RECENT DEVELOPMENTS IN MARINE DIESELS (THE DOXFORD ENGINE)

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The outstanding developments in marine Diesel engines in Great Britain during the past few years have been concerned with the supercharging of the large two-cycle engine and with the extensive use of boiler fuel.

An outline will be given of the development work undertaken in the supercharging of the Doxford engine and a brief survey of experience at sea and of the characteristics of supercharging of the two-cycle opposed piston engine. Similarly, a description will be given of the employment of boiler fuels and of the advantages to be gained and of the difficulties encountered and methods of overcoming these.

The diaphragm engine and oil cooling of the lower pistons has also been developed for the Doxford engine, particularly to enable boiler fuel to be used more successfully. A new fuel injection system has been developed, known as the timing valve injection system, and a brief description of the characteristics of this system will be given, together with a survey of the advantages.

Prospects of improving the thermal efficiency and rating of the marine two-cycle engine will be reviewed, together with methods of achieving mechanical simplification of two-cycle marine engines, leading to easier operation and reduced maintenance. From this a short outline will be given of possible future developments.

SUPERCHARGING

In 1950 the author's company began to consider the turbocharging of their large two-cycle opposed piston engines and it was decided to build a three-cylinder engine of 600-mm. bore by 2,000-mm. stroke to run at 125 r.p.m., and to supercharge this to 30 or 50 per cent increase in power over the normal engine.

A preliminary investigation was made by fitting nozzles

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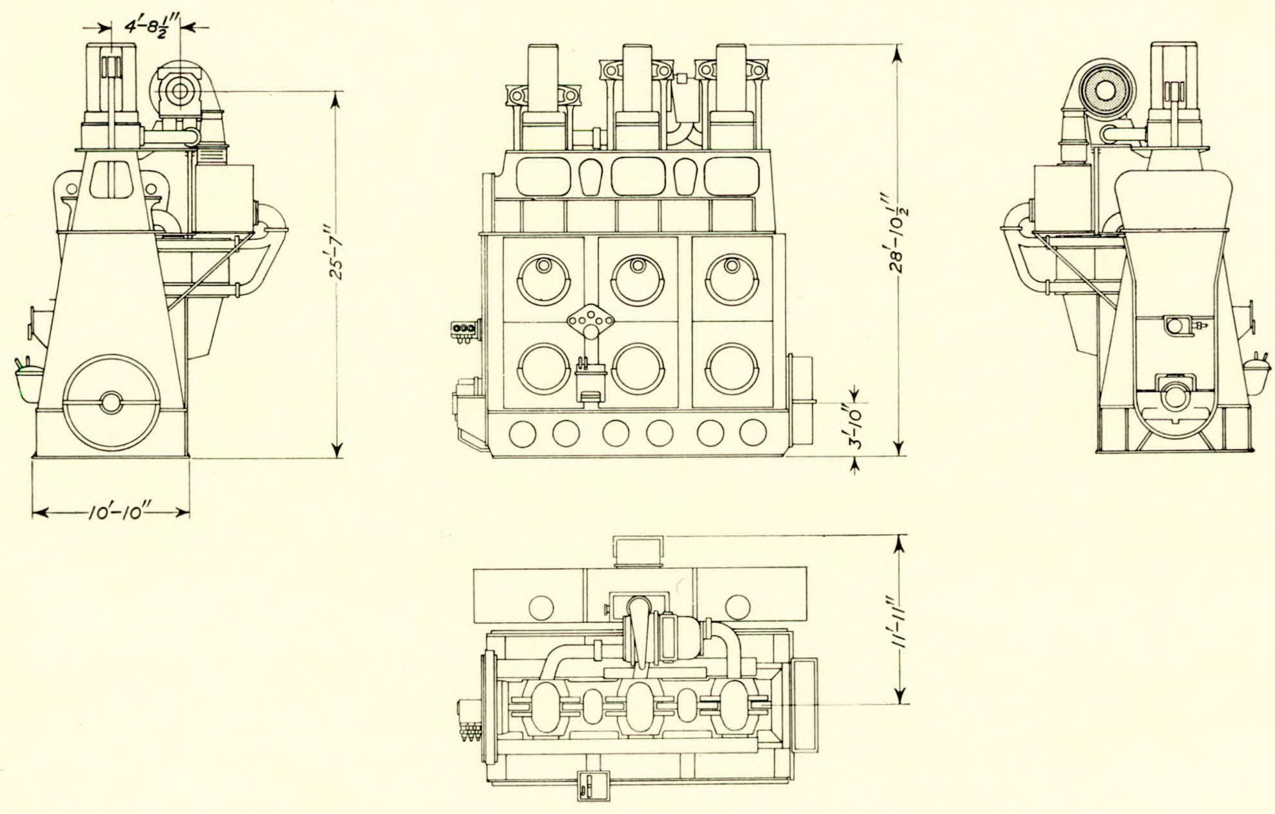


