the average domestic costs of T.2. S.E. A.1. ships with turbo-electric machinery as $3,010,703, and of T.3. S.A.1. geared turbine ships as $2,970,029, which is a difference of 1·35 per cent. in favour of the geared turbine. The machinery in both cases was designed and built by the same firm who, no doubt, would design both to the best advantage, and the shipbuilders had no reason to prefer one or the other. As a great number of each type were built, one can be sure that the relative prices are fair and represent the true facts.

It would be interesting to have comparative figures of the relative economy and maintenance costs.

(VI) The Combustion Turbine

By J. Calderwood, M.Sc. (Member of Council)

Synopsis

The paper first deals very briefly with the present position of practical development of the combustion turbine. The conditions required to make the combustion turbine a practical commercial machine for marine service are then reviewed.

Following this, the working principle of the Sulzer high-pressure cycle and the reasons that have led the Company to adopt this cycle for marine turbine development are examined.

The question of type of drive which can be adopted in conjunction with the combustion turbine for marine service is next considered. Layout drawings with tables of weights, including main and auxiliary machinery for two alternative 13,000-h.p. twin-screw proposals, are then given.

The main particulars for a 7,000-h.p. single-screw proposal are also dealt with, but not in such detail as the larger proposal.

Finally, the question of types of fuel which may be suitable for use in combustion turbines is briefly referred to.

Whereas all the other types of machinery covered in this symposium are engines which have been tried and proved in service, the combustion turbine is as yet in its infancy. Up to the present time a number of combustion turbines have been built for service in land power stations but, so far as the author knows, none of these has yet had a long period of working time in service. One locomotive with this type of power plant has now been in service for some years, but even to date its total working hours must be very short as compared to the service life of marine propelling engines. As yet no combustion turbine is in service at sea and although one or two turbines likely to be fitted into ships have been described in the press, these are comparatively small units of moderate efficiency and can hardly represent the type of turbine that must eventually be developed for marine propulsion.

One turbine of about the size being discussed in this symposium with which high efficiency is expected, is nearing completion and the author had hoped that some test results from this might be available before the symposium was presented. Unfortunately the delays, usual to-day, in supplies of material have postponed the testing date on the set, so that it is not likely to be ready before the latter part of next year. Even though nearing completion in the workshops, it is not proposed that this set should be installed in a ship until substantial experience has been gained of running on the testbed. Before the combustion turbine can compete with the other types of available prime movers, it must show the prospect of an overall performance which is not inferior to those other types and at the same time must give some commercial advantage to the user who has obviously no incentive to change from well-tried and proved machinery to plant of which there is no experience unless the new type of plant promises some marked commercial advantage.

The combustion turbine is to-day being developed by a number of firms in many countries, but the author is here able to deal only with the particular design that has been developed by his own Company, Sulzer Bros., and to explain the reasons for proposing their particular cycle for marine work. Some years ago it became evident that developments in material were tending towards the stage that would make it possible to manufacture a combustion turbine of high efficiency. As a result Sulzer Bros. started to study the particular problem of the design of a turbine to meet the requirements of marine service. A series of researches was carried out on the use of turbines in conjunction with highly-supercharged reciprocating engines, as power units developing power from gases produced by free-piston compressors and finally as a true combustion turbine. From this work it appeared very evident that the true combustion turbine was essentially a type of machine suited for high outputs. At the present stage of development of the combustion turbine, it is hardly possible to attain high efficiency in a small installation unless at the cost of great complication and excessive first cost.

The primary conditions that must be considered in the design of a combustion turbine for marine use are:—

(1) Complete reliability in service.

(2) High overall economic efficiency; this, of course, involves a low fuel consumption, reasonable first cost and low maintenance cost.

(3) Small size and low weight.

To attain all of these simultaneously is a difficult problem. The aircraft turbine for example is ideal in respect of (3) and probably partly good in respect of (2), but would be quite useless for marine service as it would have a life of only a few days. In respect of reliability in service, the main problem is one of design and materials and in the present stage of development it is very desirable that the manufacturing process and the materials used should be of types of which experience in other fields has been gained, also that the design should not incorporate completely untried features. For example, blade cooling or the use of ceramic materials for blades might offer substantial improvements in fuel consumption, but at the cost of incorporating completely untried features of design or of material.

To meet conditions (2) and (3) is the basic problem of the design of a marine gas turbine, in particular when high powers are considered. It is a difficult problem in the general design of a plant to allow of high efficiency with moderate dimensions and weight, and a great deal of study has been expended on this problem resulting eventually in the Sulzer high-pressure combustion turbine cycle which is suitable for installations exceeding 6,000 b.h.p. on each shaft.

To keep the dimensions of the gas turbine small, whilst at the same time obtaining high thermal efficiency, it is necessary to use a high pressure of 14 to 28 atm. (200 to 400lb. per sq. in.). Those who have studied the published literature on this subject will realize that the pressure ratio which gives the maximum efficiency in the simple combustion turbine is between 3 : 1 and 5 : 1. The simple open cycle is therefore not suitable for high-pressure working. The closed cycle is, of course, suitable for operation at high working pressures, but the difficult problem of the air heater must then be faced. This air heater is not only a large and complicated piece of equipment but it also reduces the efficiency of the plant. If the cycle itself has a thermal efficiency of 36 per cent. and the "boiler efficiency" of the air heater is 90 per cent., then the overall efficiency of the plant is $0{\cdot}36 \times 0{\cdot}9$, i.e. 32 per cent.

Open-cycle type turbines can reach as high efficiencies as are possible with the closed cycle, particularly in view of the fact that they do not suffer from loss caused by the "boiler efficiency". However, such efficiencies cannot be obtained with the simple open cycle, but require intercooling, interstage heating and recuperation. With these additional complications the output of the plant per lb. of air inhaled becomes considerably better than with the simple open-cycle turbine. Nevertheless, the air quantity is still high and the low-pressure part of the installation has to handle large volumes of air at or about atmospheric pressure. The question is made even more difficult in the open-cycle design, because pressure losses in all of the pipes and apparatus must be kept very low, as any loss of pressure is directly reflected in efficiency. As a result, the gas velocity through the pipes, heat exchangers, etc., must be kept low so that these parts become of very large dimensions and are relatively heavy.

These considerations led to the study of the possibility of designing a cycle in which the volume of gas to be handled at low pressures would be much less than in the open cycle, whilst at the same time the cycle could work at high maximum pressures.

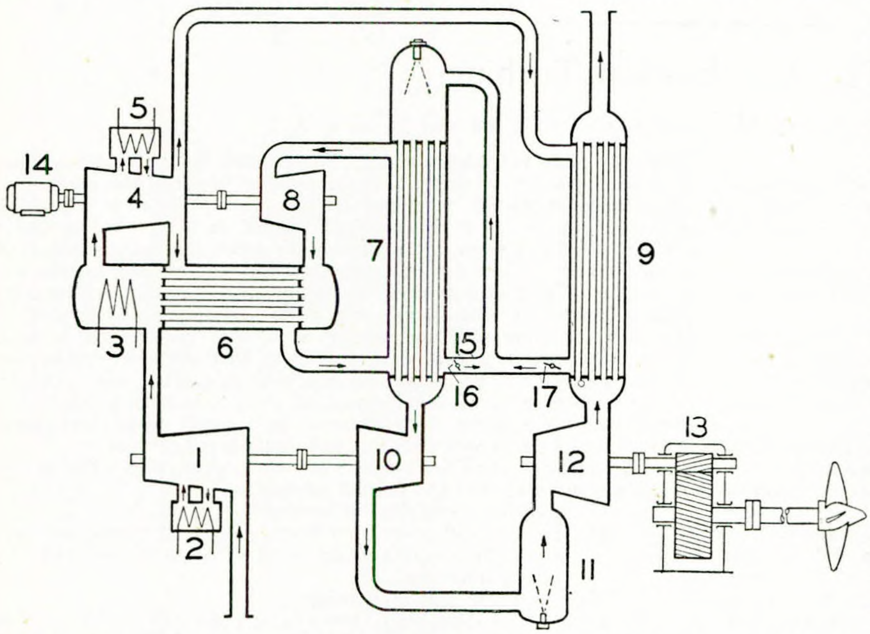


FIG. 1.—*Diagrammatic arrangement of Sulzer high-pressure combustion turbine cycle.* (1) L.P. compressor; (2) intercooler; (3) pre-cooler of circuit; (4) H.P. compressor; (5) intercooler; (6) circuit heat exchanger; (7) air heater; (8) circuit turbine; (9) exhaust-gas heat exchanger; (10) turbine for driving L.P. compressor; (11) secondary combustion chamber; (12) propulsion turbine; (13) reduction gear; (14) starting motor.

These studies have resulted in the design of a high-pressure cycle, one arrangement of which is shown diagrammatically in Fig. 1. A diagram showing the principle of the cycle was published in a paper read before the North-East Coast Institution of Engineers and Shipbuilders in Newcastle-upon-Tyne on the 5th April, 1946: "Some Researches on Internal Combustion Prime Movers". The cycle there described lends itself to a number of variations in the arrangement of different components. The diagram now given in Fig. 1 shows a particular arrangement of this cycle which fits in well with the conditions of marine service. It will, however, be understood that the arrangement and proportioning of the various units can be modified to suit particular conditions of service.

The low-pressure compressor (1) draws air from the atmosphere and compresses it to a pressure of, for example, 70lb. per sq. in. This compressor (1) is provided with an intercooler (2). The compressed air then flows through the cooler (3) which also serves as the return cooler in the air circuit. The air is then further compressed by a high-pressure compressor (4) which is fitted with an intercooler (5). Here the pressure is raised to the maximum cycle pressure, for example 280lb. per sq. in.

The discharge from the high-pressure compressor (4) is divided; one part of the air flows through and is heated in the heat exchanger (6), and finally passes through the tube stack of the air heater (7), where it is raised to the maximum temperature of the cycle. The hot air is then expanded in turbine (8) which drives the high-pressure compressor (4). The exhaust air from the turbine (8) then passes through the heat exchanger (6) and gives up most of its heat to the compressed air coming from compressor (4). Leaving the heat exchanger, it mixes with the fresh air delivered by the compressor (1) and then recommences the cycle by being cooled in cooler (3). The other part of the air discharged from compressor (4), is first heated in the heat exchanger (9) by the exhaust gases from the power turbine. It then passes to the combustion chamber of the air heater (7). The resulting hot combustion gases flow through the tube stack of the air heater, in which they are partly cooled while heating the air flowing to the turbine (8). The comparatively cool gases are then led to the turbine (10) where partial expansion takes place. From this point the gases are led to a secondary combustion chamber (11) and re-heated to a temperature suitable for the turbine blades. The expansion of the gas then takes place to atmospheric pressures in turbine (12). The turbine (12) produces the useful power and the turbine (10) drives the low-pressure air compressor. The expanded gas leaves the system after giving up part of its heat in heat exchanger (9) to the compressed air.

It will also be noted that both air circuits are connected by a pipe (15). By suitably adjusting valves (16) and (17) it is possible to control the air flow to heat exchangers (6) and (9) and so obtain the maximum thermal efficiency under all conditions. Air is delivered to turbines (8) and (10) approximately in the ratio 55 per cent. to 45 per cent.

It is hoped that the thermal efficiency of the completed machine working on this cycle will be about 35 per cent., corresponding to a fuel consumption of about 180 gr. per b.h.p.-hr. (0·4lb. per b.h.p.-hr.). This consumption is based on a maximum temperature of the blades not exceeding 680 deg. C. (1,250 deg. F.). There is no reason why similar overall efficiencies should not be reached with other types of combustion turbines, for example open-cycle turbines. However, whatever the basic cycle employed, it will not be possible to obtain so high an efficiency with an arrangement of the plant which is in any way simpler than that required for the high-pressure cycle illustrated in Fig. 1. On the other hand, the Sulzer high-pressure cycle has the great advantage that in those parts of the cycle which are working at low pressures the quantity of gas in circulation is much smaller than in the usual open cycle and therefore the size of piping, heat transfer units, etc., is very much reduced. The cycle has the additional advantage that, as only a part of the air is handled in the low-pressure section of the machine, the pressure losses in this part of the cycle have a much smaller influence on the overall efficiency. As a result, not only are there smaller quantities of air to be dealt with in the intake and exhaust, but higher velocities can safely be allowed in the pipes. This is a particularly important factor in relation to the only low-pressure heat exchanger in the system, namely, that indicated at (9) in the diagram. The heat exchanger (6), on the other hand, works with high pressure on both sides of the tubes, so that a high efficiency of heat transfer can be attained with a heater of reasonable dimensions, thus improving the thermal efficiency of the plant. A feature that should be specially noted is that in the primary heat exchanger (7) the pressure on both sides of the tubes is the same, namely, the maximum pressure of the cycle. This gives a very high rate of heat transfer, resulting in small dimensions for this heater. It has the further advantage that there are no mechanical stresses of the tubes due to pressure. This is an important factor, as these tubes are at the highest temperature of any part of the plant. It is a characteristic advantage of the cycle that it solves the problem of attaining a high efficiency of heat transfer with heat exchangers of reasonable dimensions.

The working cycle described has other particular advantages for marine service. The power and efficiency are practically unaffected by changes of ambient air temperature. They are, however, influenced by the temperature of the water available for the intercoolers, but this temperature range is relatively small so that it can be said that the change in power and efficiency with the climatic conditions is comparatively unimportant. Further studies are being made from which it is hoped to make the cycle entirely independent of water temperature so far as the output of the plant is concerned.

The effect on power, output and efficiency of fouling the turbine blades is smaller with a cycle such as described than with the usual open cycle, although in this respect it is, of course, not so good as the completely-closed cycle. It is, however, not the purpose of this paper and space is not available to make a comparison in detail of the various types of combustion turbine. However, it must be stressed that the cycle described is particularly flexible in respect of the arrangement of the various units to enable correct proportioning of the blading to meet operating conditions. The arrangement shown incorporates five rotary machines, i.e. compressors and turbines arranged on three shafts, and this is probably the simplest arrangement that can be obtained for a high-power installation of high efficiency and to enable the necessary regulating to be obtained whilst working with a correct balance between the various machines. It is also considered very important that the actual power turbine should be an independent unit, not mechanically connected to any of the compressor units, in order to ensure that speed variations on the propeller which are bound to occur in any but the smoothest weather, are not reflected back to the compressors.

The starting of an installation of this type is carried out entirely on the high-pressure compressor group, which is provided with an electric starting motor. After this group has been started by the motor, the combustion can next be started, after which the whole of the plant and rotating machines must be gradually warmed through. In normal service it will be advisable to allow about 90 minutes for heating up the plant before it is brought up to full load.

It is not possible here to deal in detail with the governing and control, but in principle the power is altered by varying the speed of the two compressor groups together with an alteration in the quantity of the fuel. In this way the maximum pressure will be lower when running at reduced power whilst the maximum temperature will be only slightly altered. As the maximum pressure is the product of compression ratios of the L.P. and M.P. compressors, wide variation in maximum pressure and consequently in power can be obtained with comparatively small variations in the speed and temperatures of the compressor units.

One of the main problems of the application of the combustion turbine for marine service is the question of reversing. The turbine itself is a non-reversible unit and the provision of an astern turbine as in steam practice is not practicable for the combustion turbine. There are therefore three schemes that may be considered for providing for the manœuvring of a ship. These are :—

(1) Electric drive.

(2) Gear drive incorporating reversing gear probably in conjunction with hydraulic couplings or electromagnetic couplings.

(3) Reversible propeller.

The first of these, namely the electric drive, is well-known and tried in service, but is rather heavier than other types of drive. The reversing gear drive is as yet a completely untried method of reversing for the power here being considered. It is, however, a very likely future development and is being closely studied, but could certainly not be considered for any installation at an early date. The reversible propeller is a simple and promising solution with which some experience has been gained in practice on ships of comparatively high power.

Fig. 2 shows the proposed machinery arrangement which has

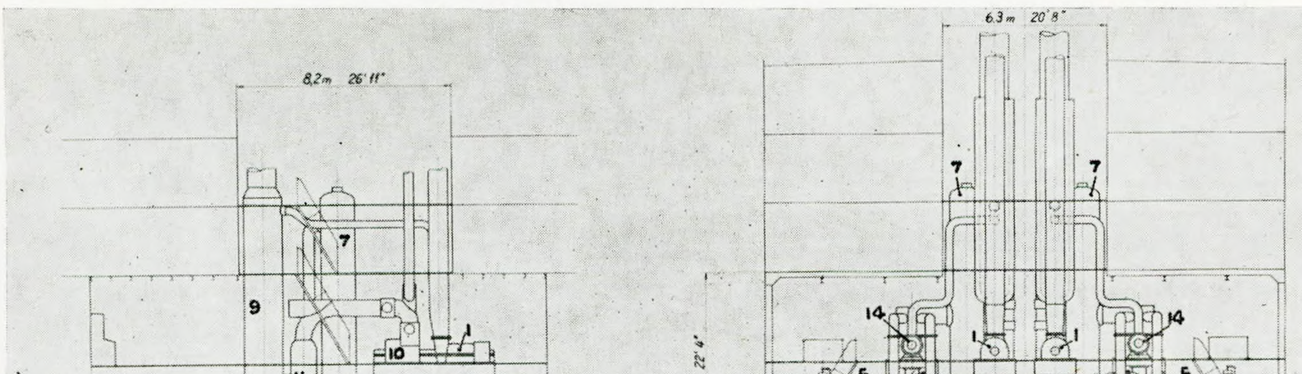


FIG. 2.—*Machinery arrangement of 13,000-s.h.p. continuous service output twin-screw combustion turbine with gear drive and reversible propeller.*

(1) L.P. compressor; (2) intercooler; (3) pre-cooler of circuit; (4) H.P. compressor; (5) intercooler; (6) circuit heat exchanger; (7) air heater; (8) circuit turbine; (9) exhaust-gas heat exchanger; (10) turbine for driving L.P. compressor; (11) secondary combustion chamber; (12) propulsion turbine; (13) reduction gear; (14) starting motor.

been prepared for the larger twin-screw installation being considered in this symposium. For this arrangement it is assumed that a straight gear drive will be used in conjunction with a reversible propeller. The various items of equipment in this diagram are numbered in the same way as the corresponding items on the diagrammatic layout Fig. 1. The auxiliaries directly connected with the turbine, i.e. cooling and lubricating oil pumps, have also been shown in dotted lines. Four auxiliary generator sets, each of 300 kW., are also shown. The various other auxiliary equipment in the engine room has not been indicated in detail, but dotted circles have been put in to show the space available for auxiliary pumps, filters, etc. Although ample space has been allowed for the equipment, it will be seen that the required engine-room dimensions are moderate. The actual size of the equipment shown in this arrangement drawing is based on a turbine of slightly larger output that is at present under construction, so that in fact the total output obtainable from the machinery shown would be greater than that called for in the symposium specification. Machinery weights for this arrangement are given in detail in column 1 of the table appended to the paper. It will, however, be appreciated that whilst the weights of the principal parts, i.e. turbines, gears and their associated equipment, are actual designed weights, the general installation weights are estimated only and might vary substantially according to the details of any particular owner's specification. A particular point of interest for direct comparison with other types of propelling machinery is that the complete turbine and gears with all directly associated equipment weighs only 330 tons which, on the designed service output of this turbine, is 22·6 kgs. per b.h.p. (50lb. per b.h.p.).

Fig. 3 shows the same installation but with electric transmission. The main engine room only is shown and a motor compartment would, of course, have to be provided in addition. The items in this diagram have not been numbered, but the whole of the layout is the same except that the generator replaces the gear box shown in the previous diagram and the exciter sets are also shown in dotted lines. Apart from the additional motor room, the space required for this arrangement is about 3ft. longer than for the geared drive with reversing propeller. However, the electric drive is thoroughly tried and a reliable system of transmission for large powers, and it is most probable that at least the earlier ships fitted with a combus-

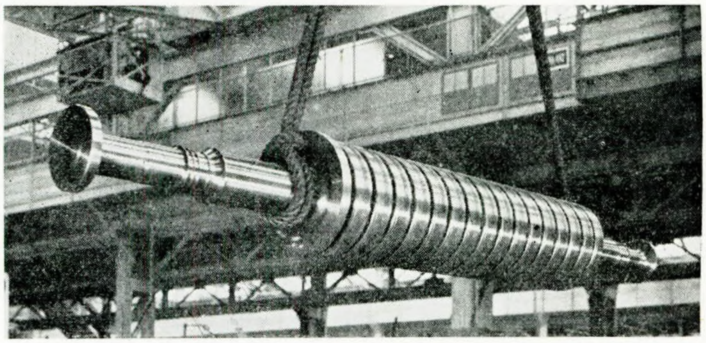


FIG. 4.—*Compressor rotor machined ready for blading.*

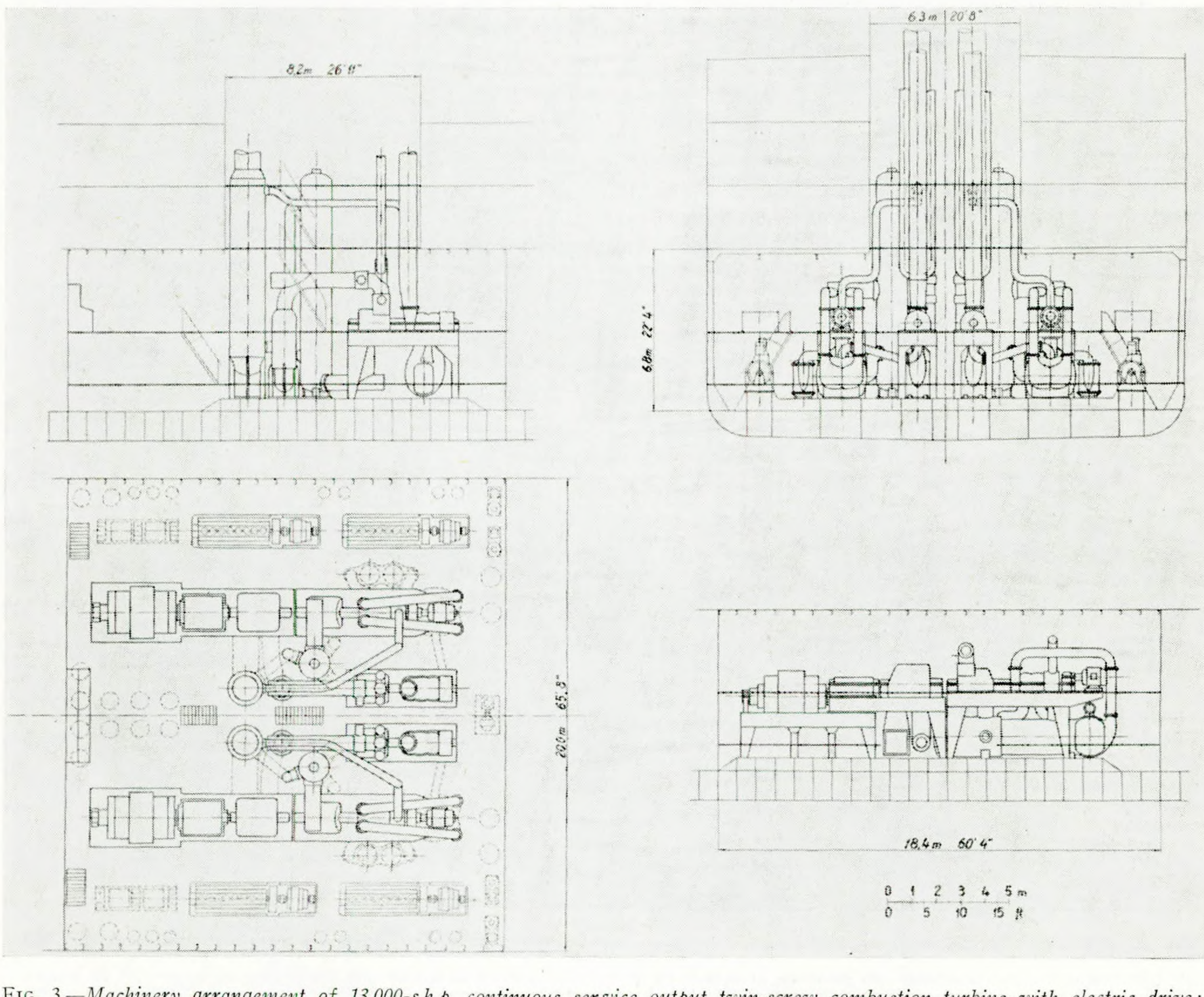


FIG. 3.—*Machinery arrangement of 13,000-s.h.p. continuous service output twin-screw combustion turbine with electric drive; propulsion motors to be arranged in separate compartment.*

TABLES 1 and 2.

	1	2
Machinery type	High-pressure cycle combustion turbine	
Transmission	Geared with reversible propeller	Electric drive
Number of engines...	2	2
Maximum working pressure, abt. ...	280lb./sq.in.	280lb./sq.in.
Maximum temperature	1250° F.	1250° F.
Propeller speed	110 r.p.m.	110 r.p.m.
Type of auxiliary machinery	Electric	Electric
Fuel consumption in tons per day (at 13,000 s.h.p.):		
Main engines	59†	62‡
Auxiliaries (assuming load at sea about 600 kW.)	3·7	3·7
Total fuel consumption at sea ...	62·7	65·7
Lubricating oil consumption per day:		
Main and auxiliary engines approx....	16 gals.	16 gals.
NOTE.—Oil consumption is mainly for auxiliaries.		
Weight:—	tons	tons
Main turbine equipment, including piping on turbines, combustion chambers, intercoolers and air heaters	230	230
Baseplates for turbines and compressors...	40	40
Two sets of reducing gears	100	—
Two sets of electrical transmission equipment, consisting of main generators, main propulsion motors, exciter sets and switch gear	—	203
Four auxiliary generator sets of 300 kW. each, with starting air tanks, cooling pumps and other accessories direct connected with the engine	64	64
Auxiliary machinery consisting of:—		
Two electrically-driven cooling-water pumps for main coolers, capacity each approx. 270 tons per hour ...		
Two lubricating-oil pumps, capacity each approx. 54 tons per hour ...		
Two governor oil pumps, capacity each approx. 18 tons per hour ...		
One set of filters for oil and fuel ...		
One electrically-driven starting air compressor for auxiliary engines, capacity 28 cu.ft./min.		
One electrically-driven centrifugal ballast pump, capacity 250 tons/hour		
One electrically-driven reciprocating general service pump, capacity 80 tons/hour		
One electrically-driven reciprocating bilge pump, capacity 80 tons/hour		
Two electrically-driven rotary replacement fuel oil pumps, capacity each 40 tons/hour		
One refrigerator $1\frac{1}{4}$ kW.		
Auxiliary boiler (15,000lb./hour) ...		
Condenser		
Evaporator		
Distiller	75	75
General installation work including pipes, valves, fittings, connections for bilge, ballast, steam, water and oil systems, tools and fittings, floors, gratings, ladders, daily service tanks, etc., including weight of water and oil in piping and machinery, approx. weight ...	190	190
Shafting and propellers, approx. weight	240	150*
Total weight	939	952

*This weight is based on the assumption that propulsion motors are arranged as far aft as is possible.

†Includes $5\frac{1}{2}$ per cent. for gear losses and loss in propulsive efficiency with V.P. propeller.

‡Includes 10 per cent. for losses in electric drive.

tion turbine are likely to have electric drive. The weights for the installation of electric drive are shown in column 2 of the table, and although the motors are not shown in the drawing they are included in the weights.

In the early stages of combustion turbine development it is unlikely that any owner would send a ship to sea with a single propulsion turbine only, and a drawing has therefore not been prepared for the single-screw installation called for in the symposium specification. However, the units of equipment required for single-screw installation would be one half of the units shown for the twin-screw installation. The arrangement would, of course, be different. The approximate total weight for the single-screw geared installation with all accessories of relative sizes to those proposed for the twin-screw installation would be approximately 555 tons.

Finally, a word may be said about fuel for combustion turbines. There is at the moment a great deal of discussion as to the class of oil that can be burnt with safety in this type of machine. A certain answer to this question can only be obtained from a long period of experience in service. In tests that have already been carried out during the development of the turbine here described, a considerable time has already been expended on fuel tests, from which it appears that the danger of fouling of the installation lies mainly in incomplete combustion. There is therefore every reason to believe that if good combustion can be maintained, the heavier grades of fuel should be satisfactory, but it must be stressed that a period of tests in the workshop or even a short period in actual service is no adequate proof of the ability of combustion turbines to burn the heavy grades of fuel. In principle this problem is equally important for all gas-turbine cycles as even with the totally-closed cycle the air boiler must be supercharged, and when this is done the fouling problem will occur in the air boiler and in the turbine and compressor of its supercharger. Even if eventually it is found that the combusution turbine cannot work satisfactorily on the heavier grades of fuel, it would with an efficiency of the order of 35 per cent. and with the small dimensions and low weight of the set shown in Figs. 2 and 3 still be an attractive proposal commercially.

Unfortunately, in the present stage of development of the combustion turbine, it has not been possible to make this paper of a practical nature. A large amount of testing and research has been done and will continuously be carried out on the development of the combustion turbine, but no high-efficiency turbine working on any of the known cycles is yet in commercial operation. It has therefore been possible only to deal with the subject on the basis of the designs which have been prepared for the first large high-efficiency turbine which is now nearing completion. It is hoped, however, that even this limited information will prove to be of interest and value as a comparison with the existing established methods of propulsion.

Summary of the Symposium Papers

By S. F. Dorey, C.B.E., D.Sc., Wh.Ex. (Vice-President)

This symposium, another in a series which have been arranged by our Papers and Transactions Committee, will I am sure be found useful not only by marine engineers but also by those whose business it is to run a ship in an economical manner. The shipowner, like all other business men, is in business to make money, and what he is concerned with most is to operate a ship at the lowest cost per ton of cargo carried throughout the ship's whole life.

The details given in the six papers that comprise the present symposium do not of course deal with all the factors which may be considered of importance to the shipowner, but they will, however, give him an opportunity to assess the merits of each of the various installations described.

The Institute is to be congratulated on being able to persuade engineers of the standing of the authors to come forward and make authoritative statements in the manner they have done. A special feature is the way in which the authors have explained in detail certain parts of machinery, which will, I am certain, be to the advantage of sea-going engineers who may not be familiar with all the machinery arrangements described.

What are the more important factors which the shipowner has to bear in mind when considering what type of machinery he will install in a vessel? The first is undoubtedly reliability, and I do not say this because I happen to be a classification society surveyor, and therefore not primarily interested in first cost and upkeep.

Reliability ensures a maximum sea service over a period, and with it is coupled low maintenance and repair costs. These two considerations alone may well be worth some extra first cost. To the shipowner the fact that a particular engine gives the lowest fuel consumption is not of great significance unless the saving is appreciable and able to outweigh some disadvantage such as extra weight. Weight and space are of direct concern to the shipowner. It would have been interesting to have had the remarks of the authors regarding the total operating costs of two vessels suitable for the two horsepowers quoted and fitted with the various types of machinery for a round trip of, say, 10,000 miles. This would bring out points which might indicate that the machinery giving the lowest fuel bill may be the heaviest to carry round, and involve higher repair costs than another with higher fuel bill and lower maintenance costs. The highest first cost may give the lowest weight and repair bill but require more bunker space, while the most efficient engine may require a more highly skilled crew.

In my opinion the four most important factors in their order of merit are therefore:—

Reliability.
Low maintenance costs.
Maximum cargo capacity.
Initial cost.

Turning now to the six papers which have been presented, it is proposed to draw up comparisons under various headings such as fuel consumption, weight, etc. from the information given, but before doing this some further comments of a more general nature may not be considered out of place.

It would appear that discussion of the relative merits of the types of machinery described must, broadly speaking, fall under two main headings, namely, the engines themselves (i.e. the motive power) and the transmission.

With regard to the engines themselves, it is evident that a very good case can be made for both steam and Diesel propulsion over the range of powers considered, and it does not seem that either of these two types is progressing relatively so rapidly that this state of affairs is likely to change in the near future. Development of steam plant is generally along the lines of increased steam pressures and temperatures with correspondingly reduced fuel consumptions, whereas progress with Diesel engines is towards greater compactness rather than any substantial improvement in economy, apart from a tendency towards the ability to use lower-priced fuels.

In this connection it is of interest to recall that experiments directed towards the use of boiler fuels in Diesel engines are by no means a new departure, and in fact a fleet of vessels using such engines was in operation just before the war. One reason for the recent revival of interest in this development is the increasing use of Diesel fuel on land. If this could be offset by using a larger proportion of boiler fuel instead of Diesel fuel at sea, the balance of output of the various grades of fuel from the oil refineries would be maintained.

The steam conditions selected for the turbine machinery can be taken to be typical modern marine practice, with the exception of Dr. Brown's third alternative in which he has shown the improvement in economy to be obtained by the use of reheat in conjunction with a boiler pressure of 1,400lb./sq. in., a condition which, up to the present, has been almost entirely confined to land practice. On the Diesel side it is interesting to note that two of the three alternative types of engine given in Messrs. Belsey and Robinson's paper operate on the four-stroke cycle, the development of this type of engine fitted with exhaust-driven superchargers now having reached a stage where it can challenge the two-cycle engine for sizes of 1,000-1,500 h.p. per unit for marine service.

The newcomer, that is to say the internal-combustion turbine, cannot be compared with the two established types of engine until such time as some idea of its first cost has been reasonably established and sufficient operating experience has demonstrated its reliability and cost of maintenance under the conditions of sea service. This state of affairs may not be too remote, as evidenced by the recent trials of the first marine unit (which, however, was not of Merchant Navy scantlings) and the possibility within the next year or so of trials of at least two merchant ships fitted with this type of propulsion. However, on paper, the internal-combustion turbine appears to be a formidable opponent, as will be seen from more detailed comparisons made later on.

Marine engineers are often accused of being unduly conservative in their selection of machinery. While there may be some truth in this at the present time, it would appear that this state of affairs is largely a legacy from the greatly reduced activity in shipbuilding between the two world wars, coupled with the development of large land power stations. It has not always been the case. The steam turbine was developed mainly for marine purposes, and going back further still, some of the first reciprocating plants were to be found on shipboard. It is suggested that at the present time the marine engineer has a great opportunity of at least keeping pace with his counterpart on land, particularly in the development of the internal combustion turbine.

Turning our attention to the question of transmissions, there appears to be a good deal of divided opinion as to the relative merits of the different types. It is thought it will be agreed there is a growing feeling that the astern steam turbine is likely to be replaced by other means of reversing. If not, Dr. Brown would probably not have considered it necessary to put forward alternative reversing means in his single-cylinder turbine scheme. This state of affairs is likely to be accentuated, rather than the reverse, by the trend towards increasing steam temperatures. This factor causes great interest to be attached to reversible transmissions in the case of steam turbine drives, an interest which is emphasized by the necessity for some such arrangement also in the case of the internal-combustion turbine. With Diesel drive, reversibility of the engines does not affect their reliability, but here again increasing interest is attached to a non-mechanical form of transmission which isolates the torsional vibrations of the engine crankshaft system from the remainder of the shafting, facilitating the use of a multiplicity of engines running at speeds greater than that of the propeller, with the possibility of varying the number of engines in service according to the power requirements. Multiple engines with purely mechanical reduction gearing have been used, but Mr. Pounder considers that some form of non-mechanical transmission is very desirable in the present state of the art.

Apart from the full electric drive in which both reversibility and speed reduction are achieved, two types of transmission for use in conjunction with mechanical reduction gearing have been described, namely hydraulic and electric couplings. Of these, Dr. Brown prefers the hydraulic type on the grounds of cheapness, robustness, simplicity and high efficiency, whereas Mr. Pounder considers the electric type to be preferable on due consideration of these same factors. It may be, however, that experience has yet to be obtained with both these forms of couplings for the powers quoted. If these couplings are incorporated to cater for astern running, then a third alternative is the reversible-pitch propeller, which is proposed by Mr. Calderwood. Dr. Brown, on the other hand, dismisses this as lacking merit, yet one hears in certain quarters that the time may not be far distant when the fitting of reversing propellers will be quite common.

At this stage it is as well to compare the values of efficiency for the different transmissions. With the full electric drive losses of 7·5 per cent. up to 10 per cent. are quoted. The hydraulic and electric coupling losses are given as 2 and 3 per cent. respectively, to which must be added 1·5-2 per cent. reduction gear losses, making totals of, say, 4 and 5 per cent. respectively. The electric transmission would therefore appear to have an extra loss of from 2·5 to 6 per cent., a portion of which can be regained in some installations from reduced line shafting losses with the motors placed right aft, the alternative type of machinery being placed amidships. From information given by Messrs. Saunders and Turner, it would appear that the use of an astern turbine instead of reversal through hydraulic or electric couplings would not necessarily effect a reduction in the loss figures, owing to the effect on turbine efficiency of design changes in order to accommodate the astern wheels.

