PROPOSALS WITH 10,500 S.H.P. SULZER DIESEL ENGINES IN A 10,600-TON CARGO SHIP

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INTRODUCTION

The object of this paper is to discuss various factors in the design of Diesel propulsion machinery that must be considered in order to take advantage of the proposed tonnage reform and to suggest alternative installations of 10,500 s.h.p. suitable for a 10,600-ton cargo ship. The design of auxiliary sets, pumps, coolers and other accessories will not be discussed but obviously much could be done to make these more compact and to lessen their overhauling space and the access gangways around them.

The following factors must be considered by the engine designer when attempting to reduce the size of propelling machinery:—

(*a*) An increase in output per cylinder by the use of higher mean cylinder pressures.

(*b*) An increase in output by raising the piston speed.

(*c*) The use of higher revolution speeds with geared or electric drive.

(*d*) A more compact design of the engine.

(*e*) The use of unorthodox installations.

Dealing with each factor in turn it is clear that various solutions to the problem of space reduction can be obtained.

(*a*) It is generally accepted that the unsupercharged two-stroke engine gives about one-and-three-quarter times the output of a similar naturally aspirated four-stroke engine, each engine having the same number of cylinders, revolution speed, and cylinder dimensions. With a moderate increase in maximum cylinder pressures and the use of air coolers, a four-stroke engine can be supercharged to give about double its unsupercharged output, whilst the output of the two-stroke engine can be increased by 35 per cent to 50 per cent with the present methods of supercharging. These increases refer only to single acting engines supercharged by exhaust gas driven turbo blowers. The approximate relative outputs of similar engines are as follows:—

Four-stroke engine	1·0
Four-stroke supercharged engine	1·5 to 2·0
Two-stroke unsupercharged engine	1·75
Two-stroke supercharged engine	2·35 to 2·6

It would appear that the most compact type of engine, without

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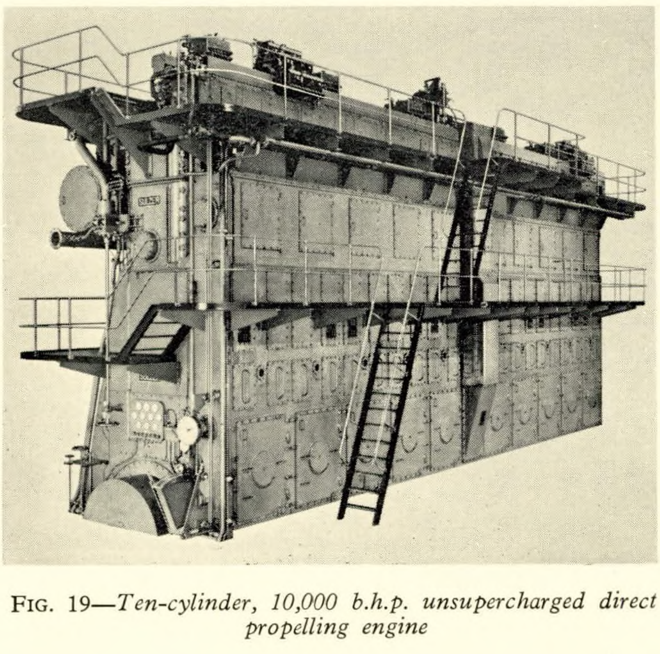