Fig. 10—*Arrangement of turboblower on Doxford three-cylinder engine*

in the exhaust pipes of a standard engine. The period of exhausting and charging of a cylinder occupied 120 degrees of crank angle but there were pressure waves in the exhaust system, particularly in the long pipes from No. 3 cylinder. It was, therefore, decided that the turbine should have two inlets, one taking the exhaust from No. 1 cylinder and the other from Nos. 2 and 3 cylinders. In this way the exhausts of Nos. 1 and 3 cylinders, which were farthest apart, were kept separate so that there would be no interference due to pressure waves. The turbine nozzle ring was divided into two almost equal sections, since the volume of gas from Nos. 2 and 3 cylinders do not combine, but follow through the turbine successively. The arrangement is shown in Fig. 10. It was considered that some 28 degrees of crank angle would be required to ensure the products of combustion being expelled from the cylinder before the air ports opened under conditions of 35 per cent increase in power by supercharging. The engine had to operate in the astern direction as well as in the ahead direction.

From the information supplied, Brown, Boveri and Co., Ltd., were able to calculate the quantity of gas and mass flow to determine their turbine nozzles and blading, and the engine was built incorporating these features. Within one month of completion it was giving an increased power of 50 per cent with a clean exhaust and low fuel consumptions. A set of diagrams is shown in Fig. 11 and the corresponding performance curves are shown in Fig. 12. The engine had been built with a lever-driven reciprocating scavenging pump and with the turbocharger delivering air into it through an intercooler and air receiver. The air receiver was fitted to minimize the effect on the rotary blower of the irregularity of the demands for air of the reciprocating pump. On a 6-cylinder engine fitted with three double acting scavenging pumps, giving a delivery of air for every cylinder once per revolution, the effect of the air impulses will be much reduced and only a small receiver will be required. The intercooler was fitted to reduce the temperature of the air delivered by the turboblower to increase the weight passing into the engine.

Preliminary tests indicated that the best arrangement was with the turboblower and engine-driven reciprocating blower in series and a further consideration was that with this arrangement the reciprocating blower could supply the engine with air should the turboblower be out of action for any reason. When the turboblower alone was used to supply the scavenging air to the engine there was insufficient energy from the exhaust impulses and the air supply to the engine was deficient, causing a smoky exhaust up to about half-speed of the engine. It was obvious that for the turboblower alone to be used there