The type of drive advocated by Messrs. Belsey and Robinson has been described previously as long ago as 1938, and it is fair to assume that in the intervening years the details have been very thoroughly explored and perfected.

Nevertheless, although an equipment of 3,750 h.p. will be ready for trials in the near future, it has yet to be given the test of experience. No doubt makers of the gas turbine will give this their careful attention.

Coming now to a more detailed comparison of the main features of the various types of machinery, it need hardly be stated that, however detailed and comparable the information given, it is not possible to place the proposals in order of merit from consideration of the machinery alone. The choice of machinery to be installed in any particular vessel is influenced by factors, some of which have been mentioned in my earlier remarks, outside the scope of these papers, such as the service in which the vessel is to be engaged, the required operating schedules, port facilities for overhauls, and so on. Further, as Mr. Pounder has said, considerations such as weights of machinery seatings and bunkers, and space taken up by the bunkers, must all be taken into account.

Fuel Consumption.

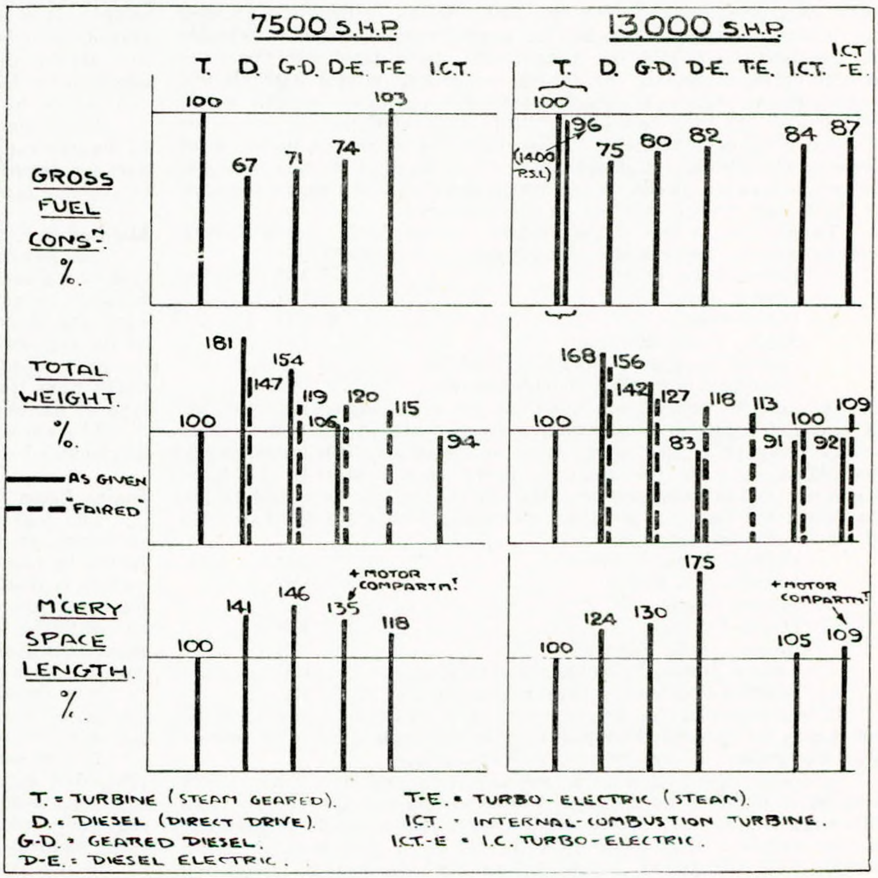
In comparing the fuel consumptions it should be noted that the same shaft horse-power values of 7,500 and 13,000 have been taken throughout, although the different proposals gave propeller speeds from 100 to 120 r.p.m. with corresponding different propulsive efficiencies, requiring different shaft powers for a given speed of the vessel. It is considered, however, that the installations could be re-designed slightly to give identical propeller speeds in all cases without affecting the fuel consumptions and, incidentally, with only a small change in machinery weights, so it has not been considered desirable to penalise a particular installation because the propeller speed selected was above the average, and vice versa. A factor which cannot be so readily dealt with, however, is the use of single and twin screws in the higher-powered installations. The inevitable loss of propulsive efficiency with twin screws must be debited against the direct-coupled and geared Diesel installations. In the case of the internal-combustion turbine, twin screws are specified on the score of reliability, but doubtless a single unit, showing a somewhat higher cycle efficiency, would be the logical selection when sufficient running experience had been gained to remove doubt on this point.

For the purpose of ready comparison the geared steam turbine proposals, numbers 1 and 2 of the first paper, read by Dr. Brown, have been taken as a basis having all-purpose consumptions of 100 per cent. Dr. Brown has shown in his third proposal that a saving of 4 per cent. is possible in the case of the higher-powered installation by the use of 1,400lb./sq. in. boiler pressure with two-stage re-heating but without increase in initial steam temperature.

On the above basis the turbo-electric propulsion, operating with slightly higher steam conditions, gives a consumption of 103 per cent. for the smaller power, that for the larger power depending upon whether one or two turbo-alternators are selected. The three Diesel engined proposals give consumptions of from 66 to 71 per cent. in the case of the smaller powers and 74 to 80 per cent. in the case of the larger. Of these figures, the lower apply to the direct and electric drives and the higher figures to the geared installation. It should be noted, however, that the figures for the Diesel-electric case do not include any allowance for auxiliaries at sea. It is not unreasonable to assume that as much auxiliary machinery will be required as in the case of the direct-drive and geared installations, for which 10 per cent of the main engine consumption was allowed. However, the use of electricity for these auxiliaries, taken directly from the main motors or the exciters will give a saving up to perhaps 20 per cent as compared with auxiliary generators running under partial load. On this basis it seems fair to add 8 per cent. to the consumptions for this case, making them 74 per cent. and 82 per cent. for the two powers respectively.

The average cost of Diesel fuel at present is 30 per cent. above that of boiler fuel, so that the relative Diesel engine fuel costs become roughly 90 and 100 per cent. of those of the geared turbine plant for the lower and higher powers respectively.

The consumption of the internal-combustion turbine installation is 87 per cent., so that if this can be maintained in service using boiler fuel, the costs will be the lowest. It is possible, however, that by the time this has been established in practice it will have become



Comparisons of the installation characteristics.

usual to run the direct-coupled type of Diesel engine also on boiler fuel.

Lubricating Oil Consumption.

The lubricating oil consumption varies from 1·7 per cent. of the fuel consumption by weight in the worst case of the Diesel-electric drive down to 0·1 per cent. with the internal-combustion turbine. Figures for steam turbines are not given but are stated to be very small. The cost of lubricating oil may, therefore, amount to an appreciable percentage addition to the fuel bill in the case of the high-speed Diesel engine. The slower direct-drive Diesel engines consume one-third to one-half the oil of the higher-speed engines.

Weight.

A fair comparison of the installation weights is unfortunately an extremely difficult task. By way of example, let us compare the geared Diesel and Diesel-electric cases. Although the combined weights of the main machinery and transmission are very nearly equal (the extra weight of electric transmission presumably being counter-balanced by the lighter higher-speed engines) the total installation weight with electric drive is only two-thirds of that with geared drive. A study of the itemised weights for the former, however, appears to show that such items as gratings, floor-plates, tanks and piping are not included. Items of this type amount to an appreciable proportion of the total as can be seen when it is considered that the combined weights of the main machinery and transmission vary from slightly over one-third to approximately one-half of the total installed weight.

Dr. Brown and Mr. Pounder have given lists of weights for both sizes of installation which, as far as can be seen, are complete and directly comparable. Mr. Calderwood's weights for the 13,000 horse-power internal-combustion turbine also appear to be comparable with these except for the possible omission of the funnel.

Messrs. Belsey and Robinson's weights for Diesel-electric machinery have already been commented on, and it would also appear that they have been a little optimistic in respect of some of the items listed; for instance, their propeller, tailshaft, and sterngear weight for the larger installation is 37 tons, whereas Mr. Calderwood's corresponding figure is 150 tons, in both cases with electric motors fitted right aft. Messrs. Saunders and Turner have not given complete installation weights for the turbo-electric drive, but, as they say, a direct comparison with the geared turbine drive is possible on the assumption that the weights of other items are the same in both cases, neglecting any saving in shafting weight with electric motors placed right aft. On this basis the aggregate weights would be 120 and 135 tons heavier than for the geared turbine drive for the two sizes of machinery. This difference is in reasonably good agreement with Mr. Calderwood's estimate of 103 tons for the difference between electrical and mechanical transmission in the case of the larger internal-combustion turbined vessel.

Taking the geared steam turbine as our basis, the following relative weights for complete installations are obtained:—

Direct-drive Diesel	170-180 per cent.
Geared Diesel	140-150 „ „
Diesel-Electric	85-103 „ „
Steam Turbo-Electric	120 „ „
Geared Internal Combustion Turbine ...	92 „ „
Internal Combustion Turbo-Electric ...	93 „ „

In order to smooth out some of the inconsistencies in weights of ancillary items, it is instructive to compare installation weights on the assumption that such items as sterngear, auxiliaries, pipes, floorplates, etc. are the same in every case, and only the main machinery and transmission weights differ. On this basis and taking the "constant" items as weighing 440 and 730 tons for the two sizes of machinery, we get:—

Geared Steam Turbine	100 per cent.
Direct-drive Diesel	150 „ „
Geared Diesel	122 „ „
Diesel-Electric	118 „ „
Steam Turbo-Electric	115 „ „
Geared Internal-Combustion Turbine	100 „ „
Internal-Combustion Turbo-Electric	109 „ „

It is appreciated that this comparison is still not strictly accurate, but it may be considered to give a somewhat truer picture by removing some greater discrepancies than it introduces.

It would appear, therefore, that the geared steam turbine is only equalled in lightness by the geared internal-combustion turbine. With either type of turbine, electric transmission adds some 10-15 per cent. to the weight. The direct-drive Diesel is 50 per cent. or more heavier than the geared steam turbine, and the geared Diesel and Diesel-electric machinery weights are intermediate between these two extremes, both of the order of 15-25 per cent. heavier than the geared steam turbine.

Space.

The relative lengths of engine rooms are as follows:—

Geared Steam Turbine	100 per cent.
Direct-drive Diesel	120-150 „ „
Geared Diesel	130-155 per cent.
Diesel-Electric	135-175 „ „
Geared Internal-Combustion Turbine ...	105 „ „
Internal-Combustion Turbo-Electric ...	110 „ „

The values for Diesel-electric installations are stated to be capable of considerable reduction at the expense of ease of accessibility.

Cost.

It is hardly surprising that little information as to relative costs is available at the present time. American experience shows that one type of turbo-electric vessel of about 7,500 h.p. cost, during the war period, 1·35 per cent. more than the corresponding geared turbine vessel. It is also stated that, in the case of Diesel engines, electric transmission becomes less than for direct drive for powers of 7,500 and above, and that there is little difference between geared and direct-drive Diesels at these powers. The big missing link is a comparison between modern steam and Diesel power units.

The above headings are those under which direct comparison of figures can be made. Under the following headings some comparative features affecting the relative merits of the different installations are brought together.

Maintenance.

The present high price of labour brings the question of ease and cost of maintenance very much to the fore. Hence the argument put forward by protagonists of the multiple Diesel-engined installation with geared or electric transmission, namely that routine overhauls of the engines can be carried out in rotation by the engine room staff during a voyage, is one that merits attention. In any case the relatively light weights to be handled would be advantageous with this type of machinery.

The amount of maintenance required with electric transmission as compared with geared drives is stated to be very small.

Engine Room Personnel.

The degree of experience and training required by engine room personnel, as affected by the type of machinery, is a factor which cannot be neglected. An advantage put forward for multiple Diesel engines is that, owing to the flexibility of layout, it would be possible for a company to have a fleet of ships of different powers and types all with the same type of engine, thus enabling training to be standardised and simplified. This scheme would also reflect on the cost of spares and maintenance.

It is stated that electric transmission does not require a staff with specialised training, but it does give a high degree of comfort in the engine room.

The tendency towards higher steam pressures and temperatures, associated with boiler equipment which is not so tolerant to careless handling, would appear to be adding to the responsibilities of the engineers with this type of machinery, or else to their faith in automatic controls.

In this summing up an endeavour has been made to give a fair picture and at the same time to restrict the evidence to that given in the six papers under consideration. It will be appreciated, however, that there are other factors which have not been touched upon up to now. For example, criticisms have been levelled against Diesel engines and reduction gears on the score of vibration and noise, with corresponding discomfort and the possibility of expensive breakdowns. It is hoped than any additional aspects such as these will be brought out in the discussion.

Discussion

Dr. T. W. F. Brown (Member) in opening the discussion stated that he proposed to deal with the papers in the order printed in the TRANSACTIONS.

Dealing first with the direct coupled Diesel engine, under the heading of fuel consumption Mr. Pounder commented on the steady deterioration in the fuel consumption of turbine engined vessels, making the statement that it amounted to 1 per cent. for each year in the ship's life. Dr. Brown suggested that this was not a figure which could be used over a number of years, i.e. that in 20 years the consumption was not 20 per cent. greater. Presumably the main reason for deterioration was increase in clearances in glands and packing and he suggested that if a fraction of the money expended in keeping Diesel engines overhauled and operating was applied to turbine machinery there would be no such deterioration in fuel consumption as mentioned by Mr. Pounder. The cost of renewing packing, and he suggested that if a fraction of the money expended would be required to keep Diesel machinery running for say 10 years in which liners would require renewing, and if proper means were taken to maintain turbine machinery in good condition there should be no pronounced falling off in fuel consumption after a period of time, although it should be pointed out that the figures given in the paper on geared turbine machinery were such as could be relied on after the machinery had been in service for some time.

On the question of weights, it seemed strange to put forward four engines to fulfil the requirements for 7,500 h.p. machinery with a change in weight of only 6 per cent. In reading his paper Mr. Pounder did make an indication as to the type of engine which he preferred, and the speaker thought that it would have been possible to produce the paper without having four engines to choose from—two double acting and two single acting in each case. It was also noticeable that when the author came to engine the vessel requiring 13,000 h.p. he put forward twin screw machinery, although it was well-known that 13,000 h.p. in a single screw vessel gave a greater speed than 13,000 h.p. in a twin screw vessel of the same dimensions due to the change in propulsion coefficient.

Dr. Brown agreed with Mr. Pounder's demurrer on the question of what weight mattered to the shipowners and agreed to take the basis suggested by Mr. Pounder. In the drawings Mr. Pounder showed a shallow type of bedplate in each engine which would add weight to the basis given in the paper due to the greater amount of seating required. However, granting Mr. Pounder all that was possible in the matter of fuel consumption, it remained a fact that with the lighter initial weight of turbine machinery it would require more than 20 days steaming before the weights were equal on Mr. Pounder's basis. The matter of lubricating oil consumption had also to be considered, including oil which deteriorated in use. The geared turbine consumption would be a very small fraction of that required for the Diesel machinery.

Dr. Brown did not think it necessary to say very much about the geared Diesel because it was clear that the author was not strong in urging its use. The best he could do was to suggest that it was equal to the direct coupled Diesel in cost and consumption in the regions of power mentioned in this paper, and the speaker would suggest that this type of machinery would only have a special sphere of usefulness in ships where the question of head-room would be of great importance, e.g. cross channel vessels.

With reference to the Diesel-electric paper, the speaker stated that it appeared that the fuel consumption in tons per day was about 10 per cent. greater than that given for the engines only. This difference was presumably to make up for the transmission losses mentioned in the paper, but it would appear that the paper was very incomplete. There was no auxiliary scheme given although certain hints were given, and Dr. Brown could not believe that they would be driven by steam raised from the exhaust gas heat, especially as some of the auxiliaries might not be suitable for motor drive. The fuel consumption figures required correction in respect of the auxiliary drives. Similarly in the matter of the table, the engine room length was given as 32ft. but this was not a workable machinery arrangement and in addition had no provision for auxiliaries.

The idea of watch-keeping engineers overhauling main propulsion units was a novel one. In an engine room with a multiplicity of engines of this type it was going to be a sufficiently difficult job to look after the running of the main propulsion units. If the watch-keeping engineers could make models when on watch Dr. Brown suggested that they were more skilful than engineers known to him when he was at sea. He thought it better that the spare engine should be on shore and overhauled as requisite, and it would be important that the machinery arrangement was such that propelling engines could be taken in and out of the vessel readily. When an overhaul was required the change in the ship should only be switching of the engines. This would be one of the advantages which would accrue with this type of machinery providing the same type of machinery was used throughout the ships in a company's fleet. If however the author's ideas had any merit, then drawings and weights should have shown extra main propelling units to be carried to sea for overhaul when the other units were running, which would materially alter the space occupied and the weights given.

Referring to the paper on turbo-electric propulsion, it was clear that the paper was only written from the point of view of engineers not capable of being responsible for a complete installation. It was this fact which more than anything else had prevented complete weights and auxiliaries including boilers being put forward in the paper. It was stated on page 280 of the paper that with 500lb/sq. in. boiler pressure the steam consumption of a turbo feed pump was 7 per cent. of the boiler output when a ship was at full power. The speaker considered the figure to be excessive and was prepared to substantiate the figures given in his own paper on feed pump consumption. In consequence the rest of the argument in the paper arising from this statement which was erroneous became meaningless.

Under the question of astern power the authors stressed electric transmission when reversing, and said there was the usual mass of trouble with geared turbine machinery. If anyone went round and picked out at random the troubles quite a list could be made, but they were not common. The authors commented on the losses with gearing, but Dr. Brown suggested that he did not look in the right place which gave the gearing losses measured accurately, and referred to the Proceedings, American Society of Naval Engineers, 1941, where they would find a series of tests with double reduction gears; the tests of the high powered gears described there showed an efficiency of 98 per cent. as measured.

Under "Astern Power" pages 280 and 281 the authors stressed the advantages of electric transmission when reversing in that the momentum of the turbine rotor helped to slow up the shafting and propeller, and that since the turbine rotor never changed direction, when steam was re-admitted it met blading going in the right direction (and not the reverse direction as in the case of an astern turbine before the direction of rotation had changed). The same advantages and more could be claimed for the hydraulic system of transmission.

Making a general comment on this paper, Dr. Brown said that the paper was produced by people who could make turbo-alternators and motors, and eventually they appeared to apologise that they were unable to design the installation complete with auxiliaries, shafting, boilers and funnel. They simply talked about the turbo-alternators, the main motor, the necessary d.c. supply and a few of the auxiliaries with the closed feed system, and made the same ingenuous remark that the shafting would be the same for all the schemes, but they knew quite well that it would not be so. Diesel machinery shafting was considerably more in diameter than that for a geared turbine. The authors mentioned the gear ratio of 26 to 1 between alternator and motor. It was quite easy with gearing of the double reduction type to get ratios of more than 50 to 1, and one then had complete freedom in the matter of turbine revolutions. The authors were restricted to revolutions for the turbines of about 3,000 with the results reflected in the paper of a cumbersome design of comparatively heavy weight.

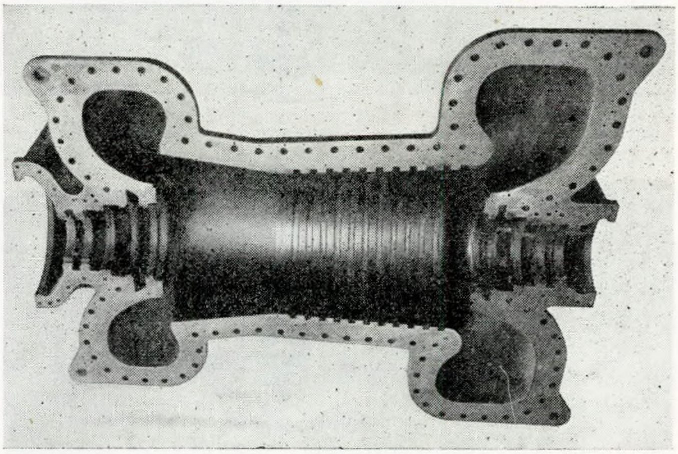
Referring to Mr. Calderwood's paper on the combustion turbine, the author stated (para. 1, page 289) that one or two gas turbines likely to be fitted into ships had been described in the Press, but that these were comparatively small units of moderate efficiency and could hardly represent the type of turbine that must eventually be developed for marine propulsion.

In addition to the units referred to by the author, Dr. Brown thought that it was necessary to draw attention to the gas turbine which Pametrada had under construction at the present time. This turbine, which had been designed specifically for marine use, operated on the open cycle and would develop 3,500 s.h.p. It was expected to have an efficiency at the turbine shaft of about 30 per cent. with a turbine inlet temperature of 1,200 deg. F. The unit was relatively simple both in construction and operation, and whilst more complex units of rather higher efficiency would no doubt be developed for higher powers, the turbine as at present designed was believed to give the best compromise possible for an installation of 3,500 s.h.p.

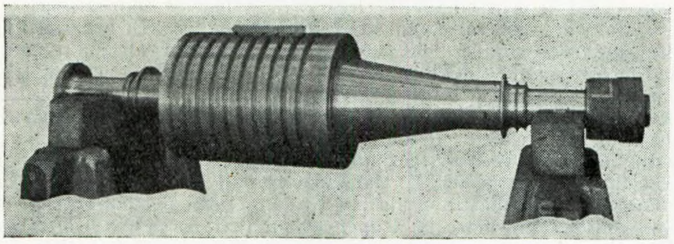
This 3,500 s.h.p. unit could be doubled in power by a comparatively simple supercharge arrangement by having an l.p. turbine drive a blower supplying air to the first stage compressor. The unit therefore in size of components, etc. was strictly comparable to that put forward by Mr. Calderwood. Pametrada had a comparatively short history but they had had the great advantage of being associated with a running gas turbine built for experimental purposes by Messrs. C. A. Parsons & Co. and The Parsons Marine Steam Turbine Co., and from this point of view had the same experience of the running of simple open cycle gas turbines as Sulzer Bros.

In the case of the marine gas turbine referred to above, it had a series arrangement of turbines with axial flow compressors driven by the h.p. turbine on an independent line to the power turbine. To show that the relative state of preparedness was somewhat about the same as that of the author, a number of slides, reproduced herein, were shown by the speaker, depicting the compressor cylinders and rotors, one of the turbine cylinders and both rotors. In the case of the author's gas turbine it was worth noting that only a single compressor rotor was shown in the paper.

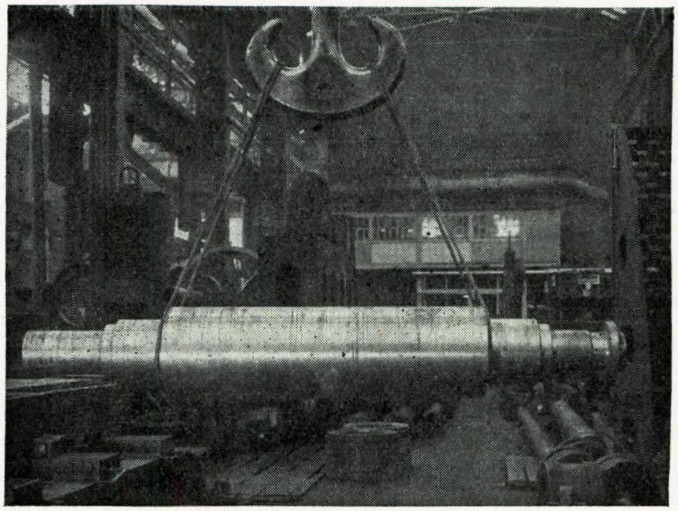
A number of changes had been made to the cycle put forward by Mr. Calderwood in his paper to the North East Coast Institution of Engineers and Shipbuilders about a year ago, and met the dis-



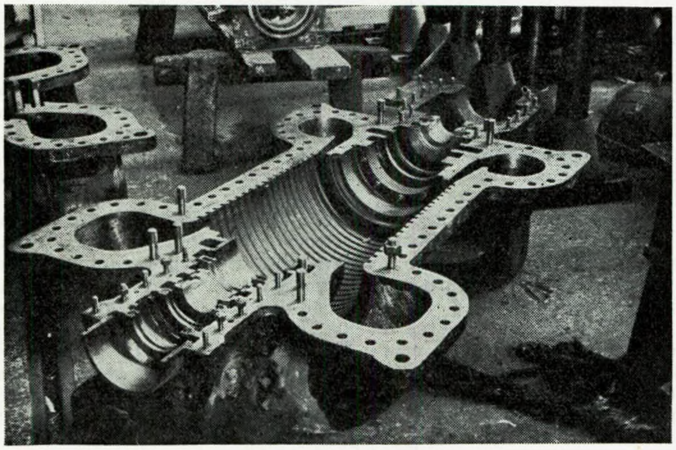
L.P. Compressor top half casing.



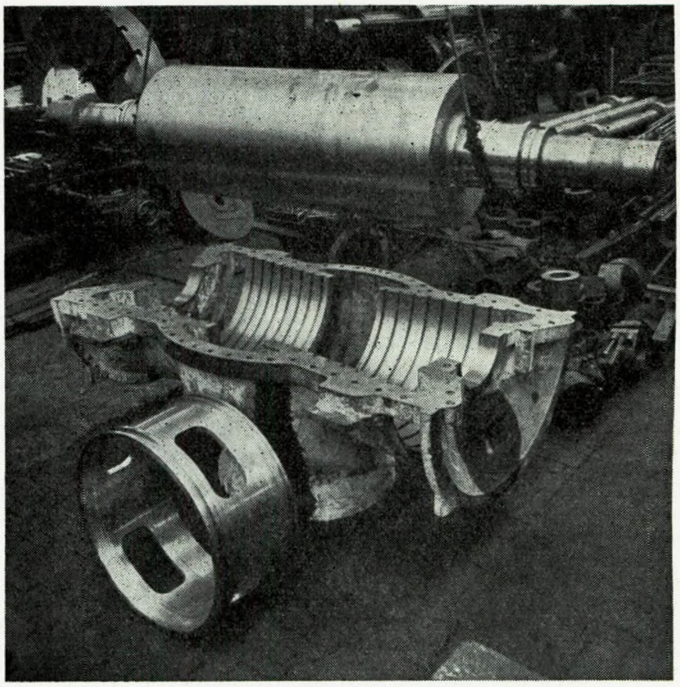
L.P. Compressor Rotor.



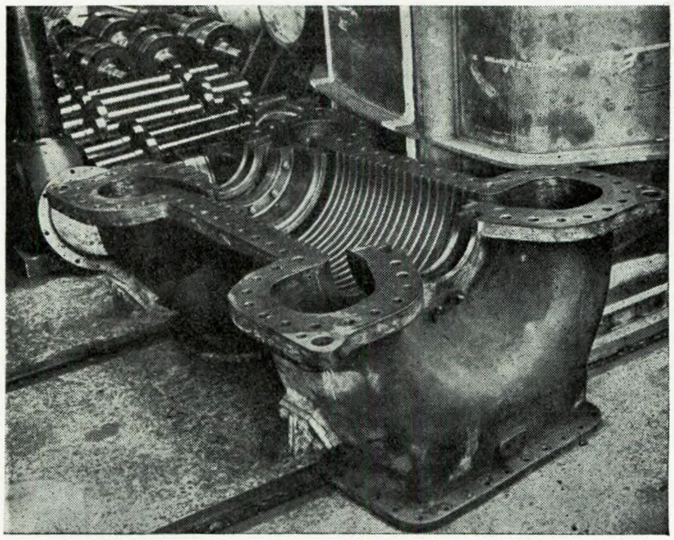
H.P. Turbine Rotor.



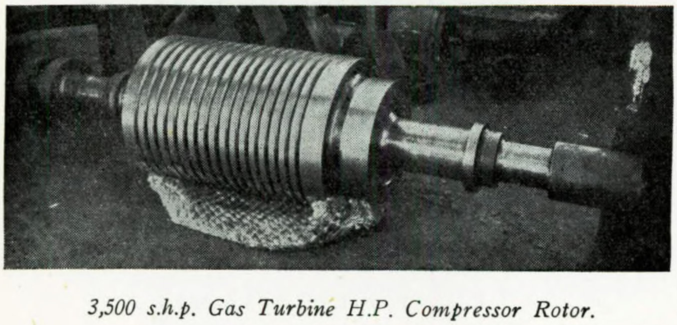
3,500 s.h.p. Gas Turbine H.P. Compressor. Cylinder bottom half.



L.P. Gas Turbine Rotor and half casing with control valve.



3,500 s.h.p. Gas Turbine H.P. Compressor. Cylinder—top half.



3,500 s.h.p. Gas Turbine H.P. Compressor Rotor.

advantages pointed out then by the speaker. The excessively high gas temperature at entry to the air heater, which was 2,700 deg. F., was now about 1,930 deg. F. and it was considered that trouble might be anticipated from scaling of the tubeplate and tube ends at this temperature. The revised arrangement would probably improve the control of the unit at reduced powers, but as a compensating disadvantage the number of heat exchangers had been increased from two to three. As the author had stated, two of these heat exchangers operated under pressure, with a beneficial effect on the heat transfer rates, but it should not be overlooked that the total size of the three heat exchangers was not less than the size of the single heat exchanger required by the corresponding open cycle, so that the semi-closed cycle did not give any reduction in size in this respect. In addition, an extra drive turbine was present and the compressors had been split, although the author assured them a year ago that the question of matching of these compressors had been fully gone into.

The author made a point (bottom of page 290) of the fact that variations in ambient temperature had little effect on the performance of the semi-closed cycle. It would be true to say that all cycles having multi-stage compression with intercooling possessed this property to a greater or lesser degree. The criterion was not whether the cycle was of the open, closed or semi-closed type, but was the proportion of compressor work which was done in the first stage compressor. This was clear from the fact that a change in ambient air temperature affected only conditions in the first stage compressor, the temperatures at inlet to the remaining compressors being determined by the water temperatures in the intercoolers. Admittedly in the case of the semi-closed cycle the work done in the l.p. compressor was quite small, but the difference from other corresponding cycles was not great. Incidentally, he thought the author tended to under-estimate the effect of the variations in water temperature likely to be met with in practice. The author devoted quite a large portion of the paper to knocking down a phantom open cycle turbine having low compression ratios, but it might be asked who stated that the open cycle gas turbine was specifically tied to low compression ratios? The open cycle gas turbine could utilise high compression ratios and benefit in efficiency thereby.

The efficiency of 35 per cent. quoted by the author appeared to be dependent on assumptions regarding component efficiencies and/or pressure losses which were distinctly optimistic. The speaker was of the opinion that for operation over a long period of time a figure of 32 per cent. would be more reasonable for this cycle.

Perhaps the best method of assessing the merits of the semi-closed cycle was to compare its performance with that of the corresponding open cycle operating with a high pressure ratio. For such a cycle having the same number of rotating machines, but only one heat exchanger instead of three, it was estimated that an efficiency of 34 per cent. at the turbine shaft would be obtained with a turbine inlet temperature of 1,250 deg. F. This efficiency was based on the same assumption as gave 32 per cent. for the semi-closed cycle. The configuration the speaker had in mind would give a good part load efficiency, and the curve of efficiency against load would be very similar to that obtainable with the semi-closed cycle. As regards weight they estimated that the dry weight, excluding gearing of two sets to deliver a total of 13,000 s.h.p. would be about 170 tons or 150 tons depending on the cycle, which compared with the figure of 230 tons quoted by the author to the same demarcation. The sets would require approximately the same length of engine room, i.e. about 58ft.

Referring to the author's diagram, Fig. 1, it should be noted that the l.p. compressor designated 1 was in fact two compressors with an intercooler and similarly compressor 4 was two compressors with an intercooler. Even although the compressors might have a common rotor and common casing they remained two compressors, as the air was led out from one compressor through the intercooler and back into an inlet on the second compressor, the arrangement presumably being similar to that shown in the Escher Wyss paper, page 781, Transactions of the American Society of Mechanical Engineers, 1946. The author's five rotary machines should therefore read seven.

The basis of these statements were summarised in the following table:

The bug-bear of large low pressure ducts was not nearly as great as Mr. Calderwood seemed to suggest. There were only three ducts at atmospheric pressure and of these two could be kept extremely short; the third was longer but had no expansion problem in it. The only component containing gas at atmospheric pressure was the heat exchanger, and as had already been stated this component was no larger than the three heat exchangers required for the semi-closed cycle. The open cycle required fewer ducts and led to quite a simple engine room arrangement. It was admitted that the air rate drawn from the atmosphere was lower for the two cycles put forward by Mr. Calderwood, the first to the North East Coast Institution of Engineers and Shipbuilders and the second in this paper, but it should be remembered that very high air rates were flowing in the closed part of the cycle, the corresponding figures being:—Cycle to the N.E. Coast Institution, air rate in open part of cycle 14·4lb./s.h.p./hr., in closed part of cycle 49·7. Similar figures for the configurations given by Mr. Calderwood in this paper were 18lb. of air per s.h.p./hr. drawn from the atmosphere and 26·3lb. of air per s.h.p./hr. in the closed part of the cycle, the corresponding figures for the Pametrada cycles A and B being 31·9 and 36·7, i.e. considerably less than the total flows for the semi-closed cycles.

On page 291 the author discussed briefly various solutions to the problem of reversing. The system of transmission used in Pametrada gas turbines was that described in a previous paper in the symposium in connection with steam turbine machinery. Briefly it consisted of a double reduction gearbox incorporating a normal hydraulic coupling for ahead drive and a hydraulic reversing transformer for driving astern. To go astern one merely opened up the oil supply to the astern coupling and closed the supply to the ahead coupling. One of the important points in connection with this method of transmission was that the combined losses in the gearing and ahead coupling were much less than the losses associated with electrical transmission. The system also had a higher efficiency than that which used a reversible pitch propeller, besides having obvious advantages regarding reliability and upkeep. Why if Sulzers could use this method did they not put it forward?

Thus when comparing two installations with say equal efficiencies at the turbine shafts, one with hydraulic transmission and the other with some other type of transmission, it should be remembered that the installation having the hydraulic transmission would have the better fuel consumption relative to the power at the propeller and that after all was the important thing.

The question of fouling the turbine with combustion products which it was earlier claimed was reduced in the semi-closed cycle had now disappeared in the configuration given in this paper, as the gas went through two heat exchangers and two turbines, and as one of the turbines was the output turbine, the speaker suggested that there was now no special merit in the semi-closed cycle.

On the question of efficiency figures for gas turbine cycles, it might well have been that a lower efficiency in association with a

Cycle	No. of turbines	No. of compressors	No. of coolers	No. of heat exchangers	No. of combustion chambers	Dry wt. exc. gearing 13,000 s.h.p. set tons	Max. temp.	Effy. at turbine shaft		Effy. at propeller shaft	
								Sulzer estimate	Pametrada estimate	Sulzer estimate	Pametrada estimate
Sulzer: Cycle 1.	2	3	2	2	1	—	1,200° F. (2700)	32½%—35%	28%	30%—32%	25·8%
Cycle 2.	3	4	3	3	2	230	1,250° F. (1930)	35%	32·3%	32·2%	29·8%
Pametrada: Cycle A.	3	3	2	1	2	170	1,250° F.	—	33·8%	—	32·5%
Cycle B.						150	1,250° F.	—	29·9%	—	28·7%

The reason for the difference between efficiency at turbine shaft and efficiency at propeller shaft was that the figures had been calculated on a 92 per cent. efficiency for Mr. Calderwood's expressed preference for electrical reduction and in the case of Pametrada designs to incorporate double reduction gearing with the hydraulic reversing device having a higher efficiency. Cycle A was an arrangement with an h.p. turbine driving the h.p. and l.p. compressors, an output turbine and a supercharge turbine being in series, the supercharge turbine exhausting into the heat exchanger. Cycle B was an arrangement with the output turbine and supercharge turbine in parallel, the other arrangements being as before and with re-heat and heat exchange applied only to that part of the gas flow passing through the output turbine.

simple cycle was particularly suitable for certain types of marine machinery. The author mentioned the time required to start the Sulzer gas turbine and stated the time required to heat up some turbines to bear this out. In marine steam turbine machinery it was probably better to start up quickly from cold, and the great advantage of gas turbine machinery for say a cross channel vessel would be that its availability for use was very good and that the starting up time was merely a matter of minutes if designed properly with proper mass relationship between rotor and casing. On starting up clearances should be increased, and in the complex matter of rotor temperature stresses in association with centrifugal loading it would appear better to heat up fairly quickly, and in such particular instance the matter of fuel consumption would be minor in comparison with the question of availability and the lightness of the machinery.