FIG. 19—*Ten-cylinder, 10,000 b.h.p. unsupercharged direct propelling engine*

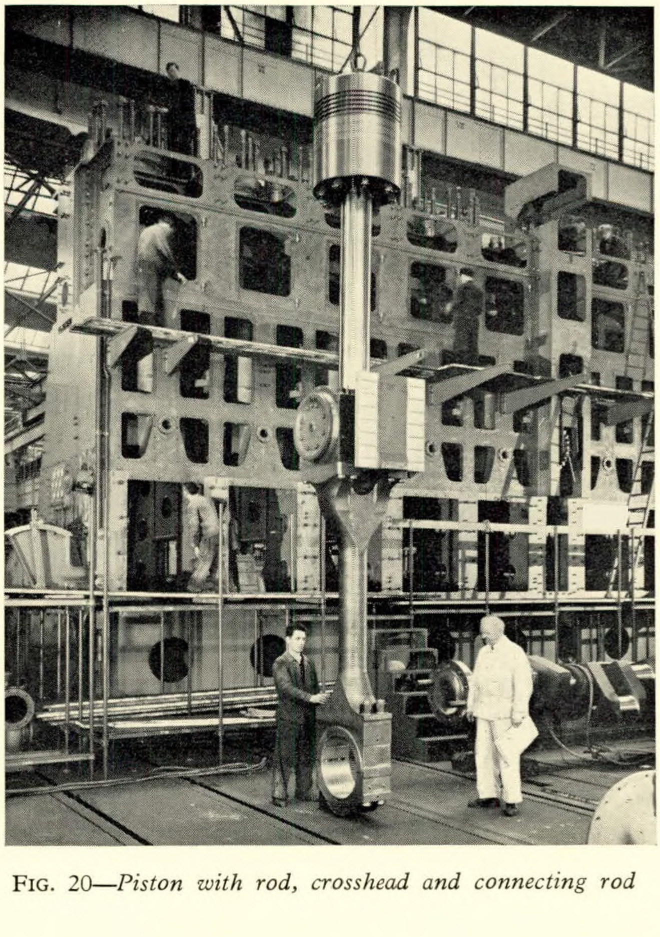


FIG. 20—*Piston with rod, crosshead and connecting rod*

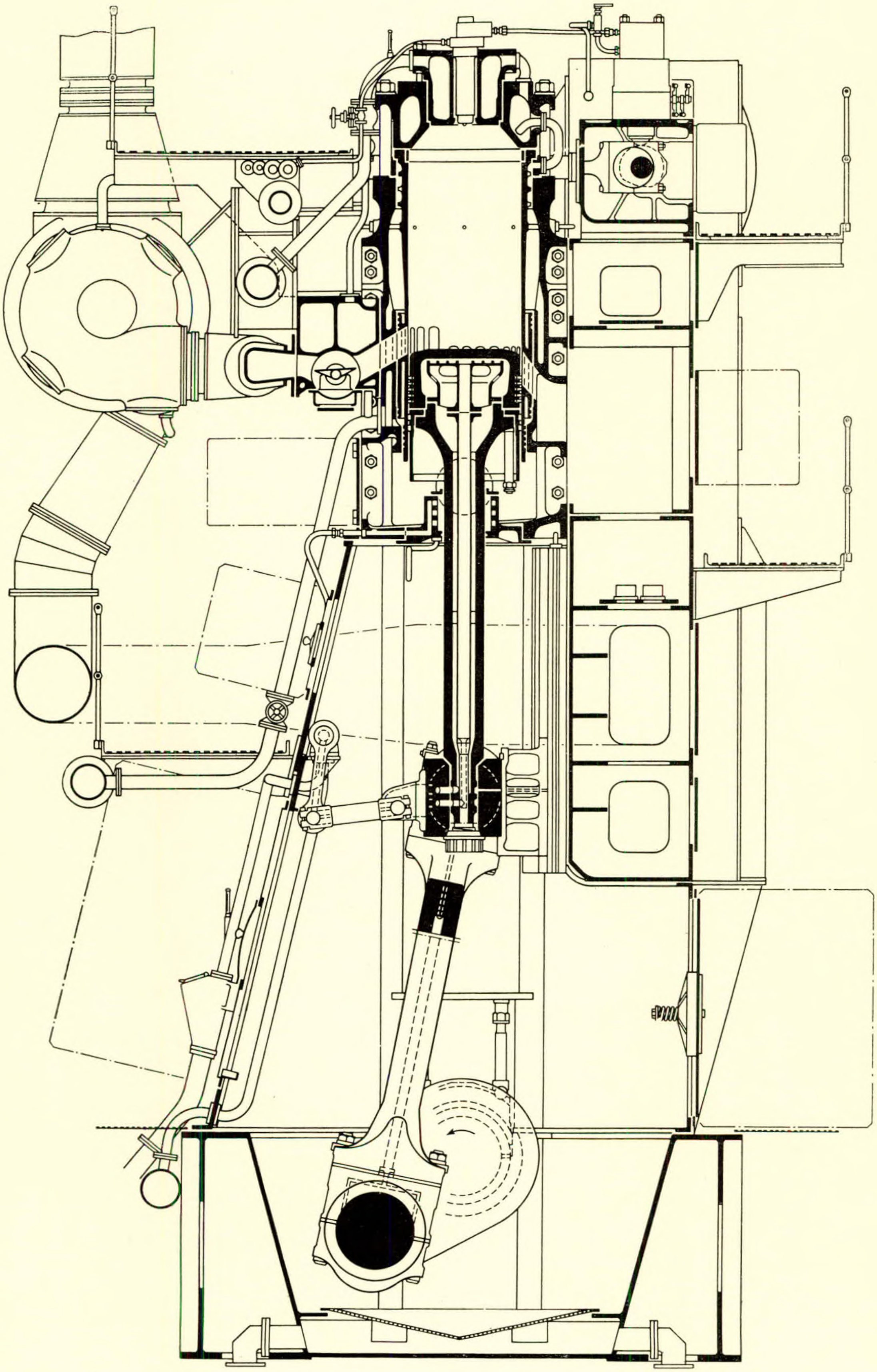


FIG. 21—*Cross-section of direct propelling supercharged engine*

the use of either engine driven or separately driven superchargers and without going to high maximum cylinder pressures, is the two-stroke supercharged engine.

(*b*) A substantial increase in the generally accepted figures of mean piston speed is not to be recommended for several reasons.

1. There would be greater liner and piston ring wear.
2. Higher and therefore less efficient propeller speeds, or alternatively, longer stroke engines would be necessary with greater height and width.
3. A possibility of greater vibration in the ship because of the greater forces and couples which may not be perfectly balanced.
4. Higher inertia stressing of the engine parts.
5. Less efficient scavenging and higher mechanical losses, hence higher fuel consumptions.

(*c*) The use of engines having higher revolution speeds makes essential the use of some form of indirect transmission. The usual methods are mechanical single reduction gearing incorporating either hydraulic or electromagnetic couplings, or alternatively, an electric drive. It is possible to dispense with the couplings in certain geared installations provided the gears are dimensioned to withstand the peak value of the cyclic torque produced by the engines and provided the torsional vibration conditions of the system are favourable. The elimination of couplings will save space and can improve conditions of balance in multi-engined installations because the crankshafts can be set in definite angular positions one with another.

(*d*) Apart from supercharging and the use of higher revolution speeds, other factors can be considered for reducing engine dimensions. Firstly, the use of the double acting principle which in theory can almost halve the length but which greatly adds to the height and the complication of an engine. Generally the double acting engine is out of favour because of the difficulties of overhauling but in view of the proposed tonnage reform, the saving, because of its compactness, may outweigh its other disadvantages. Secondly, the use of engines having two or more rows of relatively small cylinders can be considered, such as the V-type, the double row type with two crankshafts geared together, the Deltic type, etc. For cargo ships of the type under consideration these engines could only be used with an indirect drive. Proposed installations using these engines could be subdivided into two main classes: (1) in which there are relatively few cylinders and working parts allowing overhauls to be carried out on board, preferably during times of unloading and loading of the ship, and (2) for electric drive only, where a large number of small engined generators are fitted which can be taken in and out of the ship for overhauling ashore.

For installations of the former class, the size and number of cylinders should be carefully considered, the first for ease in handling pistons in the engine room and the second for the possibility of routine overhauling during normal service. Also, it is important that the engines, for both classes of installation, are sufficiently robust for long continuous running between normal overhauls.