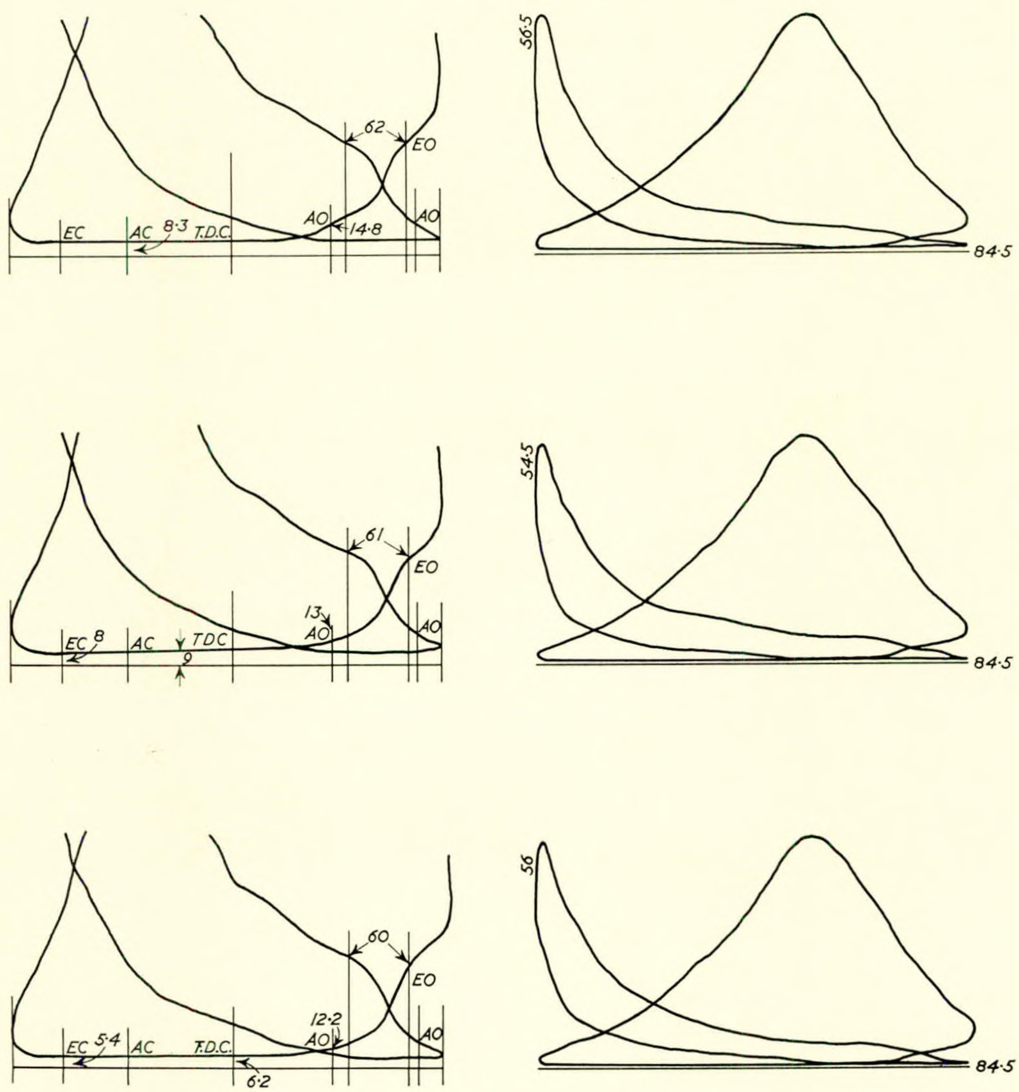


FIG. 11—*Diagrams from turbocharged engine*

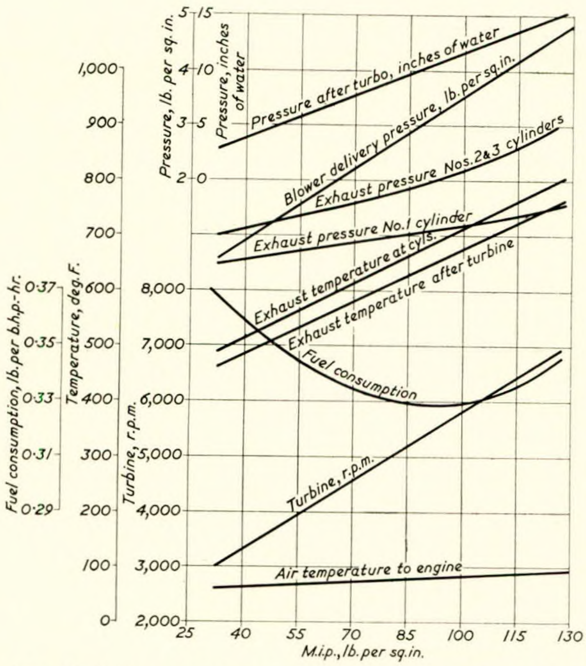


FIG. 12—*Performance curves of Doxford supercharged engine*

would have to be some auxiliary supply of air or auxiliary power to the turbine to increase the air delivery at low loads and under manœuvring conditions. The turboblower was very sensitive to the back pressure on the exhaust system and the efficiency of the turbine fell considerably with increase in back pressure, to the detriment of the power that could be developed. It was essential that the exhaust from the turbine should be available for passing through a waste heat boiler for raising steam and with the engine-driven reciprocating blower operating behind the turboblower and taking its air from the intercooler, the system is by no means so sensitive to the back pressure of the exhaust piping and exhaust boiler. It will be seen from the performance data given in Fig. 12 that a fuel

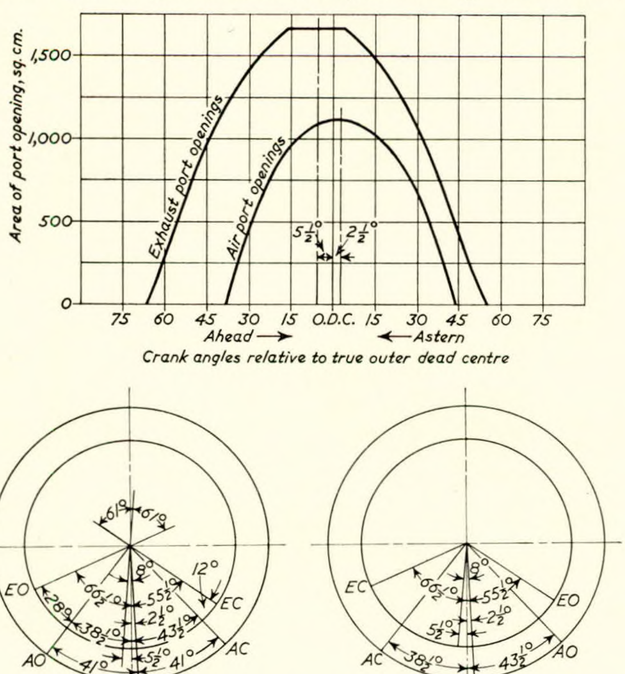


FIG. 13—*Timing diagram and port opening curves with 8-deg. lead*

consumption of 0·33lb. per b.h.p. per hr. was obtained at a mean indicated pressure of 105lb. per sq. in., and at 125lb. per sq. in. at 125 r.p.m. the consumption was little over 0·34lb. per b.h.p. per hr. It will be noted that the temperature of the exhaust gases after the turbine at full load was 760 deg. F. and at the three-quarter load 660 deg. F., which compare very favourably with the corresponding temperatures from a normally aspirated engine, so that there is just as much heat available for raising steam from the exhaust gases as from the normally aspirated engine of corresponding power. The air delivery from the blower rose to 5lb. per sq. in., at which its temperature approached 130 deg. F., and this was reduced by the intercooler down to 70 deg. F. It had been considered that the exhaust ports must open at least 28 degrees of crank angle before the air ports corresponding to a timing diagram and port opening diagram shown in Fig. 13, and with an 8 degrees lead on the crankshaft the exhaust ports remain open after the air ports have closed for 12 degrees of crank angle. During this period there is a loss of air from the engine cylinder. When running astern the exhaust ports are opened only 12 degrees before the air ports, so that it is not desirable to run astern for prolonged periods at very high loads, though this should not cause any inconvenience in operation, as high power astern is rarely required for more than a few minutes at a time. As the major portion of the work to raise the scavenging and charging air to the pressure required is derived from the energy of the exhaust gases, the mechanical efficiency of the supercharged engine is higher than that of the normal engine and mechanical efficiencies exceeding 90 per cent have been measured. This improvement in mechanical efficiency leads to a slight improvement in the fuel consumption of the order of 4 per cent.