Before concluding, the speaker desired to stress the fact that there was a great deal of gas turbine knowledge in this country and that developments were going on in different firms. It should not be necessary to use foreign designs. They were all engineers who could not hide or conceal the gas turbine machinery referred to by the author and by the speaker, which would both be tried on the test bed, and he expressed the hope that the best one might win.

Messrs. Belsey and **Robinson** stated that they had no comments to make about the other papers, but they would like to make a few remarks about Dr. Dorey's summing up and also Dr. Brown's analysis of the Diesel-electric paper.

Firstly, they had to rely on memory but they understood Dr. Dorey to say that a 3,750 s.h.p. Diesel electric drive was shortly going into service but experience would have to be gained before one could say whether the method was entirely satisfactory.

Such a reference would infer that Diesel electric drive was as yet untried and therefore unproved. They would like to refer Dr. Dorey to the technical information which had been published about the German ship "Wuppertal" of 6,800 s.h.p. and also the 16,000 s.h.p. "Strength Through Joy" ship, which had been in service several years.

Also there were in the U.S.A. seven submarine tender ships with eight prime movers generating 12,000 s.h.p. on two propellers giving the ship a speed of 17·2 knots at 15,000 tons displacement. One interesting point about these last mentioned ships was that with one engine running giving 1,590 h.p. the ship's speed was 10·2 knots.

With regard to Dr. Dorey's criticism of the weights, the authors had to make estimates of the gratings, pipes, etc., and no doubt Dr. Dorey was in a more favourable position to assess the figures than they, nevertheless, they did feel that where the prime movers were small the estimate of $\frac{1}{3}$ to $\frac{1}{2}$ of the total installation weight was more than it actually would be.

About propellers and shafting, it would appear that Dr. Dorey was confusing Mr. Calderwood's twin screw amidship installation with the authors' single screw arrangement placed right aft. The figures put forward in the Diesel electric paper were actual "as fitted" figures of a ship of similar power and type already in service. It would appear that the line of demarcation for the various items was not consistent in all the papers, and Dr. Dorey had possible thought that their *spare* cast iron propeller and tailshaft (weight 37 tons) was the sum total of all their stern gear. Actually the working propeller and shaft were included in the item "Propelling motor intermediate shaft, etc.".

Referring to Dr. Brown's comments about the electrical load allowed for the auxiliaries, if he would care to analyse the data given in Table 2, he would see that 330 kW and 320 kW had been allowed for the liner and tanker respectively, and here again these were maximum loads which were recorded from two ships actually in service.

It was not the authors' intention that the small steam dynamo which derived its energy from the heat remaining in the engine exhaust gases should take care of the auxiliary loads and it was felt that the wording made it clear that this machine could be installed to bring about added economy by assisting the auxiliary load.

With regard to the unworkable engine room, the authors put forward what they thought the Papers Committee asked for, and at the same time they made it perfectly clear that their idea of an engine room was twice the length of that shown and everything was based on this latter arrangement.

If Dr. Brown doubted that engineers at sea to-day did have time to make models, the authors would like to suggest that he should visit the exhibition of models arranged by the Anglo-Saxon Petroleum Company next year.

Finally, the authors had looked at Dr. Brown's model of the hydraulic reversing gear and they would like to know more about the size of pipes and type of control gear needed for a 13,000 s.h.p. installation.

Mr. T. Halliday Turner (Member) said that whilst congratulating Dr. Brown on the bold step taken in accepting the advantages of impulse turbines, he could not help feeling that the proposed arrangement, having a separate internal casing, would have been quite impossible with a re-action turbine. Unless the outer casing could be trusted to be free from distortion, due to the vacuum load, it would be a bad design, even for an impulse turbine. He said there was no certainty that any distortions would be symmetrical, and however carefully bracket supports and key guides were adjusted, the internal casing would be drawn out of line with the shaft if the outer casing became distorted. He suggested that making the outer casing a strength member did not seem to be economical of material, which was better applied as seating girders, which permitted the turbine to have a 3-point support arranged without stressing the casing barrel.

He stated that the exhaust flange area of the turbine illustrated in Fig. 2 was deceiving, and if the main exhaust flow was sketched in on the arrangement drawing in Fig. 1, it would be observed that the distribution of steam over the total condensing surface was not good.

The locked train gears appeared to have been adopted only for the single casing turbine, and he asked if the advantages claimed for this design of gears would also be available for the larger sizes. He also asked if it was certain that this arrangement would not increase the noise in the engine room. He added that the couplings were a neat way of obtaining astern power, but he had some doubts as to the heat to be removed, particularly when stalling the turbine during manœuvring in the manner indicated by Dr. Brown.

As Mr. Belsey remarked, it might be somewhat difficult to make effective space available for the oil pipe system.

The steam flow diagrams were exceptionally useful, but it was noted, as Dr. Brown remarked, that there were discrepancies between their ideas. He assured Dr. Brown that the percentage steam consumptions he gave for the feed pumps were those which had been used though the pumps specified for that particular job had a somewhat larger margin than usual. Dr. Brown's steam flow diagrams disclosed extraordinarily low steam consumptions for the feed pumps. He asked if Dr. Brown would indicate what kind of feed pump had been adopted to get so low a steam consumption. It seemed odd to reduce the feed pump exhaust quantities so much, when the auxiliary turbine generator had to be arranged for back pressure operation to increase its exhaust quantity.

Mention was made of the advantages of a pass-out design for the auxiliary turbine generator, but Table 11 appeared to indicate that the auxiliary turbines chosen were condensing machines designed for 27in. vacuum, but running at a limited load exhausting at 20lb. pressure. This reduction of load could, of course, be avoided by using a pass-out design, and a different ship might not be able to do with a reduced load at sea. In the case of an oil tanker it had been found that so far from being able to use a pass-out turbine, a pass-in turbine was necessary as there was always a surplus of l.p. steam, and a great improvement was made by admitting the surplus through mixed pressure governor gear, obtaining useful work from it in the l.p. part of the auxiliary turbine. This, of course, was only a pass-out turbine with a negative pass-out quantity.

A low quantity of make-up was indicated on Dr. Brown's steam flow diagrams, but the drains from the oil fuel heaters and galley, usually suspected of contamination, were led back to the main feed system. It was not usual to be permitted to do this on a high pressure intallation, and it was usually demanded that these drains go through an observation tank back to the double bottom fresh water tank. They were then evaporated before they entered the feed system as extra make-up. This raised the make-up in Fig. 3 from 200lb./hr. by 750lb./hr. to 950lb./hr., which was a more usual rate. This extra make-up, of course, would increase the fuel consumptions which had been indicated.

He added that in the case of an oil tanker the demand for heating steam was vastly greater, besides requiring a different arrangement of the low pressure system so as to isolate it completely in a separate contaminated steam circuit, and this with such heat as could be returned from the contaminated system completely altered the balance, and it would be quite different from that shown in Dr. Brown's flow diagrams.

He stated that Dr. Brown saw an advantage in driving auxiliaries from the main turbine which gave an overall improvement of fuel consumption, but the auxiliary demands did not fit very well when steaming at reduced speed and during standby. He said they had abandoned this even for lubricating oil pumps, being fully in sympathy with Messrs. Robinson and Belsey's opinion on this.

Referring to Mr. Pounder's remarks, Mr. Pounder mentioned the advantages he had with his direct access to shipyard figures. He asked if Mr. Pounder could say how much extra steel was put into the double bottom of a ship, using a large slow speed Diesel engine, over that required for a geared turbine installation, and added that his impression was that the double bottom was very much deeper and the plating much thicker. Further, he stated that the seating weight for turbo-electric was about 20 tons per set, and no extra strength required adding to the double bottom.

In regard to lubricating oils Dr. Dorey mentioned that the turbine makers gave no figures for lubricating oil consumption, claiming that it was insignificant. Mr. Pounder stated that lubricating oil consumptions ranged from 46 to 92 gallons per day. He asked Mr. Pounder if this was the daily make-up or did it include the oil found to be so polluted after a voyage that it had to be discarded; also, how much oil was carried as a first charge and reserve.

He claimed that turbo-electric had the lowest lubricating oil consumption of all, and gave figures from actual service. The ship for which the trial data had been given had two years continuous service, mostly in the Far East. The total initial charge was about 2,000 gallons, of which about 1,000 gallons only was in the circuit. The amount consumed had been one gallon per day, and no new oil had been obtained since the ship was first commissioned. The oil remained clean and bright and had only rarely had even centrifuging. On the same ship he saw a bucket of oil taken from the auxiliary Diesel engines which was so black and thick that it seemed suitable only for road mending.

He said that through the courtesy of Mr. Nelson, he was able to examine an American T.2 oil tanker, and saw in her log book that they had been recording a daily oil consumption of one gallon from a total charge of about the same as theirs.

Mr. J. Calderwood (Member of Council) said that he was very interested in the gear box design shown by Dr. Brown and would like to have some further information as to the principle on which it worked. Information would also be of interest on the question of possible heating up of the couplings. There were two particular conditions that it would seem might give rise to difficulty, one was when running ahead the astern coupling would be turning at its full maximum speed but would be empty of fluid. The other difficult condition would appear to be a long period of running astern. The efficiency of an ordinary hydraulic coupling as used for the ahead side was very high so that the heat to be dissipated was not great, but there was no indication of the efficiency of the converter used for astern running and if, as might be expected, the efficiency of this unit was low, then there would be a great deal of heat to be dissipated from a very small coupling.

Dr. Brown made a particular reference to the question of astern horsepower. In general, however, this was a less important point than astern torque and it would be better with any type of machinery to sacrifice something in horsepower astern if by doing so greater astern torque could be assured.

Referring to Mr. Pounder's paper, he said that he would like to associate himself with Mr. Pounder's remarks regarding the correct comparisons of weights and other particulars as between the figures given in the various papers. Reading the figures simply as they stood in the various tabulations one would almost get the impression that Mr. Pounder had been trying to make out the worst case for the Diesel engine as his weights appeared so high in comparison with those given in some of the others' papers. A careful comparison, however, showed that Mr. Pounder had included every item of a very comprehensive machinery specification, whereas in none of the others was the specification quite so complete and in some, very many of the weights included by Mr. Pounder had been omitted.

Dr. Dorey in his summing up had attempted to make a table of faired weights and so had gone some way towards what Mr. Pounder called for, but the only way of getting a true comparison of all the types in respect of weight and technical particulars would be to tabulate for every one an equally detailed and complete specification as shown in Mr. Pounder's table on page 273.

Dr. Dorey's method of assuming that all such items as stern gear, auxiliaries, pipes, floor plates, etc. were the same in every case does not quite make a true comparison. This auxiliary equipment is, of course, lighter on a combustion turbine or a Diesel engine than on a steam installation.

Turning to the question of weights in Mr. Pounder's paper, he did not think that Mr. Pounder had put forward the best possible case for the direct coupled engine, as the main engine weights seemed high, partly no doubt due to the low propeller speed chosen. However, for a twin screw ship of this class there would be little, if any disadvantage in going up to a propeller speed of 120 r.p.m. and certain other types of two stroke single acting Diesel engine working at a somewhat lower b.m.e.p. rating than chosen by Mr. Pounder but at a speed of 120 r.p.m. could show a saving in weight of approximately 65 tons over the lightest single screw figure given by Mr. Pounder or 110 tons over the lightest twin screw proposal shown in his table.

Mr. Pounder referred at some length to the question of scavenge trunk fires but in this he was surely raising a scare. Many years ago scavenge trunk fires were not uncommon for certain engine types but they were practically unknown of recent years in some types of two stroke engines.

Referring to the question of overhauling times, in the paper by Messrs. Belsey and Robinson, the times given appeared to be very liberal and to have allowed for carrying out the jobs under the worst possible conditions, i.e. when working at sea on an individual repair where the man had to hunt out tools before starting work. If, as was suggested, overhauling work was carried out regularly at sea on a Diesel electric installation, then there would be regular drill and the times for all the operations could be very substantially reduced. In the case of the Sulzer engine for example, the times for removing the various parts could, in regular organised routine overhaul, be reduced at least to one-quarter of the times given in the paper. Incidentally it might be pointed out that on the Sulzer engine proposed in the paper there were no inlet and exhaust valves and presumably the time mentioned in respect of these for that type of engine referred to the scavenge valves.

In respect of the MAN engine included in the Diesel electric paper it was mentioned that this engine had uncooled pistons. He said that in his experience no engine of this power per cylinder had been known to give satisfactory continuous service working without piston cooling.

In general the rating of the MAN engine included in the Diesel electric proposals seemed high by comparison with the other two types, and it seemed doubtful whether this was a type of engine that had been developed for continuous heavy duty in cargo ships.

Mr. J. Lugt (Visitor) said that they should be very thankful to Dr. Dorey for giving "a symposium on the symposium", and for making clear in a graphical way the various figures given in the papers concerning fuel consumptions, weights, and so on. All this looked like statistics drawn up in the manner of prices of commodities and cost of living figures of the present time compared with say, those for the year 1938 or 1939. One had to have a certain yard-stick to measure by. He suggested, however, that they should not look at Dr. Dorey's graph in this way as statistics were only useful if they were given for a large number of engines. For instance in Mr. Calderwood's paper they were only considering one type of turbine, and in Mr. Pounder's paper only one type of Diesel engine. He finished his remarks by telling an amusing story indicating the danger of making statistics.

Mr. J. W. Howard (Visitor) said that in Dr. Dorey's résumé of the symposium direct comparison of the various competing systems of propulsion had been made on the basis of the weights quoted by the various authors. In making such a comparison it was only fair to point out that an important factor was the circumstances prevailing at the time the designs were made.

For example, in the case of turbo-electric drives, Paper V, the installation described by the authors referred to four ships ordered by the Admiralty under war conditions for service on special and important duties in the Far East zone of operations. As a consequence in the design of the propulsion machinery emphasis was properly laid on reliability and freedom from trouble somewhat regardless of weight. It was doubtful whether similar conditions were the deciding factors in the weights quoted by the other authors.

In the system described by the authors of Paper V the auxiliaries were driven by d.c. motors and the a.c. electrical system was confined to a purely local circuit between the turbo alternator and the synchronous motor driving the propeller, the frequency chosen being approximately 50 cycles. It was well known that in the case of electrical machines the weight tended to decrease with increasing frequency, so that if weight was the controlling factor obviously the frequency in this local circuit should be kept as high as possible.

The situation was not so open for ships where the auxiliaries were driven by a.c. motors. In this case it was customary to arrange the frequency of the propeller machinery circuits at approximately the same frequency as that of the ships auxiliaries, so that in case of need the two a.c. systems could be paralleled. For instance, the large number of American ships referred to by the authors built during the war with a.c. auxiliaries had a frequency of 60 cycles on both the propulsion units and the auxiliaries.

For the scheme described by the authors, therefore, where no external limits were concerned, if minimum weights were required it appeared feasible to raise the frequency to 75 cycles giving a turbine speed of 4,500 r.p.m. with consequent reduction in weight on both the turbine and alternator as compared with the authors' speed of 3,120 r.p.m. At 60 cycles, the corresponding speed was 3,600 r.p.m.

The use of such a high speed brought into prominence a feature stressed by the authors that the first critical speed of the alternator turbine unit should be 20 per cent. higher than the maximum running speed. It was desirable that this statement should be given without ambiguity and the authors were, therefore, asked to state whether the 20 per cent. referred to the maximum speed in service or to the overspeed for which the unit was tested.

The contention that the first critical speed should be 20 per cent. above the running speed although obtained on the ship's plant described in Paper V was by no means generally accepted and expressions of opinion on the subject by other members would be most valuable. For the standard land alternator at 3,000 r.p.m. the first critical was usually about 1,800-2,000 r.p.m. To raise this first critical to 20 per cent. higher than the running speed necessitated an abnormal design which in general was heavier and more expensive than that of a normal standard alternator.

If minimum weights were required, therefore, it would appear that a reasonable compromise would be to design the steam turbine with its fine clearances with the first critical speed 20 per cent. above the running speed but to allow the alternator with its large air gap to run through the first critical in attaining its normal operating speed.

Mr. W. Nithsdale, B.Sc. (Local Vice-President, West Hartlepool), said that it was earnestly hoped that, from this most excellent group of papers, it would be possible within reasonable limits to define the types of marine machinery likely, during the next few years, to be best suited for the various ranges of power required. This should save owners the necessity of analysing variegated schemes, builders a multiplicity of tendering, and marine engineers much study that might prove redundant.

Having had responsible association with different marine engines and now being entirely free, he was not infrequently asked by owners for an independent opinion, but the position was extremely difficult to assess.

Between the two Great Wars, when coal was a commercial proposition for certain sea services, there seemed a tendency towards the following demarcation:—

Horse power per shaft.	*Machinery.*
1,000 - 2,000	Steam reciprocating and Scotch boilers *versus* some competition from Diesel engines.
2,000 - 4,000	Steam reciprocating and Scotch boilers, with reheat, or exhaust turbines *versus* strong competition from Diesel engines.
4,000 - 6,000	Diesel engines *versus* competition from geared steam turbines and water-tube boilers.
Above 6,000	Geared steam turbines and water-tube boilers *versus* slight competition from turbo-electric drive.

So much for the past, but what of the present? Before seeing these papers at all he had put down for his own edification, without bias or reference to any firm or firms, and purely from a personal unauthoritative angle, a comparison for the lower power of 7,500 s.h.p. specified by the Superintendent Engineers' Committee, between:—

A—Double reduction geared turbines and water-tube boilers. Either 2 cylinder with astern in h.p. and l.p. or 3 cylinder with astern in i.p. and l.p. Steam conditions 450lb/sq. in. and 750 deg. F.

B—Two cylinder gas turbines and double reduction gearing with reversible propeller.

C—Six cylinder direct coupled Diesel

with the following result:—

	A.	B.	C.
Steaming weight	800 tons	700 tons	1,000 tons
Length of machinery space	55ft.	55ft.	60ft.
Oil s.h.p./hr. propulsion only	·575lb.	·55lb.	·375lb.
Approximate price	£240,000	£280,000	£300,000

These figures were on a strictly comparable basis with corresponding auxiliaries electrically driven and included the necessary Diesel generators to serve also deck and refrigerating machinery. Everything for the complete machinery of a first class cargo liner was included, fitted on board, right down to workshop tools, spares, switchboard and wiring, and cost of trials.

Consumptions for A and B were boiler oil and for C Diesel oil.

The prices were what in his opinion owners would probably have to face when the respective sets were delivered ready to sail in minimum periods of:—

(a) 2 years; (b) 3 years; (c) 2½ years, from an order placed now. Basis prices would be lower but the future trend was still towards higher costs, and shortage of food would not improve production efficiency.

Mr. Pounder had undertaken that he would stand by his figures for any contract an owner cared to make, and while his (the speaker's) were completely devoid of responsible backing, he dared say he could lead a buyer to where, even in the case of the gas turbine, he could place an order very near the limits quoted by him.

It was emphasised that the foregoing comparison was on proper commercial lines and was by no means the best case that could be made out, especially for the turbine alternatives. For instance the steam conditions in A were really out of date. It was a good many years since the "Nieuw Amsterdam" went to sea with 600lb./sq. in. and 800 deg. F. steam, and surely now we should adopt at least 800lb./sq. in. and 850 deg. F. Moreover Dr. Brown put forward a much better proposition.

The gas turbine in B showed no phenomenal consumption, but was based on a gas temperature which could safely be adopted for long service with present materials. Development of special metals for higher initial temperatures would bring correspondingly improved consumptions.

While the turbines in A and B would show to better advantage at higher powers, the Diesel engine in C approached the limit of present size for a single unit. On the other hand this was not the whole story, and Mr. Pounder had rightly called attention to supplementary factors. One, so far not mentioned, against the steam turbine, was that considerably more fresh water should be carried for make-up feed for the water-tube boilers than was required for the auxiliary boiler in the gas turbine and Diesel comparisons.

Naturally the authors were shy of prices, but he would be grateful for any comments on them. They would expect Mr. Pounder, with no lack of potatoes confronting Belfast, to say the Diesel price was too high!

However, he did not think £300 per ton all in for the steam turbine and Diesel sets was far out, and £400 per ton would seem reasonable for a first gas turbine installation where the high temperature section was costly and compressors expensive. At the same time the high temperature portion was only a small fraction of the whole, and heat exchangers, which formed a large part of a gas turbine plant, were comparatively cheap to make. With experience he would expect gas turbine machinery to cost little more per ton than steam or Diesel, and under normal conditions time of manufacture should be about the same for all three.

As regards turbo-electric propulsion, his only limited experience had shown cost, weight, and consumption all to be somewhat more than the straight geared turbine. Price limits, however, were not beyond the commercial range of a buyer's market, i.e. the best turbo-electric proposal would compete with an inferior geared turbine.

Diesel-electric propulsion certainly merited consideration for special cases, such as shallow draught vessels.

To sum up the whole immediate outlook he would say the price of coal had placed a big obstacle in the way of the reliable steam reciprocator with Scotch boilers. Consequently, although the old steam combination by no means was yet finished, for powers up to 4,000 h.p. per shaft the Diesel engine held a strong position which extended in only slightly lesser degree up to 6,000 h.p.

Above 6,000 s.h.p. he would advocate the geared steam turbine and would recommend the same machinery down to at any rate 4,000 s.h.p. if the turbine and water-tube boiler makers would pay more attention to this lower range. Many of them, now that warship building was again restricted, should be free to do so.

While there was such a bottleneck for electrical equipment on land (whereas more than ample gear cutting capacity existed in the country), surely schemes with electric drive involved prohibitive deliveries, apart from other considerations in the meantime.

If, as seemed probable, Dr. Brown and others succeeded in developing a good reverse gear, the steam turbine would forge further ahead and the gas turbine get a better chance.

Engineer Rear-Admiral W. R. Parnall, C.B., C.B.E. (Member) said that it would be of great interest if Dr. Brown could state the efficiency ratio of each and all the various turbines which were described in his paper, or if that was not convenient, the efficiency ratio of the l.p. turbines in schemes (2) and (3) and of all the turbines in each scheme taken together.

In this connection one would like him to discuss what margin or scope there was for improvement of efficiency ratio by practicable change of design of marine turbines within the powers with which they had been dealing.

With regard to Mr. Pounder's demurrer against the terms of the symposium the speaker was in full agreement that the machinery comparison should take into account machinery weights plus weights of seatings, bunker steel weights, and weight of bunkers for any prescribed length of voyage and so on. In other words, one should not separate the machinery from its environment, and the environment was the ship. He did not wish to impugn anybody's opinions or methods, but it was certain that one approach to the problem was to conceive a ship—length, breadth, depth, and so on—and then to consider how one type of machinery or another would affect its carrying capacity. He would have thought that the proper basis of the problem of engining cargo vessels was to decide what weight and cubic capacity of cargo it was desired to transport, and the speed at which it was to be transported; the possible different types of machinery had their different characteristics as they had seen here, in respect of weight, space and requirements for fuel, water, stores, etc., and at the drawing board stage one would expect to see arranged about the specified provision for cargo and the specified speed, ships of dimensions slightly differing by reason of the characteristics of the different methods of propulsion.

Taking the figures of papers Nos. I and II as gospel, the overall differences which arose from the two methods of propulsion were small; neglecting for a moment anything except weight of machinery and fuel consumption per day, it would be observed that the machinery of 7,500 s.h.p. had a weight difference in favour of the steamer of about 485 tons and a fuel consumption in favour of the motor ship of 16 tons per day; that was to say that taking the weight of machinery and fuel together, equality on this basis was reached in a voyage of 30 days' passage. In the 13,000 s.h.p. ship the corresponding figure was 38 days.

He had not forgotten the other items to which Mr. Pounder referred, nor the necessity to provide some water for make-up in the case of steamers, which in these days of effective evaporator systems should not exceed 100 tons, but he contended that at the drawing board stage a slight adjustment one way or the other of a few frame spaces dependent upon the type of machinery being considered, sufficed to ensure the fulfilment of a specification to carry so much deadweight cargo at such a speed. Let him observe here that in the class of ships they had in mind, 1ft. in length corresponded to 40, 50, or even 60 tons of deadweight.

Examination of engine room lengths stated in these papers showed that the steamer required anything from 10 to 23ft. less length than the motor ship for machinery, and thus if a capacity cargo only was being considered, it would appear that for the same performance i.e. for a definite cargo at a definite speed, the steamer might be a slightly smaller ship.

The most potentially adaptable feature in the whole problem was the ship itself, and whether it turned out that the steamer should be smaller or greater than the motor ship. the original cost was affected only by a relatively small quantity of structural steel, which he supposed was the least significant factor in the price of a ship, and could be easily offset by difference in price of the machinery. He could not, therefore, unreservedly concur with Mr. Pounder when he referred to the factors which he had mentioned as necessarily operating in favour of the Diesel engine. If one approached the problem from the point of view of "Here's a ship, let's put an engine into it", he might agree, but if one approached the problem from the point of view of "Here's a cargo, let's carry it about the world", he did not.

Dr. W. A. Tuplin (Visitor) said that it had been interesting to examine the details of the gearing used in conjunction with the 13,000 s.h.p. installation described by Mr. Pounder. It was stated that the teeth were of involute form, hobbed and lapped. The tooth-form could certainly be approved as representing good modern practice. Nearly everyone who had had much experience of gear design had invented at least one tooth form only to find that it had already been tried by someone and found wanting. Others, less fortunate, carried out their own trials, but with only the same result. The involute tooth survived because its practical convenience outweighed the advantages that other tooth-forms had sometimes shown when working in ideal conditions that could not be maintained in practice.

The choice of the hobbing process for cutting the teeth was also one that must be commended as it was a method that could be relied upon to attain the necessary accuracy with economy.

One would prefer some alternative to lapping as a means of applying a small final correction to the teeth. The products of some of the older hobbing machines usually needed a subsequent finishing operation, but the use of a "running-in" oil for the first stages in the operating life of the gears was a preferable alternative to lapping. Application of the cross-axis shaving process after hobbing was the method now recommended of quickly improving even the finish produced by high-grade hobbing and of correcting some forms of slight error in the contact distribution on the teeth of wheel and pinion. There had been not a little progress since the time when a particular tooth-form was recommended because it produced so much relative sliding of mating teeth as to make lapping reasonably rapid.

It could, however, be said of lapping or running-in with suitable oil that resulting correction took into account any slight error in relative position of the shaft axes, and this led to the question of maintenance of the initial conditions through the operating life of the gears. Even in land installations, with heavy bedplates and foundations, doubt sometimes arose about stability over long periods, but there would seem to be greater uncertainty in the relatively flexible mountings in ships' hulls. Had Mr. Pounder even encountered conditions that suggested some possible value in three-point support for marine gear-cases?

The helix angle of the gear teeth was 23 deg. whereas the British Standard angle was 30 deg. In a comparison between gears cut by the same hob, the smaller helix angle led to a lower bending stress for a given torque on the gear, but the maximum surface stress (which was usually the limiting factor) was the same for both.

The normal pressure angle of 25 deg. was higher than the British Standard 20 deg., but this had little effect on load capacity. Relatively large pressure angles were permissible in gears with large numbers of teeth, but the British Standard pressure angle had to cover a wide range of conditions.

The tooth loading was specified as 1,490lb. per in., but such a figure had little meaning unless examined in conjunction with other factors. These were taken into account in the British Standard formulæ for loading capacity and the "expected life" of the gears under consideration, assuming them to be made from steel with carbon contents of about 0·55 per cent. and 0·45 per cent. for pinion and wheel respectively, was about 12 years whilst carrying 10 per cent. overload continuously.

The British Standard formulæ for load capacity of gears had been devised to cover gearing for all purposes. It was not uncommon for users of gears for particular duties to imagine that "standard" formulæ were not applicable in their cases, and that they needed special formulæ of their own. This was not the case, however. Careful study of operating conditions and determination of actual loading always caused what were apparently "special" cases to fall into line with general formulæ.

The engineering layman may have had the impression that the British Standards Institution was an austere assembly, aloof in detached Olympian majesty in Victoria Street, who were committed to the periodic issue of arbitrary laws for lesser breeds. That was however, quite untrue. In particular, the British Standard formulæ for load capacity of gears was the result of prolonged co-operative effort by a number of people with wide experience in manufacturing gears for all purposes. The formulæ could not be associated with the name of any individual, as was the case for example with the venerable Lewis formula. They were rules forced on to the British Standards Committee by the hard facts of long and sometimes bitter experience in many fields.

Pet theories that some members had formed and were disposed to cling to with passionate devotion had to be abandoned because they did not line up with established facts about gear performance. The standard formulæ were the result of co-operation even if it was for some persons only a reluctant co-operation with the inevitable. The development of the formulæ was emphatically *not* indeed the effort of a publicity-minded technician who set to work after a brisk preliminary rub of the hands with the aspiration of making the job, as the saying was "all my own". No one could truthfully claim that he devised the British Standard formulæ for the load capacity of gears. It was a fact, on the other hand, that the British Standard Specifications presented to the gear designer the concentrated benefit of a wealth of experience, and nothing better was known at present.

Mr. E. C. Warne (Member) said that Dr. Brown had referred to the suitability of turbines with reduction gear to run the propeller shaft at the reduced speed of 120 r.p.m. whereas the revolutions of direct-coupled Diesel machinery given by Mr. Pounder in his paper on this subject were very much lower.

With reference to the paper on the direct-coupled Diesel engine, Mr. Warne asked if the first of the proposals on page 257 was a proposition which it was intended to pursue, in view of the advantages of the "coverless" type of engine, which was a very distinct advance on the present design.

On the subject of the single-acting two-stroke engine which he illustrated on page 258, he asked for Mr. Pounder's reason for adopting the particular shape of combustion chamber which he had shown. One might have expected, for instance, a concave head on the exhaust piston valve, instead of the convex shape indicated by the section.

Mr. R. C. A. Dimmick (Visitor) said that he had listened with much interest to the rational approach which the authors had made to the subject of Diesel-electric propulsion using a.c. transmission. The system for successfully paralleling an unspecified number of alternators driven by relatively small, high speed Diesel engines had certainly overcome the limitation of s.h.p. at present existing with d.c. transmission. The way had been opened to Diesel-electric propulsion of 20,000 s.h.p. and higher. It was, however, a matter for conjecture whether full use of this wider application would be made, since, in the past few years, the gas turbine had, at least potentially, entered the field of ship propulsion.

Some considerable time must elapse before sufficient confidence would have been placed in the reliability of the gas turbine to make it the only type of prime mover in a ship's propelling installation, but eventually it was probable that the high speed Diesel engine would be confined to powers up to, say, 7,000 s.h.p., except for geared drives.

The gas turbine was not directly suitable for astern power, and for marine service, therefore, it would be associated with either electric transmission, reversible gearing and couplings, or reversible propellers. For the initial propulsion machinery installations using gas turbine prime movers exclusively, it was probable that electric transmission only would be favoured. Competition with the high speed Diesel engine would thereby be established, but instead of employing at least four generating units, rarely more than two would be needed, and certainly never more than two per propeller shaft. The paralleling of alternators on to a common bus-bar would, therefore, be avoided, as with existing turbo-electric schemes. The multi-Diesel engine lay-out could not readily be justified as an alternative except for powers below about 7,000 s.h.p., that is, within the region where the size of gas turbine units would be too small to operate at a competitive efficiency.

Regarding the authors' reticence to the use of a.c. for driving auxiliaries, he was a little surprised in view of their otherwise courageous attitude in the paper. He agreed that a.c. should be used with discretion, particularly when taken direct from the propulsion bus-bars, but he did not consider it in the best interests of advancement to exclude it altogether on the score of lack of suitable motors and control equipment. He admitted that a.c. was not ideal for all ship auxiliaries, and one would like to combine the best of a.c. and d.c. features, but in a paper introducing advanced thought he would have preferred a compromise by the use of some a.c. auxiliary drives. For example, those drives associated with the propelling machinery and for which the power demand varied directly as the prime mover speed might well be connected through a step-down transformer to the "synchronising" bus-bars. D.c. from the exciter sets at sea, and from the auxiliary generating sets in port, could be of limited capacity sufficient for drives with long range speed control. For the 7,500 s.h.p. scheme the refrigerated cargo spaces could be fitted with two-speed, change-pole, squirrel cage induction motors as in the "Hornby-Grange". These motors, to suit fan characteristics, operated at 230-volts, 3-phase, 100-cycles, which supply could be derived from alternators coupled to the exciter sets. The load would justify running one main alternator set in port during the cooling down period.

He would like to pay tribute to Mr. Pounder for his excellent papers, and also to say how gratified he was to note that he had selected the electro-magnetic coupling to provide the necessary "cushioning" between the Diesel engine and the gearing. He would, however, like to have seen a little more emphasis on the flexibility of control and rapidity of manœuvre when using electro-magnetic couplings with a multi-engine unit.

In this country, designs of electro-magnetic couplings had been available for over 11 years, but it was not until early in 1946 that the first contract was placed in Great Britain for an installation on board ship. With the Chairman's permission he would like to show on the screen an illustration of one of the first two electro-magnetic couplings ever to be built in this country for a ship's propelling machinery. See Fig. A. This coupling was for a single engine arrangement rated 1,520 s.h.p. at 300 r.p.m. It was of the single squirrel cage type, with a torque/slip characteristic approximating to that shown in Fig. 16, curve "Y", of the Diesel engine paper.

Excitation on the coupling was maintained during the whole of the reversing procedure on the engine, and in consequence the relative speed difference between the two coupling members, in other words the "slip" did not reach that at which the coupling torque curve was a maximum. The feature of this torque/slip curve was that should the propeller foul, the coupling drive fell right away as soon as the maximum torque was exceeded, thus protecting the Diesel engine.

The coupling illustrated had the d.c. poles on the outer member, which had the higher peripheral speed. The d.c. windings were, therefore, rather more effectively cooled than would obtain on the inner member. Also the magnetic flux distribution was better on the outer member. A further consideration was the weight distribution. He had found that when the inner member carried the d.c. poles its weight was approximately half the total coupling weight. When carrying the squirrel cage winding, however, the fraction of the total weight was about one-third. Diesel engine builders, he found, preferred this light inner member on their crankshaft.

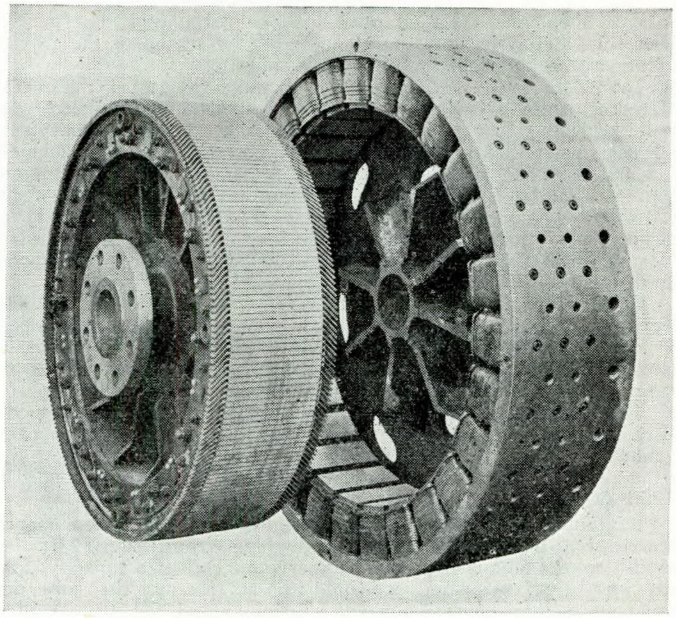


FIG. A.—*First British-built electro-magnetic marine coupling.*

The overhang of both members was kept as small as possible, and to that end, split slip rings might be provided and the brushgear mounted on the gearbox bearing cap. The side elevation in Fig. 3 of the paper on geared Diesel engines, and the subsequent views of engine lay-outs, were, therefore, unfavourable compared with recent practice in this country.

In conclusion, he wished to ask Mr. Pounder whether any further information could be given about the wear on the gearing of the largest geared engine set with which the author had had contact.

Mr. H. Sinclair (Visitor) said that in reference to Mr. Pounder's paper on the geared Diesel engine, the fluid coupling did not compare well in size with the magnetic coupling in the examples shown. Thus, in the 7,500 s.h.p. single screw vessel with a pair of engines of 220 r.p.m. the diameter of the magnetic coupling was 8ft. whereas that of a fluid coupling of the scoop-control type would be 11ft. In the alternative 7,500 s.h.p. vessel with a pair of engines at 300 r.p.m. the magnetic coupling was 6ft. diameter, whereas the corresponding scoop-control fluid coupling would be 8ft.

As regards the 13,000 s.h.p. twin screw vessel with a pair of engines on each shaft, running at 220 r.p.m. and alternatively 300 r.p.m. the foregoing comparative dimensions of coupling size also held good.

