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GAS TURBINE ENGINED INSTALLATIONS

T. W. F. BROWN, D.Sc., S.M. (Member)*

Although the title of this symposium might encourage the presentation of machinery installations which are still in the experimental stage, the gas turbine machinery described in this paper embodies only features which have been tried and proved by extensive research on full scale prototype machinery. Gas turbines for marine propulsion have had a considerable incubation period in which the problems associated with long life units capable of burning residual fuel have had to be solved. They can now be said to have reached a stage at which they can be applied with confidence to marine propulsion.

The example of *Auris* has shown that a high degree of reliability can be achieved and that maintenance costs should not be excessive.

Considerable progress has been made in the research on a liquid cooled turbine to operate at a gas temperature of 2,200 deg. F. but in the present state of the art such units must be classed as experimental. This study is based on uncooled turbines operating at a gas temperature of 1,250 deg. F. at turbine inlets at which temperature a great deal of experience is available. A 3,500 s.h.p. marine gas turbine has been operating at this temperature since 1948 at Pametrada Research Station and the cycles put forward are generally similar with the addition of supercharging. The supercharging set operates at a pressure of 32-35lb. per sq. in. absolute and a temperature of about 900 deg. F., which involves no difficulty. Supercharging enables a higher pressure ratio to be used without fear of compressor surging and leads to a reduced air mass flow and an improved part load efficiency. Another advantage applicable to the hot part of the turbine installation is that as the pressure at the h.p. turbine inlet is 194lb. per sq. in. absolute the corresponding high density of the gases leads to a small robust turbine in relation to output. The reduction in size greatly reduces thermal shock and consequent distortion.

GENERAL FEATURES

The output of the gas turbine installations in the designs for the two ships provided by the Institute are 3,340 and 10,500 s.h.p. respectively at 110 r.p.m. For the smaller power a slightly simpler cycle than that used for the larger output is employed and a single gas turbine installation with a steam turbine for emergency or "get you home" purposes is provided. For the larger power, two gas turbine units are employed which will run independently of each other, thus giving the equivalent of twin-screw machinery, although geared together in a double reduction gearbox to provide the economy of single-screw propulsion.

In considering the rest of the installation, actual heavy-oil-engined ships were examined as only the main engines had to be changed with some alteration in associated auxiliaries. It appeared that the *Welsh Trader* 3,300 b.h.p. and *Middlesex* (9,000 b.h.p. geared oil-engines) represented good installations at sea of about the specified power on which to base the schemes put forward in this paper. By the kindness of the North Eastern Marine Engineering Co., Ltd., and Alexander Stephen and Sons, Ltd., specifications, weight sheets, trial data and electrical loads measured at sea were made available for these ships. Steam turbine engined ships are not good for this purpose as so many of the auxiliaries are intimately connected with the main propulsion machinery that considerably more substitution is required.

CYCLES

The cycles employed are shown in Fig. 1 with the appropriate pressures in the upper line diagrams and the temperatures on the temperature entropy diagrams below (the points in this diagram correspond to the numbers shown above).

As already stated, if the l.p. turbines driving the l.p. compressors are considered together as superchargers, the cycle is similar to that chosen in 1946 on which the 3,500 s.h.p. marine gas turbine was built and on which extended running experience has been gained. The pressures and temperatures shown apply to the maximum power condition. A pressure ratio of 13·6 is employed, obtained by using two or three axial compressors in series driven by independent turbines. This arrangement allows the speeds of the compressors to be largely independent of each other, which reduces surging difficulties and makes for ease of starting and manœuvring. The intercoolers increase the overall efficiency and reduce the size of the installation. From the h.p. compressor the air passes through a heat exchanger where it is preheated before entering the primary combustion chamber and passing to the h.p. turbine at an inlet temperature of 1,250 deg. F. The h.p. turbine exhausts into a reheat combustion chamber where the gas temperature is again raised to 1,250 deg. F. before passing to the i.p. (power) turbine. The use of reheat increases the efficiency and reduces the mass flow (and hence size) by about thirty per cent. The power turbine is mechanically independent and

* Research Director, The Parsons and Marine Engineering Turbine Research and Development Association.

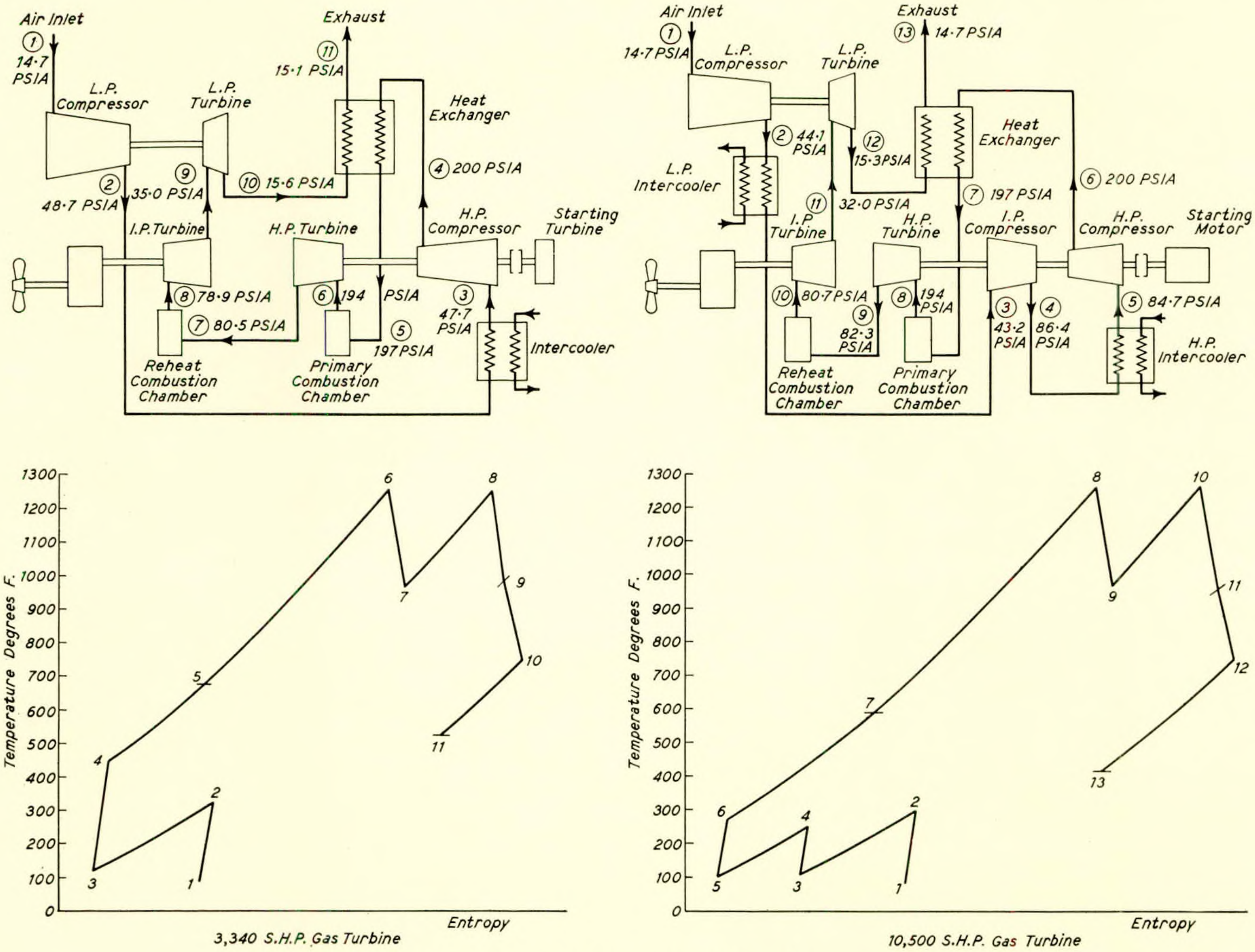


FIG. 1—*Cycle and temperature-entropy diagrams*

therefore imposes no speed limitation on the compressors at low powers. From the power turbine the gases pass to the l.p. turbine which exhausts to the heat exchanger. In the smaller unit the gases then pass by way of a composite boiler to the funnel. In the larger unit, with the addition to the cycle of a second intercooler, the temperature of the exhaust gases from the heat exchanger is too low for a waste heat boiler to give an appreciable gain and the gases pass direct to the funnel. (This explains the difference in pressure after leaving the heat exchanger in the two schemes.)