(*e*) Under the heading of unorthodox installation, one using a large number of small-engined generators as mentioned above could be contemplated. There would be no need for the usual engine room containing practically all the ship's machinery but instead the generating sets may be placed anywhere in the ship, possibly where the space may not be suitable for cargo.

INSTALLATIONS

For the type of ship under consideration any machinery installation chosen must be reliable and suitable for long continuous service without frequent overhauling. The propeller speed must be sufficiently low for efficient propulsion and the installation should not be too complicated. The overall efficiency must be good, and if at all possible, the engines should be capable of burning boiler oils without risk of crankcase contamination.

With these objects in view, three modern compact installations of 10,500 s.h.p. for a 10,600-ton cargo liner are compared, in which the engines are of standard design with generally accepted normal mean indicated pressures and piston speeds. In all three arrangements the space required for overhauling the main engines, auxiliary sets and accessories has not been cramped, and the usual pumps, fuel purification plant, air compressors, coolers, tanks, exhaust gas and oil fired boilers, etc., are shown.

Installation A

By far the greater majority of Diesel driven cargo ships have direct propelling machinery and over 80 per cent of these have single-acting two-stroke engines. This installation has been planned with an engine of this type fitted with exhaust gas driven turbo blowers giving about 35 per cent supercharge. The design of the unsupercharged version of this engine (Fig. 19) was described in Mr. W. Kilchenmann's paper*. Briefly, it is a crosshead engine with loop scavenge in which the cylinders are sealed off from the crankcase by a division plate and by stuffing boxes on the piston rods. To keep down the height of the engine, short pistons are fitted, as shown in Fig. 20, necessitating eccentric operated oscillating valves in the exhaust branches from the cylinders. Scavenge air is provided by individual double acting pumps driven by arms bolted to the engine crossheads.

The supercharged version of this 760 mm. bore engine is shown in cross-section in Fig. 21. The scavenge pumps of the normal engine are removed and in their place a scavenge air receiver or trunk is fitted. This receiver is supplied with air delivered through coolers by the turboblowers which are independent of electric motor drive. The receiver discharges upwards through low resistance streamlined non-return valves into the under piston spaces which are in direct connexion with the scavenge ports in the cylinders and which are separated one from another. Fig. 22 shows the arrangement and the scavenge pressure obtained on a test engine of smaller cylinder size.

With this arrangement the underside of the piston supplements the scavenging effect, gives higher scavenging pressures throughout most of the scavenging period, especially at the beginning, gives higher pressure impulses to the exhaust gas turbo blower and hence gives greater volumes of scavenge air with lower thermal loading of the engine. An equally or perhaps more important feature of this arrangement is that ample air is pumped by the undersides of the pistons to enable the engine to start and manœuvre equally well as a non-supercharged engine. Even with all turbo blowers out of action the engine can be operated at about 70 per cent of full speed, thus ensuring the safety of the ship in this unlikely event.

Fig. 23 shows the proposed layout of the eight-cylinder supercharged engine in a ship of 10,600 tons. Two turbo blowers are fitted, each working on the impulse system. These are placed at cylinder block level, leaving access to the cylinder heads, pistons, cylinder ports, etc., quite unobstructed, as shown in Fig. 24. The scavenge air is led to coolers at the forward and aft ends of the engine and then to the scavenge

* Kilchenmann, W. 1953. "New Designs of Large Two-stroke Marine Diesel Engines". Trans.I.Mar.E., Vol. LXV, p. 137.