The inside of a fluid coupling was not, however, full of metal

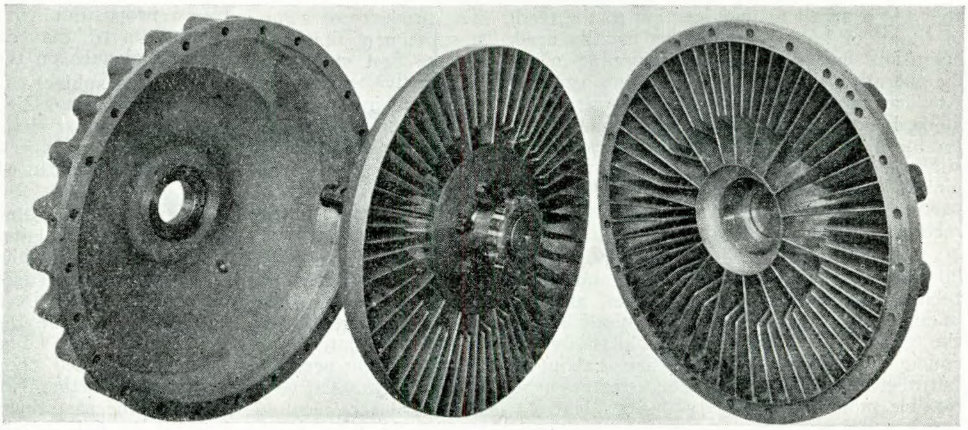


FIG. B.—*Vulcan-Sinclair fluid coupling with working circuit of the multi-wave baffle type.*

to the extent of a magnetic coupling, hence the weight factor would not be so high as these figures suggested.

Since the power transmitting capacity of a fluid coupling varied as the cube of the engine speed, it followed that with higher engine speeds the conditions became more favourable for the fluid coupling, though as regards the size factor for the engine speeds selected, he appreciated Mr. Pounder's preference for the magnetic coupling.

There were features in the design of modern fluid couplings that overcame shortcomings of the magnetic coupling referred to by Mr. Pounder. For example, the scoop-control coupling was almost independent of auxiliaries, and its construction avoided the heavy overhang on the crankshaft and pinion shaft due to the primary and secondary members of the magnetic coupling. Furthermore, the fluid coupling offered the possibility of a lower Wr^2 of the primary part rigidly connected to the engine crankshaft, and this was at times most important from the aspect of torsional vibrations.

Mr. Pounder had made the encouraging remark that whereas the hydraulic coupling in the examples shown was not so favourable, the time was overdue for the design of such couplings to be brought up-to-date, and made more favourable. Also, that the geared Diesel system could be made a sound combination equivalent in fuel economy with the direct Diesel, and more favourable as regards facility of overhaul.

With regard to Dr. Brown's paper on the geared steam turbine, it was interesting to see the fluid coupling come forward in the form proposed by Dr. Brown in association with a reversing converter, and a uni-directional turbine. The layout was similar to the transmission demonstrated first in 1924 in the M.S. "Vulcan", a geared Diesel vessel of 2,000 tons with non-reversing engines.

He suggested that the use of the term "astern coupling" employed by Dr. Brown, was less desirable than "astern converter", since the unit in question could not be a coupling of the two-element type. It had been for many years the practice internationally of those working on fluid transmissions to use the term "coupling" to identify the two-element type wherein no torque conversion took place, and the term "converter" for the three-element type wherein a stationary reaction member was introduced to change the torque ratio between the input and output shafts. Hence he hoped the more truly descriptive term "astern converter" would be adopted.

The transmission proposed by Dr. Brown reminded one of the introduction of the same type of ahead and astern drive in several geared Diesel vessels with non-reversing engines in 1925/1928: after which time it became the practice to use reversible engines instead of the reversing **converter.**

He believed there were several occasions where one or other of these vessels with non-reversing engines had gone aground and for some hours full power astern operation had been called for, thus bringing to the fore the amount of heat generated in the astern converter due to its low efficiency.

Dr. Brown had given in his example of a 7,500 s.h.p. turbine, a figure of 70 per cent. for the astern converter efficiency, and hence when going astern at full power the heat dissipated would be about 2,250 h.p. or 5½ million B.T.U's. per hour.

One had, of course, to bear in mind that the time interval during which the change between "ahead" and "astern" running of the vessel took place was relatively short, being only a matter of minutes, but even so, the rate of heating was severe.

If the vessel was going full speed "ahead" when the order to go "astern" was given, the turbine stop valve would be open, and the astern converter would come into action with the propeller rotating ahead: hence for maybe a minute the average rate of heating would be of the order of 10,000 h.p. dissipated in the transmission. This was acounted for by 7,500 h.p. input to the converter impeller due to the turbine at full power, plus the negative power at the converter runner arising during the continued forward rotation of the propeller. During this period, the rate of heating would be about 25 million B.T.U's. per hour.

As regards the fluid coupling for "ahead" propulsion, if this was of the kind shown in Dr. Brown's paper which had the proportions of the standard Vulcan coupling, attention had to be given to the occurrence of severe vortex surges when passing through the partially filled condition. It would appear that when changing from "ahead" to "astern" drive, and vice versa, there would be a period in which the conditions of torque and slip and partial filling of the fluid coupling while emptying or filling, would be such as to create heavy torque surges that would apply severe overload stresses on the reduction gears.

Information relating to torque surges in a fluid coupling of 1,000 h.p. with which the first unhappy experience of this trouble was gained, was given in a paper read before The Institution of Mechanical Engineers in April 1935. The problem was one that had since been solved in such a way that undesirable surges were eliminated and furthermore a valuable reduction in the stalling torque capacity had been attained without impairing the percentage slip at normal load. This result was achieved with the multi-vane baffle design of vortex circuit established in recent years.

It would seem that the best solution of the reverse gear problem was to use mechanical clutches of the synchro-self-shifting type which were reliable, easy to control and entailed no transmission loss in operation.

Major General A. E. Davidson, C.B., D.S.O. (Member) said that resulting from a recent visit to Scandinavia it would appear that Scandinavian shipowners and builders had settled for themselves the problem confronting the meeting in that they had all decided in favour of straight 2-stroke Diesel main drive with 4-stroke Diesel auxiliaries for both powers under review.

Possibly the choice of the straight Diesel drive was due to there being but little steam turbine industry in Scandinavia while electrical industry was fully employed on other work owing to electrical supplies from Germany having ceased.

The type of main Diesel engine in greatest favour appeared to be the one with exhaust valves in the head similar to Mr. Pounder's No. 4 design.

He was disappointed that so little had been said from the point of view of the shipowner. When he mentioned the Scandinavian position to a British shipowner, the latter had stated he had decided that a geared steam turbine with Diesel auxiliaries was the most suitable for his particular trade route. Dr. Dorey's slide on the screen appeared to sum up this owner's views, namely, that the cost of the turbine was so favourable while the shorter engine room and lesser weight together with lower maintenance costs outweighed the advantages of lower fuel costs and smaller engine room crew.

This owner did not agree that the consumption of the turbine went up with increasing age; in fact, the best efficiency was often not reached till five years' service, and with improvements to auxiliaries did not drop for twenty years. The same shipowner would be glad to have a proven reversible propeller so that he could fit a one-way steam turbine and realise a 10 per cent. better economy. Like others, he was not prepared to be the first to try out an 8,000 h.p. installation of this sort. He was, however, informed that a 19½ knot cargo liner with twin 7,000 h.p. Diesel engines fitted with reversible propellers was likely to be on test in 1948. The largest variable pitch propeller which he had himself seen was a 900 b.h.p. on a ship while doing her basin trials, and he had also handled a vessel with a ten year old 400 b.h.p. variable pitch propeller and found the complete control of the vessel from the bridge easy and exceedingly interesting.

Dr. R. W. Bailey (Visitor) said that Mr. Calderwood was to be commended for not having made any extravagant claims for the combustion turbine and also for having fairly displayed its present position, vis-a-vis established propulsion power plant. The statement in his second paragraph concerning the prospects of combustion turbines would be difficult to improve, and it should help those interested in applying combustion turbines to ship propulsion to keep their feet on the ground. A formidable task involving much development faced the combustion turbine in equalling, and more so in surpassing, the Diesel engine in the field of merchant ship propulsion. Steam turbine engineers had a very considerable respect for the Diesel engine applied to merchant ship propulsion and they were unlikely to underestimate its position and capabilities when examining the prospects of gas turbines. This they had been, and were, doing, not without finding encouraging features.

Commencing some years ago, when the steam turbine showed signs of losing ground to the Diesel engine in merchant ship propulsion, his firm examined the problem of what might be the most promising line of counter development. Although they favoured the conventional engine construction to the free piston type, the possibilities of combining a piston type gas generator and gas turbine, as later tried by Mr. Calderwood's firm, were closely analysed, some initial experimental work was carried out by Dr. Ricardo which was encouraging but it was difficult to see a competitive place for such a plant in the range of power normally required for merchant shipping. Their calculations showed that the overall thermal effiicency would be somewhat less than that of a comparable Diesel engine, but no question of high temperature was involved at the turbine, and they considered the gas temperature was unlikely to exceed 750 deg. F., unless quite novel means were employed to reduce the heat given up at the engine during exhaust. This indeed was one of the problems to be faced in increasing the thermal efficiency. Their examination led to the idea of combining steam and gas power. Here the case was found to be peculiarly favourable in a number of respects and he would like to deal briefly with the gas steam

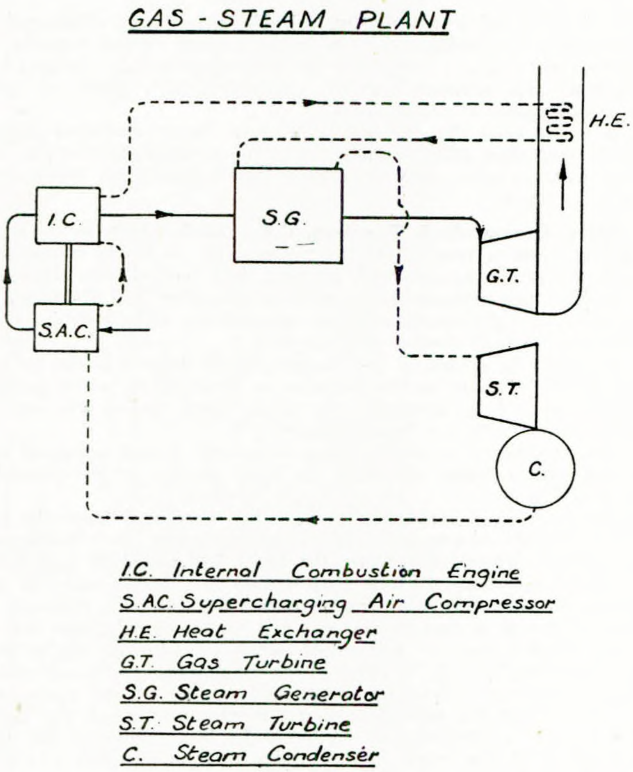


FIG. C.

arrangement as he believed it offered, at the present stage, important advantages over the straight combustion turbine, and good prospects of surpassing the Diesel on practically all counts at powers of 10,000 s.h.p. and above.

The principle of the plant in a simple form was shown by slide Fig. C. It comprised a compression ignition engine driving an air compressor which supercharged the engine, making it a self-acting gas generator, delivering gas at approximately 55lb./sq. in. (gauge) and 390 deg. C. (734 deg. F.). Only about 22 per cent. of the oxygen of the air supplied was burnt at the C.I. engine, so that the gas exhausted was capable of burning much more fuel—about three times the amount burnt at the C.I. engine. The gas passed to a pressure combustion steam generator where fuel was burnt, making the total fuel burnt up to 85 per cent. of the theoretical amount for complete combustion of the air used. The gas then passed to the gas turbine. The steam generated was employed in a steam turbine. Feed heating was carried out by heat taken up at coolers, at the jackets of the C.I. engine and at the gas turbine exhaust. Regarding this heat as a gift to the steam side, the thermal efficiency of the steam side, as expressed by the ratio of steam turbine power to fuel heat at the steam generator, came out very high, since in addition there was no stack loss at the steam generator. This steam efficiency would generally exceed that of the gas side and depending upon the circumstances might reach 40 per cent. The plant produced 20-30 per cent. of its full power on gas and the remainder, i.e. 70-80 per cent. on steam; with an overall thermal efficiency exceeding that of a long life combustion turbine, and with gas and steam temperatures within the capacity of established ferritic steels of steam turbine practice.

Slide Fig. D referred to plant for 12,000 s.h.p. on a single propeller shaft, under steam conditions, 800lb./sq.in. and 850 deg. F., which would be admissible, and showed the overall thermal efficiency of the plant against the gas turbine inlet temperature. A line was drawn representing a combustion gas turbine corresponding with the author's hoped-for 35 per cent. thermal efficiency, at 680 deg. C. (1,250 deg. F.) reduced by 9 per cent. to allow for double reduction gearing, the use of hydraulic couplings and auxiliaries, as had been assumed for the gas-steam plant. It was assumed that the quoted temperature of 680 deg. C. (1,250 deg. F.) referred to the gas before the first guide blades. For the gas-steam plant, if the gas turbine inlet temperature was 1,000 deg. F. (538 deg. C.) which would permit ferritic steel to be used throughout, the overall thermal efficiency, fuel to propeller shaft, would be 33·7 per cent.

It would be seen that the thermal efficiency of the gas-steam combination was much less influenced by gas turbine inlet temperature than was the combustion turbine. If, in the former case, the gas turbine inlet temperature was reduced from 1,000 deg. F. to 850 deg. F., the overall thermal efficiency would only fall from 33·7 per cent. to 33·1 per cent. For the latter efficiency the combustion turbine inlet temperature, according to the line on Fig. D would need to be 1,296 deg. F. i.e. 446 deg. F. higher than for the gas-steam plant.

A point which was of considerable importance was that with the gas-steam plant no special provision need be made to minimise heat given up at the C.I. engine. Also, the overall thermal efficiency was comparatively insensitive to air compressor efficiency if the air was cooled and the heat extracted was taken up on the steam side as could readily be done. The influence of compressor efficiency was shown by Slide Fig. E.

With regard to weight and space, without considering a case in detail it could be seen that there were several basic factors heavily in favour of the steam-gas plant compared with steam power alone, or with a gas turbine plant of comparable efficiency. The I.C. engine cylinder capacity was small and little more than of auxiliary order, as was indicated by the brake m.e.p. equivalent to the total power. This was about 550lb./sq. in. for a 12,000 s.h.p. plant.

The air compressor, if of the piston type, had been a problem, as no doubt Sulzers had realised. This receded into the background with the gas-steam scheme, because of the relative insensitivity of overall efficiency to air compressor efficiency. A light compact high

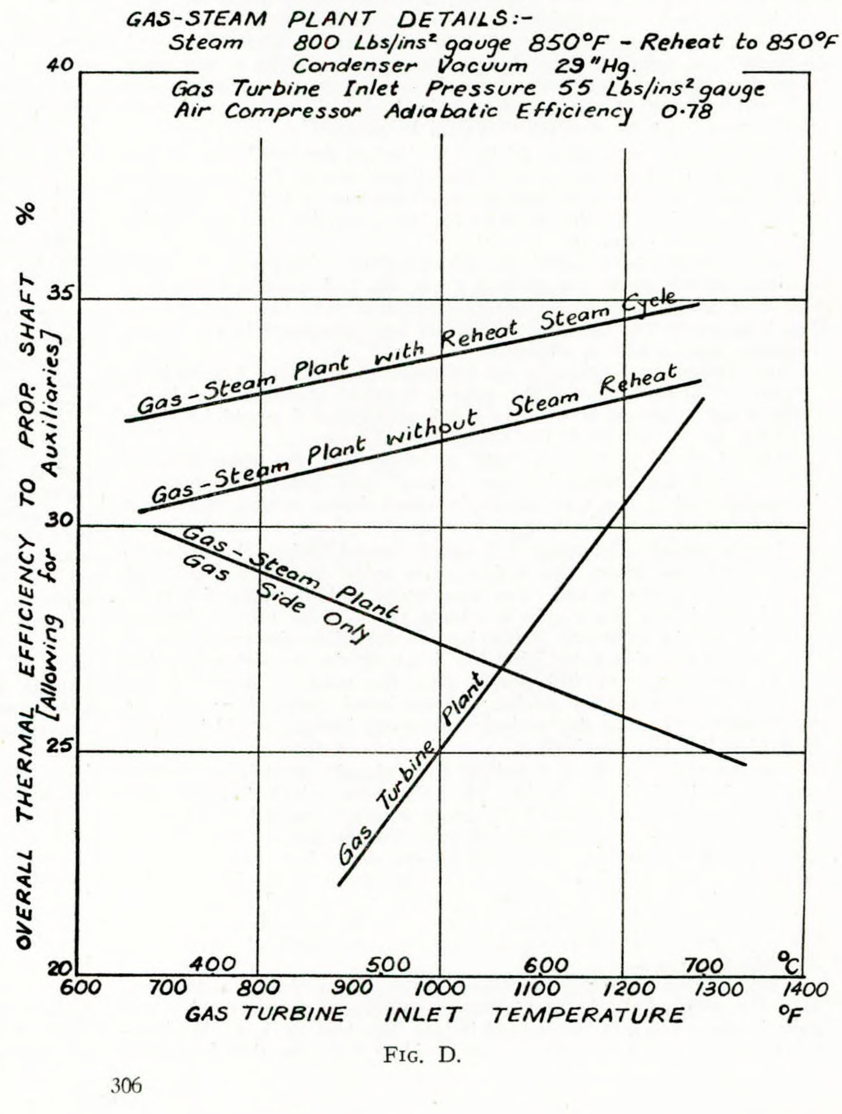


FIG. D.

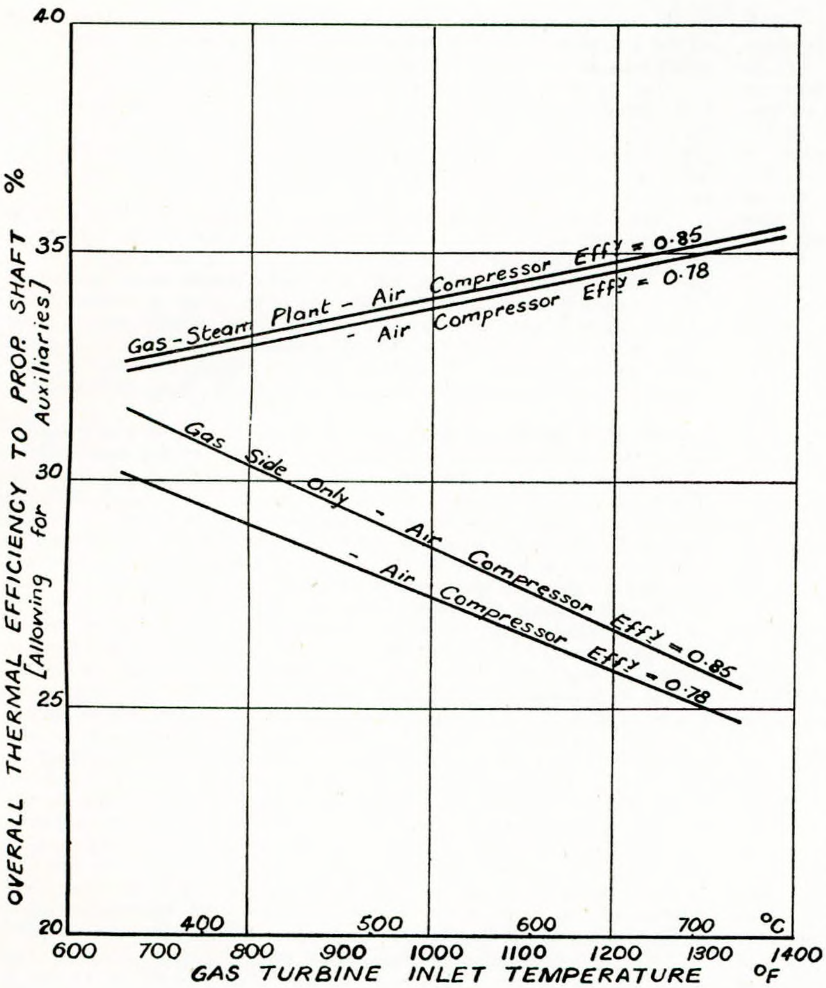


FIG. E.

speed axial or radial flow compressor was therefore suitable, thus reducing the compressor to quite secondary proportions and great reliability.

The steam generator was small and light for the following reasons:—

(a) Pressure combustion at a fair pressure.
(b) Heat transfer under a large mean temperature difference, most taking place at steam generating surfaces.
(c) Air pre-heater not required.
(d) Thermal efficiency of nearly 100 per cent.

Their estimate of the overall cylinder casing dimensions of the steam generator, superheater and steam reheater for the 12,000 s.h.p. case they had referred to, were: steam generator 6 feet dia. × 16 feet high; superheater 5 feet dia. × 13 feet; steam reheater 5 feet dia. × 13 feet. These three components would be assembled together near the turbines, reducing piping and conduits to a minimum. Steam reheating was simplified.

A case they had investigated where light weight and compactness were highly important indicated that for a 20,000 s.h.p. plant the specific dry weight for gas and steam generators, turbines, condenser, gearing was substantially less than 20lb./s.h.p. At this stage they believed that a more normal design such as would be favoured for the merchant ship the corresponding weight would not exceed 30lb./s.h.p.

Referring to auxiliaries, whilst Diesel-electric generating sets might be used for port needs, the case for using gas turbines for some of the auxiliary drives where steam turbines in a steam ship would be employed, appeared attractive, both on the score of economy and simplicity.

On the point of reliability, in the event of a failure of the gas turbine, the plant could continue under steam power by blowing off the gas. If desired the gas inside only could be used by by-passing the steam generator. In this event waste heat would be employed to generate sufficient steam for sealing cooling and maintaining a vacuum in the steam turbines.

The power of the gas turbine being only 20 per cent. to 30 per cent. of the total power, it would not present the same material, design, and production difficulties as would a gas turbine of full power and it could probably operate safely at a higher temperature.

With regard to further development, the plant they had considered could take advantage of higher temperatures. The development of successful reliable gas turbines of competitive thermal efficiency was bound to lead to higher operating temperatures for steam turbines. For their present purpose they had limited their steam temperature to 850 deg. F. and as they had stated, the gas turbine initial temperature could be the same, and the overall thermal efficiency fuel to s.h.p. including auxiliaries would, they believed, exceed 33 per cent. With very large power, and with further development, overall thermal efficiencies of 38 per cent. appeared feasible.

Mr. J. H. Trickey (Member) said that he could not agree with the remarks made by Major-General Davidson that the Scandinavians were entirely in favour of direct Diesel drive, as the "Werna", "Wiros", and "Freja" on the Stockholm-London run were all multi-engined with the drive through a single propeller shaft. He had recently visited M.V. "Saga" on the Stockholm-London service which had four engines of 1,700 b.h.p. each, driving a single propeller shaft through slip couplings and a reduction gear. He was informed that this ship was giving every satisfaction.

Previous speakers had mentioned the advantage of the multi-engine geared drive having the same unit size throughout the fleet and, most important, that it allowed absolute freedom in selecting the most suitable propeller speed for the economical performance of the ship.

In comparing geared and direct-coupled engines he would like to make one further point in favour of the geared engine. In these days when production was all-important the larger number of smaller engines which would be required for the geared system or Diesel electric propulsion would lend itself to speedier or more economical production. This would have the advantage of lower first costs in favour of the smaller units.

Mr. R. J. Welsh (Member said that he was currently connected with the manufacture of all the types of machinery dealt with in the symposium (with the exception of direct-drive Diesels) and was, therefore, able to make unbiassed comment on the various proposals.

Dr. Brown, he thought, deserved congratulation for the boldness of his proposals, which contrasted refreshingly with the unprogressive outlook formerly alleged to be associated with marine steam turbine development. It was, in fact, possible that Dr. Brown had gone too far in his Scheme 3, where he put forward a higher steam pressure than was normally used for so small a power even on land. The difference between the fuel consumptions of Scheme 2 and Scheme 3 was only 4 per cent., and a shipowner might well ask himself whether so small a saving was worth the risk of adopting machinery of such extreme characteristics.

Other investigators, and Dr. Brown himself as recently as January of this year (N.E. Coast Trans. 1947, Vol. 63, p. D.15), had expressed the view that the economic advantages of reheat were very small in installations of the size now under consideration. The sole justification for it was generally regarded as being the additional dryness it gave at the l.p. end. This was particularly true of steam reheat, regarding which it was frequently said that the use of steam to heat steam could never effect a thermal improvement.

It would be helpful if Dr. Brown could give some further explanation to support the change to his present view that reheat was of important economic benefit.

Other speakers had mentioned the question of heating in the hydraulic couplings and converters of Dr. Brown's reversing mechanism, and it would be interesting if figures could be given for the rate of oil circulation visualised and the oil temperature rise expected during full power stalled condition.

Mr. Halliday Turner had suggested that Dr. Brown's double casing design was impracticable but, by a coincidence, this design feature was employed by, amongst others, the General Electric Company of America. Apart from its cost and the difficulty of arranging for access to the bolts on the horizontal joint of the inner high pressure portion, it had much to commend it.

Messrs. Belsey and Robinson had selected engines of which two out of three were foreign. This seemed to require some explanation in view of the large number of excellent makes of British engines available. The selection had obviously not been done on the basis of satisfactory experience, as the terms in which the authors referred to the M.A.N. engine showed clearly that they were unfamiliar with it. The M.A.N. engine had apparently been chosen for its high performance, light weight, and relatively low speed. The association of light weight and low speed was, however, merely an indication of relatively flimsy construction. A much more reliable and robust engine would be obtained if the same weight of 33lb./h.p. were put into a unit of higher speed.

Higher speed engines would be more suitable for multi-engine drives, as the 18in. pistons suggested by the authors were too large to man-handle, and once one required a crane then the size of the lift became of minor significance. It was difficult to imagine 18in. pistons being withdrawn at sea for purely routine maintenance (as opposed to emergency conditions) and undoubtedly Dr. Brown was on the right lines when he suggested smaller engines suitable for easy complete removal for overhaul ashore. By the use of three-point mountings and quick-break connections, the removal of complete engines could be made surprisingly simple.

As Mr. Calderwood had already pointed out, the authors were on somewhat dangerous ground in praising air cooling for pistons of this size. Even with exhaust turbo-supercharging, which was of great help in piston cooling, the size was uncomfortably close to, if in fact it was not above, the limit at which air cooling could be regarded as thoroughly reliable.

The authors' proposals for retaining the engines in synchronism during idling periods involved doubling the number of main alternators. A lighter and simpler procedure was merely to permit the engines to drift out of phase if they wanted to; they would be found to fall into step again quite smoothly as soon as excitation was re-applied.

Mr. Calderwood was a friend whom everyone respected but his selection as an author in this symposium had the effect of emphasising the Swiss, as against the British, gas turbine developments. There were at least four British firms building marine gas turbines of their own design at the present time and in most cases the work was at a comparable state to that illustrated by Mr. Calderwood.

Mr. Calderwood was being less than just to H.M.M.G.B. 2009 in dismissing it lightly as merely a boat, not a ship. The same could have been said of the "Turbinia", yet look at the developments that arose from her. During these discussions H.M.M.G.B. 2009 had been described as being purely experimental, as being already out of date, and as not being a real marine job. All this might be true but those who had had the privilege of a demonstration run on the vessel would agree that if her performance was an example of what could be done with so imperfect a gas turbine, then the outlook for good gas turbines was extremely rosy. The starting up and bringing up to full power of the 2,500 h.p. unit on H.M.M.G.B. 2009 took fewer seconds than Mr. Calderwood's suggested minutes. Larger gas turbine machinery with longer life characteristics would not start so quickly, but with open cycle machines it ought to be possible to improve greatly on the figure of 90 minutes.

The open cycle gas turbine was not limited to the low pressure ratios of 3 : 1 and 5 : 1 suggested by Mr. Calderwood. These figures would apply not to a *simple* gas turbine but to one *with* a large heat exchanger but *without* intercooling or reheat. In actual fact it seemed possible, as Dr. Brown hinted, to make the open cycle more compact than the closed or semi-closed cycle, at least in powers up to about 15,000 h.p. In the closed cycles the savings in the sizes of the rotating parts tended to be more than offset by the size of the heat exchangers and air boilers; and even the small turbines had their own disadvantages, as in Mr. Calderwood's proposal where the speed of the small output turbine was too high for direct coupling to an electrical generator—thus introducing the cost and weight of reduction gears which would otherwise have been unnecessary.

The diagram of the Sulzer cycle was interesting in comparison with that included in the 1945 North East Coast Paper. One addition since that date was a reheat combustion chamber, and it would be interesting to know whether this had yet been tried under practical conditions. With a semi-closed cycle there was much less excess air available as secondary and tertiary air for the cooling of ordinary combustion chambers, and conditions were even worse for a reheat chamber where the air supply had already had part of its oxygen consumed, and such air as could be spared for wall cooling was, in any case, at a temperature where its value as a "cooling" medium was problematical. The validity of Mr. Calderwood's arguments about the unimportance of pressure drop in his heat exchangers was open to question as in Exchanger 9 the losses on the low pressure side would be much more important than those on the h.p. side; and the conditions on the l.p. side were atmospheric, exactly as in an open cycle machine.

The author's statement about the effect of ambient temperature had already been challenged by Dr. Brown but it was possible to go even further and point out that, with a pre-cooler at the compressor inlet to an open cycle machine, there would be absolutely no difference between it and a closed cycle so far as concerned sensitivity to ambient temperatures.

The future of marine gas turbines would probably be found to lie with sets of greater simplicity than those shown in the paper. It might be that for cargo vessels it would be some time before they could be fully justified, but they had a much more immediate application for cross-channel vessels and other ships in which it was of advantage to have a high power/weight ratio, a high power/space ratio, an absence of stand-by losses, simplicity, reliability and ample power in single units—all without any appreciable sacrifice of fuel economy, or any appreciable increase in first cost, noise or vibration.

Mr. H. F. Sherborne, M.C., M.A. (Associate) said he was speaking late in the discussion as befitted one who represented the hand maiden of the designers and builders, namely the suppliers of material. The papers and the discussion had been productive of much valuable information. He wished to make one or two observations in regard to Mr. Calderwood's paper in which any reference to materials to be used was conspicuous by its absence. It was not to be expected that Mr. Calderwood should attempt to deal with the points to which he desired to draw attention in his oral reply to the discussion, but he hoped the matter might be mentioned in the written reply. Mr. Calderwood had put on the screen the very interesting diagram shown on page 290. This was a diagrammatic arrangement of the Sulzer high-pressure combustion turbine cycle. He had referred in his remarks to "the difficult problem of the air heater" which had to be faced, and went on to say "This air heater is not only a large and complicated piece of equipment but it also reduces the efficiency of the plant". This is on page 289. Then on page 290 occurred the following: "A feature that should be specially noted is that in the primary heat exchanger the pressure on both sides of the tubes is the same, namely, the maximum pressure of the cycle. This gives a very high rate of heat transfer, resulting in small dimensions for this heater. It has the further advantage that there are no mechanical stresses of the tubes due to pressure. This is an important factor, as these tubes are at the highest temperature of any part of the plant".

He suggested that in regard to the material of which the tubes were to be made they were in the infancy stage, or to put it otherwise, they were groping their way. Most important questions of creep and of corrosion were involved. He did not know yet whether the answer was to lie in the ferrous or the non-ferrous field. Dr. Brown had told him that he hoped to see tubes abolished altogether in these heat exchangers, but he gathered that Dr. Brown did not intend to abolish tube makers altogether as, no doubt, he would want them to manufacture tubes for the condensers of his geared turbine installations. There were present in the room some of those veteran experts who twenty to twenty-five years ago had trodden the long, hard and painful path of corrosion troubles with condenser tubes. At that time there was not a material readily available to do what was required of it. We did not want a repetition of that sort of thing. He expressed the earnest hope that Mr. Calderwood would address himself quite directly to the problem of the material for the tubes of these heat exchangers and give the Institute the benefit of having the nature and scope of the problem clearly recorded.

Mr. H. T. Meadows, D.S.C. (Member) said that in Messrs. Belsey and Robinson's paper another case had been made for the multi-engined Diesel-electric ship, this time on the score of making an engineer officer's life at sea more congenial, and thus presumably to make the profession more attractive. He suggested that this object might well be defeated, because, no matter how palatial an engine room might be, the marine engineer disliked repetition work, and it was this very dislike which was one of the fundamental differences between the potential marine engineer and the fitter and turner. Some years ago a particular motor ship had great difficulty in keeping junior engineers, their complaint being that their whole lives were spent between the top platform and the workshop, refitting inlet and exhaust valves. This ship had only 64 inlet and exhaust valves; one wondered whether the less physical effort required to refit the 252 inlet and exhaust valves entailed in one of the schemes suggested by the authors would not be outweighed by the monotony of such refitting.

It was claimed that major overhauling and servicing could be done during the sea passage, thus enabling the engineer to go ashore while in harbour, such work being done concurrently with the watch-keeping duties. It was the experience of many that no matter how much work one did on watch, the Second Engineer was always able to keep one fully occupied in harbour. Further, one wondered whether the engine-room at sea would be as pleasant as it was made out to be, and whether in fact verbal orders could be heard, in the presence of seven M.A.N. supercharged engines, or was it the intention to duct air to the blowers? It would seem, therefore, that the advantages from a personnel point of view were doubtful.

A recognised disadvantage of the Diesel engine compared with the steam turbine was the cost of maintenance, due to the number of moving parts. One reason why the two-stroke cycle engine was so popular to-day for direct-coupled propulsion (there were 570 on order, against 28 four-stroke cycle) was because of its fewer working parts. Surely then it was a retrograde step to increase this number of working parts, however small and manageable they might be. It seemed fantastic to have an installation with 63 pistons, 63 gudgeon pins, 63 connecting rods, 63 bottom end bearings, 77 main bearings and 252 inlet and exhaust valves.

Could the authors (1) give their ideas on the number of engine-room staff which would be required to run the 13,000 s.h.p. installation, bearing in mind that presumably one man would be fully employed writing up the engine room register and taking the 63 exhaust temperatures, etc. and (2) say whether in the design of the M.A.N. engine put forward, it was possible to remove a connecting rod through the crankcase doors. If not, then it would appear that the time required to change a connecting rod would be nine hours instead of the three hours shown in Table 1, i.e. six hours for piston and cover plus three hours for connecting rod. This would bring it into line with the other engines mentioned. It had been known to take almost six hours to replace and make pressure tight the supercharge trunking on such an engine, and one wondered if these times might not be a little optimistic.

Attention was drawn to the lubricating oil consumption; the average for the three types of engines was 180 gallons per day—that was over 100 gallons a day more than the direct-coupled job, for about the same overall fuel consumption. He suggested that this extra running cost would be of the order of £4,000 per sailing year of 200 days. Further the figures given were probably in the "as new" condition, and would get progressively higher as those 315 piston rings and 63 cylinder liners became worn.

By the same token it was interesting to note that Mr. Calderwood could accept a loss in overall efficiency of his combustion turbine equivalent to £6,600 per sailing year, and still remain on level terms with Diesel-electric propulsion.

Finally, it was not considered fair of Mr. Belsey to compare submarine depot ships, which spent most of their time in harbour, with oil tankers, which spent most of their time at sea.

The President, referring to Table 9, asked Dr. Brown if, where he gave ·58lb./s.h.p./hr. for the very interesting single cylinder turbine, he could give the corresponding consumption for a 2 or 3 cylinder turbine of the same power.

He added that he could not understand the following intriguing extract from Mr. Pounder's paper on the direct-coupled Diesel engine: "With the passing of time, for reasons which will duly arise, the swing will again be away from steam machinery". It would be interesting to have the author's explanation of this remark.

BY CORRESPONDENCE

Commander (E) L. Baker, D.S.C., R.N. (Member) wrote that it was refreshing to read the opening paragraphs of the paper on the Combustion Turbine. At the present time it was all too rare to find authors admitting quite readily that the gas turbine required development and that the future of the plant was dependent upon considerable expenditure on research effort of both time and money.

With regard to the particular cycle proposed, it was not possible with the somewhat scanty data presented to comment with any degree of certainty, but it appeared from inspection and some data provided by other calculations, that the potential weakness of the semi-closed cycle lay, as it did in the marine adaptation of the closed cycle, in the air heater (7). With the cycle proposed the incoming air would have been heated before combustion and very high combustion temperatures were probable, so that the heat transfer rates to the walls of the combustion chamber and to the first lengths of the tubes of the air heater (7) must result in very high metal temperatures. It was certain that the temperatures reached by the tubes would be considerably in excess of the 680 deg. C. of the blades. Whilst it was admitted that there was no serious pressure differential, there would be very considerable stresses induced by the heat transfer and probably by differential expansion. In a compact unit these two together would almost certainly result in conditions for which no material was at present suitable.