DESIGN CONDITIONS

The machinery is designed to give full output at ambient air and sea temperatures of 80 deg. F. The effect on efficiency when operating under tropical conditions is small although power is reduced if the inlet temperature of the turbines is maintained constant. Taking the unfavourable case of tankers operating to the West Indies and the Persian Gulf, an analysis of log records shows that about 40 per cent of their time at sea is spent with ambient air temperatures between 80 degrees and 90 degrees, less than 10 per cent with air temperatures between 90 deg. F. and 100 deg. F., while the time spent with air temperatures above 100 deg. F. is negligible. For the remaining 50 per cent of the running time the air temperatures are below 80 deg. F. For at least 70 per cent of the whole run the difference between air and sea temperatures is less than 10 deg. F. Fig. 2 shows the slight increase in maximum cycle temperature required to restore the power loss due to high ambient air temperatures in relation to load. It shows that full service power can be maintained with ambient air temperatures between 90 and 100 deg. F. with an increase in maximum temperature of only 10-30 deg. F. As the machinery only operates for some 10 per cent of its time at sea under such conditions, the effect of this increase in temperature on the life of the machinery is small.

The designed life of the machinery except for some minor expendable parts in the combustion chamber is 100,000 hours at maximum power. Maximum power in these designs is 12 per cent above the full service power mentioned earlier.

The turbine and compressor design avoids the very light

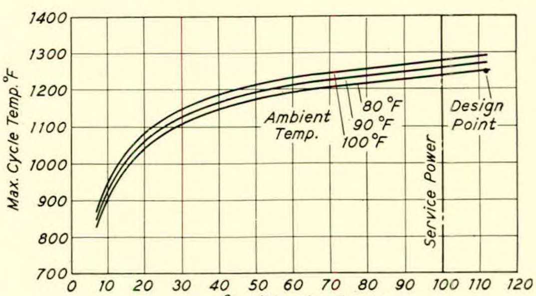


FIG. 2—*Variation of turbine inlet temperature with load and ambient conditions*

scantlings of the aircraft type of construction which even if derated to improve life are unsuitable for marine gas turbine machinery for merchant ships. Such designs have inherent high losses which are accepted to reduce bulk and weight at all cost. On the other hand the scantlings are considerably lighter than those associated with marine steam turbine design owing to the lower pressures employed and the necessity to control thermal stresses and distortions. With the given life requirements, stresses in the hot parts are low, hence turbine blade speeds are low and consequently the compressors driven by these turbines have many stages for comparatively low pressure ratios. Fig. 4 showing the l.p. turbine and compressor illustrates this point. Design particulars are summarized in Table I.

The specific fuel consumptions for the two units for the turbines only, over the range of power from low load to full load, are given in Fig. 3. These curves are based on a gross calorific value of 18,500 B.Th.U's per pound. It will be seen how flat the fuel rate is from 50 per cent to full power and is a consequence of the use of compounded compressors in association with a high pressure ratio at full power.

TABLE I.—SUMMARY OF TECHNICAL DESIGN PARTICULARS

Size of unit, s.h.p.	3,340	10,500
Maximum, s.h.p.	3,760	11,800
Maximum, r.p.m.	114·4	114·4
Service, s.h.p.	3,340	10,500
Service, r.p.m.	110	110
Maximum astern, s.h.p.	2,460	7,670
Maximum astern, r.p.m.	90	90
H.P. turbine: r.p.m. (service power)	8,750	7,000
No. of stages	5	5
I.P. turbine: r.p.m. (service power)	5,850	4,500
No. of stages	4	4
L.P. turbine: r.p.m. (service power)	5,450	4,350
No. of stages	2	2
H.P. compressor: No. of stages	24	13
Compression ratio	4·19	2·36
I.P. compressor: No. of stages	—	10
Compression ratio	—	2
L.P. compressor: No. of stages	16	15
Compression ratio	3·31	3
H.P. intercooler effectiveness	—	0·85
L.P. intercooler effectiveness	0·85	0·85
Heat exchanger thermal ratio	0·7	0·7
Air mass flow (maximum power), lb. per sec.	37·1	104·6 (2 units)
Air rate (maximum power), lb. per s.h.p. hr.	35·5	31·9

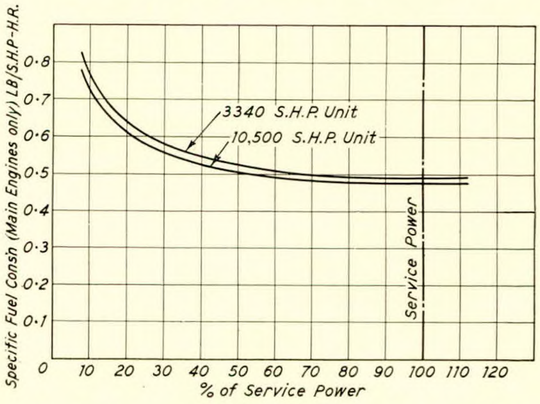


FIG. 3—*Specific fuel consumption in relation to power*

RESIDUAL FUEL BURNING AND OIL FUEL SYSTEM

Sufficient experience has been gained to show that residual fuel can be used at sea in gas turbine machinery. Long runs have been carried out using additives and it would now appear that the cheapest additives required to prevent corrosion and fouling of the high temperature parts are in powder form. The fuel system designed for operation with residual fuel carrying these additives is shown in Fig. 5. Two mixing tanks are provided in which the dry additive is mixed with heated purified fuel from the ready use tank in the proportion of about 5lb. of additive per ton of fuel. It is anticipated that the additional cost to treat each ton of residual oil will be about tenpence. Turbulent mixing is effected by circulating the fuel through a transfer pump and injecting it back into the tank through a nozzle. While mixing is taking place in one tank,