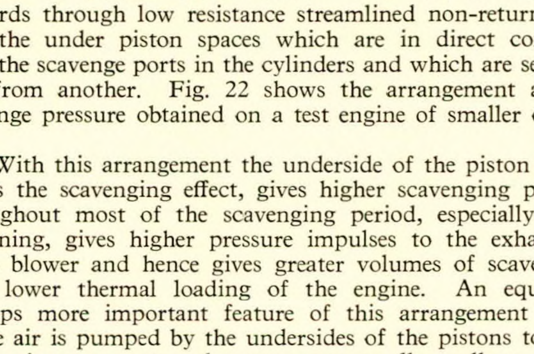


FIG. 22—*Supercharging tests (pulse-system) diagram*

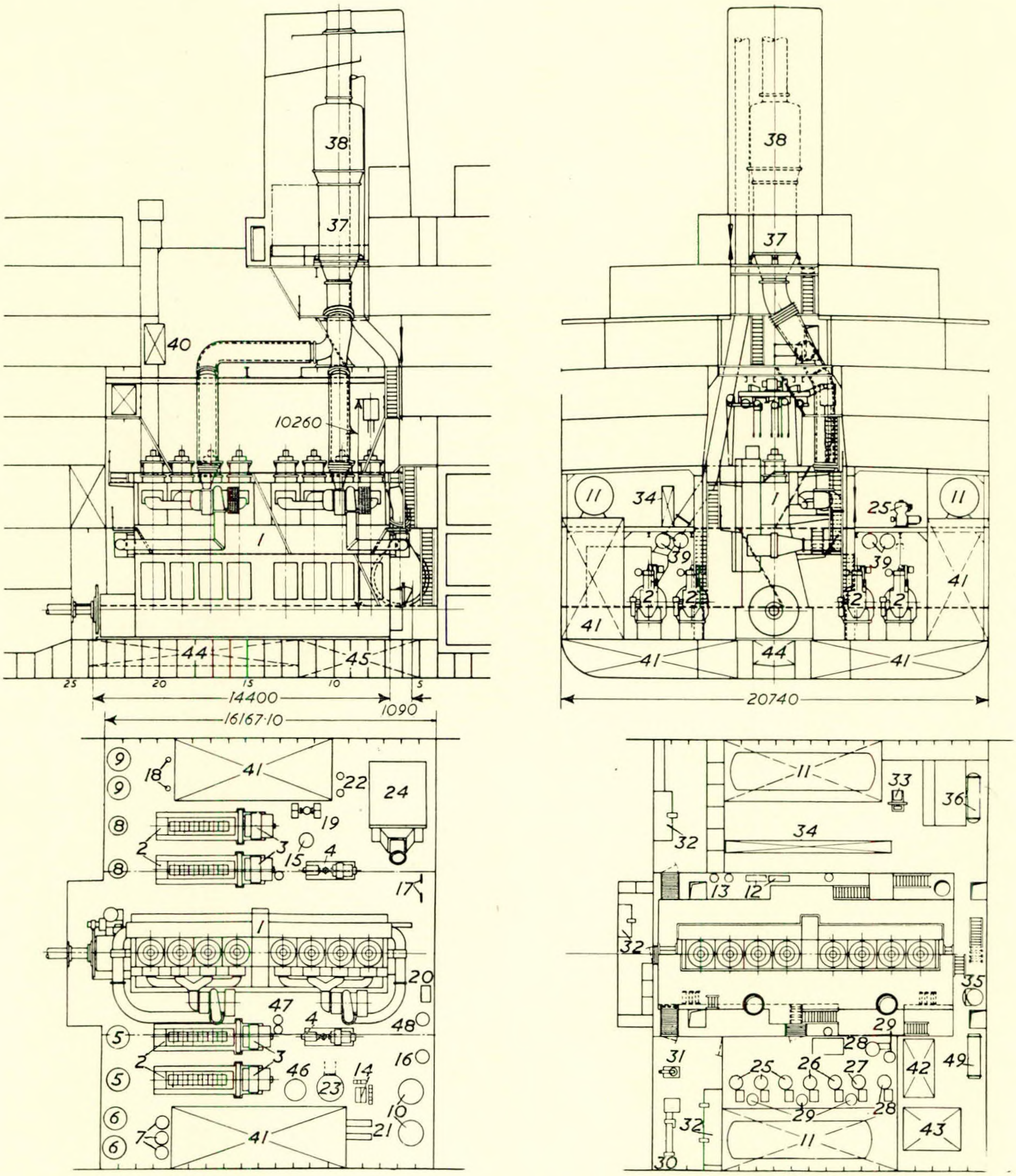


FIG. 23—*Single-screw two-stroke machinery: 10,500 s.h.p., 115 r.p.m.*

(1) Main engine, 10,500 b.h.p., 115 r.p.m.; (2) Two auxiliary engines, 450 b.h.p., 500 r.p.m.; (3) Four generators, 300-kW; (4) Two air compressors, electric driven; (5) Two oil pumps for piston cooling and bearing lubrication; (6) Two oil coolers; (7) Three oil filters; (8) Two fresh water pumps for cylinder cooling; (9) Two fresh water coolers; (10) Two sea water pumps for oil and fresh water coolers; (11) Two starting air tanks; (12) Two fresh water pumps for fuel valve cooling; (13) Two fresh water coolers for fuel valve cooling; (14) Bilge pump; (15) Ballast pump; (16) Fire pump; (17) Two boiler feed pumps; (18) Two fuel oil service pumps; (19) Fuel oil transfer pump; (20) Sanitary pump; (21) Two drinking water pumps; (22) Two warm water pumps; (23) Oily water separator; (24) Oil fired boiler, La Mont type; (25) Three heavy oil purifiers; (26) Two heavy oil clarifiers; (27) Diesel oil separator; (28) Two lubricating oil separators; (29) Four heaters for heavy fuel and lubricating oil; (30) Lathe; (31) Drilling machine; (32) Three working benches; (33) Steam driven air compressor; (34) Switchboard; (35) Evaporator; (36) Auxiliary condenser; (37) Exhaust gas boiler; (38) Exhaust gas silencer for main engine; (39) Four exhaust gas silencers for auxiliary engines; (40) Fresh water expansion tank; (41) Fuel oil bunkers; (42) Fuel oil daily service tank; (43) Lubricating oil storage tank; (44) Oil drain tank, 27 cu. m.; (45) Reserve oil tank, 23 cu. m.; (46) Residue tank for separators; (47) Starting air bottle for auxiliary engines; (48) Sea and fresh water harbour pump; (49) Drinking water filter.

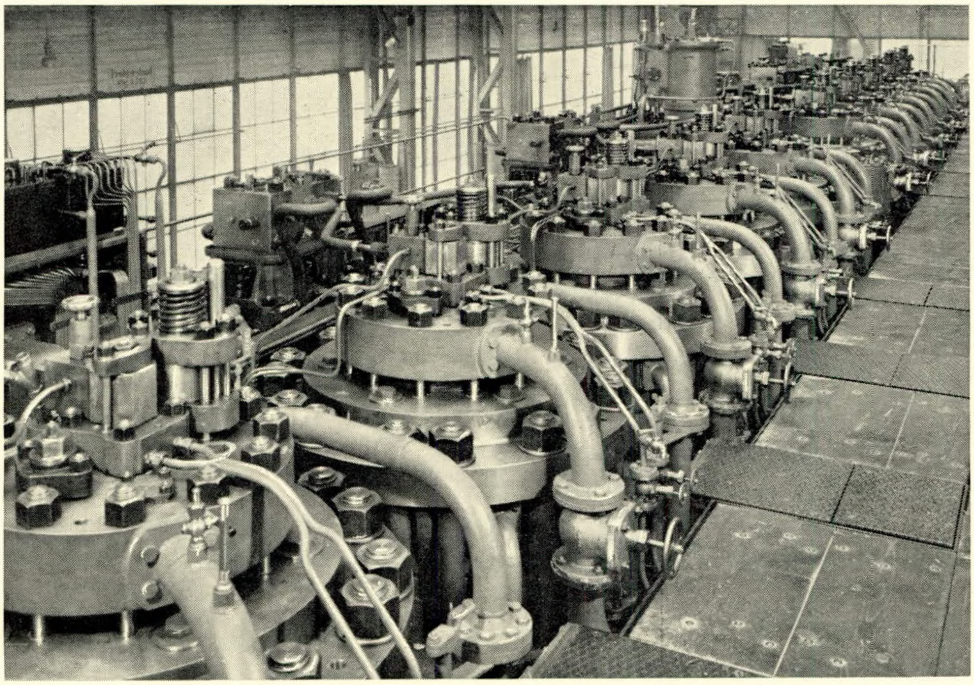


FIG. 24—*Cylinder heads of ten-cylinder direct propelling engine*

receiver at the port side. Otherwise, the installation is normal and similar to others with non-supercharged engines planned during the last few years.