The remark that the effect on power, output and efficiency of fouling the turbine blades was smaller with this cycle, did not appear to be founded on reliable data. If the fouling was *solely* due to the conditions of the inlet air to the cycle this would be correct, but fouling might also have been caused by the fuel and by the combustion processes, and the higher specific output of this cycle would make the chance of fouling due to combustion greater, whilst that due to the fuel was the same for the same cycle efficiency. If, however, the fuel was restricted to a distillate such as a high grade gas oil, the risk of fouling was much reduced and the problem then was likely to be limited to that of dealing with the deposits experienced on the heating surfaces of the air heater (7) and the exhaust gas heat exchanger (9).

Mr. J. E. Church, F.C.M.S., A.M.I.N.A. (Member) wrote that he had read the various papers which formed the Symposium with the greatest interest and complimented the authors on having put forward what must have undoubtedly been the last word in up-to-date propelling machinery, which would prove of inestimable value to shipowners and their superintendents in their choice of engines for the immediate future.

Writing both as a shipowner and a superintendent engineer he said he found the most important point when comparing the merits of the various types of machinery under review was made by Mr. Pounder at the end of his first paper when he said that steaming weight of machinery plus weight of bunkers for any prescribed length of voyage was what the shipowner was most concerned with. This was absolutely right and almost the first consideration in the choice of machinery except in special circumstances, and to it must be added the loss of cubic capacity which was taken up by such bunker spaces in addition to their deadweight loss. He suggested that a comparison of the basis of say a 10,000 ton deadweight vessel having a cubic capacity for bale cargo of 600,000 cubic feet when fitted with direct Diesel engines on the basis of 10,000 miles voyage from bunker port to bunker port would be most interesting, giving the corresponding cargo deadweight and cubic capacities for the various engine arrangements including bunkers. From that it would soon become apparent that the cargo cut out by bunkers in the case of all steam machinery would far outweigh other considerations, for the extra freight thus earned per annum would pay for many repairs and replacements and still leave a balance in favour of Diesel engines in some form.

Paper I. Turbines. There were of course cases where the above would not apply, such as on shorter runs where the total bunkers required even at the higher daily rate did not amount to an appreciable reduction of cargo deadweight, or in the case of passenger vessels where the lifting of maximum cargo was not of first importance. For these the turbine and the turbo-electric proposals were most attractive. He was particularly impressed with the single cylinder turbine arrangement with its simplicity and compactness. Such an installation, but with an orthodox astern turbine, built by the General Electric Company in America in 1922 was still running remarkably well under his supervision. He would, however, like to ask Dr. Brown for further details and a sketch of the astern reversible fluid coupling, for he found it hard to believe that after all these years of astern turbines and their difficulties, so simple a solution had been found. He thought this was quite a remarkable device deserving of more than a passing reference if they were to be persuaded to adopt it. Alternatively, if it was agreed that the fluid coupling was a good means of obtaining reversal without an astern turbine, it was suggested that an equally attractive arrangement would be to replace the astern coupling with an orthodox fluid coupling which would be attached to an additional pinion meshing direct with the main wheel, so that filling of the after coupling with the forward one empty would couple the turbine by single reduction to the propeller shaft, thereby giving astern operation.

The gearing arrangement would lend itself to this with little modification, and careful selection of pinion and wheel sizes would probably give 70 per cent astern power at a turbine speed not too low to be quite useful as compared with previous astern turbine arrangements.

It was noted that quill shafts were adopted in all the gearing arrangements proposed; were these highly stressed shafts of special materials absolutely necessary? They had a habit of breaking every ten years or so, and he would rather do without them wherever possible.

He would also like to enquire the reason for designing the casing so that it was completely embraced by the exhaust belt, for surely the exhaust temperature at high vacuum was such that it would cool the steam casing excessively, whilst the elimination of the usual lagging was not thought to be an advantage of any consequence.

Finally, he was of the firm opinion that the steam turbine could only be used to best advantage in conjunction with Diesel generators for port use. The cargo ship of to-day unfortunately spent many days in port loading and discharging cargo when proper engine room watches could not be kept, and it surely would not be right to keep a large water-tube boiler alight for the purpose of steaming turbo-generators. More important still from the superintendent's point of view was the need of being able to shut down all steam in port at this end at least. Both boilers could then be cleaned in comfort and any part of the steam system opened for inspection without the bugbear of hurried and troublesome "shut downs", which were the bane of the superintendent and ship repairer alike. As boilers had to be surveyed annually the mere fact of having Diesel generators to take over cargo winches and all power requirements completely would save many days delay without doubt before the ship was very old. He suggested one turbo-generator and two Diesel generators for each of the present schemes.

Paper II. Direct-coupled Diesel. Mr. Pounder's proposals brought to light an interesting feature, namely, that a standard two-stroke *single*-acting engine would give as much power per cylinder as the alternative two-stroke *double*-acting engine, it apparently being convenient to build a single-acting engine with a considerably larger cylinder diameter than was possible for the double-acting type. Being convinced that simplicity and fewer working parts should have always been their aim he enquired why build *double*-acting if it could be avoided? The engine referred to, the two-stroke single-acting "coverless" engine illustrated in Fig. 3 seemed to be the most attractive proposition in Mr. Pounder's paper, but why did they not call it what it was—an opposed piston engine? The piston of any orthodox two-stroke engine acted as a "valve" when it covered and uncovered the cylinder ports, but it was still called a piston, and surely when a "piston valve" became as large as the main piston and full working pressure was transmitted *by* it to the crankshaft it should be looked upon as a piston opposed to the main piston. He hoped Mr. Pounder would agree, for he was convinced there was a very bright future for opposed piston engines.

Regarding extraction of oil vapour from the crankcase by means of a fan and ducting, he was not sure that this was advisable. Extraction of vapour would mean ingress of a certain amount of air which no amount of sealing would successfully exclude, and an explosive mixture must more readily result. He believed that a better approach to the problem would be to seal as much as possible without extraction or ventilation, thereby maintaining the contained vapour at far too high a density to be explosive.

Mr. Pounder's comments about engine repairs were only too well known to superintendents, and further confirmed his opinion that engines must be made as simple as possible, a further very strong point in favour of the single-acting opposed piston type of engine where pistons could be exposed and rings attended to in a matter of a few hours without disturbing any valves or high pressure joints.

Paper III. Geared Diesel Engines. This type of machinery represented the most attractive in the present series and the type to be immediately developed to fill the gap at least until the gas turbine became an established fact, for the following reasons:—

(1) It had the main advantage of the Diesel engine, which was lowest fuel consumption resulting in minimum bunker space and maximum cargo deadweight and cubic capacity, without its most serious disadvantage, namely the always present possibility of crankshaft failure, which in the case of geared engines was not nearly so serious as the vessel could complete her voyage on the remaining engine.

(2) Replacement of shaft was quicker and less costly, and the vessel could actually complete another voyage whilst waiting for the new shaft to be made.

In the case of a single direct-coupled engine, one shaft failure in the ship's life of 25/30 years might be a most serious matter to her owners, yet what engine builder could guarantee that it would not happen?

(3) Engines could be properly run and tried after overhaul before leaving port without disturbing the propeller, which dock-masters were increasingly disinclined to allow and which in any case was not very satisfactory. Without doubt almost every motorship at some time or other missed a tide on sailing with cargo and passengers on board, due to some slight maladjustment which only became apparent after starting up. The writer was convinced that they *must* in future arrange all machinery so that engines could be properly tried before sailing.

All other advantages claimed by the author were of course very considerable and he looked forward to learning of more general adoption of this type of machinery.

In this connection, however, it was suggested that hydraulic couplings might have been more suitable for ship work in view of their satisfactory use in other spheres and the troubles with over-heating of electric couplings referred to in the paper. It was also felt that advantage should be taken of lower propeller revolutions—say 85/90 in order to offset the otherwise slightly higher fuel consumption as compared with the direct-coupled engine.

Paper IV. Diesel Electric. This paper was most interesting and might indicate the shape of things to come, but it was suggested that the Diesel-electric did little that the geared Diesel did not do better until this form of machinery became so general that engines became standardised for all types of ships which would use them in greater or lesser numbers, so that they could be mass produced to such an extent that it became cheaper to fit new engines than replace cylinders, pistons, etc. This would of course be ideal, but it called for considerable development in the direction of higher speeds and lighter weight and a very great reduction in noise, as mentioned by the authors. The lubricating oil consumption was also a serious problem with these engines, and it was noted that 240 gallons per day was expected for one proposal and this when the engines were new!

At the same time the large number of cylinders—in one proposal 63, together with 4 valves to each cylinder, in all 252—to be maintained and kept in good order presented quite a problem, even if most of this could be done at sea, which was doubtful.

Paper V. Turbo-electric. These proposals seemed most attractive, and it was noted with interest that Diesel generators were shown in all cases. It was suggested, however, that each scheme would be more attractive to shipowners if two turbo generators were provided, for in the case of these high powered vessels it was frequently required to run at reduced speed and as this was one of the principal advantages of electric machinery it seemed a pity not to use it. Otherwise, there was little advantage, if any, over the geared turbine arrangements put forward in the Symposium.

Paper VI. Combustion Turbine. Mr. Calderwood's paper, whilst not promising gas turbines for ships of the immediate future, gave some interesting news of present developments. There was, however, one point about which the writer was most definite in connection with the apparent tendency for advocates of gas turbines to dismiss their starting and reversing difficulties by adopting wholeheartedly the reversible propeller. This, he believed, had not been tried to date on any high powered installation, although there was one ocean-going vessel of comparable power on twin screws at sea with *variable-pitch* propellers using Diesel engines, which however stopped and reversed in the usual way. Whilst these propellers no doubt would soon be developed as reliable fully reversible propellers, he was not at all sure that shipowners would like them. The point which had to be made was that a disconnecting clutch of the hydraulic or similar type would be necessary in the case of gas turbines in order that starting up could be done without the propeller turning, and also so that the propeller could be stopped revolving when manœuvring, for when docking through and moving in and out of wet docks and harbours, "Stop" frequently and urgently meant "Stop the propeller turning" because stern moorings were being let go and would foul the propeller, because a boatman was under the stern picking up a line, or that the stern was perilously close to a barge, buoy, quay wall or other obstruction and a few revolutions would mean serious damage. Under these conditions there could be no question of the propeller running around at 100 or so revolutions even though set at no pitch and in the stop position Therefore a hydraulic clutch or coupling must be considered a necessary adjunct to the reversible propeller and would have to be added to Mr. Calderwood's arrangement Fig. 2. If it was agreed that this coupling was necessary, it would seem reasonable to suggest that another astern coupling be fitted also for reversing with a solid propeller, in which case what could have been better than the arrangement of gears and couplings proposed in Dr. Brown's paper for the single turbine arrangement, or two couplings and two pinions giving double reduction for ahead running and single reduction for astern, the gas turbine then operating uni-directionally and continuously?

Mr. S. H. Dunlop (Member) wrote that all the types of marine propulsion included in the Symposium, except the last, had established reputations and the persons under whose control they were being operated would be influenced to support the unit which had combined the qualities of reliability, simplicity, accessibility and ease of maintenance. The first cost, fuel cost, wages, etc., were factors which could decide the final choice of a propulsion unit. Consideration of personnel was also becoming exceedingly important and the successful and remunerative operation of modern machinery installations was dependent to an increasing extent upon the skill of the operating engineers and to a decreasing extent upon the particular type of prime mover.

The geared steam turbine and turbo-electric systems of propulsion depended upon the same element for producing power. The difference arose in the method of transmitting this power to the propeller. One of the advantages claimed for the turbo-electric scheme was the absence of the astern turbine. This advantage did not exist in the geared steam turbine systems and the main argument for or against revolved around gearing versus electric reduction. The return of confidence in gearing, especially double reduction, would tend to influence the selection of a geared unit. It would be of interest to know the range of powers covered by the single turbine unit.

The Diesel direct-coupled engine of the double-acting type was a proven propulsive unit but the masses to be handled during overhaul made maintenance both costly and lengthy, The single-acting two-stroke engine of the coverless type had the redeeming feature of not possessing a lower unit and although the masses remained, the maintenance charges should have shown a reduction due to a better regulated maintenance routine.

The modern tendency in Diesel engine design was to increase the m.e.p. by pressure charging, causing a reduced cylinder size and higher speeds, resulting in the progress of the geared Diesel and the Diesel-electric systems for propulsion. With the naval architect's quest for cubic capacity rather than deadweight, space and length became valuable, which made the geared Diesel a very suitable choice for propulsion. From the manufacturer's point of view, the indirect drive enabled the production of a unit covering the greatest range of powers with one cylinder size. The inclusion of a paper by Mr. Pounder outlining this type of installation could be accepted as evidence of the future prospects of this type of propulsion unit. The conservative rating and revolutions of the Diesel assured its reliability and met the claims mentioned.

The electro-magnetic coupling was a very suitable elastic coupling, but with the revolving masses reaching tons, it was assumed that dynamic balancing was employed, thereby reducing to a minimum the effect of overhang. The incorporating of the means of reversal in this coupling would remove that detrimental feature of employing the engine for manœuvring purposes.

Mr. J. F. R. Ellison (Member) wrote that the single casing uni-directional turbine scheme described in Dr. Brown's paper appeared to be an attractive proposition from the point of view of simplicity and saving in space and weight if fuel economy was not the first consideration, although the "propulsion only" figure of ·58 was better than the performance of the more or less standard 8,000 s.h.p. three casing design evolved during the war, the latter being in the region of ·6 to ·61lb./s.h.p./hr. under service conditions.

The reversing fluid coupling, if it proved robust and reliable, would certainly be a welcome alternative to astern turbines with their attendant disadvantages. It would appear that very high relative speeds were involved in the change of direction, and it was to be wondered whether there was a risk of erosion taking place on the blades and nozzles, especially when churning with both couplings filled. Perhaps Dr. Brown would enlighten them on this point.

The elimination of fine clearances such as obtained with end-tightened blading was advantageous both from the building and maintenance point of view, in particular when high superheat temperatures were employed. He had known at least one case where permanent growth occurred in one section of an h.p. turbine casing after some six months in service, possibly due to faulty annealing. This reduced the clearance of ·020in. over several rows of blades to nil, and the only remedy, short of re-machining the roots, was to fit a shim behind the adjusting gear case, housing the thrust block, thus leaving excessive clearance on all the other rows of blades with a consequent increase in consumption.

The gashing out of solid forged rotors to form the discs, which involved cutting a number of grooves about 12in. deep 2in. wide seemed to be an expensive method but certainly more reliable than keyed-on bucket wheels.

Scheme 3 might be attractive from a consumption point of view, but a fair amount of trouble was found with main steam joints on installations using moderate pressures in the region of 450lb./sq. in. so that one would imagine that the maintenance required with a working pressure of 1,400lb./sq. in. would be considerable.

He was glad to note that Mr. Pounder had not over-rated the engines described in his paper, as was so often the case with builders of both main and auxiliary Diesel engines. The assumption of too high a mechanical efficiency (possibly obtained in certain cases under ideal conditions on the test bed) had frequently led to motor vessels being underpowered, with the result that the engines were run at their maximum output all the time, with consequent heavy upkeep costs. The fitting of torsionmeters of the Siemens Ford type to motor vessels had in his experience furnished valuable information as to the true output of Diesel machinery under sea-going conditions.

He noticed that Mr. Pounder quoted high mean indicated pressure figures with moderate piston speed, and it was presumed that he found these conditions conducive to minimum liner wear.

It was interesting to note that as the turbine consumption had been progressively reduced over the past 10 to 15 years, and the Diesel had remained more or less static, the former type of machinery could now compare favourably with the Diesel installation when weight of machinery plus bunkers was taken into consideration, as the following table showed:—

Type.	*Machinery* Total weight, tons.	*Weight of bunkers tons.*	*Total weight, tons.*	*Cost of bunkers at Curaçao.*	*Engine room length.*
Single casing turbine (Scheme I) ...	592	2,270	2,862	£5,430	46′ 6″
Double-acting coverless B. & W. Diesel	1,041	1,535	2,576	£6,000	61′ 0″

The figures quoted were for a 7,500 s.h.p. general cargo vessel plying between the U.K. and New Zealand via Panama and bunkering at Curaçao. The bunkers taken were for the outward passage from Curaçao to Auckland and back to Curaçao. Times allowed: 20 days outward, 5 days coasting, 20 days homeward, 40 days in port. The machinery weights were as quoted by the authors, but if Diesel generators were fitted in the turbine vessel, the reduced consumption would bring the total weight at the start of the voyage slightly below the Diesel installation while the cost of bunkers would still be £300 less.

There were, of course, other factors to consider, but the foregoing explained in some measure the present swing over to turbine machinery which was evident to-day.

Mr. H. M. Gemmell, B.Sc. (Member) wrote that it would have been helpful if the specification laid down by the Superintendent Engineers' Committee had been given to show the degree of freedom of choice remaining to the contributors. The various proposals were not all strictly comparable, particularly in respect of shafting and spare gear weights due to choice of machinery aft or amidships and different ways of dealing with or not dealing with spare gear.

Dr. Brown's proposals embodied many attractive features and represented up-to-date practice in turbine design. Attention would undoubtedly be focussed on the proposal to use a uni-directional turbine with hydraulic transmission and reversal on account of its promise in association with gas turbines. Dr. Brown ascribed to it every merit which had been claimed for electric drive. The writer suggested every merit excepting proved performance in the field of application under discussion. As recently as January, 1947, Mr. R. J. Welsh, in discussing Mr. Davis's paper on the Beaver ships, stated with regard to electric drive as a means of manœuvring with uni-directional turbines that till then no really satisfactory alternative had proved itself on installations of more than very moderate power. Could Dr. Brown point out any cases of combined hydraulic transmission and reversal where there had been satisfactory service over a period of years under conditions comparable with those for which he proposed the arrangement?

It was thought that an efficiency of 98 per cent. was an optimistic estimate for the ahead coupling. Assuming this efficiency, would the author state the physical proportions of the couplings which would be large enough to ensure the corresponding slip, also efficiency figures, estimated or recorded, and size of the astern element?

Regarding the conditions obtaining when both ahead and astern couplings were filled it was understandable that the system from the pinion to the main shaft would come almost or completely to rest, but the writer would appreciate a brief explanation of the factors which led to turbine conditions such that "at least twice the normal full load torque is available for manœuvring".

In view of the temperature troubles which could occur in astern turbine wheels as instanced by Stodola (Vol. II, p. 774, 1928 Edn.) and in l.p. ahead casings when running astern, the possibility could be imagined of similar effects in the astern element rotating at 4,200 r.p.m. in a direction contrary to normal operation and containing air at atmospheric conditions which was much more dense than the fluid in the eduction passage of a turbine even at reduced vacuum. Presumably the peripheral speeds were lower in the fluid couplings but the condition was one of continuous operation, day in—day out. Would Dr. Brown give an assurance that no such possibility need be feared?

A steam-sealed vacuum joint between the outer casing and the condenser was mentioned. Further particulars would be of interest as the sectional views of the turbine showed no details.

A minor error in Scheme I was the inclusion of two l.p. feed heaters in Fig. 4, and the omission of an h.p. heater, resulting in contradiction with Fig. 3.

In Scheme II it was noted that deflectors were fitted for the astern steam flow, changing any axial component of leaving velocity into an equal radial component but leaving unaffected the tangential component which might be of considerable magnitude from astern blading. Would this not result in one end of the condenser receiving the bulk of the steam at a speed much in excess of the mean value through the eduction opening, while the other half was relatively starved? For condensers having an eduction opening symmetrically disposed about a vertical through the shaft centre it was suggested that stationary blades might with advantage be arranged to direct an even share to each side of the centre line.

With reference to steam reheat it was stated that the reheat temperature was limited to a value below the saturation temperature corresponding with the boiler pressure, but surely suitable contra-flow arrangements could be devised to take advantage of superheat in the high pressure steam. It might be that the added complication in the heater would not be justified, and would Dr. Brown say if considerations of this nature imposed the limit to which he referred?

Some doubt might be entertained regarding the likelihood of attaining the make-up feed figures appearing in the flow diagrams for Schemes I, II and III, which were 200, 300 and 350lb./hr. respectively, corresponding with a rate per day per thousand horse power of 0.286, 0.287 and 0.288 tons.

Scheme II operated at 600lb./sq. in. compared with 450 for Scheme I and had two additional main engine glands, and in Scheme III the pressure was up to 1,400lb./sq. in., the main engine glands up to six in number, and there were glands on each of two drain pumps operating with water at 580 deg. F., 1,400lb./sq. in. at suction and a higher pressure at discharge. In spite of a two pocket gland with the outer connected to the feed pump suction in the case of the drain pumps, the addition of 50lb. per hour to the make up figure seemed very inadequate. It should be noted that a pressure above atmosphere must have existed in the outer pocket on account of the feed temperature of 232 deg. F. in the pump suction. The figure for make up given by Mr. Saunders and Mr. Turner, although admittedly high, and subsequently reduced, was of interest in this connection.

It was observed that the feed temperatures of 300 deg. F. and 390 deg. F. had been adopted presumably on account of the thermodynamic benefits to the turbine cycle arising from bleeding, and this necessitated the adoption of airheaters in the boiler installations if high boiler efficiency was to be attained. Incidentally in Scheme III how were leaving temperatures of 318 deg. F. and 390 deg. F. obtained from l.p. and h.p. feedheaters which had steam pressures in their casings of 75 and 200lb./sq. in. respectively, the corresponding saturation temperatures being 307·5 deg. F. and 381·8 deg. F. in view of the stated limit of temperature in connection with the steam reheaters?

With this need for airheaters and the adoption of the feed temperatures quoted, which could be increased if desired by a different choice of bleeding points, it was questionable whether the double complication of airheaters and economisers could be justified, except perhaps in high pressure installations where the saturation temperature was relatively high and bearing in mind that the introduction of economisers in addition to airheaters changed the matter of induced draft fans from one of choice to one of necessity.

With the simple tubular airheater having the flue gases passing up the tubes and discharging direct to atmosphere up the funnel it was found at mercantile boiler ratings that the natural draft created by the hot gases was sufficient to overcome the resistances imposed by the boiler and superheater tube nests and the airheater, resulting in a pressure in the furnace which was approximately atmospheric. The increase of resistance imposed by an economiser would cause a pressure in the furnace unsuited to open stokehold conditions even with very well made boiler casings, assuming that reasonable means of access for examination were retained. Induced fans, involving weight, cost, power, consumption and maintenance, then became a necessity.

A minor correction in the flow diagram of Scheme III was the condensate flow from fuel oil heaters and galley given as 13,000 instead of 1,300lb./hr.

Would Dr. Brown state if his turbo generators could operate as self-contained units, as otherwise his single main condenser would be continuously under steam at sea and in port excepting when the atmospheric exhaust line was used.

Mr. Pounder in his contributions dealing with direct and geared Diesel machinery had given particulars of his proposals in considerable detail, and the effort involved would be appreciated by every reader. Examination of the tabulated synopsis at the end confirmed the assertion that the ratings were well below those for maximum continuous output, and it might be asked if this had not been carried too far to the disadvantage of the Diesel proposals in comparison with the others, particularly in regard to weights. Weight, however, should be accorded the attention it deserved and no more, for in the class of vessel with which we were concerned weight was not of primary importance. Mr. Pounder's concluding remarks as regards costs in relation to weight were worthy of study in this connection as they were equally relevant in comparing all the contributor's proposals.

It was noted that Mr. Pounder's direct-coupled proposals had crankshafts about 10 per cent. in excess of Lloyd's Rules, tunnel shafting 5 per cent., thrust shafts 10 per cent. and tail shafts 10 to 15 per cent. Owners and superintendent engineers often specified margins of strength above Lloyd's Rule sizes, which were minima, but it was considered that any requirement of this nature should have been mentioned in the Superintendent Engineers' Committee's basic requirements. This factor alone must account for a certain amount of the weight differences, an outstanding example being the stern gear shafting and propeller for Dr. Brown's Scheme I at 75 tons as compared with Mr. Pounder's figures for the same item plus spare propellers and tail shaft ranging from 120 to 133 tons. Would Mr. Pounder state his shafting weight excluding the spares, and Dr. Brown give the length of shafting on which his weight was based?

With reference to the proposals for Diesel-electric propulsion made by Mr. Belsey and Mr. Robinson, a most important question appeared to be the probable life between overhauls of the engines they put forward, when operated at the ratings they gave. To maintain the specified powers all units must be in continuous operation and there were many long voyages undertaken by the class of vessel under consideration, at the end of which every cylinder might demand attention. Would the authors state their estimate of the length of time which the different types of engines would last between routine overhauls of pistons and valves, when rated as proposed? Although power was a direct function of mean pressure and piston speed, it was felt that rates of revolution must be held in mind simultaneously with piston speed in assessing longevity. The ratings adopted by the authors were probably well suited for auxiliary generating sets, but such sets were not on continuous duty, as the installed capacity was well in excess of average and maximum requirements, and this point was well exemplified by Mr. Pounder's figures on page 261.

It would be noted too that Dr. Brown in his schemes gave an installed capacity of roughly four times the average sea load. With such reserves it was possible to maintain a routine of duty and overhaul periods, but no such margin was provided in the main propulsion units as proposed. It was suggested that at least one engine in the low power scheme and two in the high power scheme would need to be provided as stand-by if the specified powers were to be available as consistently as was considered reasonable in average marine practice. Such provision of course would interfere with the model making!

The authors' points regarding the overhaul of engines having small parts were appreciated, but the numbers of them did not receive any attention—63 of each in the 13,000 s.h.p. tanker with M.A.N. engines.

The large number of cylinders inevitably raised the question of spares, and the authors' schedule of weights definitely stated Lloyd's spares. Surely a more realistic attitude should have been adopted and a more generous provision made in the authors' proposals.

The paper by Mr. Saunders and Mr. Turner on turbo-electric propulsion was of special value for the particulars of actual performance that it contained. As so often happened some assumptions must be made to enable an analysis to be completed; in the case of the fuel would the authors give the considerations which led to the adoption of a value as low as 18,000 B.T.U.'s per lb. as the calorific value of the oil. It was stated that the turbine nozzles had been carefully calibrated—was this in respect of flow or only of size? If the latter was the case, what coefficient of discharge had been used?

In the summary of trial results a steam/oil ratio of 13·2 was derived from the first four hours' performance, but a figure of 14·75 was used for the second four hours. How could this be justified, as no observations of CO_2 were made nor of funnel temperature, in either of which an improvement would warrant the use of the higher boiler efficiency?

In the new design for 7,500 s.h.p. was not an allowance of 10 tons per day of make up unnecessarily generous in view of the authors' earlier comments on this matter?

Table 3 referred to three 180 kW. Diesel engines which appeared to be indicated in Fig. 6—were these installed in a separate compartment?

Mr. H. M. Hart (Member) wrote that while it was true that no major improvement in marine machinery was possible without the need for courage on the part of a particular shipowner in ordering machinery of advanced and relatively untried design, a symposium on engines for cargo vessels should surely have confined attention mainly to machinery arrangements, designs and sizes which had stood the test of actual service at sea. For this reason the combustion turbine, while indicative of possible future developments, could as yet find no place in a general cargo fleet and, similarly, the use of electric propulsion was difficult to justify except for special services. The choice was, therefore, mainly between the more usual forms of steam or Diesel propulsion and it was unfortunate that the steam turbine arrangement selected for the 7,500 s.h.p. proposal should contain such unusual features as hydraulic reversing gear, thus invalidating any comparison with the well tried designs put forward for Diesel drive.

Again, not every shipowner would wish to venture into the realms of super-pressure and re-heat as proposed for the 13,000 s.h.p. turbine vessel, Scheme III. The steam turbine proposal for 13,000 s.h.p., Scheme II, was of orthodox and acceptable design, and it would be of interest to know if a single-screw Diesel proposal could be put forward for this power on the basis of conservative and well-tried designs similar to those indicated for twin-screws.

Mr. F. J. Mayor (Member) wrote that he was very much interested in Messrs. Belsey and Robinson's excellent paper. The feature in particular which impressed him favourably was the lay-out comprising several prime movers. He was glad to see this trend, and was of the opinion that its development would improve the all-in cost of maintenance, minimize the lay up time, and give the very best service factor, the greatest economy of all.

Mr. J. E. M. Payne (Member) wrote that the choice and design of the suitable prime mover and the best method of application of the power to a ship were complicated by the varieties of choice and the uncertainties of the hour.

The future trend of prices of the different grades of fuel oil, the time occupied on annual overhauling, availability of spare parts for replacement, and probable trend of future research and development all played a part in the resolution of the problem. The wealth of data here presented came, therefore, at an opportune moment and would enhance the value of the TRANSACTIONS.

In the view of the writer one of the most acute problems to be faced at the present time was the serious loss of earning power represented by the annual overhauling, which occupied from three to four or five weeks depending in part on factors outside the scope of this discussion. However, any factors in design tending to facilitate the expeditious completion were amply repaid during the subsequent life of the vessel.

With the exception of the gas turbine every type of installation described in the symposium had been tried in service and had proved reliable.

Referring to the paper on Diesel electric propulsion it was observed that the authors had chosen engines developing their designed output at rates of revolutions between the ranges 340-450. This was important in this context, and the point made here was that even if brake mean effective pressure was kept within conservative limits, the rate of revolutions had an important bearing on the cost of maintenance and also the important factor of noise. At speeds in the region of 1,000 r.p.m. and above, maintenance of pistons, rings, liners and valves became increasingly onerous.

Messrs. Belsey and Robinson stressed their view that ease of maintenance depended on physical size. This view was not accepted; it was all a matter of accessibility and the proper appliances (vide Mr. Pounder's comments).

Trouble had been experienced with Diesel generators (though not the particular machines described) owing to lack of adequate rigidity and maintenance of alignment between engine and generator. The change of section of the bedplate in way of the coupling to allow for the stator clearance necessitated compensation to avoid concentration of bending stresses which had been a contributory cause of pedestal bearing failures. Machines of this type running for long periods at the designed full load and subjected to the vessel's motion in a seaway gave rise to appreciable gyroscopic torque. This presented no exceptional problem, but required consideration in the design to avoid failure and excessive wear of the machines and the hull. Analogy with the auxiliary generators was misleading, these being subjected to varying loads and normally running at say 60 per cent. or 65 per cent. of full load.

Referring to the electrical equipment, the writer would stress the importance of correlating the design of the engine room ventilating system with the air cooling circuits of the main alternators. The inlets of the latter should be remote from downcast ventilator outlets.

Earlier protagonists of this general scheme advocated the periodical removal of a generator set to the shops and Messrs. Belsey and Robinson visualised routine maintenance at sea. It should be observed, however, that maintenance repair costs and wages bills formed a heavy item in the operating of ships, and these factors influenced to a considerable extent the ultimate choice.

It might well be argued that Messrs. Belsey and Robinson had put their strongest case in the last paragraph of page 276 of the papers, and it was in perhaps ships destined for special duties, and in which the auxiliary load was relatively high, that their admirably presented scheme would find its most useful field of application.

Referring to Dr. Brown's paper, the incorporation of hydraulic couplings for ahead and astern propulsion simplified an already simple job—the manœuvring of the ship under conditions of stress. The author stated that the hydraulic losses were largely offset by the elimination of the losses occasioned by driving the astern turbine in the ahead direction. The writer would ask whether this loss was known with accuracy. The point made here was that this loss was not so serious as its constant recurrence in print would suggest and mainly consisted of what was usually termed "windage", working as a compressor. Astern power and torque, of which power was a function, was adequately catered for in cross-channel vessels where conditions were onerous and one wondered whether on balance there was a gain in altering established practice.

The turbine was the cheapest type of propelling machinery to maintain, and the overall thermal efficiency during the last twenty years, when associated with suitably matched auxiliaries, had increased considerably. With adequate supervision this could be maintained and an important factor was the prevention of priming and the maintenance of clean blading.

Cylinder wastage mentioned by the author had been experienced by the writer in a minor degree confined to the inner surface of the stator between the diaphragm landings of the last two stages in the low pressure cylinder. This took the form of surface attrition, the appearance presented being similar to that of wood which had been planed against the grain. The trouble did not recur after surface dressing and treatment with a graphitic coating; the vessel concerned was 27 years old.

The sleeve nozzle control valve shown in Fig. 2 appeared to the writer suited to vessels engaged on short voyages and subjected to frequent changes of speed depending on traffic exigencies. The view was put forward that if this valve was not frequently used in its different settings the localised heat stresses in way of the packing rings would give rise to local corrosion.

The machinery arrangement shown in Fig. 4 brought to mind a factor concerning the annual overhauling. The turbine would be opened for inspection every two years, and at the same time work would proceed on boiler maintenance; repair of refractories, cleaning, tube inspection and so forth and in its indicated setting some trouble was likely to accrue in securing cleanliness and ensuring the uninterrupted concurrent performance of work on boilers and engines. Operationally the arrangement had very strong advantages, and the removal of the boilers from the cellular double bottom removed a prolific source of hull corrosion.

Mr. Calderwood's paper on the gas turbine, while giving an adequate conception of the position to-day, led one to suspect that some considerable economic problems would arise when this prime mover had proved its reliability in service. The question of open or closed cycle was to some extent bound up with the ability to maintain the blades in a clean condition. Future details would be awaited with interest and expectancy.

Mr. W. H. Purdie (Member) wrote that he was confining his comments entirely to the contribution of Mr. Pounder dealing with Diesel engines, and would like to compliment the author on the excellence of his illustrations and on the completeness of the data he had given for all items in his four typical installations. Every detail seemed to be covered. There were only one or two small points on which he would like to comment.

Reference was made on page 258 to the eccentricity of main bearings of 1 mm. in order to facilitate their removal. He would like to know if this had been found sufficient for older engines, since some of their early installations were now operating with a wear down of between 2 and 3 mm.

It was interesting to note on page 259 that the author adhered to plain Ramsbottom rings. In spite of the variety of patent types which had been evolved in the past 30 years, it was surprising how few of these survived in oil engine practice with two cycle engines having ports in the cylinder liner.

Regarding pegging of rings, they had tried this in a variety of ways in the early days, but ultimately gave it up and for many years their rings had been free to rotate in the grooves.

He could fully support the author's views on the necessity for power-operated lifting gear of the most convenient type for all heavy main engine parts. On all their vessels completed since 1921, with the exception of two very small installations, they had fitted power-operated cranes. In the early days considerable criticism was encountered, their critics contending that they, as builders, evidently expected frequent breakdowns and overhauls.

For their wartime cargo vessels of low power they succeeded, after considerable discussion with the authorities concerned, in retaining the electric overhauling crane, suggesting to these authorities that a considerable amount of spare gear which was seldom required in the early life of a vessel could be dispensed with rather than lose the undoubted advantages of power driven lifting gear.

He fully supported the author's views on the advantages of ingot mild steel for bolts, as this was one of the few instances in the engineering world where the best material was actually the cheapest. He could confirm that under-tightening of bolts was much more dangerous than over-tightening, and where the bearing body which was to be under compression by the tightening of the bolts could be made of really substantial cross section so that it was incompressible compared with the extensibility of the bolts, the initial tension on the bolts could be of the order of 12,000/14,000lb./sq. in. since a fluctuation of bearing load produced comparatively little change in the tensile stress of the bolt.

He also concurred fully with the author's comments at the top of page 266 as his definition of real machinery weight was the correct one.

Too much stress had in many cases been laid on the dry weight of a particular type of propelling plant or on the small space it occupied. For long voyages such as U.K. to the Far East or Australia, the extra fuel bunkered for a less efficient type of propelling plant could soon outweigh any advantage gained by a plant with a less dry weight but a heavier fuel consumption.

Mr. E. F. Souchotte (Member of Council) wrote that it was probably true to say that the speeds of ships in most liner trades had so increased in the last two decades as to double the power requirements, without an appropriate increase in ship dimensions. That was to say, that it became increasingly difficult to select main and auxiliary machinery which could be accommodated in the best harmony with the many desiderata, e.g., passenger and crew accommodation, cargo stowage and handling gear, cubic capacity, balance of working, and so on. In short, the marriage (the word was quite expressive) had to be one of convenience and the choice of propelling machinery was seldom governed only by the matters which specially concerned the engineer and which formed the subjects of this symposium, notwithstanding their importance. Mr. Pounder had drawn attention to a number of pertinent factors outside the terms of reference, which affected the issue.