FIRING ORDERS AND TURBINE ARRANGEMENTS

In the past the orders of firing of normally aspirated two-cycle engines have been determined by considerations of the balance of the engine and of the critical torsional vibration characteristics. In the opposed piston engine, the rotating forces and the primary forces of the upper and lower pistons can be equalized and balanced for each cylinder, and the orders of firing have been such that the out-of-balance secondary couples have been reduced to a minimum consistent with good torsional characteristics. With the supercharged engine, however, another feature must be taken into account—the satisfactory spacing of the exhaust impulses to the turbine and the piping arrangement.

The pipes from the cylinder to the turbine have to be as short and free from bends as possible to conserve the energy of the exhaust gases for use in the turbine. It has already been seen that the scavenging and charging of the cylinders occupies some 120 degrees of crank angle and, therefore, the spacing of the exhaust impulses into any one section of the turbine nozzles must be at least 120 degrees apart, to avoid interference. For these large engines the area of the turbine nozzles has to be adequate to prevent high back pressure of the exhaust impulses, so that there is little resistance in the cylinder when the air ports open for the scavenging and charging period. Provided that the impulses into any one section of nozzles are spaced 120 degrees apart, the area of the nozzle is the same whether it has to deal with the impulses from 1, 2 or 3 cylinders. It is this factor which governs the size, weight and cost of a turboblower.

POSSIBILITIES OF SUPERCHARGING BY TURBOBLOWER

The first three-cylinder Doxford supercharged engine was installed in a tanker, the *British Escort,* of 12,000 tons. Tankers of this size are normally fitted with four-cylinder engines of the same type, so that the three-cylinder engine doing the same work has to develop 33 per cent more power per cylinder.

The largest Doxford engine being built at the present time is of 750-mm. bore × 2,500-mm. stroke, capable of developing 1,500 h.p. per cylinder at 110 r.p.m. so that the six-cylinder engine will give 9,000 h.p. at a mean indicated pressure of 90lb. per sq. in. When the supercharging developments are applied to this largest engine, it will be possible to develop up to 13,000 s.h.p. with a supercharged six-cylinder engine, as shown in Fig. 14. The estimated weight of this engine is 570 tons. Three turboblowers are shown, each taking the exhaust gases from two cylinders only, though on smaller six-cylinder engines two blowers only may be used. 750-mm. bore engine which today is rated at 9,000 s.h.p. in six cylinders is 64 feet, but with the application of supercharging an engine of 650-mm. bore × 2,320-mm. stroke, running at 116 r.p.m. would give the same power. The weight of the engine would be about 420 tons as against 525 tons, with a reduction in length of 10 feet. The higher mechanical efficiency of the supercharged engine gives a reduction in fuel