Provision is made for burning boiler oil of viscosities up to 3,000 seconds Redwood No. 1 at 100 deg. F. or higher if required. With this relatively cheap fuel, the high thermal efficiency of this type of engine, the practically non-existent transmission losses and the efficient propeller speed, it will be extremely difficult to devise any other type of acceptable propelling plant with lower fuel costs.

As regards space requirements, the volume of the engine room accounts for 12·6 per cent of the gross registered tonnage and the weight of the engine, generating sets and accessories is about 666 tons. Better figures can be obtained for both space requirements and weight, as the following examples will show. For ten-cylinder non-supercharged engines of comparable power, the space and weight figures are about 14·5 per cent and 774 tons respectively.

Installation B

This installation makes use of higher revolution speeds in addition to supercharging as described for the engine in proposal A. The installation is shown in Fig. 25 and consists of two eight-cylinder supercharged two-stroke engines running at 240 r.p.m. driving through hydraulic couplings and a single reduction gear to the propeller shafting turning at 100 r.p.m. The engines are of similar design to the engine described in proposal A but of smaller cylinder dimensions and stroke-bore ratio. Unsupercharged engines of this design are installed in several geared ships and are running on heavy fuel oils (see Fig. 26).

The design in service has proved to be robust and quite suitable for ships having long voyages. The sizes of parts are convenient for easy handling by the engine room lifting gear and there are relatively few cylinders, thus allowing routine overhauling to be carried out during loading and unloading of the ship in overseas ports.

With this arrangement the length of the engine room is about the same as for proposal A, but some of the space above the engine room can be used for accommodation or other purposes. The engine room volume is reduced to 10·7 per cent of the gross registered tonnage and the total weight of engines, gears, auxiliary sets and accessories is reduced to 548·6 tons.

The auxiliary sets comprise two 650-kW sets driven by 1,000 b.h.p. twelve-cylinder double-bank, four-stroke supercharged engines running at 700 r.p.m., and one 200-kW set driven by a 300 b.h.p. six-cylinder four-stroke supercharged engine running at 750 r.p.m.

Installation C

This installation (Fig. 27) goes one stage further and makes use of V-type main propelling engines of compact design running at an even higher speed of 420 r.p.m. The arrangement of the drive and the auxiliary sets are as described for proposal B.

The main engines are of the single-acting supercharged two-stroke type with cylinders 420 mm. bore, arranged with one exhaust gas turbo blower per engine. Again, there are relatively few cylinders which make it possible for routine overhauling to be carried out with the ship in port for loading or unloading. The parts on this type of engine are much lighter than those for proposal B, which more than compensates for their additional numbers and the more awkward piston handling with the inclined cylinders.

A considerable saving in space and weight over that taken for proposal B results with this arrangement, as will be seen from the particulars given in Table VIII. The space taken is about 9 per cent of the gross registered tonnage but the reduction in length of the engine room is small, i.e. 10·7 per cent of the length b.p. of the ship as against 11·3 per cent for both proposals A and B.

CONCLUSIONS

In comparing the installations with others, allowance should be made for the fact that in all proposals, A, B and C, the figures given include the boiler, the workshop, and the stores. The high percentage of space occupied by them and by the auxiliary sets and accessories causes the saving by the use of geared and more compact engines, to be less than may be expected. For proposal C, and to a lesser extent for proposal B, further saving in space could be achieved, not by further reductions in size of the propelling engines but by careful study