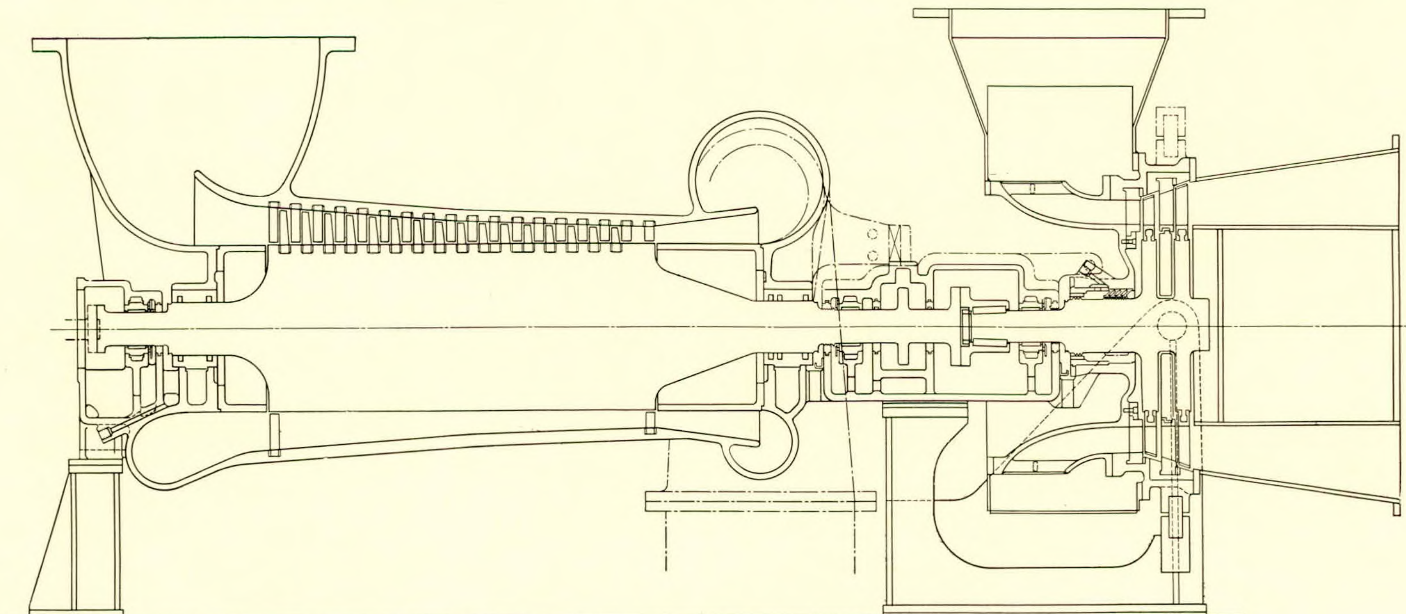


FIG. 4—*L.P. compressor and l.p. turbine*

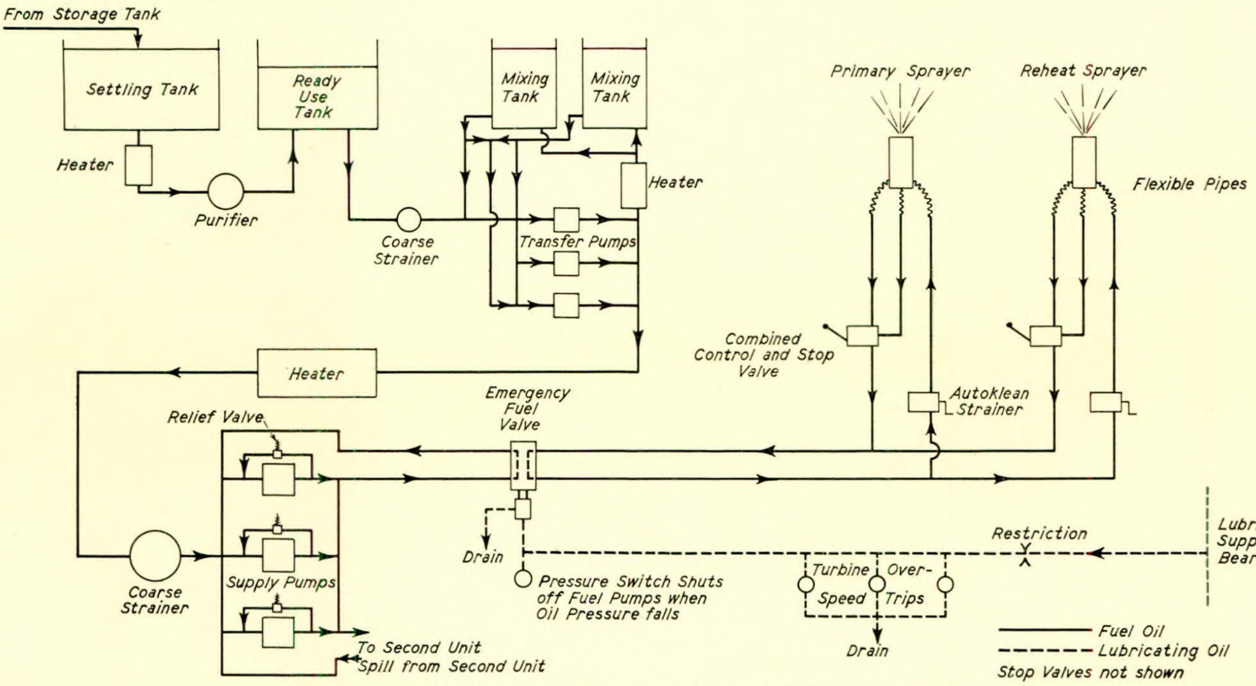


FIG. 5—*Diagrammatic arrangement of fuel oil system*

fuel from the other is passed by a second transfer pump to the service pumps. Standby pumps are provided.

The fuel is heated to about 250 deg. F. before entry to the service pumps, from which it is delivered at a constant supply pressure of 600lb. per sq. in. Each gas turbine unit is supplied by a separate service pump with standby pumps provided. The filters do not remove the additive and no undue wear has been experienced in the pumps after extensive trials. The additives have been proved to remain in suspension in the fuel oil for periods up to twenty-one days after mixing.

The compressors can be washed with water to which a little detergent has been added while running. This has been carried out many times and is completely effective in the industrial atmosphere at Wallsend. The turbines can similarly be washed when cold and the swallowing capacity, which may have been slightly reduced during running with additives, completely restored.

The sprayers are of the spill control type with a spring-controlled piston-operated tip valve, held open during normal operation by the fuel supply pressure. A small quantity of fuel is permitted to leak off past the piston to pump suction via the shut-off spill line. By this means the fuel supply to the combustion chamber is shut off either by loss of supply pressure or by closing a stop valve in the shut-off spill line, thereby allowing the pressure to build up behind the piston until the tip valve is closed by the spring. An emergency valve is fitted in the system to shut off the fuel if the turbines overspeed or if failure of the lubricating oil pressure occurs. The sprayers can be changed without shutting down the set by accepting a momentary reduction in power.

HYDRAULIC TRANSMISSION FOR MANŒUVRING

No astern turbines are provided, the power from the i.p. (output) turbine being transmitted to the primary pinion shafts through fluid couplings and direct drive clutches for ahead running and through a fluid torque converter for astern running. Control is effected by a changeover valve supplying oil to the appropriate coupling. The arrangement is shown in Fig. 6.

The driving members of the friction clutch, ahead fluid flywheel and the torque converter are all connected to the quill shaft, which is connected through a fine tooth coupling to the drive turbine. When driving ahead the quill drive is in operation and the fluid flywheel and torque converter are empty.

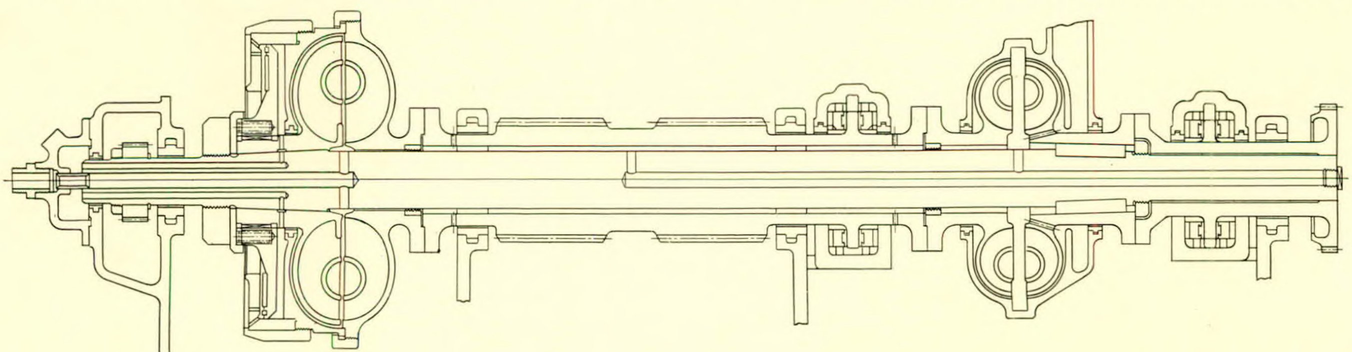


FIG. 6—*Hydraulic transmission system: section through primary pinion and quill shaft*

Lubricating oil at the same pressure as the bearing supply is admitted to the clutch and the pressure generated by the spinning of the clutch holds it against a sintered ring which forms the clutch lining. With increase in speed the clutch load increases and the design is such that more than twice the torque corresponding to the speed can be transmitted. There is no loss due to slip and the gearing efficiency at full load has been proved in many tests to be better than 97 per cent.