There were others, among which might be included availability of repairing facilities, the time required for repairs, the familiarity of those employed in the repairing industry with the work to be done and the numbers required to do it. The importance of the cost of fuel, whether considered as a price per unit of heat or of weight, would vary in different trades. Figures published recently showed that the average price-difference between furnace and Diesel oil was now smaller than it was before the outbreak of war, yet, as Mr. Pounder also remarked, there was presently an appreciable swing towards high pressure turbines. It would be very surprising if he was wrong in declaring, as he did, that the swing would be reversed. The history of marine engineering suggested that it would be so and that distinctive denominational problems would continue to arise. Comparisons between this author's very complete contributions and others showed that the estimates of total weight in various parts of the symposium were not similarly based. Comparisons with existing ships confirmed Mr. Pounder's figures and one must endorse his advocacy of powered lifting tackle and provision for direct lifts of heavy parts without avoidable stripping. The writer was concerned with the operation of the geared Diesel installation described by this author and found that the operating personnel liked it, in spite of the multiplicity of parts requiring periodical inspection. The consumption of lubricating oil in multi-cylinder engines was always prodigious and increased rapidly as soon as wear was apparent; this remark was applicable to all the indirect Diesel installations described; crosshead engines had much to commend them even at some penalty in weight.

The Diesel electric proposals of Messrs. Belsey and Robinson showed a remarkable possible economy of space and weight, though the absence of comparable first cost figures prevented economic valuation. The ratings proposed seemed to be higher than would be acceptable for continuous operation, even in temperate zones and the engine governing arrangements, if the engines were to be manœuvred reliably, presented a problem which required much consideration and assurance. Their schemes were attractive and would be the more so if the stage of Diesel development permitted higher ratings at higher rates of revolution.

There was a limit, fairly quickly reached, to the amount of first class work which could be done at sea. The standard of maintenance of even moderate speed engines must be very high if they were to operate continuously and first class facilities were required.

With indirect Diesel drive where the speed of the engines varied with that of the propeller, it was necessary to take steps to avoid overloading of the remaining cylinders in the event of failure of one (or more) to carry the load. It would be apparent that for example, if only three engines out of four were available, the realisable proportion of power would be 68 per cent., if two out of three, 58 per cent.

The powerful advocacy of the uni-directional turbine suggested that the power loss in astern turbines (while running ahead) was greater than we had believed to be the case. It offered an attractive simplicity in design, as did all Dr. Brown's proposals.

When considering the proportion of astern power to be made available, there would be some differences of opinion. The important point was the torque available, which depended upon the time taken to get the engine going astern.

If the gas turbine had the future which many expected, the reheat cycle would have only a short vogue and the high temperature turbine might be the longer term development.

The heat exchanger seemed, in gas turbine development, to occupy the place of the condenser in the early stages of development of the water tube boiler—high pressure turbine combination and the satisfactory operation of a high power high efficiency gas turbine to depend upon the production of another type than one containing a great many closely pitched small tubes.

Mr. G. F. Temple wrote that Mr. Pounder was to be congratulated on the very lucid papers which he had presented, particularly the section on the electro-magnetic coupling, which should clear up any doubts which might exist in the minds of marine engineers on this form of transmission.

The pioneers in the commercial development of this form of transmission, A.B. Atlas Diesel, Stockholm, had now had considerable experience in both two- and four-engined installations. The first geared installation was completed by them in 1937 and since then 36 vessels fitted with 80 Diesel engines of 93,600 b.h.p. had been delivered or ordered. Of these, 26 vessels aggregating 59,400 b.h.p. had been delivered and were in service. One installation with hydraulic couplings had also been delivered and seven others were on order. These installations were included in the figures given above, the balance being equipped with electro-magnetic slip couplings. Of the 26 vessels in service, 20 had been in operation for five years and over. The Company's experience with these geared units had been entirely satisfactory. Certain difficulties were, of course, experienced with a few of the first installations, but these were confined to faults in the machining of the gears. Only two breakdowns of the gear were, however, experienecd, one of these being due to loss of lubricating oil. In all the installations where they had been used, the electro-magnetic couplings had given absolutely no trouble.

The reliability of a geared installation with electro-magnetic slip couplings was drastically tested in the autumn of 1944, when a Swedish cargo vessel, m.v. "Wiril", of 1,250 tons, fitted with two geared engines towed the mine damaged 2,900 tons m.v. "Camelia" through storms and heavy seas from Istanbul round the Azores and the Faroes to Gothenburg, a voyage of 7,000 nautical miles. The speed in fair weather during this tow was 8·5 to 9 knots and no trouble whatsoever was experienced with any part of the propelling machinery.

The advantages of the geared drive mentioned by Mr. Pounder were entirely supported by the writer's Company's opinions and experiences and below he had set out further points which might be of special interest.

The cushioning effect of the slip coupling was very great and so far as torsional vibrations were concerned, it actually cut the vibrating system into two parts, i.e. engine and shafting and the one node vibration between engines and propeller was practically eliminated. The one node vibration within the engine itself might be effectively damped by means of a suitable damper and, in fact, one installation of four engines was at normal speed, working on the sixth order critical damped with hydraulic dampers of his Company's design.

A Swedish shipping company had had four small ice-breaking cargo vessels operating in the Baltic for the past nine years, these vessels being equipped with geared engines and electro-magnetic slip couplings. The experience gained with these vessels showed that the drive was excellent for ice breaking conditions, their best performance being when running astern. The fact that no trouble had occurred with any of the propellers might certainly be credited to the properties of this particular form of drive.

Mr. Pounder mentioned the possibility of manœuvring a geared installation by running half the engines ahead and half astern, changing the direction of the propeller by energizing and de-energizing the respective electro-magnetic couplings. Tests made on one of their installations showed that such manœuvres were more rapidly effected by keeping the couplings energized and reversing the engines, rather than in the manner suggested by Mr. Pounder.

Mr. Pounder only occasionally mentioned the use of hydraulic slip couplings and gave as his opinion that they were substantially more expensive than electro-magnetic slip couplings. This opinion was contrary to the writer's Company's experience, as it had been found that hydraulic couplings of their design were cheaper, including in both cases the costs in connection with changes and additions which must have been made to the auxiliaries in any particular installation.

Hydraulic couplings were also, according to his Company's experience, not as heavy as the electric type if they were made without quick emptying devices, which, in the light of the experiences mentioned above, were entirely unnecessary. Mr. Pounder was apparently thinking of a type of hydraulic coupling built into the casing of the reduction gear with heavy surrounding covers. A relatively light fabricated sheet steel covering round the overhanging coupling halves was sufficient to gather the oil leaving the coupling.

He said they had had experience with the multi-engined hydraulic installation in the case of eight rescue tugs built during the late war for the Admiralty. These vessels, the largest of their type in the world, were known familiarly as the Bustler Class and were fitted with two eight-cylinder engines 340 mm. bore by 570 mm., each developing 1,620 b.h.p. at 320 r.p.m. coupled to Vulcan hydraulic couplings and single reduction gearing, the propeller speed being 140 r.p.m. The excellent work done by these vessels had been referred to from time to time in the national press.

When comparing electric and hydraulic slip couplings one must remember that there was a great difference in action between these two types of coupling. The slip in the electric coupling during normal running conditions was approximately proportional to the torque passing through the coupling but was independent of the actual rotational speed of the coupling halves. The slip in the hydraulic coupling was also approximately proportional to the torque at a given speed, but with decreasing rotational speed the slip increased approximately in inverse proportion to the latter. The torque of a hydraulic coupling as a function of the rotational speed followed the same law as the propeller, and thus the slip remained constant at all speeds, provided that all engines were running. If some of the engines were cut out at a lower speed, the torque on the remaining couplings was increased, but remained unchanged at the propeller. The only way in which the balance could be restored was by increasing the slip of the hydraulic coupling until it corresponded to the increased torque.

The increased slip when running with any one engine stopped resulted in more heat being delivered to the oil flowing through the engaged couplings. In the Swedish installations this oil was taken from the engine lubricating system with the result that the heat delivered to the oil from the engine was decreased at the lower load, and the existing engine coolers were still sufficient to handle the total heat.

There was, however, some doubt if the fuel saved by running the remaining engine or engines at better economy would not be neutralized by the increased slip in the hydraulic coupling or couplings which required higher speed and b.h.p. from the remaining engines with increased fuel consumption as a result.

Mr. O. Wans (Member) wrote that the papers presented by Mr. Pounder dealing with direct and gear coupled oil engines were of considerable value to those interested in marine propulsion. A further advantage of geared engines not stated in the paper was the freedom allowed in selecting the best propeller and engine speeds to give the maximum overall efficiency.

During the last ten years or so, engine speeds had risen considerably, and whereas formerly, rotational speeds of over 250 r.p.m. would not be considered for marine auxiliaries, nowadays many auxiliaries were giving reliable service running at over double that speed. The up speeding of propulsion engines had been a natural corollary and engines running at considerably higher speeds than those demanded by direct coupling were giving reliable running under sea going conditions. The engines given in the schemes 1 and 2 ran at 435 r.p.m. and showed a marked reduction in weight and space compared with the direct coupled installation of corresponding power. By present day standards, this engine speed was not excessive for the powers in question.

The accompanying summary of data had been tabulated in the form given by Mr. Pounder so that direct comparisons might be made. The weights for shafting, propellers, piping, etc., given by Mr. Pounder had been used as they were sufficiently representative and applied equally to the engines considered.

Engines in schemes 1 and 2, giving installation layouts for 7,500 s.h.p. and 13,200 s.h.p. were Ruston four-cycle pressure charged engines of 17in. bore 18in. stroke rated as follows:—

Scheme	*Total b.h.p. B.S.I. Rating*	*B.m.e.p. lb./sq. in.*	*Total s.h.p. Service*	*B.m.e.p. lb./sq. in.*
1	9,170	120	7,500	108
2	16,160	120	13,200	108

A reduction of 18 per cent. from the b.h.p. was made to cover service running and transmission losses in gearbox, hydraulic coupling and tailshaft.

The engine rating at 108lb./b.m.e.p. was conservative. An overload at 135lb./sq. in. could be carried for short periods. The engines were direct reversing and controlled from one station. Hydraulic couplings were shown but where there was a preference, electric couplings could be used.

With regard to the cost of multi-oil engine propulsion, he agreed with Mr. Pounder that it would not exceed that of a direct coupled installation, but he thought that ultimately it would be less because the smaller engines allowed of the quantity production of smaller details.

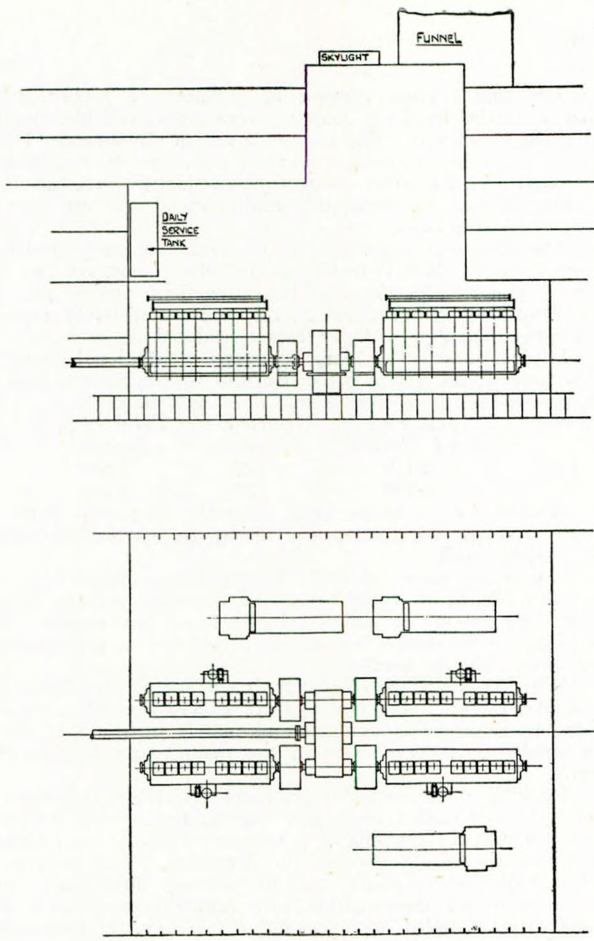
Looking ahead, one could not but feel that the present massive direct coupled engine must give way to the smaller higher speeded type and that there would be a tendency towards even higher speeds than now considered feasible for Merchant Naval service. It was appreciated that reliability was of primary importance, but higher speeds were not incompatible with reliability any more than slow speed direct coupled engines were necessarily an assurance against breakdown.

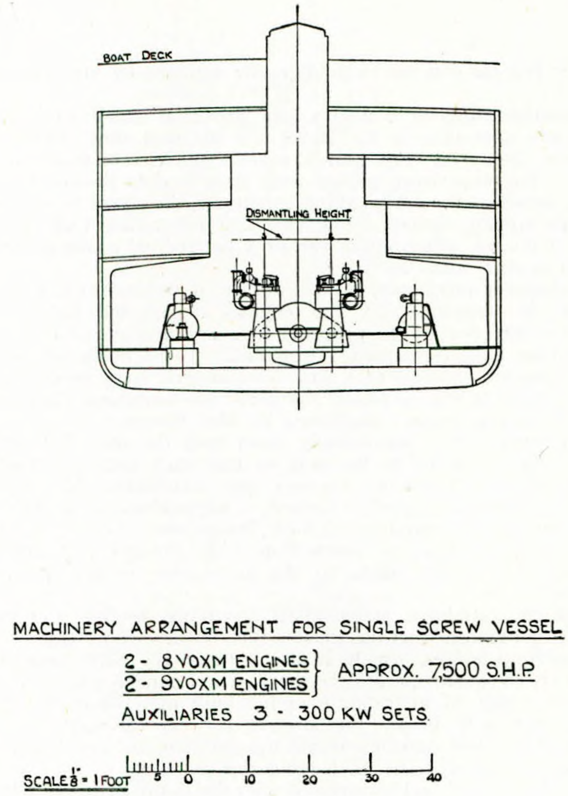
Turning now to Dr. Brown's paper on the geared steam turbine, the fuel consumptions for all purposes given in Table 9 showed these to be appreciably higher than those of geared oil engines, ranging from 36 per cent. higher in scheme 1 to 18 per cent. higher in scheme 3. These differences would have an appreciable effect upon the running costs, depending of course upon the prices of the fuels used. Also, the bunker capacity would be proportionally greater for a given steaming range or conversely the oil engined ship would have a greater range for a given bunkerage. The pros and cons of steam versus oil engines were many, but it was considered that the oil engined installation was more economical and simpler.

The speaker said he would like to make a correction in the paper on Diesel-electric propulsion by Messrs. Belsey and Robinson. In Table 1, giving the weights of parts to be handled during overhaul, the actual weight of each cylinder cover of the Ruston engine was 820lb., not 1,120lb. In the design of these heads, every endeavour was made to reduce the weight in order to make them as handable as possible. So far as the engine builder was concerned, Diesel electric propulsion presented no difficulties.

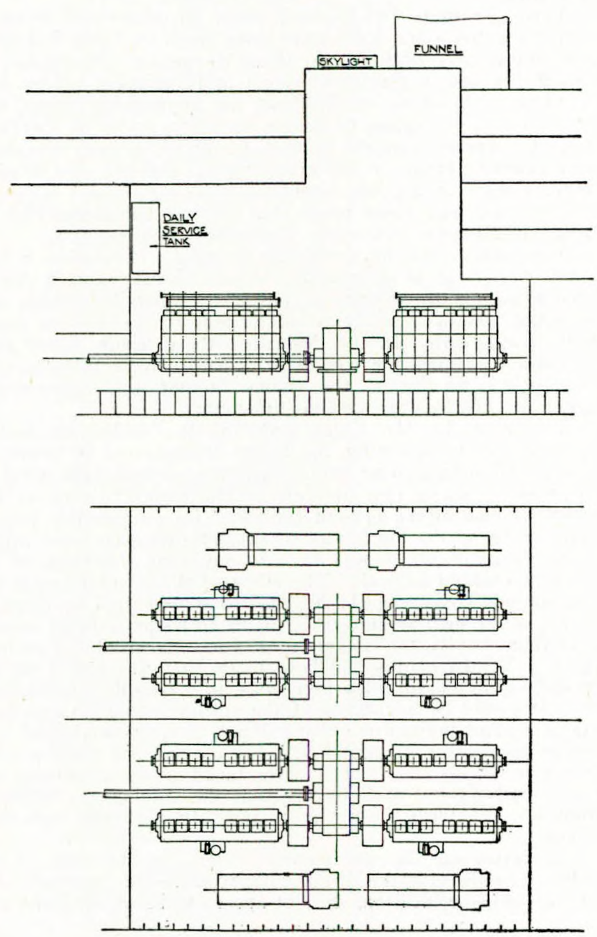
The paper by Mr. Calderwood on the combustion turbine was interesting as representing the latest development in prime movers. Its application to marine propulsion was probably the most difficult and presented many new problems. The main criticism of the combustion turbine at the present time was its comparative low thermal efficiency due to the limitations of temperature, and from information at present available it was safe to say that the price of such an installation would be high. The effect of the lower thermal efficiency upon running costs could be lessened to some extent by using a lower grade fuel oil such as bunker C, owing to its price being considerably lower than that of the higher grade fuel oils usual for reciprocating engines. Whilst claims had been made that this fuel could be used successfully in gas turbines, there was little reliable information available. It would be interesting if the author would give some definite data as to the present position. Much pioneer work had yet to be done in developing the combustion turbine and one could not but wish success to those who were boldly tackling the problems involved. It was perhaps not rash to say that the combustion turbine would ultimately supersede both the reciprocating engine and the steam turbine.

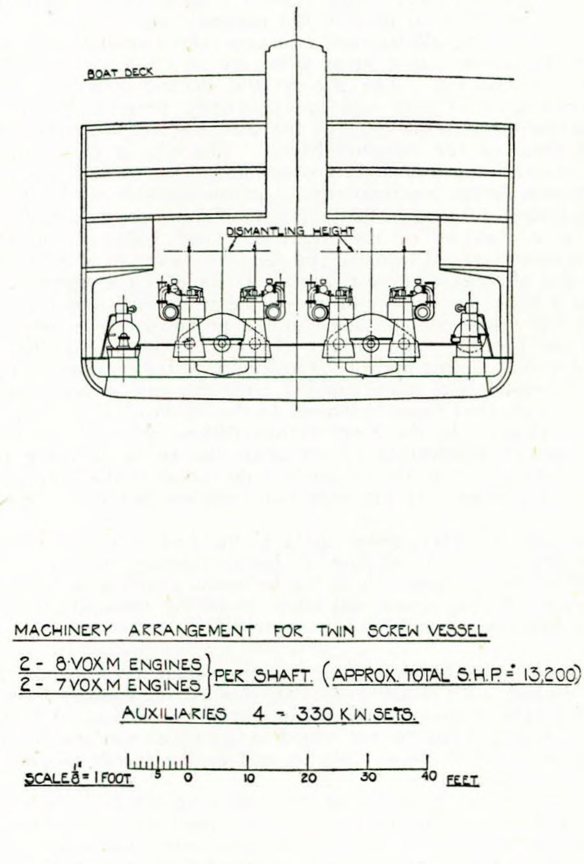
In reviewing the six papers under consideration, it was not readily possible to make direct comparisons—the methods of tabulation differed, specifications did not appear to be on all fours and there





SCHEME 1.





SCHEME 2.

was a general lack of uniformity. He had followed the lead given by Mr. Pounder which appeared to him to be the clearest and most direct tabulation. It would make comparisons easier and more reliable if a standard form of data sheet could be devised, and this, it was suggested, might be a fitting work for the Institute. The particulars called for should be those required by the shipbuilder and should be used in all relevant papers presented for reading. Data sheets of this description would make a valuable file for those interested.

Installation size	7,500 s.h.p.	13,200 s.h.p.
Type of drive	Geared	Geared
Number of screws	1	2
Engine type	SA4C	SA4C
Engines per screw	4	4
Cylinders per engine	2–8 2–9	2–7 2–8
Bore—ins.	17	17
Stroke—ins.	18	18
Engine—r.p.m.	435	435
Propeller—r.p.m.	100	100
Piston speed—ft. per min.	1,300	1,300
B.m.e.p.—lb. per sq. in.	108	108
Auxiliary machinery, engine room and deck	Electric	Electric
Fuel consumption, main engines—tons per day	*33	*58·5
Fuel consumption, auxiliary engines—tons per day	3	$4\frac{1}{2}$
Total fuel consumption at sea—tons per day	36	63
Lubricating oil, main engines—galls. per day	51	85
Lubricating oil, auxiliary engines—galls. per day	7	11
Total lubricating oil at sea—galls. per day	58	96
Weight of main engines, gearing and hydraulic coupling, thrust blocks, grating, etc., spares to Lloyd's schedule—tons	298	563
Weight of all shafting, sterntubes, propellers, plummer blocks, etc., also spares including bronze propellers, and tail-shaft—tons (180ft. shafting for 7,500 s.h.p. and 190ft. for 13,200 s.h.p.)	133	217
Weight of auxiliary engines, pumps, boilers, air receivers, and spares—tons	195	240
Weight of pipe arrangement work, including floors, gratings, ladders, tanks, silencers, funnel, workshop, storeroom, ventilation, etc.—tons	177	292
Weight of water and oil in circuit in running condition—tons	35	62
Total weight of machinery in running order—tons	838	1,374
Length of engine room—ft.	70	67
Width of casing at successive decks	30/22/10	50/10
Overhauling height, tank top to crane-hook—ft. (straight lift)	22	22

* These consumptions are based upon a maximum figure of ·38lb./b.h.p./hr. Actually consumptions 2 to 3% lower should be obtained in service.

The Authors' Replies to the Discussion

Dr. Brown, in his verbal reply, stated that during the course of the discussion he had been considerably aided by some of the speakers and did not propose to refer to the discussion in favour of the geared turbine. Mr. Turner had referred to the construction of the single cylinder turbine shown in Fig. 1 as being weak, while a later speaker pointed out that it was similar to a known American construction. They did not in fact realise this, but it was clear that the main strength in the casing was a deep side girder which only had to deal with the variations in temperature due to the minor variations in vacuum. The inner casing had in fact three-point suspension which Mr. Turner recommended.

The bibliography referred to in the paper gave further information on the matter of consumption of auxiliaries, etc. Dr. Brown stated that he was not trying to sell any type of auxiliary, but it was quite clear that the feed pump consumptions shown in the diagrams of paper No. 1 could be achieved, and that the consumption figures given for feed pumps in Mr. Turner's paper were wrong for modern types of feed pumps.

The operation of fluid couplings to achieve ahead and astern manœuvring had been demonstrated to the members present at the meeting, and the only further information given at present was that the astern converters could be of various forms. Pametrada had patented several constructions, one in which the astern converter was considered as a pump discharging oil through nozzles on to an impulse turbine, the pump element being coupled to the quill shaft and the impluse turbine to the pinion. In the other case the pump discharged through nozzles and the fluid was carried round in channels to a couple of reaction elements, the pump being coupled as before to the quill shaft and the reaction elements to the pinion. Mr. Sinclair spoke about the terrifying result of operating one of these couplings when a shin took the ground and also the amount of heat to be dissipated with say a 7,500 h.p. turbine rotor in which the kinetic energy by rotation had to be taken up by the astern element. The large amount of heat to be taken up in such circumstances would cause difficulty if in fact the period in which the energy had to be dissipated was not short. When a ship grounded the astern power would be applied in jerks which would be more effective than a steady grind. Similarly, in absorbing the kinetic energy from the rotor running at full speed the period in absorbing this energy would be quite small. In the circulating system of a geared turbine vessel of this type, the weight of oil might be of the order of 12 tons and many million B.T.U.'s could be taken out in a matter of seconds with a temperature rise of say 10 deg. F. If a long period of astern running was required, the heat generated by the coupling would be dissipated in the lubricating oil cooler, oil coolers having large heat dissipation capacity within reasonable size. It had been suggested that because the oil supply pipes in the model were $\frac{1}{4}$in. bore that very large oil supply pipes were required to the couplings. In point of fact the oil supply pipes to a 7,500 h.p. geared turbine set for the couplings would be rather more than 3in. in diameter. The problems in a small scale model of the type demonstrated due to scale effect involved greater difficulty than those in the full size couplings. Electric couplings had been shown in the paper on the geared Diesel to be smaller than the hydraulic couplings for the same power and speed. It should be noted, however, that with the configuration shown at the meeting and drawn in the paper for the 7,500 h.p. set, the couplings operated at the main turbine shaft revolution speed, and at these high revolutions the hydraulic coupling was smaller and simpler than any electrical coupling yet on the market. The lubricating oil used in the couplings was of the same quality as used in the sprayers and bearings in the gear case.

As an interim reply to Rear-Admiral Parnall, the author pointed out that the overall efficiency ratios could be obtained from the figures given in the paper, providing it was clear that they also included the transmission and gearing losses, as the whole of the consumptions for turbines were referred to the power at the main shaft. Mr. Warne pointed out that the turbine consumptions given in the paper did not have the same value at half as at full power, although this was claimed as one of the advantages of geared turbine machinery. In point of fact it was one of these compromises which could be reflected in the cost of the machinery. If the turbine was given more blading for the same heat drop, as the efficiency curve plotted against mean blade speed/steam speed ratio was extremely flat, it only involved having the turbine with enough blading to be over the peak of the efficiency curve and then at half power the turbine would give the same consumption as at full power. In the paper the designs were those which were considered to be the most economical to construct, i.e. the best compromise, and even under such conditions the half load consumptions were not seriously greater than those for full power.

Turbine machinery was extraordinarily flexible in meeting particular requirements. For example, in a whale catcher turbine machinery could develop 400 h.p. under extremely economical conditions and be capable of developing 1,200 h.p. at short bursts during the period of chase. Mr. Welsh considered that the 1,400lb./sq. in. reheat turbine machinery shown in Scheme III was overdone. Dr. Brown did not agree as the difference in weight between it and Scheme II was not great and the consumption achieved in Scheme III involved only the use of ordinary materials increased in scantlings to meet the higher pressure, and in fact the cost of Schemes II and III would be practically identical. He suggested that the paper would have been seriously incomplete without an investigation of the merits of steam or gas reheating. The author however recommended the straight expansion of steam from boiler to condenser in association with high temperature providing this could be achieved without jeopardising the primary marine requirement of reliability. Mr. Welsh asked for more information on the steam reheat installation; Dr. Brown suggested that he should read the articles referred to in the bibliography in the paper and if there was then anything missing he would be pleased to supply it.

In his subsequent written reply, Dr. Brown congratulated Dr. Dorey on his appreciation of the papers, which was a most valuable one. There were only two minor matters in connection with the geared turbine. A further statement might be useful. The statement that the third alternative involving the use of 1,400lb./sq. in. pressure showed the improvement to be obtained by the use of reheat under this condition missed the point that this high boiler pressure was associated with a comparatively low total temperature of the steam and that while the scantlings had to be heavy to suit this pressure, the temperature was such that ordinary materials could be used for the casings, blading, rotor, etc. Dr. Dorey suggested that the reversing method by use of hydraulic couplings had been forced on marine turbine designers by difficulty of designing astern turbines. In fact this method of reversing was developed for use with a uni-directional output turbine as part of the complete gas turbine installation where astern turbines could not be fitted. It was then realised that improvements in design of the steam turbine could be made, as the elimination of the astern turbine would allow higher rotating speeds with the same critical speed margin to be achieved, and that the hydraulic couplings were cheaper than the astern turbine together with the ahead and astern steam manœuvring valves and piping. In the matter of use of hydraulic couplings, in addition it would be found that they had their best field at high revolutions when they were smaller, lighter and cheaper than alternative forms of couplings, and that was why in the design put forward hydraulic couplings were fitted where the power was transmitted at the highest r.p.m.

Messrs. Belsey and Robinson had raised the question of size of pipes and type of control gear needed for hydraulic reversing gear suitable for a 13,000 h.p. installation. The size of pipes required should be proportioned from the size for the 7,500 h.p. installation given in the author's verbal reply. The control gear required was extremely simple, admission of oil to the ahead and astern elements being controlled by a single piston valve operated by a lever having two alternative positions, one for ahead and one for astern. This operating lever would be interlocked with the engine-room telegraph.

In reply to Mr. Halliday Turner's remarks regarding the use of locked train gears, this system was particularly advantageous for the arrangement proposed in Scheme 1 since it allowed the power from a single cylinder turbine to be transmitted to the main wheel through two primary wheels and secondary pinions, thus permitting a smaller size of gears to be used for a large gear ratio.

The adoption of locked train gears for the remaining two schemes would lead to a reduction in the size of the gearing, but when more than one cylinder was involved the additional complication was a factor to be considered, bearing in mind that the numbers of primary wheels, secondary pinions, and associated bearings would all be doubled. This arrangement had, however, great advantages when really high powers had to be transmitted. With regard to the question of noise, experience with locked train gears had shown that they did not in fact give rise to more noise than other arrangements providing the usual requirements of accuracy, rigidity and alignment were maintained.

Mr. Turner had suggested that the drains from the oil fuel heaters and galley should be re-evaporated before being returned to the main feed system. This was a debatable point, but if Mr. Turner would re-examine the flow diagrams he would see that the steam supply to the evaporators was more than sufficient to provide the additional make up that would be required, and that therefore no increase in fuel consumption would be required or in fact change in the diagram.

The author agreed that an oil tanker would constitute a special case as regards the requirements for heating steam but this would in general be covered by fitting an auxiliary boiler as high pressure superheated steam was not required or in fact usable. To meet special requirements all the information was given in the paper to enable the requisite changes to be made, unlike the information in Mr. Turner's paper

Auxiliaries driven from the main turbine would obviously be limited to those pumps whose requirements varied in sympathy with the output from the main unit, for example, the lubricating oil pump, circulating pump and extraction pump. Such pumps normally had a larger margin so that no difficulty should be experienced when operating at reduced speed. During standby and when manœuvring, the standby pumps, which would be fitted in any case, would be in operation but such standby pumps would not require to be duplicated.

In addition the engine driven pumps having a simple mechanical drive with only the pumps themselves would be robust and simple and the power would be generated at the lowest steam rate.

The remaining points raised by Mr. Turner had been answered in the author's verbal reply to the discussion.

Most of the points raised by Mr. Calderwood in connection with the system of transmission put forward in Scheme I were dealt with in the author's verbal reply to the discussion. Regarding possible heating up of the astern converter when the machinery was running full ahead, the amount of heat to be dissipated was quite small since the ratio of air density to oil density was so small. In addition, the design of the converter was such as to minimise windage when running under the conditions referred to. As an added precaution a continuous small amount of oil passing through the astern converter had been provided for when running ahead in order to ensurue that overheating would not occur.

He was in agreement with Mr. Calderwood that astern torque was, in general, more important than astern power, but would point out that the astern torque available from a turbine was higher than for any other prime mover when passing through the condition of zero main shaft speed. Moreover the use of hydraulic reversing gear did not prevent a high value of astern torque from being realised, although the problem was rather more complicated.

Mr. Nithsdale's estimate of comparative figures for three different types of machinery were of considerable interest, and served to emphasize the advantages possessed by the geared steam turbine. Even so, the author believed that the figures for the steam turbine installation could be improved on without difficulty. The author was pleased to see that Mr. Nithsdale would advocate the geared steam turbine for powers down to, at any rate, 4,000 s.h.p. In this connection it should be stated that the marine turbine builders were prepared to design geared turbines for any power for which there was a demand, and that this type of machinery was well known for powers down to 2,000 s.h.p. with proved reliability and low upkeep costs.

In giving the information asked for by Rear-Admiral Parnall the author wished to give a warning that the evaluation of the efficiency ratio of any turbine depended on the manner in which the losses were assessed; for example the gland leakage losses affected the efficiencies of the different turbines differently. The figures included the leaving losses, mechanical losses, etc. Care should be taken before comparing these figures with others which might not be calculated on the same basis. The values of efficiency ratio on the basis indicated above were:—

Scheme I.	S.C.	Turbine,	77·5	per cent.
Scheme II.	H.P.	"	79·3	" "
	L.P.	"	80·2	" "
Scheme III.	H.P.	"	71·5	" "
	I.P.	"	83·8	" "
	L.P.	,	82·0	" "

Rear-Admiral Parnall's second point relating to the scope for improvement in turbine efficiency by practicable change of design raised questions difficult to answer fully. It was not appropriate to deal here in detail with the various factors in which improvement could be effected, although it was clear that as the steam turbine was already a well developed instrument further gains would require great efforts. The turbine designs presented in the paper were believed to show a definite improvement and it was considered that further improvements in efficiency would follow the trends given in this paper.

Referring to Mr. Warne's remarks, there seemed to have been a certain amount of confusion amongst the various authors regarding choice of propeller shaft speed and some authors had evidently chosen values of speed which they considered to be most suitable for their machinery. The point Dr. Brown had wished to make was that in any particular case if a low value of propeller speed were desired, this could be achieved easily with the geared turbine installation simply by providing a main wheel of appropriate diameter with a relatively small correction in weight.

In the discussion on the paper on the geared steam turbine a number of speakers referred particularly to the arrangement shown in Scheme I for ahead and astern running my means of a hydraulic coupling and astern running hydraulic converter. The author was pleased to have Mr. Sinclair's assurance that this type of manœuvring device had already been used in geared Diesel machinery. The question of heat dissipation in going astern at full power, having $5\frac{1}{2}$ million B.T.U.'s per hour to dissipate, had been referred to in the reply to the verbal discussion. The author was pleased to have Mr. Sinclair's warning about running the ahead coupling in the partially filled condition. There was no intention in the arrangement shown to run the couplings other than either full or both full.

The author was glad to have the assurance from General Davidson that Mr. Pounder's statement about the falling off in turbine efficiency with passage of time was not borne out in practice.

Dr. Brown was pleased to have the statement of Mr. Welsh that the proposals were progressive. He could not agree, however, that he had gone too far in Scheme III, as the option put forward between Schemes II and III was whether a slightly greater weight of ordinary materials would form the better compromise than the use of special materials in association with a direct expansion of steam in the turbines involving the use of higher inlet temperature. The use of steam reheat *per se* might not effect the thermal improvement but the scheme had to be taken as a whole and steam reheat involving the use of 1,400lb./sq. in. boiler pressure did improve consumption. The total capacity of the lubricating oil pump for the installation of 7,500 h.p., including lubricating oil supply to the turbine bearings, gear case bearings, sprayers and oil couplings was 50,000 gallons per hour. The oil temperature rise expected during full power stalled condition was about 30-35 deg. F. As already explained, it was not expected that this stalled condition would occur for more than a limited period and the temperature rise in the way of oil in the circulating system would accommodate it until the second lubricating oil cooler had been brought into operation. The author was pleased to have Mr. Welsh's commendation on the design of the single cylinder turbine.

Mr. Sherborne had referred to the problems involved in the development of a material suitable for heat exchanger tubes. Dr. Brown did not wish to enlarge on the problem at this time, but would assure Mr. Sherborne that his wish to see tubes abolished altogether in heat exchangers referred to his hopes for the development of the regenerative type of heat exchanger which gave promise of marked reductions of size and weight compared with the tubular type, and it was not evidence of any ill-will towards tube makers! they would as usual be an important factor in the construction of parts for gas turbine installations.

The President had requested information regarding fuel consumptions for 2- and 3-cylinder installations of the same power as the single cylinder turbine of Scheme I. Comparative steam consumptions for the arrangements in question were given below, and the corresponding fuel consumptions would be very nearly in the same ratio as the steam consumptions.

	Turbine steam consumption.
(a) 1-cylinder turbine with fluid coupling	Datum.
(b) 2-cylinder turbine without fluid coupling	$4\frac{1}{2}$ per cent. improvement over (a).
(c) 3-cylinder turbine without fluid coupling	$2\frac{1}{2}$ per cent. improvement over (a).

The consumption above was affected by the 98 per cent. efficiency of the fluid coupling. In other words the changes in consumption were such that they might be difficult to measure under seagoing conditions and it would be clear that the reasons for the choice of arrangement to be adopted would not be sought in fuel consumption alone but in robustness, simplicity, reliability, space occupied and weight.