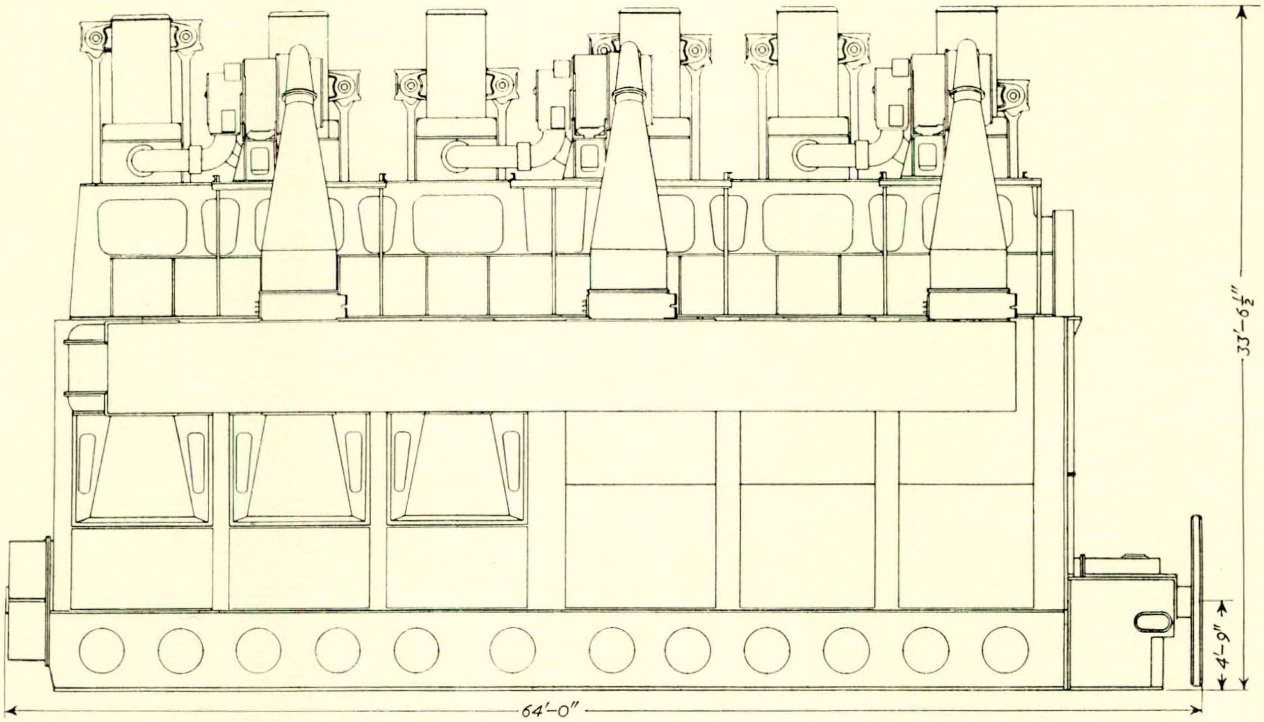


FIG. 14—*750 × 2,500-mm. engine supercharged by three turboblowers*

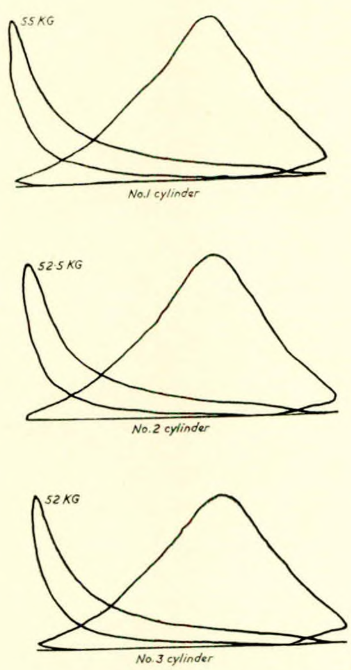


FIG. 14(a)—*Set of indicator cards*

Another advantage of supercharging is the reduction in engine room space, and a smaller engine of lighter weight can be offered to do the same work. The length of the largest

consumption of the order of 4 per cent, which on a vessel carrying bunkers for 40 days means a reduction of 55 tons in the weight of the fuel carried.

The first engine has now been at sea for nearly 18 months. The average speed of the ship has been half a knot faster than given by the normal four-cylinder engine, indicating that the power developed has been more than 30 per cent over that of the service power of the normal four-cylinder engine and the log has shown that there has been a reduction in fuel consumption of over one ton per day. There have been no difficulties with the engine or with the supercharging, except that on one occasion the air cooler between the blower and the engine driven scavenging pump was choked up with a fine oily dirt and a larger and more efficient air filter has been fitted.

The table below gives a set of readings taken from the sea trials and from a typical run of the vessels, and a set of indicator cards are reproduced (Fig. 14(a)).

	Sea trials	On voyage
Horsepower	3,580	3,150
Engine r.p.m.	120	115
Average m.i.p.	116	104
Exhaust temperature at the cylinders, deg. F.	780	695
Exhaust temperature after the turbine, deg. F.	740	660
Turbine r.p.m.	6,700	6,000
Maximum cylinder pressures, lb. per sq. in.	750	660
Entablature pressure, lb. per sq. in.	7·3	6·3
Pressure after the blower, lb. per sq. in.	4·6	3·8