In manœuvring, this clutch is emptied and lubricating oil is supplied to the ahead hydraulic coupling of high efficiency or the astern converter (efficiency 65 per cent). On full away the friction clutch has only to take up a small slip and then drive solid. A baffle is fitted to the hydraulic coupling to prevent excessive torques being transmitted to the gearing on taking up the drive. The whole manœuvring control, which is interlocked with the oil fuel control, consists of a single piston valve with appropriate ports for admitting oil to the clutch, ahead coupling and astern torque converter At stop all ports are closed and the couplings empty. Windage in the ahead coupling when running astern is small and windage in the torque converter when running ahead is negligible, as the two rotating elements are turning in the same direction. This system has been well tried out in many experiments.

Fig. 7 shows the compact arrangement of the hydraulic transmission in conjunction with a locked train gearbox for the 3,340 s.h.p. unit. The double reduction gear is of the normal articulated type with a quill drive between the primary wheel and secondary pinions. For the 10,500 s.h.p. unit a normal double reduction gear with two primary drive pinions, including the hydraulic transmission, is used. The schedule of gearing particulars for both schemes is given in Table II. The emergency steam turbine used in the smaller unit is attached to a pinion which is shipped for emergency drive and is shown on the side of the primary gear in Fig. 7. This pinion can be mounted in eccentric bushes which normally hold it out of engagement and which can be turned to bring the pinion into mesh when required for emergency drive. The turbine will be fitted to the gearcase with steam pipes attached on the gearbox where shown in the general arrangement of the installation. The turbine then engages with this extra pinion by pulling over the dog clutch. It will be seen from Fig. 8 that this steam turbine can without other means go ahead or astern using the same turbine wheel. With 30,000lb. of steam per hour this unit develops 1,200 s.h.p. The steam supply valve cannot be opened until the dog clutch is engaged. It exhausts to the winch condenser.

TABLE II.—GEARING SCHEDULE

Size of unit, s.h.p.	3,340	10,500
Primary gears		
Pinion P.C.D., inches	6·70	10·5 (2 off)
Wheel P.C.D., inches	41·9 (2 off)	54·3 (2 off)
Face width, inches	12·40	21·15
Gap, inches	3·0	3·0
Secondary gears		
Pinion P.C.D., inches	12·35 (2 off)	18·32 (2 off)
Wheel P.C.D., inches	105·0	145·0
Face width, inches	23·0	35·45
Gap, inches	3·0	3·0

When manœuvring, the heat dissipated from the astern coupling (efficiency 65 per cent) can be absorbed in the oil cooler. If this were not in use and emergency astern required, the heat capacity of the oil in the lubricating oil system is such that full astern could be maintained for about a quarter-of-an-hour without undue temperature rise. This gives ample time to bring the coolers into operation.

These preliminary considerations have been dealt with before proceeding to the complete installations.

3,340 S.H.P. INSTALLATION

As previously stated, the gas turbine set proposed for this installation comprises a single unit driving through hydraulic couplings and a double reduction gearbox of the locked train type. Cycle particulars at full power have been given in Fig. 1.

In the *Welsh Trader,* used as a basis, the auxiliaries were all steam driven with the exception of minor items, such as

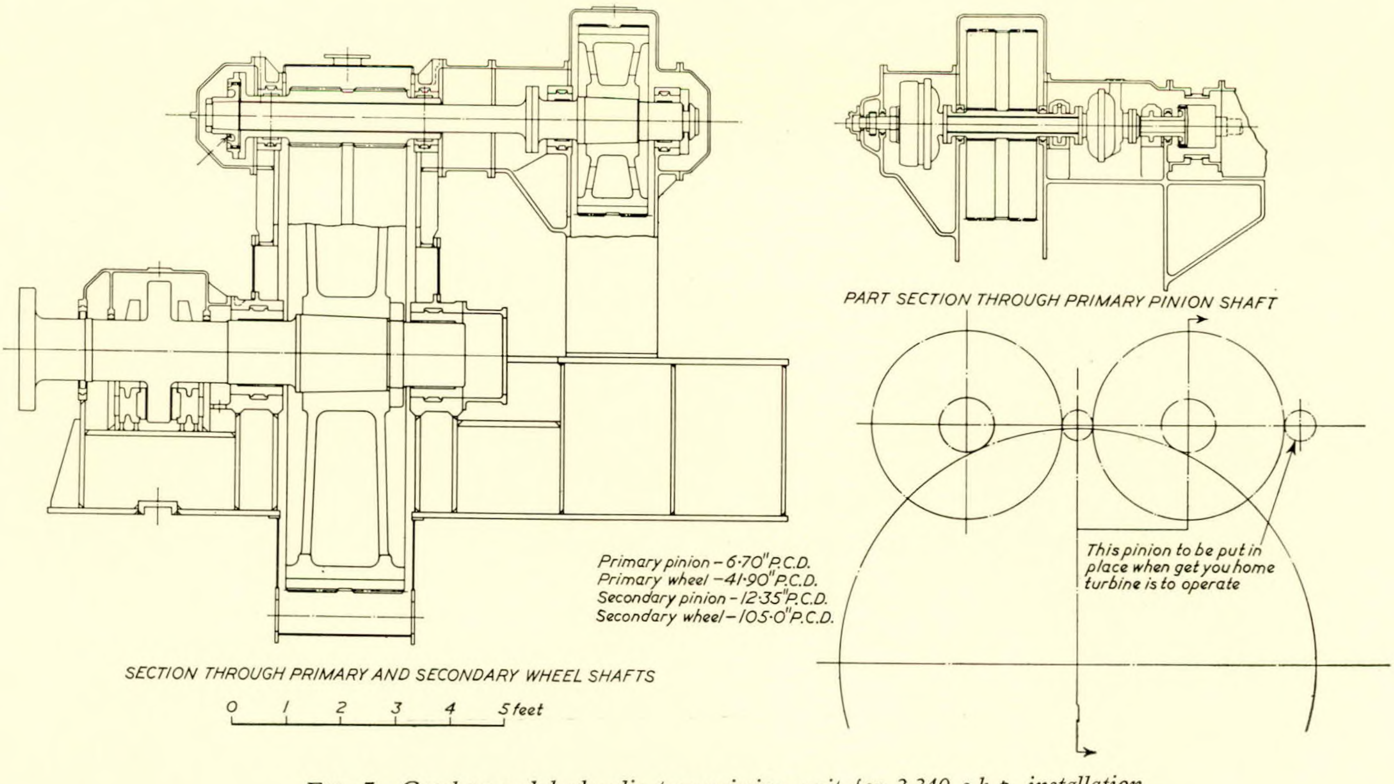


FIG. 7—*Gearbox and hydraulic transmission unit for 3,340 s.h.p. installation*

TABLE III.—AUXILIARIES REQUIRED FOR 3,340 S.H.P. INSTALLATION

A—Steam auxiliaries

Auxiliary	Duty	Max. steam consumption, lb. per hr.	Load factor	Average steam consumption, lb. per hr.
1 Scotch boiler (for port or emergency use only)				
2 oil-burning units for above				—
1 F.D. fan for above				—
1 Cochran composite boiler				
1 oil-burning unit for above				—
2 feedwater pumps (1 standby)				60
1 feedwater heater				
1 feedwater filter				
1 auxiliary condenser				
1 evaporator	10 ton/day	1,300	0·1	130
1 distiller condenser				
1 ballast pump (standby for seawater circulating pump	250 ton/hr. at 30 lb. per sq. in.		0	—
1 general service pump	95 ton/hr. at 70 ft. head	1,750	0·15	260
1 bilge pump	95 ton/hr. at 70 ft. head	1,750	0·15	260
1 oily-water separator				
1 oil-fuel transfer pump	25 ton/hr. at 70 ft. head	620	0·15	90
1 oil-fuel heater				150
1 30-kW steam-driven generator		1,500	0·6	900
Ship's heating				150
Total				2,000

Steam supplied from waste heat section of composite boiler.