Mr. Church referred to the arrangement shown in Mr. Davis's paper—"The Application of the Reheat Steam Cycle to Marine Propulsion with Special Reference to the C.P.R. 'Beaver' Class Turbo-Electric Cargo Liners" in the bibliography. It was agreed that this would give a good means of reversing but it involved having a high reduction ratio in the main speed box with a corresponding small reduction ratio in the first reduction box, leading to a sleeve type of double reduction gear. In addition, it involved some difficulty in the design of the uni-directional turbine. At this stage where they had put forward single cylinder turbines for particular installations, in particular steam installations, they had been able to arrange for an orthodox astern turbine and he was pleased to have Mr. Church's statement that such an installation built in 1922 was still running well. Mr. Church's criticism of the quill shafts as being highly stressed was a mistake as they were in fact only stressed to about the same extent as the line shafting to the propeller. In the case of the locked train gears, the resilience provided by these shafts reduced the stress in the gearing by cushioning the loads. There should be no necessity to renew any more than renewing the line shafting. On the question of the working turbine cylinder being enclosed by the exhaust belt, the heat transmission through the cylinder to the exhaust steam would be small as the area for heat transmission was restricted by means of a shield shown in the drawing, and in addition the exhaust steam in way of the high temperature belt was comparatively inert. It was noted that Mr. Church preferred Diesel generators for port use. These could be added to the arrangement shown in the paper and the figures for weight and engine room space modified accordingly. This change could however be made without making any change in the features put forward in the paper.

In reply to Mr. S. H. Dunlop, who asked the range of powers covered by the single cylinder turbine unit, this to some extent would vary with the vacuum required, i.e. at 29in. vacuum the top range of power would be about 7,000 h.p. and at 28in. vacuum about 10,000 h.p. at this stage. If the main argument for or against geared steam turbine machinery or turbo electric systems of propulsion revolved around the gearing versus electrical reduction, it would appear that the geared turbine had all the advantages of reduced weight, higher efficiency and lower cost, simplicity and robustness.

Mr. Ellison asked about the question of erosion taking place on the blades and nozzles, especially when churning with both couplings filled. At present their experience in the matter was of short duration, but it would not appear to be a serious factor in association with suitable materials for the drivers and driven members, particularly as the fluid in operation was lubricating oil. The author confirmed that the reliability in obtaining the impulse rotor by gashing out the discs was worth the extra cost compared to the rather cheaper method of building up from a series of discs fitted to a shaft, although in fact the costs were not far apart because of the great deal of labour in machining and fitting involved in the older method of construction. Trouble with main steam joints at any pressure might occur unless the piping was designed to have reasonable working stresses, and when full calculations of expansions and forces induced had been made. It was suggested that if the piping was properly designed for 1,400lb./sq. in. and if heavy flanges were employed with seal welding round the actual joint, there should be very little trouble. The author was pleased to see the comparison between the Diesel and geared steam turbine, which showed that the latter merited serious consideration in running costs with the Diesel machinery, and it should be remembered that it would have very much lower costs of maintenance.

The author was appreciative of Mr. Gemmell's favourable comments on the turbine proposals generally, and noted his interest in the hydraulic manœuvring system. Although this system had a more general application, he agreed it was of particular importance for the gas turbine, and it had in fact been developed in the first instance for that purpose.

Regarding proved performance of the hydraulic manœuvring system, he would refer Mr. Gemmell to Mr. Sinclair's contribution to the discussion for evidence that the system had already given considerable service in ships of moderate powers. In addition there was a great amount of experience behind the ahead coupling which was very similar to the usual design, and the system generally was of so simple and robust a character that it was not considered that it would require a prolonged period of service before its reliability could be accepted. He could not agree that 98 per cent. was an optimistic figure for efficiency of the ahead coupling, since the slip was a function of the size of the coupling, and could be reduced to almost any desired value by suitably increasing the dimensions. In this connection the effect of the high rotational speed in greatly reducing the size of coupling for a given slip should not be overlooked. It was not proposed to hand over detailed designs of the couplings, but the required dimensions of the ahead coupling for Scheme I could be scaled from Fig. 2 and of the astern element from Fig. 1. An approximate figure for the efficiency of the astern element was quoted in the paper.

Mr. Gemmell's query regarding heating up of the astern converter had been dealt with in the author's reply to Mr. Calderwood. The reference in the paper to twice the full load torque being available for manœuvring concerned the well established property of a turbine that with rotor stationary and steam supply fully open, the torque on the rotor was of the order of twice the normal full load torque. Similarly, high values of torque could be transmitted by the ahead and astern elements under conditions in which the slip was large. When manœuvring by means of the fluid couplings the turbine rotor would not actually come to rest, but the torque available when the propeller shaft speed was in the region of zero was still in the region of twice the full load torque as stated. (See also reply to Mr. Calderwood).

The steam sealed vacuum joint referred to on page 246 was that between atmospheric pressure and condenser pressure, namely, the low pressure gland in the outer casing.

With reference to the deflectors shown at outlet from the astern blading in Fig. 7, these were fitted for the purpose stated in the paper, and it was not considered that they affected one way or the other the question raised by Mr. Gemmell, namely, the effect of the tangential component of the outlet velocity on the flow distribution across the exhaust opening. This problem remained unaltered, but it was not thought to be a very serious matter since it only related to astern running, where economy within limits was not sought.

Mr. Gemmell was correct in suggesting that with a suitably designed contra-flow reheater it would be possible to reheat the steam to a temperature a little above the saturation temperature of the heating steam. The increase which could be obtained in reheat temperature by this means would, however, be small as Mr. Gemmell could prove, since by far the greater part of the heat given up by the heating steam was in the form of latent heat. In consequence the additional complication and expense was not justified.

Regarding the make-up feed quantities given in the paper, Mr. Gemmell's arguments based on the number of glands were not valid, since with the arrangement proposed the outer-most gland pockets of the turbine glands would be connected through the gland condenser to an exhauster fan, with the object of reducing the pressure in these pockets to slightly below atmospheric pressure. Leakage of steam to atmosphere would thus be eliminated and would, in fact, be replaced by a slight "leak-in" of air to the pocket. A warning was given against comparing directly the figures put forward by the author with those given in the paper by Saunders and Turner, since it appeared from Mr. Turner's contribution to the discussion that his figures included allowances for re-evaporating the drains from O.F. heaters, galley, etc.

It was agreed that the bled steam pressures shown in the flow diagram for Scheme III were too low to give the desired leaving temperatures without some contra-flow arrangement, and that a slight adjustment of the bleed points was required to provide the necessary temperature difference in the feed heaters. Such adjustment would have no appreciable effect on the other figures given in the flow diagram. The question whether or not to fit both air heaters and economisers to the boilers in any particular case was a matter for discussion with the boiler manufacturers, but the author considered that the proposals embodied in the paper were typical of good modern practice. For the purposes of the paper the question was not a vital one, since the design of boiler could be changed without materially affecting the particulars given.

In reply to Mr. Gemmell's question, Dr. Brown stated that the turbo-generators would be provided with a separate closed feed system for port use. It was agreed that it was not advisable to have steam on the main condenser in port as, apart from other considerations, this would involve keeping sealing steam on the main turbine glands.

Mr. Hart had raised the all too familiar objection that no features should be put forward for marine machinery which had not been tried and tested in service over a long period. Whilst the author agreed that due caution was necessary in assessing the value of any design, he believed that the features suggested in the paper were such as might be considered favourably. In addition a high pressure reheat installation was at present operating successfully at sea (see bibliography). Some comments on the reliability of the hydraulic reversing gear were made in the author's reply to Mr. Gemmell.

In reply to Mr. Payne's query regarding astern windage, the author stated that the loss due to this cause varied from about ½ per cent. in a normal design to about 1 per cent. in vessels, such as cross-channel steamers, in which an unusually large amount of astern power had to be provided. He agreed that an adequate amount of astern power could be provided by means of a conventional astern turbine, but pointed out that elimination of the astern turbine also eliminated the problem of casing distortion during astern running—a problem which was assuming greater importance with the adoption of higher steam temperatures—and also gave a more compact and cheaper installation as the question of rotor critical speed might limit the revolutions of the turbine and hence lead to an unnecessarily large casing (see the author's comments on paper No. V).

He was pleased to have Mr. Payne's confirmation that with adequate supervision, turbine efficiencies could be maintained, and noted with interest that he had found cylinder wastage to be only a very minor problem. Corrosion of the nozzle control valve shown in Fig. 2 could be prevented by the use of a suitable material, bearing in mind that it operated in an atmosphere of superheated steam.

He also agreed that with the arrangement of Fig. 4 precautions would have to be taken to secure adequate cleanliness when carrying out work concurrently on boilers and turbines, but thought this was a small price to pay for the advantages of the arrangement during the whole of the running time of the installation.

Mr. Souchotte's remarks on power loss in astern turbines were noted, and he was referred to the author's reply to Mr. Payne.

While not referring to this paper, the author was pleased to read the statement by Mr. Temple that his Company found hydraulic couplings to be cheaper, including the costs involved in the auxiliary machinery. The experience gained in service with hydraulic couplings should be useful in answering Mr. Ellison's query about erosion.

Mr. Wans had pointed out that the consumptions for the geared steam turbine were aprpeciably higher than those for the geared oil engines. It should, however, be pointed out that the geared steam turbine would utilise oil of any quality which could be burned in the boilers, and that the geared oil engine would only burn the lower grades of oil in association with high maintenance costs. In addition, the Diesel fuel was credited with higher calorific value than that allowed for boiler fuel in the outline specification given by the Superintendents Committee on which the various papers were prepared. He agreed with Mr. Wans that the paper would have been easier to write if a standard form of data sheet had been issued to the authors.

In conclusion, a good deal of information had been given in the paper in order to allow readers to modify any section to suit their own experience without affecting the main outline of the schemes or invalidating the general conclusion.

The author was pleased at the interest shown in the hydraulic manœuvring gear outlined in the paper, but he was surprised that the features had aroused so few comments, except in the case of the single cylinder installation.

A good deal of support for turbine machinery had been forthcoming in the discussion, and he had the satisfaction of knowing that so many agreed that the geared steam turbine was a very stong competitor for cargo vessels of high power.

Mr. C. C. Pounder said that in the course of summarising and comparing the various papers, Dr. Dorey had stated that the inevitable loss of propulsive efficiency with twin-screws must be debited against the direct-coupled and geared Diesel installations. He dissented. How many single-screw ships of 13,000 s.h.p. were there on the seas and how many twin-screw ships? Dr. Dorey also stated that Dr. Brown and himself had given lists of weights for both sizes of installation which, as far as could be seen, were complete and directly comparable. Again he dissented.

Regarding space occupied, the diagram particulars of savings could be illusory. For the typical ships which formed the basis of his work the 13 per cent. rule would not allow of other than minor reductions.

He understood Dr. Dorey to suggest, at the meeting, that all the papers might well be re-cast on a basis of 120 r.p.m. He dissented yet again. Countless times had it been said, by steam enthusiasts that one of the advantages of the geared turbine was that the conditions at the turbine and at the propeller ensured that the highest efficiency was obtained for both ends, whereas the Diesel engine must always be a compromise, with the propeller revolving faster than it ought to revolve. The Symposium was an excellent opportunity to put precept into practice, because the terms of reference stated: "the propeller speed should not exceed 120 r.p.m." But everybody chose the top revolutions—everybody, that is, except the Diesel man!

Dr. Brown had said that it seemed strange to put forward four engines to fulfil the requirements for 7,500 s.h.p. machinery with a change in weight of only 6 per cent. This point, however, was forestalled at the meeting. When reading the paper his exact remarks were as follows:

"Some of you may be critical of the fact that four different alternatives are offered for the propelling engines. The explanation is simple enough. Harland and Wolff are shipbuilders on a very large scale; in the Belfast and Govan yards, ships are built for many purposes and in many sizes. And so it comes about that, for the powers specified in this Symposium, two double-acting engine sizes and two single-acting sizes can be offered. I have included them all, if only to show what influence the various types have upon the overall design of the machinery installation. Had the powers been different, the same breadth of choice would not have been possible".

Dr. Brown referred to the fact that he put foward twin-screw machinery for 13,000 s.h.p. Certainly he did! At the present time a number of turbine vessels were being built with 13,000 s.h.p. on a single screw—they were building some in Belfast—but they were very much the exception. By far the great majority of ships of this power, whether Diesel-driven or steam-driven, were required by the owners, for various reasons, to have twin-screws; and so, whereas Dr. Brown had seized upon the exception and presentel his data accordingly, he had put forward proposals in line with the requirements of the majority.

With regard to the deterioration in fuel economy of the modern turbine vessels to which he referred at page 266, column 1, what the consumption would be after a service life of twenty years neither Dr. Brown nor the superintendent engineer nor anybody else could say, until the fleet was twenty years old. There were many components, other than the main turbine, involved in the deterioration. The Diesel engine fuel consumption figures were averages for many engines, which had been in service for many years. From the nature of things, he doubted if Dr. Brown could be in a similar position with regard to most of his proposals .

Dr. Brown referred to the shallow type of bedplate shown in Figs. 6 and 7. At the meeting he mentioned that the tabulated weights were on the basis of deep flat-bottomed bedplates. The point made by Dr. Brown, therefore, was devoid of meaning.

Dr. Brown said it remained a fact that, with the lighter initial weight of turbine machinery, it would require more than twenty days steaming before the weights were equal on his basis, i.e., before Diesel machinery showed to any advantage. The weight given by Dr. Brown for the 7,500 s.h.p. turbine installation was 592 tons. In column 1 of the direct-driven Diesel engine proposals on page 273, as example, the total weight was given as 1,065 tons. Therefore, the balance in favour of the turbine machinery was 473 tons.

It was instructive to pursue the comparison. To avoid controversy, let Dr. Brown's figure for the main machinery and boilers be uncritically accepted, namely, 236 tons. For the remaining items—apart from water and oil—his weight was 324 tons. He had compared this weight with the corresponding items of recent high-pressure-temperature single-screw installations built in Belfast of 485lb./sq. in. pressure, 800 deg. F. temperature, at turbine, and he found that the Belfast weights were 74 tons greater, the steam installations to which he referred having the same general practice as the single-screw Diesel machinery. It was clear then, that Dr. Brown was working to a different specification. It might, however, be said with accuracy and justice that Dr. Brown's level was quite satisfactory. He agreed; and so he reduced his Diesel machinery weight by 73 tons—it should be nearly 90 tons, but never mind, 73 tons would serve the purpose. This reduced the disparity from 473 tons to 400 tons. Then there was a difference in weight between boiler seatings plus turbine and gearcase seatings, plus doublers and ship attachments for feed heaters, uptake supports and so on which had no counterpart in Diesel engines; the contra items were the thick plates under the main Diesel engines. The algebraic sum was 33 tons against the turbine, say 30 tons. This reduced the disparity to 370 tons. Admiral Parnall—who was an eminent steam engineer—assessed the reserve feed water at 100 tons. Accepting this figure, again uncritically, the disparity in weight was now 270 tons. The smallest bunker capacity in any comparable ship known to him was 35 days; the largest was 60 days. Let them adopt 45 days. The difference in fuel bunkers to be provided for would be the difference in daily consumption between Dr. Brown's figures and his for 45 days. This additional bunker space for the steam machinery could not be obtained without making the ship larger, and a trifle harder to propel. Then there was the difference in propulsive efficiency between Dr. Brown's revolutions and his. This meant more effective power, which meant more fuel, which meant more space, which meant more weight, and so on. There was also the reserve feed water to be carried. Leaving the shipbuilder to pursue all these and kindred matters to their end, the increase of steel to the ship was 184 tons—say 170 tons. This further reduced the weight disparity to 100 tons in favour of the turbine. In the steamer, which had main boilers, condenser, feed tank, presumably also a distilled water tank separate from the hull, and so on, the weight of water and oil was given by Dr. Brown as 32 tons, whereas in the Diesel proposal—which had none of these things—oil and water was assessed at 36 tons. It was true that the terms of reference stated that lubricating oil in the storage tanks was not to be included; but as this seemed to him to be unsatisfactory he had included the reserve amount of crankcase and cylinder oil—in fact he had included everything necessary to enable the installation to put to sea. So it was reasonable to make some adjustment on that account. The reserve lubricating oil accounted for 15 tons, which reduced the disparity in favour of the turbine to 85 tons. Thus one could go on. If the matter was pursued and all points were dealt with in equity—still leaving untouched the main machinery items—the weight which started as 473 tons in favour of the turbine finished up at about 40 tons adverse to the turbine. In other words, before an ounce of fuel had been put on board, the results were against the turbine installation—and this was for the most advanced type of turbine machinery! Whether the voyage was long or short, therefore, the results on weight plus fuel were adverse to the turbine. So far as he was concerned fifty tons was neither here nor there in the argument. Thus was the bottom knocked out of Dr. Brown's contention that it would require more than 20 days steaming before the Diesel engine showed an improvement on the turbine, reckoned on a basis of weight.

It would perhaps be clear now, why the demurrer was introduced at the end of the paper on direct-coupled Diesel engines.

Regarding the geared Diesel engine, Dr. Brown was in error when he suggested that he was not very strong in urging its use. The essential strength of the position lay in the figures themselves. The whole range of ship propulsion could advantageously be covered by the geared Diesel engine—and not simply cross-channel vessels, as suggested by Dr. Brown.

Regarding Mr. Halliday Turner's remarks on main engine seatings, whereas Figs. 6 and 7 in the direct-coupled engine paper showed shallow bedplates, with seatings incorporated in the double-bottom, the tabulated engine weights assumed deep bedplates, requiring a normal double-bottom, with doubling plates in way of the holding-down bolts. The combined weight of seatings and double-bottom for a geared turbine installation was greater than that of the double-bottom and doubling plates for a deep-type Diesel engine bedplate. A considerable number of Diesel engines being built in their Belfast and Finnieston establishments had deep-type bedplates.

The lubricating oil figures given were daily make-up quantities. Centrifugal purifiers were available to deal with the lubricating oil, as for turbine installations. With the crosshead-type engines described in the paper there was no pollution. Although the terms of reference asked for reserve oil to be excluded from the weights, he had, in point of fact, included reserve oil; otherwise the position would have been illogical. In the 7,500 s.h.p. installation a complete charge, i.e., oil in sump plus oil in coolers, pipes, etc., weighed 11 tons average. The storage tanks, for crankcase and cylinder oils, contained an average reserve amount, when full, of 15 tons.

Mr. Calderwood suggested that it would seem as if he were making out the worst case possible for the Diesel engine. Superficially—and in comparison with data put forward by other contributors—this might appear to be so, but when the matter was analysed with care it would, he thought, be found that the basis adopted was the only sound one.. It was at least free from the personal equation, from guesswork, from factors of false optimism and from general humbug.

Mr. Calderwood suggested that a better case could have been made for the direct-coupled engine by adopting the top limit of revolutions allowed. He thought not—not if every factor was carefully weighed. He had given what experience showed to be the best all-round schemes for the shipowner, and there he was content to leave the matter.

The men of critical judgment, on the shipowner's side, whose opinions were those which ultimately mattered, were not likely to be misled by shallow superficialities. With such men was to be numbered Mr. Calderwood himself, and he took this opportunity of paying tribute to his excellent paper on internal combustion turbines.

With reference to Mr. Nithsdale's comments, he must decline to give any figures regarding prices. He would however say this, for Mr. Nithsdale's guidance: for a Diesel engine installation of the level of practice indicated in the paper, the price was, on an average, 6 to 7 per cent. less than for a high-class steam turbine installation. Occasionally the differential in favour of the Diesel engine was more. That was not surprising. Turbine installations, with their ever-growing ancillary equipment and with the increasingly expensive materials required, tended to become more elaborate and more costly, whereas with Diesel machinery the trend had been towards simpler designs and the almost exclusive use of mild steel and cast iron. It was not a question of diet differences between Belfast and elsewhere as Mr. Nithsdale suggested; it went deeper than that. The hull of a Diesel ship, being smaller, was also cheaper.

If he understood Mr. Nithsdale aright, he placed the competitive limit of the Diesel engine at 6,000 s.h.p. per shaft. He would suggest 10,000 s.h.p. per shaft.

Engineer Rear-Admiral Parnall took for gospel the particulars given in the papers on geared turbines and direct-coupled Diesel engines. That was a pity. It would be more satisfactory if he were to follow the epistolian advice and prove—that is, test—all things in the papers to see whether they be sound or whether they be unsound. In principle the proposition was simple. A shipowner required a ship to carry a stated weight of cargo or a stated cubic capacity of cargo at a stated speed. That was a problem for the shipbuilder; he determined the size of ship and its first cost. The running of the ship, with all the factors entering into the matter, was a problem for the shipowner and for the superintendent engineer. The only part the engine manufacturer played was to provide the shipbuilder with particulars of weights and spaces for alternative types of machinery, to indicate the fuel consumption per day and to prescribe the amounts of reserve feed water, lubricating oil, and so on, to be carried on board.

It was fortunate that most shipowners had clear notions regarding the kind of machinery they required, but once in a while a shipowner with an unusually acute analytical mind came along and required proposals to be worked out on the basis of all acceptable types of propelling machinery. It was upon such carefully worked-out proposals, over the years, that he had based his judgment. The results had led him to draw attention to matters, the significance of which did not readily occur to a marine engineer, nor to an engine designer who was absorbed only in prime-mover problems.

On the basis of the foregoing remarks the figures given in Admiral Parnall's fourth paragraph would not bear scrutiny. To neglect everything except weight of machinery and fuel consumption per day was, in his opinion, to adopt unsatisfactory premises, and the conclusion, therefore, was invalid.

Adjustments of a few frame-spaces in ships which already had the machinery spaces as small as the tonnage regulations would allow was not always practicable. He could not recall any ship of the category with which the Symposium was concerned in which the engine room length could be compressed by from 10ft. to 23ft., as indicated by Admiral Parnall, without the shipowner being compelled to forego the 32 per cent. deduction—which he would not do! The sentence which occurred at the fourth paragraph from the bottom of column 1, page 264, was influenced by the factors indicated above.

To him, it was always a genuine pleasure to read Dr. Tuplin's contributions to the Press and to technical discussions. On the present occasion, as usual, his comments were of high value; a great deal of useful information was embodied in them. He had never encountered conditions which might indicate that three-point support for marine gearcases was desirable. With substantial construction of gearcases, soundly-built foundations, and a suitable design of flexible coupling there was no difficulty.

Mr. Warne asked if manufacture of the engine type shown at Fig. 1, p. 257, was likely to be continued. The reply was in the affirmative, until clients ceased to ask for it.

Regarding Mr. Warne's second question, the rings in the main piston could be arranged at the desired distance from the top, hence the design of the screwed-on concave crown permitted of adequate cooling at the inner circumference; but in the exhaust piston, where the rings must be near the piston end, the screwed-on end only permitted of the required amount of cooling if the crown was made convex. The shape of the combustion chamber was very satisfactory.

In Mr. Dimmick's excellent statement there was only one point for him to answer; it was in his last paragraph. His recollection was that the wear on the gearing was about one-quarter of the tooth thickness; certainly it was an inordinate amount.

Mr. Sinclair's remarks were clear. As stated at the meeting, it was his opinion that the time was overdue for an advance to be made in hydraulic coupling design for the powers with which the Symposium was concerned. The hydraulic coupling as at present offered was not so favourable as the electric coupling; at least that was his experience from an analysis of the various proposals received from time to time.

Major-General Davidson suggested that possibly the choice of the straight Diesel engine drive by Scandinavian shipowners might be due to there being but little steam turbine industry in Scandinavia. He did not think so. At the present time they were building many installations for Scandinavian owners, and in every instance Diesel machinery had been specified. This had been done in instances where British owners, in competition with them, had specified steam turbines. It had always seemed to him, from contact with Scandinavian shipping people, that they were extremely shrewd business men.

The President stated that he could not understand a cryptic remark which occurred near the end of the paper on direct-coupled installations, to the effect that in due course the swing would be away from steam machinery. Speaking for himself, he was not very happy with regard to certain trends at the present time. They bore too close a resemblance to unsatisfactory practices of twenty-five and thirty years ago.

It was a refreshing change from the views of men interested only in the building of machinery to read the remarks of Mr. Church, as an owner and superintendent engineer.

Mr. Church asked why Fig. 3, p. 258, was not called an opposed piston engine. By an opposed piston engine one rather visualised a design in which the top piston-stroke was a larger proportion of the main piston-stroke than was so in Fig. 3. The evolution of the design was simple. Initially the engine had cam-operated valves arranged in cylinder covers. The central exhaust valve was replaced by a small piston, whose movements were regulated by a layshaft, later by two eccentrics. In due course the cylinder covers were discarded and the piston valve was enlarged in diameter to that of the main piston. The eccentric gear remained.

On the subject of oil vapour extraction from crankcases, there was little to be said that was not already known. A certain amount of ventilation was necessary, or the work done by the moving masses in creating frictional heat inside the crankcase would increase the contained air temperature until the rate of heat dissipation through the doors and adjacent framing balanced the rate of production. The internal air pressure would also become greater.

Regarding geared Diesel engines, he agreed with Mr. Church that hydraulic couplings, as an alternative to electro-magnetic couplings, could be made more attractive.

Mr. Ellison would gather from replies made to other participants in the discussion that, in his opinion at least, machinery weight in accordance with the terms of reference of the Symposium plus weight of bunker fuel consumed in the voyage was not a criterion. The crux of the matter lay in the first eight words of Mr. Ellison's last paragraph.

Mr. Ellison correctly summed up the relationship between piston pressure, engine speed and liner wear.

Mr. Gemmell's remarks on direct-coupled and geared Diesel machinery were very sound. He agreed with him that the terms of reference of the Symposium might well have indicated whether crank, tunnel, thrust and tail shafts were required to be made to classification rule sizes or to some margin above them. In the absence of such direction he had adopted the requirements specified by clients of high standing.

Mr. Hart asked if a single-screw Diesel engine proposal could be put forward for 13,000 s.h.p. on the basis of well-tried designs similar to those indicated for twin-screws in the direct-coupled engine paper. The reply was in the affirmative. Such a proposal would comprise a 10-cylinder double-acting two-stroke engine 660 mm. (26in.) bore, 1,500 mm. (59in.) stroke, running at 103 r.p.m., the rating at this power being 80 per cent. Six engines of this size had been in service for over ten years, and the wartime record of the machinery was excellent. The total weight of such an installation in running condition, computed as in the paper, was 1,558 tons.

The remarks of Mr. W. H. Purdie were characteristically sound. Amongst other details he referred to the eccentricity of main bearings. In steam engine days, many of their engines were given one-eighth of an inch, i.e., 3 mm., eccentricity, and this practice was carried forward to Diesel engines. Later, the eccentricity was reduced to 1 mm. Many Continental-built engines were made with concentric bushes—which an engine builder would naturally prefer; but when he had tried to abolish the eccentricity there had been opposition from superintendent engineers. He agreed that 1 mm. appeared small, but it seemed to work!

As might be expected, Mr. Purdie fully appreciated the significance of what he called *real* machinery weight—which was that mentioned on page 266 of the paper on direct-coupled Diesel engines.

Mr. Souchotte, as the superintendent engineer of a very important and progressive firm of shipowners, was fully alert to all the issues involved in present-day machinery installations. The remarks in his first two paragraphs were an excellent summary of the position.

Mr. Temple's admirably clear statement of his experience with geared Diesel engines was much to be appreciated.

Regarding the hydraulic coupling, his actual remarks at the meeting, when presenting the paper, were as under: "The weights, spaces, and so on, are on the basis of the electro-magnetic coupling. This seemed to me to be the prudent thing to do. In my opinion the hydraulic coupling, as at present offered, is not so favourable as the electric coupling. This, however, is not to say that the hydraulic coupling could not be made as fully favourable. In fact it is my opinion that the time is overdue for an advance to be made in hydraulic coupling design, for the powers with which this paper is concerned".

He agreed with the comments of Mr. Wans on geared Diesel engines, also with his opinion regarding costs.

Messrs. J. G. Belsey and **J. G. Robinson,** in replying to the discussion, noted that Mr. Calderwood considered their overhauling times conservative; this was in line with their intention throughout the paper where they had endeavoured to be moderate and give as broad a picture as was possible from the limited data at their disposal. They confirmed that the times for changing head valves referred to in the Sulzer engine were for scavenge valves.

With regard to the uncooled pistons of the M.A.N. engines they would state that these engines were known to have given satisfactory continuous service during the war years and were operated at a power 30 per cent. in excess of that given in the paper, and it was also known that the engines had on occasions run with overloads up to 100 per cent.

Mr. Dimmick's views on the arrival of the gas turbine into the field of marine propulsion were, they considered, more of a pious hope at present than a reference to an actuality. It remained to be seen whether the multi-engine Diesel-electric method would not take and maintain the lead; in any event there was no technical limit to the size of Diesel-electric schemes.

The arguments in favour of the use of direct current for the cargo liner and the tanker were developed in the paper, and since d.c. was required for excitation purposes as well as for the wide range of speed control required by the particular cargo handling plant, the choice appeared to be logical for the ships under consideration.

Mr. Trickey's helpful remarks regarding multi-engine drive were concurred in, and the authors regretted that present day conditions did not permit a proper investigation and assessment of relative costs.

The authors would like Mr. Welsh to read the paragraph "Choice of Prime Movers", where he would find their reasons for putting forward the engines referred to. They were not biased one way or the other—all they aimed to do was to provide material from which members with their likes and dislikes could form their opinion of the flexibility and merits of this type of propulsion. The M.A.N. engine was chosen for the sound reasons which Mr. Welsh gave, but the charge of flimsy construction was not substantiated and it was evident that if the contributor was familiar with the engine, he held a very different opinion from that of the many technical people who examined the types fitted in German long range U-boats and analysed the performance records.

Their reasons for not considering higher rotational speeds had already been given to Mr. Church.

It was thought that Mr. Welsh was possibly confusing the state of emergency that existed when a piston of ten times the weight on a single large slow turning engine had to be changed and the ship was immobilised whilst all of the engineers turned to. With proper equipment and the ship still under way, the authors considered it perfectly feasible to withdraw pistons at sea from the engines put forward.

Mr. Welsh, no doubt, was thinking of the Brown Boveri system of paralleling alternators using heavy damping windings; he had, however, missed the important point that it was intended to use the main bus bars for auxiliary purposes, and this could be achieved by the method described, but it was not possible if the alternators were allowed to fall out of step. The number of alternators was not doubled, the windings were merely divided. Mr. Welsh's simple statement, they felt sure, would not satisfy the contributor who had asked "What will happen if one alternator loses speed?" and they now more than ever regretted that they had been unable to describe fully the governing arrangements.

The authors noted that Mr. Church shared their view that the indirect Diesel-electric method of propulsion might indicate the shape of things to come. They entirely agreed with him in regard to the question of noise, and their reasons for raising the matter was to give the engine makers a further prompting to do even more than the very creditable efforts they had made during recent years.

They purposely avoided the very high speed engines for they were at present very selective in their fuel requirements, and the authors felt that in the not very distant future engines of the rotational speeds put forward would be operating on the more viscous and cheaper fuels than the present day Diesel oil.

In common with one or two other contributors, Mr. Church raised the question of the 63 cylinders for the larger power. They did put forward a 35 cylinder arrangement which also had no cylinder head valves, for members who opposed the greater number of cylinders, but alas! criticism was fired at the other man's likes without any apparent technical objection.

Referring to Mr. Gemmell's contribution, as stated previously the authors had endeavoured to give a paper on as broad a basis as possible, covering two different types of ships and three makes of prime movers. A limit to the variety had to be reached, and in the case of the large engines they had put forward the minimum number of units required. The appropriate number for any specific ship would be decided by the shipowner's requirements. If good maintenance facilities and time were avilable at the particular ports concerned, then obviously a spare engine would not be justified, for it should be noted that all of the engines put forward had been well tried in service, and the overhauling periods were no more frequent and the equivalent work no greater than that necessary with direct drive engines.

If the ship's trade and the port servicing facilities did not allow this method of operation, then the fitting of a spare unit and the employment of additional engine room personnel to do the overhauling at sea would prove the logical solution. The authors favoured this latter method, and this was the reason for their reference in paragraph 6 under the sub-heading "Personnel". Mr. Gemmell was right in drawing attention to this point, and they regretted that they did not cover for a spare unit in each of their arrangements. They did not agree, however, that two spare units should be carried, as the "yardstick" would be to a great extent determined by the tools provided for carrying out the overhauls and the number of personnel available for doing the work.

By "installed capacity", did Mr. Gemmell mean the maximum power available? It appeared to read this way, but surely it could not be meant, for apart from economic considerations, there was no justification for providing machinery of four times the service power required. It was suggested that the reference given in Dr. Brown's paper had been misread or misunderstood.

Finally, on the question of spare parts, the instructions laid down by the Superintendent Engineers' Committee of the Institute made it clear what breakdown of weights was required and they had complied with those instructions.

Referring to Mr. Hart's remarks, as already stated in their reply to Dr. Dorey the authors felt that it was incorrect to say that the indirect Diesel-electric drive was as yet untried. They agreed that in this country this form of drive had not been developed for the higher powers, and the reason might be that the shipbuilders with their attendant or associated engine works preferred to produce machinery with which they had had long experience rather than set up new electrical departments for a type of propulsion which they considered was still in its infancy.

The authors were glad to see that Mr. Mayor entirely shared their views.

They were pleased to note that Mr. Payne was in general agreement with them regarding the present high cost and time spent in port for periodical and incidental repairs on ships. This view was no doubt shared by many others, and should be given the consideration it deserved when owners were considering building future ships.

When choosing the rates of engine revolutions they also had in mind the question of fuel discrimination as well as the heavier maintenance, which Mr. Payne pointed out took place with engines having rotational speeds of 1,000 r.p.m. This again was a point demanding attention from the individual owner, for under certain conditions it might be advantageous to install smaller high speed units which could be removed and serviced ashore.

In reply to Mr. Souchotte, the ratings given appeared to the authors to be reasonable and in line with proved practice with the class of engines put forward.

Regarding the governing mechanism (electrical and mechanical), as mentioned in their presentation remarks, they had hoped to make their paper more interesting by including a full description of the governing mechanism of a 4-unit Diesel-electric installation now under construction in this country, but due to the prevailing delay the machinery was well behind schedule and it had not yet been possible to test the new arrangements, and a description here was not therefore appropriate.

The limit to the amount of work done on board a ship was to a large extent determined by the facilities available and the number of personnel available. The economics of this could only be determined by the shipowner who built a particular ship for a specific trade.

The point made regarding the overloading of the prime movers if one or more failed was no more serious if suitable governors and switch gear were provided than would be the case if one or more cylinders in a direct drive engine were cut out.

Mr. T. Halliday Turner, in reply to Dr. Brown's remarks, regretted that he was unfortunately mislead by the author's choice of words in the paper, which having been intended to excuse the omission of data which the authors were not in the habit of preparing, had been turned into a testimonial of their capabilities. There were varying degrees of specialization, and only Mr. Pounder claimed to be capable of covering the entire machinery installation within his organisation. Dr. Brown, on his own admission, obtained his boiler particulars from a boiler maker. He (Mr. Turner) and his co-author preferred not to confine themselves to one design of boiler, as boilers differed so much in weight and dimensions, but their statement stood that the same boilers would be adequate for both geared and turbo-electric machinery. They felt sure that no shipyard estimator would have difficulty in adding the correct shafting, according to Lloyd's Rules, and all the other details which had been excluded from their weights. They would not consider it an indication of incapability for those who were in the habit of making and selling no more than the design drawings. They also served the shipbuilders very well in some circumstances.

They might have been misinformed about the amount of trouble experienced with astern geared turbines, but one chief engineer, who struggled across the Indian Ocean on h.p. and i.p. turbines with a bent l.p. rotor disconnected and not safe to run, was now in a turbo-electric tanker and had told them that his only worries now were maintaining the purity of his feed water and the behaviour of his Diesel driven auxiliaries.

Reverting to the type of feed pump, Mr. Turner was aware that it was possible to get feed pumps with as low steam consumptions as those stated by Dr. Brown, but when a superintendent engineer preferred and specified a centrifugal feed pump, then its higher steam consumption must be included in the heat balance and appear in the fuel consumptions, and his figures were based upon steam turbo-driven centrifugal feed pumps.

Mr. Howard had explained the somewhat low speed reduction ratio of 26 to 1, which was stated for turbo-electric proposals, as being due to the conditions for which those particular machines were designed, and that higher reduction ratios could be obtained. It was quite possible to go further by using an arrangement similar to Mr. Calderwood's Fig. 3, where gears were interposed between the turbine and alternator, and such an arrangement might be well worth while if the greater noise were permitted.

The first critical speed need not be more than 20 per cent. above the maximum running speed, which was the full power service speed, as any momentary rise above the maximum running speed was only transitory and would not reach the critical speed. The harm which could arise from reaching the critical speed developed from dwelling on the critical speed, and was not inevitable with a well balanced rotor.