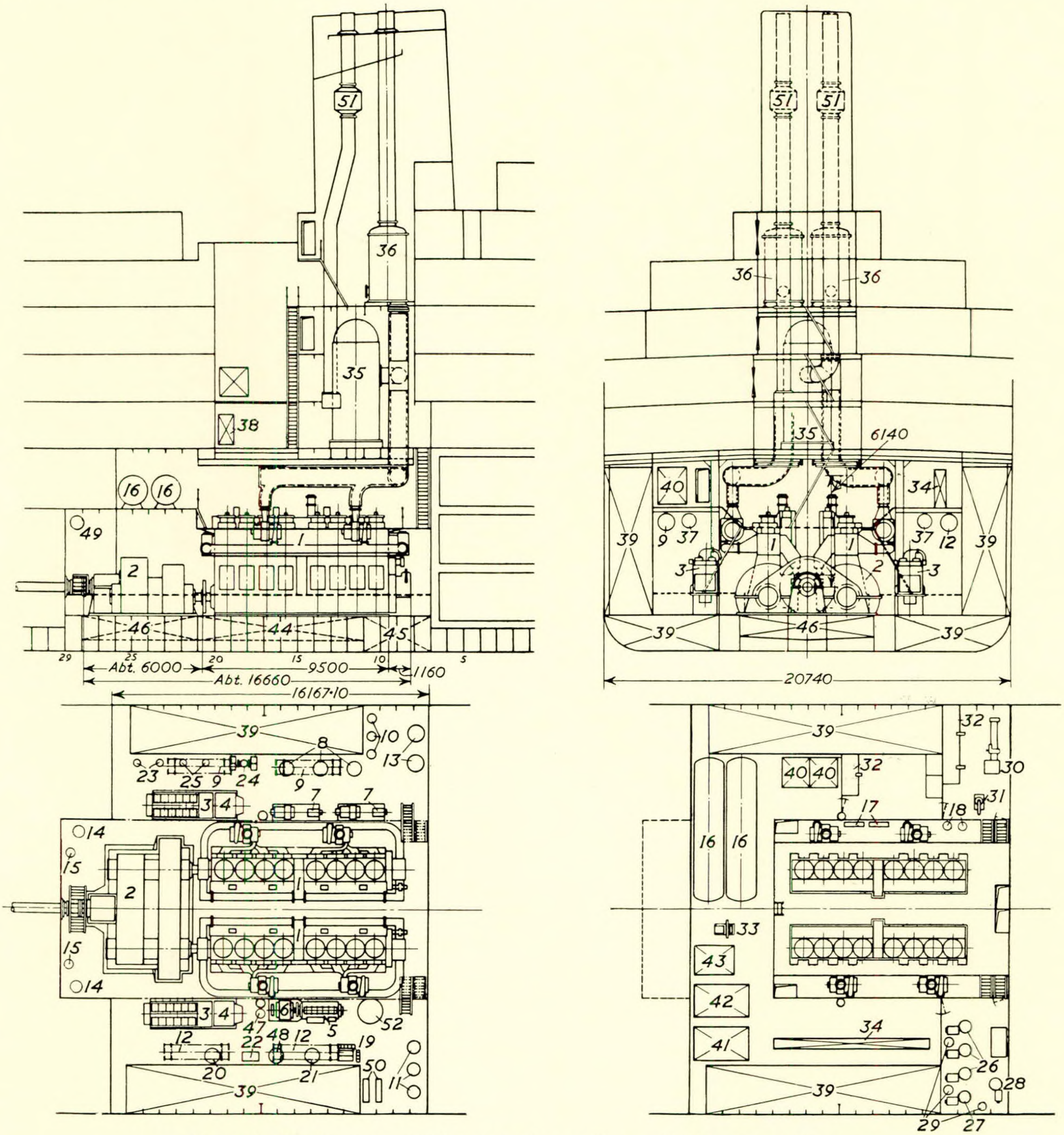


FIG. 25—*Single-screw geared two-stroke machinery: 10,500 s.h.p., 100 r.p.m.*

(1) Two main engines, 11,000 b.h.p., 240 r.p.m.; (2) Hydraulic reduction gear; (3) Two auxiliary engines, 1,000 b.h.p., 700 r.p.m.; (4) Two generators, 650-kW; (5) Harbour set, 300 b.h.p., 750 r.p.m.; (6) Generator, 200-kW; (7) Two air compressors, electric driven; (8) Three oil pumps for piston cooling and bearing lubrication; (9) Two oil coolers; (10) Three oil filters; (11) Three fresh water pumps for cylinder cooling; (12) Two fresh water coolers; (13) Two sea water pumps for oil and fresh water coolers; (14) Two oil pumps for hydraulic reduction gear; (15) Two oil filters for hydraulic reduction gear; (16) Two starting air tanks; (17) Two fresh water pumps for fuel valve cooling; (18) Two fresh water coolers for fuel valve cooling; (19) Bilge pump; (20) Ballast pump; (21) Fire pump; (22) Sanitary pump; (23) Two fuel oil service pumps; (24 Fuel oil transfer pump; (25) Two fuel oil booster pumps; (26) Three heavy oil separators; (27) Diesel oil separator; (28) Lubricating oil separator; (29) Three heaters for fuel oil and lubricating oil; (30) Lathe; (31) Drilling machine; (32) Two working benches; (33) Steam driven air compressor; (34) Switchboard; (35) Combined exhaust gas and oil fired boiler; (36) Two exhaust gas silencer and spark arresters for main engines; (37) Three exhaust gas silencers for auxiliary engines; (38) Fresh water expansion tank; (39) Fuel oil bunkers; (40) Two fuel oil daily tanks; (41) Lubricating oil storage tank; (42) Cylinder oil storage tank; (43) Gravity tank for reduction gear; (44) Two oil drain tanks for main engines; (45) Reserve oil tank for main engines; (46) Oil drain tank for reduction gear; (47) Starting air bottle for auxiliary engines; (48) Sea and fresh water harbour pump; (49) Oil cooler for hydraulic coupling; (50) Two drinking water pumps; (51) Two spark arresters for oil fired boiler; (52) Oily water separator.

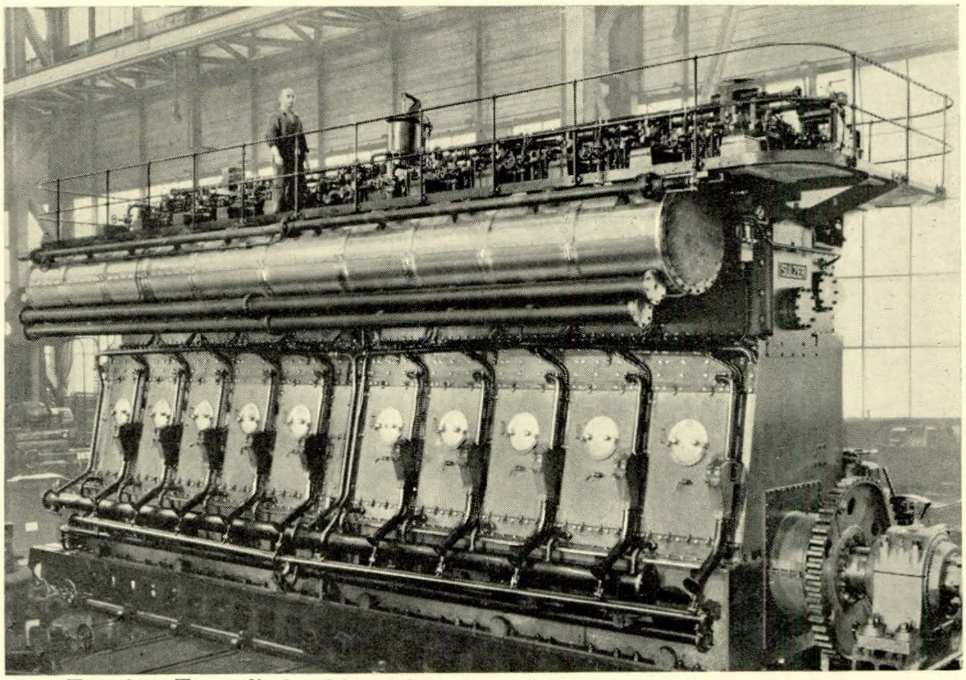
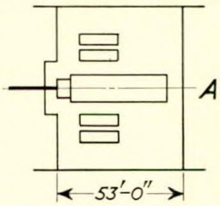
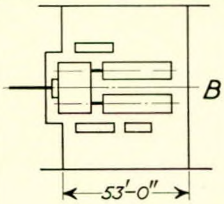
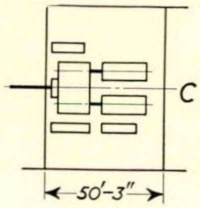