B—Electrical and other auxiliaries

Auxiliary	Duty	Motor h.p.	Load factor	Average sea load, h.p.
3 oil-fuel circulating and boost pumps (1 standby)	3 ton/hr. at 50 lb. per sq. in.	2×1	0·9	1·8
2 main oil-fuel pumps (1 standby)	300 g.p.h. at 600 lb. per sq. in.	4½	0·9	4·0
1 set oil-fuel filters				
3 oil-fuel purifiers (1 standby)	300 g.p.h.	2×2½	0·5	2·5
2 lubricating-oil pumps (1 standby)	6,000 g.p.h. at 50 lb. per sq. in.	9	0·9	8·1
1 lubricating-oil cooler				
1 set lubricating-oil filters				
1 lubricating-oil purifier	300 g.p.h.	2½	0·5	1·25
1 hydraulic coupling oil pump (standby)	6,000 g.p.h. at 50 lb. per sq. in.		0	—
1 hydraulic coupling oil cooler				
1 hydraulic coupling oil purifier	300 g.p.h.	2½	0·5	1·25
1 set hydraulic coupling oil filters				
1 seawater circulating pump for hydraulic coupling oil coolers	80 ton/hr. at 30 lb. per sq. in.	10	0	—
1 main seawater circulating pump	250 ton/hr. at 20 lb. per sq. in.	20	0·9	18·0
2 60 kW Diesel generators				
Ventilating fans		7	0·9	6·3
Workshop machinery		3½	0·1	0·35
		75		43·55
Equivalent electrical load, kW		66		38
Ship's services, kW				15
Total, kW				53

18 kW supplied by steam-driven generator.
35 kW supplied by Diesel generator.

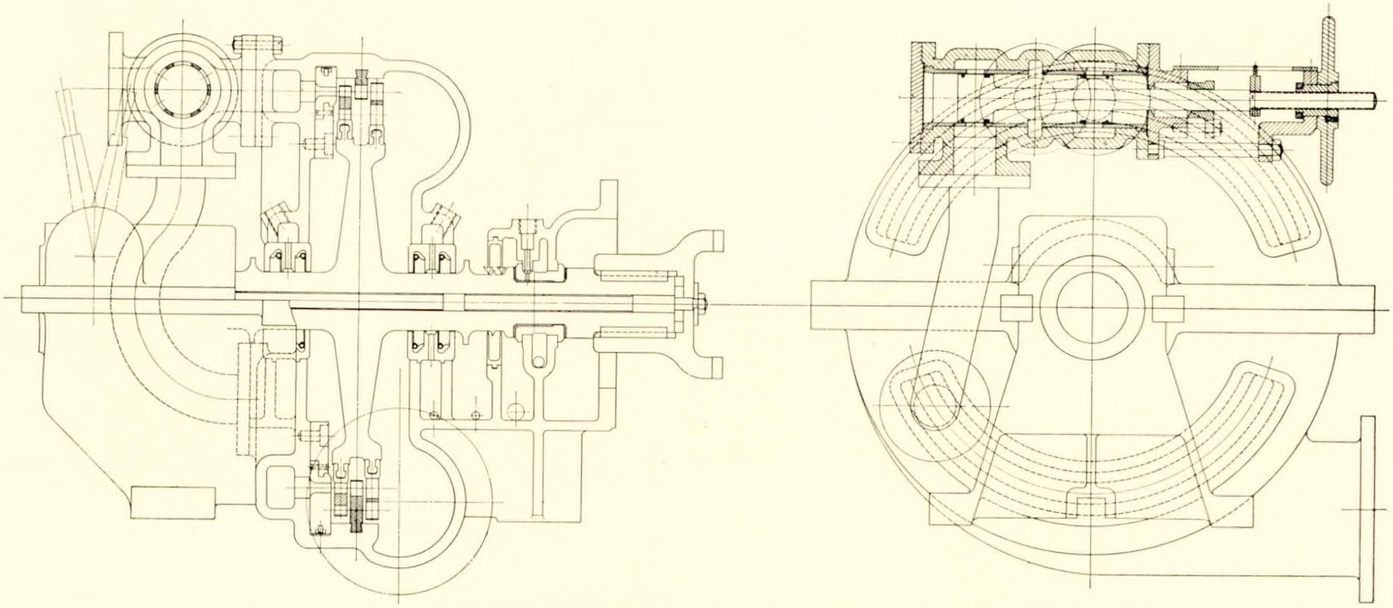


FIG. 8—*"Get You Home" steam turbine for 3,340 s.h.p. gas turbine installation*

ventilating fans, oil separators and workshop machinery. At sea steam is supplied by a composite boiler working partly on exhaust gas and partly by oil fuel. The generators, two at 30 kW, were steam driven and a small Diesel generator provided for emergency use.

An installation of this power, although heavier and less efficient than one using electrically driven auxiliaries, should be better from the point of view of the engine room staff as maintenance requires less skilled attention. The replacement of a heavy oil engine by a gas turbine required the following additional auxiliaries: —

1. Sea water circulating pump.
2. Lubricating oil pump.
3. Oil fuel circulating pumps.
4. Oil fuel service pumps.

In the oil engine items 1, 2 and 4 were engine driven and 3 was not required. If these pumps were steam driven the requirements would be too much for the composite boiler. Some of the auxiliaries must, therefore, be motor driven and a Diesel driven generator plus standby is required. With these changes some of the simplicity and robustness of the original installation is sacrificed but as many steam reciprocating auxiliaries are retained as possible and all standby pumps are of this type. In operation it will be best to use all the steam generated in the composite boiler by exhaust gas heating, using the steam driven generators to absorb all the steam not required by heaters or pumps, and to generate all the additional power required in the Diesel generators rather than by burning additional oil in the composite boiler.

The following comparison shows this: —

Total electrical power required, kW...	53	53
(*a*) Generated by steam generator, kW	30	18
(*b*) Generated by Diesel generator, kW	23	35
Steam generated—		
(*a*) Exhaust heat section, lb. per hr.	2,000	2,000
(*b*) Oil fired section, lb. per hr.	600	—
Fuel consumption—		
Main engines, lb. per s.h.p. hr.	0·494	0·494
Composite, boiler	0·016	—
Diesel generators	0·005	0·008
Total	0·515	0 502

Running under the second condition there is a reserve of about 2,000lb. per hr. of steam generating capacity obtainable in a short time by burning oil fuel in the composite boiler and about 25 kW of power from the Diesel generator.

Table III summarizes the auxiliaries required for this proposal. The gas turbine set is started up by a steam turbine similar to the emergency steam turbine already referred to, except that unidirectional blading is fitted. The starting motor is coupled by dog clutch to the h.p. turbine-compressor line. The drive is only required for a short time and while the Cochran boiler could not give the steam flow continuously it is of ample capacity in the steam space when starting at full boiler pressure to give the flow required for half a minute by a drop in pressure from 120lb. per sq. in. to 60lb. per sq. in. with oil firing also in operation. In this installation ample boiler capacity is provided for both starting purposes and continuous emergency steaming by the provision of the Scotch boiler used normally for port use. With the h.p. compressor line running, all manœuvring can be carried out without any further starts being required from the starter turbine. The Scotch boiler is only used for port or emergency use. The consequent engine room arrangement for this installation is shown in Fig. 9.