In reply to Mr. Welsh, Mr. Turner corrected his belief that The General Electric Co. Ltd. of England was associated with the General Electric Co. of America. They had seen the 7,500 s.h.p. turbo-electric tanker made by the General Electric Co. of America, and it had a single casing turbine and not a double casing turbine such as was favoured by Dr. Brown and Mr. Welsh. He suggested that if Mr. Welsh looked back 25 years, he would find that Mr. Pochobradsky had designed a turbine with a separate internal casing, although only the h.p. part of the turbine was arranged with an internal casing. The advantages claimed were much the same as those stated by Dr. Brown. These advantages, however, proved to be illusory and the disadvantages real, and only a few were built to this design. As a matter of interest, two 10,000 kW. turbines of this design had been running for 20 years at Sunderland Corporation Power Station.

In reply to Mr. Church, they would say that the number of auxiliary turbo-generators included in the weights stated in the paper were:—

Two for the 7,500 s.h.p. single screw proposal,
Two for the 7,500 s.h.p. twin screw proposal,
Three for the 13,000 s.h.p. single screw proposal, and
Four for the 13,000 s.h.p. twin screw proposal.

In reply to Mr. Gemmell, the calorific value of 18,000 B.T.U's per lb. had been agreed to be appropriate to the known specific gravity of the oil, 0·982, and was assumed, there having been no sample tested. The turbine nozzles were calibrated for size, and the discharge co-efficient for this type of nozzle had been established as 0·975 on previous more elaborate tests on many turbines.

The steam/oil ratio of 13·2 which was stated in the trials summary was derived from the total heat to steam of 1,096 B.T.U's per lb., the calorific value of 18,000 B.T.U's per lb., and the boiler efficiency at 80·3 per cent. The ratio of 14·75 was, however, for different steam and feed conditions, requiring 1,090·5 B.T.U's per lb. with oil having 18,500 B.T.U's/lb. calorific value, and for 87 per cent. boiler efficiency, these conditions having been chosen only to bring the result to the agreed basis of calorific value and boiler efficiency, neither of which were attained on the trial.

The make-up allowance of 10 tons per day which was made in the new design for 7,500 shaft horse power was chosen as a round figure covering all the losses which occurred on a voyage and not on a trial trip. Each superintendent engineer had his own practice in boiler treatment and blow-down, and soot blowing might use up to 5 tons per day. The amount of make-up was, therefore, unpredictable, but their experience showed that 10 tons per day make-up for 7,500 shaft horse power was a fair if somewhat generous allowance.

The three Diesel engines referred to by Mr. Gemmell were installed in a separate compartment, but in Fig. 8B three were shown accommodated in the engine room.

Mr. J. Calderwood, replying to the discussion, wrote that in his early remarks Dr. Dorey mentioned the question of first cost and at the same time, in relation to the combustion turbine. the fact that service experience had not yet been gained. The latter must, of course, be agreed and, as was stressed in the combustion turbine paper, this type of machinery could not be directly compared with the others until more experience had been gained with it. However, within the next few years there should be considerable opportunity of gaining service experience with this type of plant and it was to be hoped that shipowners, with a view to possible future savings, would be prepared to try out the combustion turbine at sea on a large installation, which was the type of plant for which it was best suited.

As regards cost, at the moment the cost of the combustion turbine was likely to be comparatively high as very heavy development charges were involved, not only on the part of the actual turbine manufacturer but on the part of their sub-contractors, such as steel suppliers. Even so, it would appear that the cost of a combustion turbine of large power was not likely to be greatly different from the cost of other types of machinery that were available.

Dr. Dorey further suggested that the marine engineer had at the present time a real opportunity of keeping pace with, or getting ahead of his counterpart on land, particularly in the development of the combustion turbine. This type of machinery seemed for many reasons particularly suited to marine work, whereas in this country at least it did not show to the same advantage on land until such time as it might be possible to consider it as a coal burning plant. It was to be hoped that marine engineers would take advantage of this and assist in the development of this new type of prime mover.

Dr. Dorey referred to the introduction of hydraulic or electro-magnetic couplings in geared systems in relation to the question of torsional vibration. Whilst his remarks on this question were not directly relevant to the combustion turbine, it should be pointed out that such non-mechanical forms of transmission were effective only to a limited extent in isolating the torsional systems of the engine crankshaft from the shafting with propeller and gears; in fact any known type of transmission could carry a torsional vibration from the driving machine to the driven unit. It was true that the characteristics were not the same as for a direct-coupled unit; the phasing between the two parts of the system would differ and also the amplitude would be modified. However, whether a hydraulic or a magnetic coupling was used, any fluctuation in speed in the driving element of the couplings must be reflected as a fluctuation in torque in the driven element and this factor should not be overlooked in considering such installations.

Dr. Dorey and certain other speakers seemed to be under the impression that the reversing pitch propeller was put forward in the combustion turbine paper as the principal method of reversing for this type of machinery. It should be made quite clear that this was only one of the three proposed methods and it was pointed out in the paper that little experience was yet available with large reversing propellers. It seemed almost certain, at any rate in the early stages of development of the combustion turbine, that electrical transmission would be preferred as being a fully tried system which had the further advantage that two turbines could be used on a single screw vessel, so that the safety of the two propulsion units with the new type of machinery could be obtained, combined with the improved efficiency of single screw propulsion. This was not easily possible either with reversing gear box or with reversible propeller.

Dr. Dorey referred to the difference in efficiency for single and twin screw propulsion and this was fully appreciated. However, so far as the specification for the Symposium was concerned, only a definite power was called for and as all particulars were available for twin screw combustion turbine electric set, this was used in preparing the table included in the paper. It was interesting, however, to note that if the electric drive proposal were corrected for single screw but still maintaining the two separate power turbines, and the corresponding improvements in propulsive efficiency taken into account, then the consumption with electric drive would be the same as or slightly lower than the consumption for twin screw geared drive.

As mentioned above, this would be one of the important considerations in deciding on the type of drive to be used with combustion turbines, particularly in the early stages of development.

Incidentally this alteration would also bring the weight for the electric drive down to a figure slightly below the weight for twin screw gear drive.

In his comparisons of fuel consumption and weight, Dr. Dorey had, no doubt quite unintentionally, been rather too favourable to the steam turbine in relation to the other types of equipment. To deal with fuel consumption first, he had, in the case of the Diesel electric drive, added auxiliary consumption to bring this into line with the direct Diesel and combustion turbine proposals, but similar corrections to the consumption figures had not been made for the geared steam turbine nor the turbo electric, although in neither of these papers had the author made as large an allowance for auxiliary power for ship service auxiliaries as had been made in the case of the direct and geared Diesel engine and in the combustion turbine. It was rather difficult to determine exactly the power allowance given in the fuel consumption figure in the steam proposals for ship service auxiliaries, but it would appear to be about 150 kW. or the equivalent in heating steam. On the other hand, in Mr. Pounder's two papers the surplus auxiliary power allowed over and above that required for main engine auxiliaries was approximately 610 kW., whilst in the combustion turbine paper the corresponding figure was 550 kW.

If the auxiliary power for services other than propulsion was in every case corrected to correspond with that allowed in the steam turbine papers, i.e. 150 kW., then there was an improvement of some 5 per cent. in the consumption figure in Dr. Dorey's table for all of the other types except the turbo electric. The figures in Dr. Dorey's table would then become :—

Geared steam turbine	100 per cent.
Direct coupled Diesel engines	71 „ „
Geared Diesel engines	76 „ „
Diesel electric	78 „ „
Combustion turbine geared with variable pitch propeller	80 „ „
Combustion turbine electric drive	83 „ „

As would be seen from the footnote added to the table in the combustion turbine paper subsequent to the reading of the Symposium, allowance was made in giving the fuel consumption with reversible pitch propeller for the lower efficiency of this type of propeller, and if the combustion turbine were used with a reversing gear box, such as proposed by Dr. Brown, the consumption relative to the steam turbine would improve to about 78 per cent.

In comparing weights it must be agreed that Dr. Dorey's tabulation was a much fairer comparison than a direct use of the total figures as given for each type in the Symposium; however, as Dr. Dorey himself pointed out, a true comparison would require considerably more detailed study. For example, the weight of the geared internal combustion turbine on Dr. Dorey's corected figure was the same as that of the geared steam turbine, assuming that the weight of auxiliaries for both was equal, but in fact the auxiliary weight for the combustion turbine would be considerably lower. Circulating pumps and piping, for example, were of very small size compared to the large pumps and pipes on the steam installation; further, no steam pipes and fittings were required, all of the high pressure gas piping and fittings having been included in the turbine equipment weight.

Finally, the combustion turbine paper allowed 100 tons for gearing as against the 77 tons allowed by Dr. Brown for the steam turbine. Whilst these figures for the gearing were correct for the twin screw combustion turbine against the single screw steam turbine, it was evident that the weight comparison would be more favourable if a single screw combustion turbine were compared. In relation to all of the electric drive proposals Dr. Dorey's weight comparison required some correction. Whether the motors were arranged directly behind the engine room or right aft, there would be come considerable saving in shafting weight with an electric drive and, as would be seen from the table in the combustion turbine paper, the difference in weight between geared and electric drive for that type of unit was very small, whereas in Dr. Dorey's curves the difference in weight was shown as being 9 per cent.

Regarding the question of space, Dr. Dorey's comparison might have been carried a little further, as it dealt only with engine room length. The steam turbine proposal had taken advantage of boilers arranged above the main deck level, whilst none of the other proposals, except the Diesel electric, had similarly taken advantage of space in the upper part of the machinery room.

In the case of the combustion turbine it was felt that the majority of sea-going engineers would prefer to have the machinery on one level and this arrangement was drawn up at the cost of some sacrifice in simplicity of piping, and very considerable sacrifice in engine room length. The combustion turbine could very conveniently be arranged with some of the machines at a higher level, i.e above the main propulsion turbines, and if this were done the overall length of the engine room would be much shorter than with any other type of machinery.

There was, however, another consideration which must not be overlooked, namely, that in the class of ship for which machinery of this and higher powers was required, space on the upper decks was usually considered to be of great value and the majority of owners of this class of ship would prefer to sacrifice some space with a greater engine room length below the main deck rather than to sacrifice more valuable space above main deck level. It was felt that a fairer comparison of space would have been on total engine room volume, and in this respect the geared steam turbine would not have shown to such good advantage in comparison with any of the other types.

Dr. Dorey mentioned that the funnel appeared to have been omitted in the weights for the combustion turbine. This was not correct, as an allowance for the funnel was included, although it was not specifically mentioned in the tabulated list.

In criticising these points in Dr. Dorey's summing up it was fully realised that it was an almost impossible task for anyone to make a correct and detailed comparison of this wide variety of machine types without lengthy discussions with the various people involved, for which, of course, there was no opportunity, and it must be agreed that in spite of the above criticisms, Dr. Dorey had made a very fair and impartial summing up of the situation so far as the information given in the various papers had allowed.

His final remarks regarding maintenance and engine room personnel were particularly valuable, as these two questions must be among the main considerations of any shipowner to-day.

In his remarks on the combustion turbine paper, Dr. Brown drew attention to the fact that Pametrada had now a 3,500 b.h.p. set in an advanced stage of construction. It was pleasing to see his remarks on this as it was hoped in presenting the paper that others who had similar plant under development would come forward and describe it. It was, however, rather disappointing that Dr. Brown had said so little about the machine that had been developed, and had confined his remarks mainly to contentious arguments as between the relative advantages of the normal open cycle and of the Sulzer cycle. One would have preferred in the present stage of the development of the combustion turbine to keep such arguments in the background, as it was evident that only experience of running would finally show which arrangement was best suited to marine service, and particular care was taken in the combustion turbine paper to point out that there was no reason why similar overall efficiencies to those claimed in the paper could not be reached with other types of combustion turbine. No attempt was made to claim that the assets of the Sulzer cycle for marine work were all peculiar to that cycle. However, Dr. Brown having opened the arguments on the relative merits of the two types of cycle, there was no option but to reply to his criticism as fully as the information that he gave would permit.

In the first case he stated that the 3,500 s.h.p. set being built by Pametrada could be converted to a 7,000 s.h.p. set simply by fitting a third shaft unit in the form of a supercharger. At the same time in other parts of his comments he had laid great stress on the proper matching of components, and to alter a machine designed for 3,500 s.h.p. and make it suitable for 7,000 s.h.p. would be by no means so simple as he suggested if proper matching of components was to be maintained.

It was also not quite clear how the simple fitting of a supercharger unit, which would not alter the basic efficiency of the main cycle, could improve the overall efficiency from 30 per cent. to 33·8 per cent. as one gathered he claimed in the latter part of his remarks.

Dr. Brown next remarked that in the paper only one photograph —of the compressor rotor—was shown, and in his reply had shown a series of photographs of various turbine parts. Whether any of these had actually run or not was not stated. It would have been quite easy to have shown in the paper a whole series of photographs of parts under construction for the 7,000 h.p. unit and also of various test turbines that had previously been built and run, including a previous unit of 7,000 h.p. which was supplied with its gas by free piston compressors. However, it had not been felt, and still was not felt, that to show such photographs would be of any value at the present time.

During the progress of work on the 7,000 h.p. turbine the policy had been to try out various parts individually, and illustrations had from time to time appeared in various technical papers showing the photographs of parts that had been tried out. The Symposium, however, dealt with an installation of specific size and output, and until an installation of this size and output had run successfully with efficiencies which would be attractive to the shipowner, it was not thought that any useful purpose was served by showing a number of photographs to illustrate previous researches or the stage of the work in progress.

Dealing next with the heat exchanger question, Dr. Brown made a somewhat rash statement that the three heat exchangers shown were as big as the single heat exchanger for the open cycle. A statement of this kind was extremely vague and could not be discussed at all unless more accurate data were supplied, particularly with regard to the efficiencies of the recuperators, tube dimensions, velocities and the general arrangement of the heat exchange surfaces (longitudinal flow, transverse flow, etc.). A statement which could be made, however, was that the Sulzer cycle provided especially favourable conditions for the attainment of high efficiency with heat exchange equipment of moderate dimensions. This was due to the facts that the coefficient of heat transfer depended on the pressure and that small tubes could be used for recuperator 6 (whose efficiency was particularly important) because no combustion gases were handled. The heat exchange surfaces of elements 7 and 9, on the other hand, were composed of tubes of large diameter, special importance having been attached to ease of cleaning. If it be attempted to apply the same large tube design to an open 7,000 h.p. plant of the normal type, a heat exchanger efficiency of about 70 per cent. being assumed, it was found that, while it was not impossible to build the apparatus as a single unit, its dimensions made a subdivision into two or even three units advisable. To substantiate this statement a body of accurate data would, of course, be required. It had been cited only to illustrate that Dr. Brown's argument was irrelevant as it gave no facts on which a true technical comparison could be made.

Dr. Brown then referred to the paper read last year before the North-East Coast Institution, and the differences in the cycle now shown. It was clearly pointed out, at the time when the paper was read in Newcastle, that the cycle there shown was purely diagrammatic to show the general principle involved, and it was stated that in practice the arrangement would be modified. This was emphasised again in reply to criticisms at that time of the matching of the units in the cycle as then shown. At the time of the reading of that paper it was impossible to show the cycle as now described, as certain patents which had been applied for were still pending, but the reply to the discussion at the North-East Coast Institution should have made it clear that the various criticisms of the then published cycle, which Dr. Brown now repeated, had already been considered, and that with the cycle actually adopted the question of the matching of the various units, the combustion chamber temperature, etc., would be cleared up; it was hoped that the cycle now shown had completely cleared Dr. Brown's doubts on these points.

In relation to the comment at the foot of page 330 in the paper, Dr. Brown seemed to have assumed that all of the claims there put forward were stated as being peculiar to the Sulzer cycle. This was by no means the case, and it was quite appreciated that with each cycle there were certain advantages and disadvantages. It was quite true to say that other cycles could obtain the particular advantage to which he referred, being affected only slightly by changes of ambient air temperature. He also stated that the effect of water temperature was under-rated in the paper. In this connection it might be said that with this or any other cycle having several stages of intercooling, the influence of water temperature on efficiency was approximately the same as was the influence of condenser circulating water temperature on the efficiency of the steam turbine.

Dr. Brown then stated that a large portion of the paper was devoted to knocking down a phantom open cycle turbine having low compression ratios. If this impression had been created it was certainly not the intention in writing the paper. What was intended was to point out that the very simple turbine about which there had been so much discussion in the Press in recent years was not suitable for large installations of high efficiency, i.e. that without intercooling, multi-stage combustion and the arrangement of the turbine and compressor units on two or more shafts it was not possible to design a suitable installation for marine service. With this statement there appeared no doubt that Dr. Brown agreed, as he himself with the open cycle proposed a three shaft unit as was suggested in the paper for the Sulzer cycle.

Dr. Brown stated that in his view the efficiencies given in the paper must be dependent on assumptions regarding component efficiencies which were distinctly optimistic. It could only be said that very extensive tests had been carried out on all equipment involved and the assessment of efficiency given in the paper was the closest that could be arrived at based on actual test results of various components. It would, however, be noted that this efficiency was very cautiously stated in the paper, viz., only in the form that it was hoped that this thermal efficiency would be reached and until full size machines were running in service, it was not possible to assess definitely what their efficiency would be.

It should be pointed out that the efficiency was based on the lower calorific value of the fuel, and it might be that Dr. Brown's suggested 32 per cent. had been based on higher calorific value, in which case there would be little difference between the two figures.

The next part of Dr. Brown's remarks, where he stated that the best method of assessing the merits of the semi-closed cycle was to compare its performance with that of a high pressure open cycle, seemed to be of questionable value as it consisted of blank statements unsupported by any technical facts or data. It seemed that he himself must, in this comparison, have used rather optimistic values for the efficiency of components of the open cycle and pessimistic values for the efficiency of components of the Sulzer cycle.

As already mentioned, in any such comparison the assumptions made in respect of heat exchangers were a very important factor and unless all such facts were stated, comparisons were quite valueless.

In the final efficiency column of Dr. Brown's table the Sulzer cycle had been debited with the loss of efficiency of electric drive, whereas the Pametrada cycle had been debited only with the loss of efficiency of gear drive, for which latter incidentally the figure seemed rather optimistic. The preference in the combustion turbine paper for the electric drive was stated on the grounds of practical considerations at the present stage of development. If efficiencies were to be compared it was not fair to debit one system with a less efficient drive than another.

In respect of the dry weight of the complete installation the combustion turbine design was amazingly elastic and Dr. Brown's weight comparisons could not be conclusive without a mass of other information. In the first case it would be necessary to know the relation between actual stresses and time load strength of the materials in the various rotating parts that were working at high temperatures. Secondly, various coolers in the system could be designed with tubes of any size from a small fraction of an inch up to 2 or 3 inches in diameter. With small tube design there was an enormous saving in weight. The light weights given by Dr. Brown for his proposals only suggested that he was working with high stresses on the rotating parts and with tubes rather small to be conveniently cleaned in service. At the present stage of development a claim for extremely light weights was the strongest argument against the adoption of any particular design of turbine for marine service.

In his remarks on the number of rotating machines, Dr. Brown considered a two stage compressor with intercooling to be two separate compressors. It was noted that in the photographs he showed, one of his turbines was a double flow turbine, but he did not apparently, for his own design, consider this as two machines. In fact whether the compressor was in a single stage without intercooling or in two stages with intercooling introduced no complication in the compressor design, but to design a double flow turbine with central entry for working with gas at temperatures at 1,250 deg. C. was a very difficult problem, and such a machine which was in fact two separate turbines in one casing was likely to be more troublesome than two separate turbines in separate casings. Here, as in the case of heat exchangers, mere enumeration of the elements as shown on a layout diagram was no measure of the relative simplicity of different designs.

Dr. Brown considered that with the cycle as described in this paper all the advantages of the Sulzer type turbine in respect of freedom from fouling were lost, but this surely was not correct, as the largest compressor turbine and the heat exchanger which dealt with the greatest volume of heat were both working with air so that fouling would only influence the turbine of the low pressure compressor and the final heat exchanger, which latter it might be mentioned was arranged with large easily cleaned tubes.

Dr. Brown minimised the disadvantages of large ducts on the normal open cycle arrangement but in the layout of an engine room such large ducts were, to say the least of it, inconvenient.

Regarding Dr. Brown's remarks on time for starting up, no comment would be made except to say that it must be left to service experience to show what was the safe and convenient starting up time for a combustion turbine.

Referring to Mr. Nithsdale's remarks, the only point which called for any comment in relation to combustion turbines was his comparison table. The figure that he had taken for oil consumption for the combustion turbine seemed very high in relation to the estimated figures given in the combustion turbine paper, and it was not quite clear how he had arrived at this high figure. No doubt he had assumed working at much lower gas temperatures in order to ensure the maximum safety for long life in service. On the other hand it was fair to say that with present materials the temperatures given in the paper should be reasonably safe and would result in consumptions such as are quoted in the paper.

On the price comparison it did seem that Mr. Nithsdale's figures were rather unduly favourable to the steam turbine in relation to the Diesel engine. It was, of course, difficult to make an absolutely fair comparison as so much depended on the specification, and one wondered whether a high efficiency steam turbine giving as low a consumption as 0·575lb./s.h.p. would be so low in price in relation to the heavy oil engine as was shown by the figures quoted by Mr. Nithsdale. Other comparisons had shown very little difference in price between similar installations with Diesel engines and with steam turbine machinery of high efficiency.

So far as the price of the combustion turbine was concerned, there could be no real basis for an accurate estimate of this until more experience was gained in manufacture. At the present time it was probably true that for an installation of the larger of the two sizes under consideration the price of the combustion turbine would not be greatly different from the price for the other types of machinery. There should, however, be a tendency to a reduction in prices as the combustion turbine developed and more experience was gained in the manufacture of the turbine itself and of the various components. At the present time all that could be said was that for a small installation the cost of the combustion turbine was likely to be substantially higher than the cost of the other types of machinery under discussion. As the power increased the price of the combustion turbine would be relatively more favourable, and for very large installations, even to-day, it would probably be cheaper than other types.

Mr. Dimmick remarked on the development of the combustion turbine in relation to the probable development of Diesel electric propulsion, and suggested that the field for the latter would be very restricted as the combustion turbine became a tried system of marine propulsion. It would, however, seem possible that whilst this position might be true for the high powers, the Diesel engine either direct coupled or with indirect drive was likely to remain a cheaper installation than the combustion turbine for the moderate powers, and the main field in which the combustion turbine had a real prospect of commercial success was for higher powered ships, so that it would be mainly competing with the steam turbine and to a lesser extent with the Diesel engine.

His remarks on the question of reversing combustion turbines amplified the remarks made in the paper, namely, that at any rate for the immediate development of the combustion turbine the electric drive was the only fully tried out method of handling the reversing and manœuvring problem. Other systems of reversing and manœuvring might, however, shortly become available and it would then be a matter of cost, weight and space, as to which method of reversing was finally adopted for the combustion turbine.

Dr. Bailey's contribution was mainly confined to very interesting proposals for a combined plant consisting of a reciprocating engine, reciprocating compressors, steam turbine, and combustion turbine. There was no doubt that theoretically such an installation would have very considerable advantages and as Dr. Bailey had shown, efficiencies approaching that of the Diesel engine could be obtained at more moderate temperatures than were necessary in a true combustion turbine. However, this system did involve the use of a highly supercharged recicprocating unit; whether this unit was of the free piston type or a normal engine with crankshaft, it was still subject to the problems of very high rates of supercharging, and in spite of the theoretical advantages of the system proposed by Dr. Bailey, it seemed that a very long period of development would be required to make this type of installation into a satisfactory commercial plant. A considerable amount of work had been done by Sulzer Bros. and by other firms on the combination of the reciprocating compressor unit and the gas turbine, which formed a considerable part of the system as proposed by Dr. Bailey, and a unit of 7,000 h.p. of this type was running some time ago.

Experience had shown, however, that very considerable development problems were involved and the ultimate limit of efficiency was no better than that of a straight combustion turbine plant, provided materials available would withstand the temperature of about 1,250 deg. F. The plant itself was more complicated than the straight combustion turbine and therefore it would seem that the commercial possibilities of such a system were not so great as the theoretical promise would leave one to believe.

Mr. Welsh considered that the remarks passed on HMMGB.2009 were less than fair to this development and compared it with the "Turbinia". However, there was one important difference, viz. that whereas the "Turbinia" was fitted with steam turbines of much the same type as would be installed in a merchant vessel, the combustion turbine machinery of the HMMGB.2009 was of a type which neither in efficiency nor in probable working life could be used for merchant ship propulsion. In other words the development from the "Turbinia" to merchant machinery was a small step, whereas the development from HMMGB.2009 to merchant ship machinery was a long step. These remarks did not in any way discredit the quality of the 2,500 h.p. machinery in HMMGB.2009, and in fact such an installation was a very useful and important step in the general development of this type of machinery.

Mr. Welsh referred to the question of starting up time and suggested that an open cycle machine could improve greatly on the figure of 90 minutes given in the paper. It was true that a simple open cycle turbine without recuperators could be started very quickly. Whether this could be done frequently without risk of damage to the turbine was more doubtful. In a high efficiency combustion turbine, whatever the cycle, there was the additional question to be considered of heating through the heat exchangers; the heat exchangers of an open cycle, being larger and more bulky than those on the Sulzer cycle, would probably require a longer time for heating through. In any case it should be stressed that the figure of 90 minutes given in the paper was not stated as the time that was essential to starting, but as a safe working figure for the time that would be taken before full load would reasonably be put on the plant by a careful and competent operating engineer.

Regarding the compression ratio of the open cycle, Mr. Welsh's remarks on this, and also his remarks on the dimensions of the plant involved, had been replied to in answering Dr. Brown's contribution to the discussion.

Mr. Welsh's remark that the cooling of the reheat chamber might give rise to difficulties was not entirely unfounded. In actual fact the questions of heat transmission and radiation had to be studied in detail in the design of a combustion chamber of this kind. The problem was the same, however, for all combustion turbines incorporating reheating, as fuel/air ratio and gas entry temperature to the reheater would not vary greatly with different cycles.

Regarding the question of pressure drop in the heat exchangers, it was agreed that in relation to exchanger 9 the losses on the low pressure side were much more important than in the other exchangers. However, the quantity of gas flowing through this exchanger at low pressure was very much smaller than the quantity of gas flowing through the low pressure side of the heat exchanger in an open cycle design, and the argument put forward in the paper of the relative unimportance of pressure losses in the heat exchangers as compared to the importance of those pressure losses in a normal open cycle was perfectly valid.

Regarding the effect of ambient temperature, this also had been dealt with in reply to Dr. Brown. Mr. Welsh pointed out that if a precooler was used there would be no difference with either type of cycle: this was true, but with the volume of air to be handled the precooler required on the open cycle machine would be of very much larger dimensions than that required for the Sulzer cycle.

Mr. Welsh remarked on the use of gears between turbine and alternator and assumed from this that a very high speed turbine was used. In fact the speed of the power turbine was closely similar to the speeds of the steam turbines proposed by Dr. Brown, and with both steam and combustion turbines of this size the best turbine speed would require rather a high number of poles on the propulsion motor if the alternator was direct coupled. The cheapest arrangement was a geared alternator, with which the cost of the gears was more than offset by the reduced cost of the low speed alternator (1,500 r.p.m.) and the propulsion motor with less poles.

Finally Mr. Welsh referred to the use of the simpler types of combustion turbine for cross-channel vessels and other types of plant where power space ratio was more important than fuel ecenomy. This probable application of the combustion turbine was, of course, very interesting and very likely to be one of the early developments. However, it could not be dealt with in the paper, as the subject matter was machinery for cargo liners of high power.

Mr. Sherborne referred to the question of material to be used in the various heat exchangers. At the present time much development work was going on on this question, and it was quite impossible to say what would be the ultimate type of material for this service. There were obvious practical advantages in the use of non-ferrous metal which he suggested. Provided that suitable non-ferrous metal with the necessary strength and creep resisting properties was available, and also one which was capable of resisting any corrosive effects which might arise on those tubes subject to the combustion gases, it would seem that non-ferrous tubes were likely to be considered advantageous. At the moment, however, there did not seem to be sufficient research information available on non-ferrous tubes to allow their adoption with certainty of satisfactory results. It was understood, however, that considerable research was being done on this subject by the non-ferrous tube manufacturers, and it was hoped that the results of these tests would soon become available.

Commander Baker discussed the question of the combustion chamber conditions for the cycle described in the paper. However, it must be pointed out that for a similar overall efficiency of cycle the combustion air entering the combustion chamber must be at much the same temperature whatever the cycle adopted. The only point in which the type of cycle described in the paper differed was that in that combustion chamber the temperature would be higher after mixing with the secondary air and before entering the air heater tubes. For this reason it was important that these air heater tubes, which were at a higher temperature than any other part of the cycle, should not be subject to high stresses; this object had been achieved with the particular cycle described. The temperature was still low enough to ensure freedom from the danger of scaling with materials that were now available.

Commander Baker pointed out the possibility of stresses due to differential expansion, and it was agreed that this was a problem which must be considered in the design of this heat exchanger. It was believed that with the design that had been prepared, this trouble would not occur. Tests on this unit had already been made, but as was stressed in the paper it was only the result of a long series of tests under service conditions that would finally prove reliability and serviceability of any particular type of plant.

Commander Baker remarked on the effect of fouling and apparently considered that because the ratio of fuel to air was higher in the cycle described than in the open cycle the risk of fouling would be increased. It was unlikely that the danger of fouling was greater, for the degree of blade fouling was known from experience to depend less on the "percentage" of residues in the combustion gases than on the particular nature of these residues. This fact had also been confirmed by results obtained with large numbers of exhaust gas turbines used with Diesel engines. Even if the fouling of the turbine blading were somewhat more pronounced, the plant efficiency and the output in particular were less influenced by fouling than in orthodox open cycle turbines, as the greater part of the compressor input was provided by a turbine working with air only.

With reference to Mr. Church's remarks on the use of the variable pitch propeller with combustion turbine, as had already been mentioned, it was certainly not the intention in the combustion turbine paper that this should be considered as the most suitable method of providing for the reversing. The ultimate ideal would seem to be a reversing gear box, as this would offer the highest overall efficiency. However, of the available methods it could be said with certainty at the moment that the electric drive was completely tried out and known to be reliable. Variable pitch propellers of the Kamewa design had been used and were reported to have given good results on a number of vessels, so that they might be regarded as a prospective means of reversing a combustion turbine. The large power reversing gear box was still only in the design stage, although it was understood that one was likely to be fitted to a steam turbine and would have been tried out in service within the next year or two.

Mr. Church next remarked that an hydraulic or similar coupling would be essential to enable the combustion turbine to start up without the propeller turning. This would only be true of a small unit where compressor and turbine were on a single shaft which also provided the power. With any of the multi-shaft arrangements, which must be provided with any system of high pressure high efficiency combustion turbine, starting up could be done on the compressor turbines, and there was no more need for the propelling turbine to be coupled through an hydraulic coupling than there was for a steam turbine to be so coupled. Under the manœuvring conditions that Mr. Church visualised the propelling turbine, if coupled through gearing to a variable pitch propeller, would be stopped, the compressor turbines cotinuing to run at reduced speed.

Mr. Hart considered that the shipowners could only consider types of machinery that had been fully tested out in sea service, but if all owners adopted this policy there could be no development. With any new type of machinery someone must take the step of putting the first marine installation into service. At the same time, progress with a new type of machine must be cautious, as was stressed strongly in the paper. The greatest danger to the development of any new machine—and this, of course, applied to the combustion turbine—lay in excess of optimism on the part of its advocates.

Mr. Souchotte suggested that the future of the combustion turbine lay in a simpler type of turbine working at much higher temperatures and so enabling the heat exchangers and other accessory equipment to be eliminated. It was true that this would be the ultimate ideal, but the problems which faced the metallurgist became increasingly difficult as temperature rose. To obtain high efficiency without the use of heat exchangers would probably mean working up to temperatures of 1,600 or 1,700 deg. F.

Some twenty years ago blading and rotor materials were available which would have stood up to temperatures of 1,000 deg. F. Twenty years of development in metals had improved this figure by some 250 deg., so that it would seem that the really high temperature combustion turbine was a problem of long term metallurgical research, and in the meantime if this type of plant was to be put into efficient service at sea, there seemed no alternative to the use of one or other of the systems involving heat exchangers. It was up to the designer so to arrange this system as to ensure that the heat exchangers would require little attention in service, or alternatively, that those that did require attention should have tubes of sufficient size and sufficiently widely spaced to ensure ease of cleaning and upkeep. It was in fact the same as the problems of boilers and condensers in a steam installation.

Mr. Wans asked for some definite data on the present position of the combustion turbine in respect of its ability to burn heavy fuel oils. Such information unfortunately could not be given until such time as a large high efficiency machine had been running in actual service for some time on a variety of heavy fuel oils. It was known that considerable work had been done on test bed research with the heavier oils, but it was safe to say that the results of these tests were as yet inconclusive. One test at least with a very bad grade of oil was known to have given very satisfactory results and little indication of fouling after some days of running, and there was certainly every reason to believe that after sufficient experience had been gained it should be possible to burn the heavier grades of fuel. It would, however, be impossible to-day for anyone to claim that heavy grade fuel could be burnt without difficulty, as no one had yet had time or opportunity to carry out extensive tests with a wide variety of different grades of fuel.

Mr. Wans considered that although much more development work was still to be done on the combustion turbine, it might be said that ultimately it would supersede both the reciprocating engine and the steam turbine. This one felt was a very optimistic forecast. It must be many years before a really small turbine of high efficiency could be designed to compete with the Diesel engine, and in spite of latest results with coal burning combustion turbines it was not likely that these would compete with the large coal fired steam power stations for a long time to come. The most immediate field for the development of the combustion turbine seemed to be for marine work in the higher ranges of power where oil was almost universally used as a fuel, and where there was good hope that efficiencies could be obtained substantially better than those of steam turbines, and probably that the same grade of fuel might be used. For the smaller sizes the combustion turbine at a reasonable first cost was not likely to attain efficiencies that could make it compete favourably with the Diesel engine, at any rate during the next few years.

In conclusion the author wished to acknowledge with thanks the assistance afforded in the preparation and checking of the paper and reply to the discussion by the turbine research staff of Messrs. Sulzer Bros.

THE LATE ENGINEER REAR-ADMIRAL SYDNEY RUPERT DIGHT, C.B.E.
(Member of Council).

Portrait by] [*Walter Stoneman, F.R.P.S.*

OBITUARY

Engineer Rear-Admiral S. R. Dight

With deep regret we record that Engineer Rear-Admiral Sydney Rupert Dight, C.B.E., Member of Council, died at his home in London on Thursday, 1st January, 1948, at the age of 62.

Born in 1885, he began his engineering career as a civilian apprentice at Devonport Dockyard, and later joined the Navy as engineer sub-lieutenant in 1905. After a two-year course at the Royal Naval College, Greenwich, he served in the Britannia and in the Admiralty Controller's Department. During the 1914-18 war he served in the cruiser "Weymouth" and at the Admiralty as assistant secretary of the Board of Invention and Research, and also in the Engineer-in-Chief's Department. Promoted to engineer commander in 1920, he served for two years as Instructor in Marine Engineering at the Royal Naval College, Greenwich, and later for two years as engineer officer in charge of the machinery of the aircraft-carrier "Furious". He was appointed in charge at the Admiralty Fuel Experimental Station at Haslar in 1928, remaining there until he retired in April, 1939. During this period the standard of oil fuel burning in the Royal Navy made great advances, largely as the result of his efforts. He was also intimately connected with the improvement in design of the boilers now in use in modern warships. Promoted to engineer captain in 1929, and to engineer rear-admiral seven years later, he was made a C.B.E. in the Birthday Honours of 1935.

During the recent war he served at the Admiralty in the Local Defence Division and also with the Petroleum Warfare Department, where his expert knowledge of oil fuel burning was of the greatest assistance, and he was largely responsible for the development of the smoke protection of our cities from aerial attack. In December, 1932, he delivered a paper to the Institute on "Experience with High-Pressure Steam Installations in the Royal Navy", and was elected a Member in the following year. In that year he was awarded the Institution of Naval Architects' Gold Medal for his paper on "Naval Water-tube Boilers, Experiments and Shock Trials".

He was elected a Member of Council of the Institute in 1945. During the summer of 1946 he was engaged in Abadan as consultant to the Anglo-Iranian Oil Company, investigating the working and efficiency of their shore boiler and power plants, which task he carried out with his characteristic ability and thoroughness. His genial personality and outstanding attainments will long be remembered by all who knew him.

His first wife, married in 1913, was Mabel, daughter of Mr. J. J. Martin, of Plymouth. She died in 1937. Their family now consists of two married daughters, his only son, John Rupert Dight, having been killed in action in 1944 while serving as lieutenant-commander (E), R.N. In 1947 he was married to Ione, daughter of Engineer Rear-Admiral Edward Owen Hefford.