FIG. 26—*Ten-cylinder 5,000 b.h.p. unsupercharged engine for geared drive*

TABLE VIII.—10,600 TONS DEADWEIGHT CARGO LINER. COMPARISON OF SUPERCHARGED MACHINERY OF 10,500 S.H.P.

Proposal	 Fig. 23	 Fig. 25	 Fig. 27
Output, s.h.p.	10,500	10,500	10,500
Propeller speed, r.p.m.	115	100	100
Efficiency of hydraulic reduction gear, per cent.	—	95·5	95·5
Output on engine coupling flange, b.h.p.	10,500	11,000	11,000
Engine type	8 RSAD 76	2×8 RSAG 58	2×12 VQAG 42
Engine speed, r.p.m.	115	240	420
Mean piston speed, ft. per min.	1,170	1,196	1,377
Brake mean effective pressure, lb. per sq. in.	105·3	92·7	102·2
Mechanical efficiency, per cent.	88	82	76
Mean indicated pressure, lb. per sq. in.	128·5	116	134·5
Product m.i.p. × piston speed × 10^{-3}	150·3	138·7	185
Weights of main engines, tons	510	307	111
Weights of hydraulic reduction gear including auxiliaries, tons	—	130	100
Weights of auxiliary engines, tons	96	41·6	41·6
Weights of pumps, coolers, filters, exhaust silencers and starting air tanks for main and auxiliary engines, tons	60	70	70
Weights of water, oil and fuel oil in circulation, tons	56	72	52
Fuel consumption of main engines, lb. per b.h.p. hr.	0·335	0·37	0·38
Fuel consumption of auxiliary engines, lb. per b.h.p. hr.	0·39	0·39	0·39
Daily fuel consumption: main engines, tons	37·8	43·5	45·5
Daily fuel consumption: auxiliary engines, tons	5·0	5·0	5·0
Total daily fuel consumption, tons	42·8	48·5	50·5
Lubricating oil consumption: main engines, lb. per hr.	23·2	29·1	73
Lubricating oil consumption: auxiliary engines, lb. per hr.	3·75	7·70	7·70
Daily lubricating oil consumption, tons	0·293	0·400	0·867
Underdeck volume of engine room, cu. ft.	89,600	103,000	62,200
Total engine room volume, cu. ft.	122,200	107,500	87,500
Engine room volume in percentage of gross tonnage, per cent.	12·55	10·7	9
Length of propelling machinery	50ft. 10in.	54ft. 8in.	40ft. 10in.
Length of engine room	53ft. 0in.	53ft. 0in.	50ft. 3in.
Length of engine room in percentage of ship's length b.p., per cent.	11·3	11·3	10·7

Length of ship b.p., 470ft.; Total deadweight, 10,624 tons; Gross tonnage, 9,717 tons; Displacement, 17,600 tons; Loaded speed, 16½ knots