10,500 S.H.P. INSTALLATION

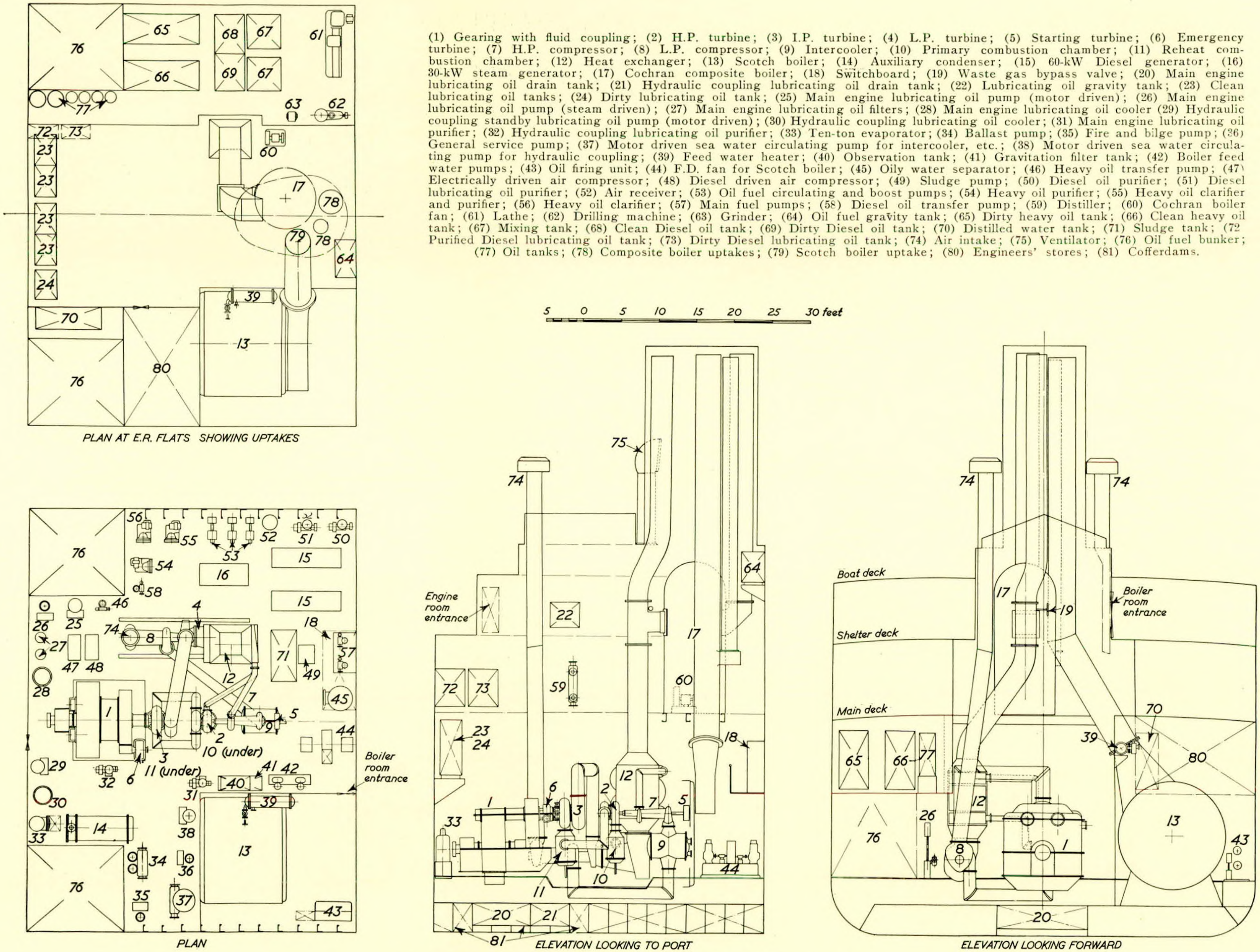
The gas turbines proposed for the larger installation comprise two separate and similar units driving into a single gearbox through hydraulic couplings of the type already described. The gearing is of the double reduction articulated type, and particulars have been given in Table II.

The gas turbine cycle, particulars of which have been given in Fig. 1, is similar to that proposed for the smaller installation except for the addition of a second intercooler. This additional complication is justified for a larger unit, and produces some improvement in efficiency and reduction in size.

As previously stated, an exhaust-gas boiler is not used owing to the low exhaust temperature. A small oil-fired auxiliary boiler is fitted for port use.

Motor driven auxiliaries with electric power supplied by Diesel generators are employed. Table IV summarizes the auxiliaries, duties and loads.

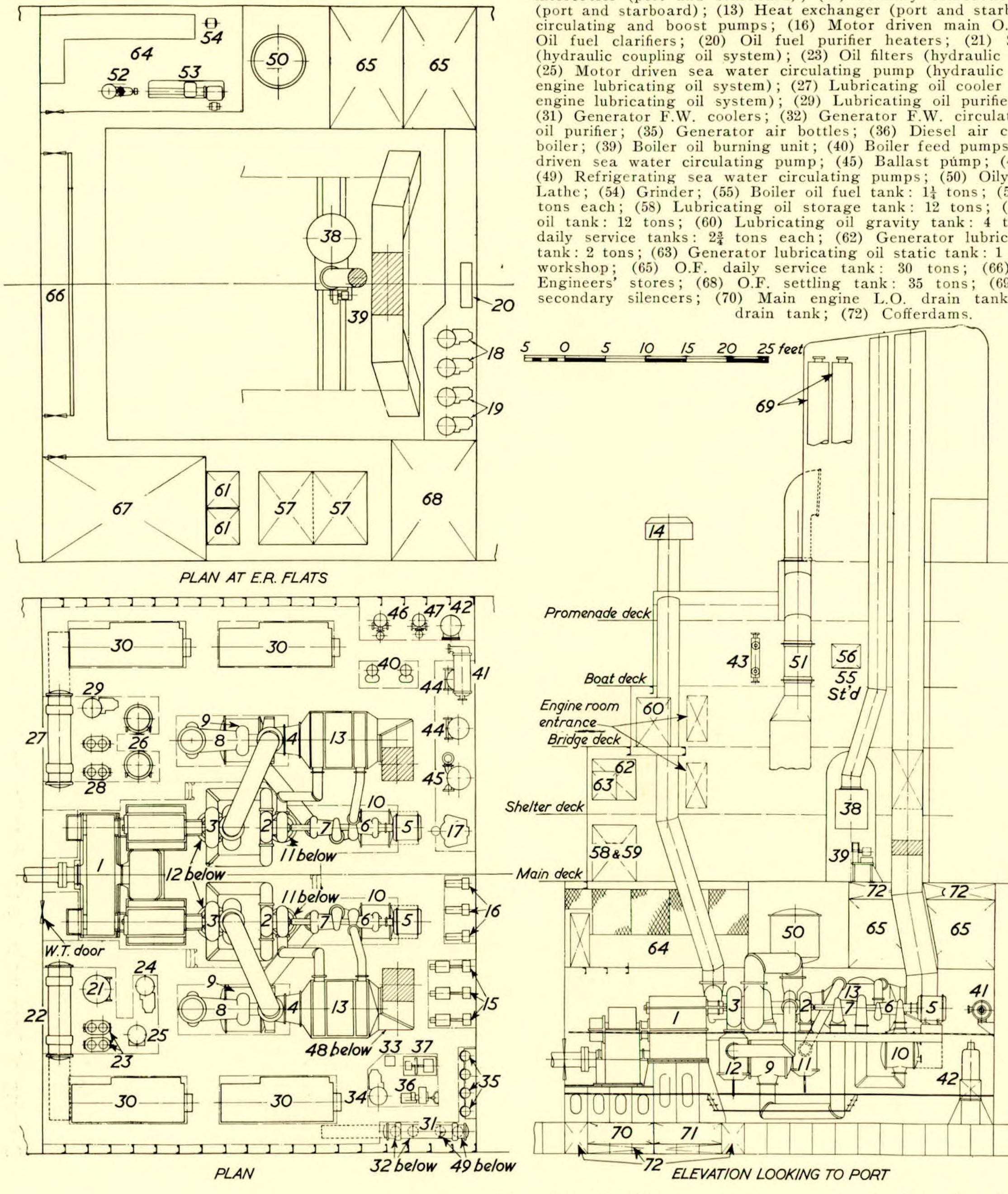
It will be seen that 145 kW is included for ships' purposes. It would have been helpful if a definite figure could have been provided by the Institute as it would have enabled the different machinery schemes to be treated on a comparable



(1) Gearing with fluid coupling; (2) H.P. turbine; (3) I.P. turbine; (4) L.P. turbine; (5) Starting turbine; (6) Emergency turbine; (7) H.P. compressor; (8) L.P. compressor; (9) Intercooler; (10) Primary combustion chamber; (11) Reheat combustion chamber; (12) Heat exchanger; (13) Scotch boiler; (14) Auxiliary condenser; (15) 60-kW Diesel generator; (16) 30-kW steam generator; (17) Cochran composite boiler; (18) Switchboard; (19) Waste gas bypass valve; (20) Main engine lubricating oil drain tank; (21) Hydraulic coupling lubricating oil drain tank; (22) Lubricating oil gravity tank; (23) Clean lubricating oil tanks; (24) Dirty lubricating oil tank; (25) Main engine lubricating oil pump (motor driven); (26) Main engine lubricating oil pump (steam driven); (27) Main engine lubricating oil filters; (28) Main engine lubricating oil cooler (29) Hydraulic coupling standby lubricating oil pump (motor driven); (30) Hydraulic coupling lubricating oil cooler; (31) Main engine lubricating oil purifier; (32) Hydraulic coupling lubricating oil purifier; (33) Ten-ton evaporator; (34) Ballast pump; (35) Fire and bilge pump; (36) General service pump; (37) Motor driven sea water circulating pump for intercooler, etc.; (38) Motor driven sea water circulating pump for hydraulic coupling; (39) Feed water heater; (40) Observation tank; (41) Gravitation filter tank; (42) Boiler feed water pumps; (43) Oil firing unit; (44) F.D. fan for Scotch boiler; (45) Oily water separator; (46) Heavy oil transfer pump; (47) Electrically driven air compressor; (48) Diesel driven air compressor; (49) Sludge pump; (50) Diesel oil purifier; (51) Diesel lubricating oil purifier; (52) Air receiver; (53) Oil fuel circulating and boost pumps; (54) Heavy oil purifier; (55) Heavy oil clarifier and purifier; (56) Heavy oil clarifier; (57) Main fuel pumps; (58) Diesel oil transfer pump; (59) Distiller; (60) Cochran boiler fan; (61) Lathe; (62) Drilling machine; (63) Grinder; (64) Oil fuel gravity tank; (65) Dirty heavy oil tank; (66) Clean heavy oil tank; (67) Mixing tank; (68) Clean Diesel oil tank; (69) Dirty Diesel oil tank; (70) Distilled water tank; (71) Sludge tank; (72) Purified Diesel lubricating oil tank; (73) Dirty Diesel lubricating oil tank; (74) Air intake; (75) Ventilator; (76) Oil fuel bunker; (77) Oil tanks; (78) Composite boiler uptakes; (79) Scotch boiler uptake; (80) Engineers' stores; (81) Cofferdams.

FIG. 9—*Machinery arrangement for 3,340 s.h.p. installation*

(1) Gearing with fluid couplings; (2) H.P. turbine (port and starboard); (3) I.P. turbine (port and starboard); (4) L.P. turbine (port and starboard); (5) Starting motor (port and starboard); (6) H.P. compressor (port and starboard); (7) I.P. compressor (port and starboard); (8) L.P. compressor (port and starboard); (9) Primary intercooler (port and starboard); (10) Secondary intercooler (port and starboard); (11) Primary combustion chamber (port and starboard); (12) Reheat combustion chamber (port and starboard); (13) Heat exchanger (port and starboard); (14) Air intake (port and starboard); (15) Motor driven O.F. circulating and boost pumps; (16) Motor driven main O.F. pumps; (17) Oil fuel transfer pump; (18) Oil fuel purifiers; (19) Oil fuel clarifiers; (20) Oil fuel purifier heaters; (21) Stand-by oil pump (hydraulic coupling oil system); (22) Oil cooler (hydraulic coupling oil system); (23) Oil filters (hydraulic coupling oil system); (24) Oil purifier (hydraulic coupling oil system); (25) Motor driven sea water circulating pump (hydraulic coupling oil system); (26) Motor driven lubricating oil pump (main engine lubricating oil system); (27) Lubricating oil cooler (main engine lubricating oil system); (28) Lubricating oil filters (main engine lubricating oil system); (29) Lubricating oil purifier (main engine lubricating oil system); (30) 200-kW Diesel generators; (31) Generator F.W. coolers; (32) Generator F.W. circulating pumps; (33) Generator transfer pump; (34) Generator lubricating oil purifier; (35) Generator air bottles; (36) Diesel air compressor; (37) Motor driven air compressor; (38) Cochran oil fired boiler; (39) Boiler oil burning unit; (40) Boiler feed pumps; (41) Auxiliary condenser; (42) Evaporator; (43) Distiller; (44) Motor driven sea water circulating pump; (45) Ballast pump; (46) Bilge pump; (47) General service pump; (48) Fresh water pump; (49) Refrigerating sea water circulating pumps; (50) Oily water separator; (51) Ventilating fans; (52) Drilling machine; (53) Lathe; (54) Grinder; (55) Boiler oil fuel tank: $1\frac{1}{4}$ tons; (56) Boiler Diesel fuel tank: 200 gallons; (57) Oil fuel mixing tanks: 12 tons each; (58) Lubricating oil storage tank: 12 tons; (59) Dirty lubricating oil tank: 12 tons; (60) Lubricating oil gravity tank: 4 tons; (61) Generator daily service tanks: $2\frac{3}{4}$ tons each; (62) Generator lubricating oil renovating tank: 2 tons; (63) Generator lubricating oil static tank: 1 ton; (64) Engineers' workshop; (65) O.F. daily service tank: 30 tons; (66) Switchboard; (67) Engineers' stores; (68) O.F. settling tank: 35 tons; (69) Diesel generators secondary silencers; (70) Main engine L.O. drain tank; (71) Coupling oil drain tank; (72) Cofferdams.