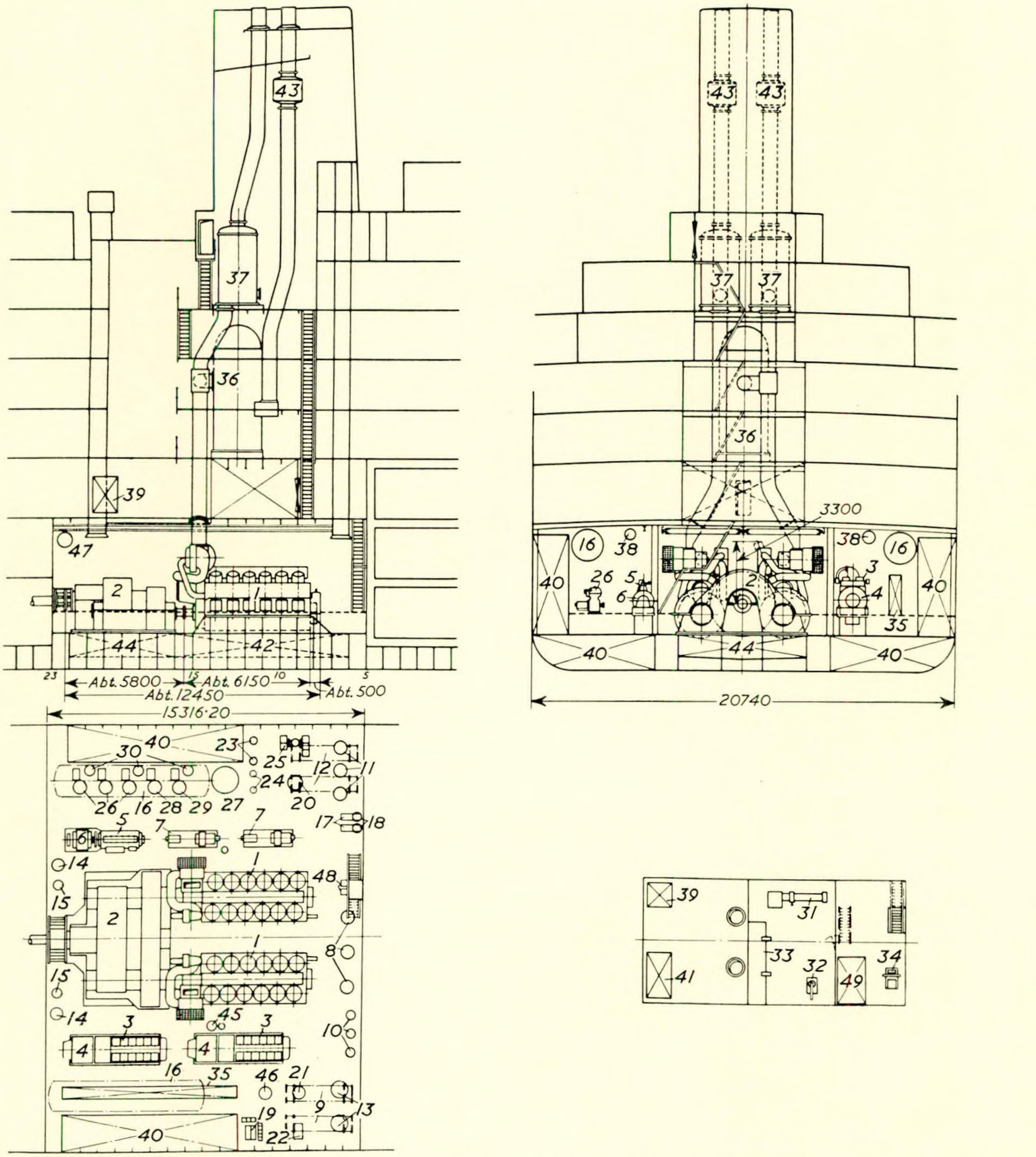


FIG. 27—*Single-screw geared two-stroke machinery: 10,500 s.h.p., 100 r.p.m.*

(1) Two main engines, V-type, 11,000 b.h.p., 420 r.p.m.; (2) Hydraulic reduction gear; (3) Two auxiliary engines, 1,000 b.h.p., 700 r.p.m.; (4) Two generators, 650-kW; (5) Harbour set, 300 b.h.p., 750 r.p.m.; (6) Generator, 200-kW; (7) Two air compressors, electrically driven; (8) Three oil pumps for piston cooling and bearing lubrication; (9) Two oil coolers; (10) Three oil fiters; (11) Three fresh water pumps for cylinder cooling; (12) Two fresh water coolers; (13) Two sea water pumps for oil and fresh water coolers; (14) Two oil pumps for hydraulic reduction gear; (15) Two oil filters for hydraulic reduction gear; (16) Two starting air tanks; (17) Two fresh water pumps for fuel valve cooling; (18) Two fresh water coolers for fuel valve cooling; (19) Bilge pump; (20) Ballast pump; (21) Fire pump; (22) Sanitary pump; (23) Two fuel oil service pumps; (24) Two fuel oil booster pumps; (25) Fuel oil transfer pump; (26) Three heavy oil separators; (27) Oily water separator; (28) Diesel oil separator; (29) Lubricating oil separator; (30) Three heaters for fuel oil and lubricating oil; (31) Lathe; (32) Drilling machine; (33) Working bench; (34) Steam driven air compressor; (35) Switchboard; (36) Combined exhaust gas and oil fired boiler; (37) Two exhaust gas silencer and spark arresters for main engines; (38) Three exhaust gas silencers for auxiliary engines; (39) Fresh water expansion tank; (40) Fuel oil bunkers; (41) Gravity tank for hydraulic reduction gear; (42) Two oil drain tanks for main engines; (43) Two spark arresters for oil fired boiler; (44) Oil drain tank for hydraulic reduction gear; (45) Starting air bottle for auxiliary engines; (46) Sea and fresh water harbour pump; (47) Oil cooler for hydraulic reduction gear; (48) Two drinking water pumps; (49) Fuel oil daily service tank.

of the arrangement and design of the auxiliaries and the layout of the workshop and stores. For example, at sea, part of the electrical load could be carried by generators driven by the main engines through the gearbox; thus the size of the auxiliary generating sets could be reduced.

A study of Table VIII shows that savings in space and weight of the engines are obtained at the expense of higher fuel and lubricating oil consumption. Taking the fuel consumption of the direct propelling engined plant, 42·8 tons per day, as 100 per cent, the consumptions of the geared engined installations, B and C, are 113 per cent and 118 per cent respectively, neglecting fuel for the boilers. The relative lubricating oil consumptions are 100 per cent, 142 per cent, and 311 per cent, neglecting losses in the couplings and gears.

In deciding whether it is worth while to have an installation with fast running compact engines of lower first cost similar to proposal C, the increased running costs should be deducted from the gains resulting from the increased cargo carrying capacity. For example, the increase in cost of fuel and lubricating oil of proposal C over A is about £2,500 for a voyage of thirty days whilst the increase in cargo space underdeck is 27,400 cu. ft. and the extra weight which can be carried is about 360 tons. These comparisons rather favour proposal C as they are based on the same quality of boiler oil for each installation. Whilst the propelling engines of C will burn boiler oils quite satisfactorily, it may be considered better to use a Diesel oil in order to minimize the crankcase contamination of these trunk-piston engines. The use of Diesel oil for the engines in proposal C will increase the fuel costs considerably and thus lessen any gains that might be achieved by the use of this compact machinery.

For the type of ship under consideration it would appear that, using orthodox machinery, it will be difficult to reduce the engine room volume below 9 per cent or 10 per cent of the gross tonnage and even these figures are only possible with some sacrifice in efficiency. Therefore, it is suggested that the direct-coupled supercharged two-stroke engined installation similar to that shown in proposal A will be, for some time to come, the most favourable, as the advantages gained from higher efficiency, reliability and long running periods between overhauls, outweigh the gains in space and weight achieved with geared installations of which proposals B and C are examples.

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