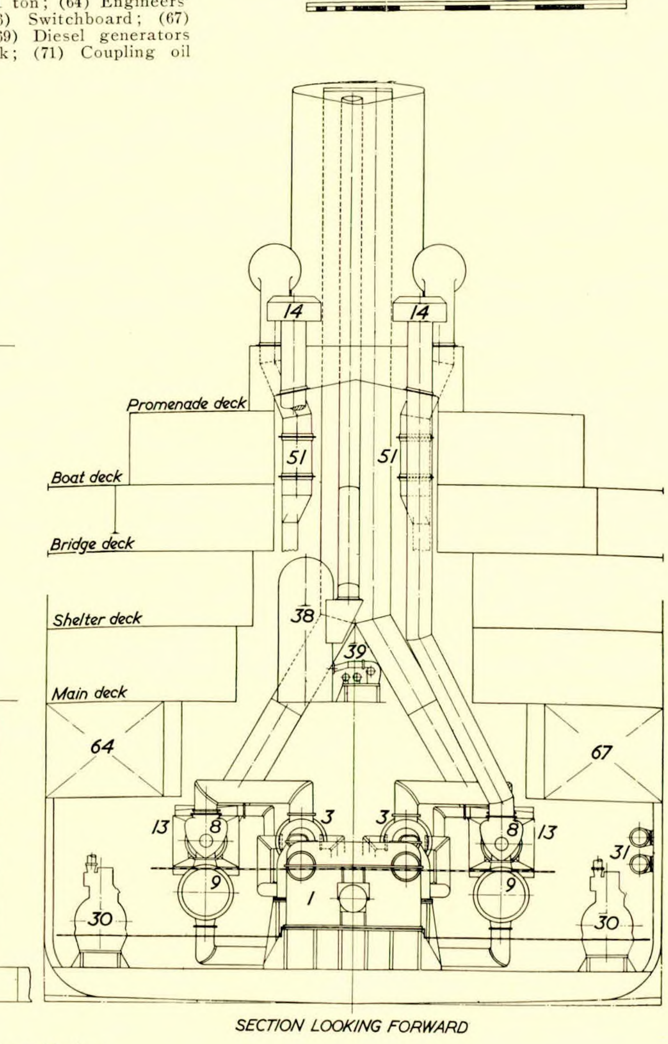


FIG. 10—*Machinery arrangement for 10,500 s.h.p. installation*

TABLE IV.—AUXILIARIES REQUIRED FOR 10,500 S.H.P. INSTALLATION

Auxiliary	Duty	Motor h.p.	Load factor	Average sea load h.p.
1 oil-fuel transfer pump	40 ton/hr. at 30 lb. per sq. in.	10	0·15	1·5
3 oil-fuel circulating and boost pumps (1 standby)	8 ton/hr. at 50 lb. per sq. in.	2×3	0·9	5·4
1 oil-fuel heater				
3 main oil-fuel pumps (1 standby)	800 g.p.h. at 600 lb. per sq. in.	2×12	0·9	21·6
1 set oil-fuel filters				
3 oil-fuel purifiers (1 standby)	2½ ton/hr.	2×7	0·5	7·0
1 Diesel oil transfer pump (standby for oil-fuel transfer pump)	40 ton/hr. at 30 lb. per sq. in.	10	0·05	0·5
1 Diesel oil purifier	300 g.p.h.	2½	0·2	0·5
2 lubricating-oil pumps (1 standby)	20,000 g.p.h. at 50 lb. per sq. in.	30	0·9	27·0
1 lubricating-oil cooler				
1 set lubricating-oil filters				
1 lubricating-oil purifier	300 g.p.h.	2½	0·5	1·25
1 hydraulic coupling oil pump (standby)	20,000 g.p.h. at 50 lb. per sq. in.	30	0	—
2 hydraulic coupling oil coolers				
1 hydraulic coupling oil purifier	300 g.p.h.	2½	0·5	1·25
1 set hydraulic coupling oil filters				
1 seawater circulating pump for hydraulic coupling oil coolers	260 ton/hr. at 30 lb. per sq. in.	36	0	—
2 main seawater circulating pumps (1 standby)	1,000 ton/hr. at 30 lb. per sq. in.	120	0·9	108·0
1 bilge pump	120 ton/hr. at 30 lb. per sq. in.	21	0·15	3·2
1 general service pump	120 ton/hr. at 30 lb. per sq. in.	21	0·15	3·2
4 200-kW Diesel generators		—	—	—
2 air compressors (for generators)		5	0·15	0·75
1 air receiver				
2 generator freshwater cooling pumps	25 ton/hr. at 30 lb. per sq. in.	6	0·9	5·4
2 generator freshwater coolers				
1 Cochran auxiliary boiler (for port or emergency use only)				
1 set oil-burning equipment				
2 boiler feed pumps				
1 feed water heater				
1 auxiliary condenser				
1 evaporator				
1 distiller condenser				
1 oily-water separator				
Ventilating fans		35	0·9	31·5
Workshop machinery		6	0·15	0·9
		381½		218·95
Equivalent electrical load, kW		335		192
Ship's services, say kW				145
Total electrical load, kW				337

basis. Only a standby lubricating oil pump is provided for the couplings as the working oil pumps are engine driven at the end of the output turbine quill shaft and are therefore running when the output turbine is turning even without load. A dynamo could also be coupled here if required and would be available for generating power in port by running the gas turbine with the clutches empty.

For survey and overhaul the gas turbine rotors are very light, for example the h.p. and i.p. rotors in the 10,500 s.h.p. scheme weigh 410lb. and 1,250lb. respectively. Such weights can easily be handled with normal facilities. Spare gear will be strictly comparable with that provided for a steam turbine installation.

Starting of both units in this installation is by a motor-driven starting motor. The engine room arrangement is shown in Fig. 10.

SUMMARY OF WEIGHTS, FUEL CONSUMPTIONS AND SPACE REQUIREMENTS

Detailed weights of the two installations are given in Table V. The weights of all the items except the gas turbine propulsion machinery are based strictly on running installations and no attempt has been made to cut weights, although with a total weight of main engines of 60 and 148 tons respectively in a dry weight of 343 and 579 tons, the scope for saving in cocks and valves, ladders, gratings, etc., is considerable.

Table VI summarizes the "all-in" specific and total fuel consumptions and is based on the use of residual fuel. The

TABLE V.—DETAILED MACHINERY WEIGHTS

Size of unit	3,340 s.h.p.	10,500 s.h.p.
H.P. turbine	0·92	2×1·30
H.P. compressor	1·73	2×2·00
I.P. compressor	—	2×2·28
H.P. line supports	0·78	2×1·60
I.P. turbine	1·78	2×2·52
I.P. turbine supports	0·36	2×0·52
L.P. turbine	1·29	2×1·82
L.P. compressor	3·31	2×4·50
L.P. line supports	1·83	2×2·58
H.P. intercooler	—	2×1·35
L.P. intercooler	1·76	2×2·50
Heat exchanger	5·85	2×8·25
Primary combustion chamber	0·50	2×0·71
Reheat combustion chamber	0·79	2×1·12
Ducting	8·00	2×12·25
Control gear, etc.	0·36	2×0·53
Reduction gearing (including hydraulic transmission)	30·0	56·0
Total weight of main engines	59·26	147·66
	—say 60 tons	—say 148 tons
Shafting, stern gear and propeller	56	120
Auxiliary boilers, uptakes and funnel	71	34
Auxiliaries, including generators	41	110
Pipes, cocks and valves	47	68
Remainder (ladders and gratings, tanks, ventilators, workshop, outfit, etc.)	47	59
Spare gear	21	40
Total dry weight	343 tons	579 tons
Water and oil	30	50
" Steam-up " weight	373 tons	629 tons

TABLE VI.—FUEL CONSUMPTION

		lb. per s.h.p. hr.	Tons per day
3,340 s.h.p. installation	Main engines	0·494	17·7
	Diesel generators	0·008	0·3
	Total	0·502	18·0
10,500 s.h.p. installation	Main engines	0·474	53·3
	Diesel generators	0·021	2·4
	Total	0·495	55·7

TABLE VII—SUMMARY OF MACHINERY WEIGHTS AND ENGINE ROOM SIZES

	Machinery weights, tons		Engine room length, ft.	Area at floor level, sq. ft.	Engine room volume, cu. ft.		
	Dry	" Steam-up "			Below main deck	Above main deck	Total
3,340 s.h.p. installation	343	373	44	2,460	51,000	20,300	71,300
10,500 s.h.p. installation	579	629	53	3,600	106,400	30,400	136,800

values quoted are not merely estimates since similar consumption figures have been measured in numerous tests. Lubricating oil consumption is negligible and this fact should be taken into account in any comparison with heavy-oil-engine machinery installations.

The final table, Table VII, gives a summary of machinery weights and engine room sizes for both schemes. The engine room volumes have been divided in the same way as the data provided by the Institute.

It will be seen that the engine room lengths, areas and volumes compare favourably with those in the ships specified. It is submitted that the engine room layouts provided in this paper are not crowded and that ample space for operation and overhaul is given. No doubt they could be improved in detail by working out an actual scheme with the superintendents of a ship-owning company.

CONCLUSIONS

From the large amount of experience gained in steam turbine work, it is felt that these gas turbine schemes have lower consumption, less weight and smaller engine room spaces than steam turbine installations designed for the same power and operating with usual steam conditions. Even at this stage the constructional cost of marine gas turbine machinery is not excessive. A careful estimate of cost of the complete installation of a marine gas turbine propulsion set for a power rather less than the larger of the two installations described here carried out by a member firm showed that if the cost of heavy-oil-engine machinery be taken as unity, then steam turbine machinery of the same power would be 1·09 times the amount and gas turbine machinery 1·16 times the amount. This latter figure, however, does not include the cost of shore trials which would be necessary in the same way as shore trials of heavy-oil-engine machinery were necessary when such machinery was in the same state of development as the gas turbine is at present. Maintenance cost of gas turbines, however, should be considerably less than for heavy oil engines as there should be no expensive parts corresponding to liners, piston rings, etc., requiring renewal at definite intervals in the life of the machinery.

While the gas turbine machinery described in this paper leads to a significant improvement in efficiency, space occupied, and weight, as compared with steam turbine machinery, research is being vigorously pursued in the development of the liquid-cooled high temperature gas turbine. Its potentialities are such that the air rate per horsepower hour would be one-third of those described, with corresponding reduction in size, and efficiencies better than those possible with present-day heavy oil engines.

REFERENCES

These references apply only to the continuous research on combustion of heavy fuel, part load running and other features on which the installations described in this paper are based.

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