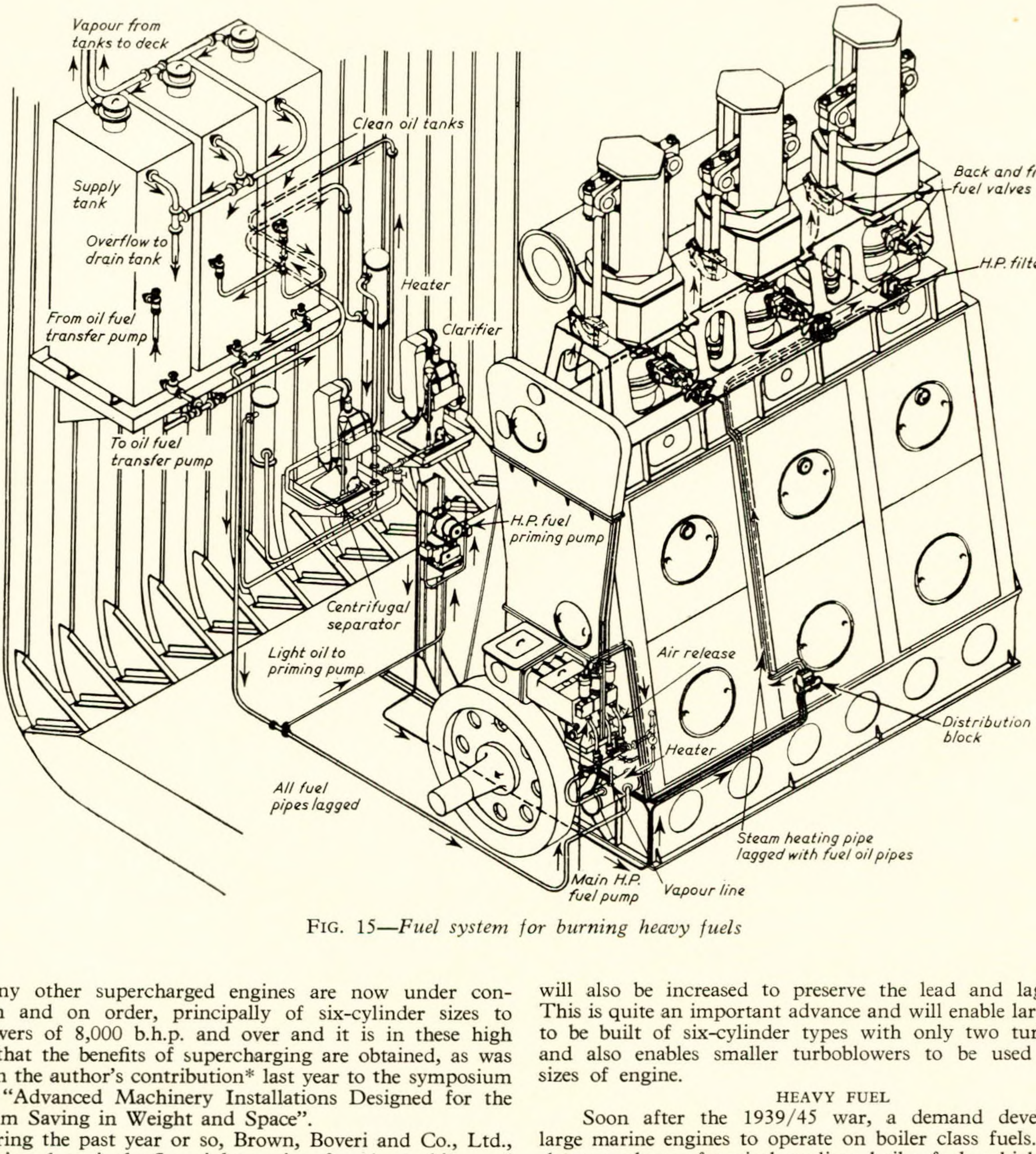


FIG. 15—*Fuel system for burning heavy fuels*

Many other supercharged engines are now under construction and on order, principally of six-cylinder sizes to give powers of 8,000 b.h.p. and over and it is in these high powers that the benefits of supercharging are obtained, as was shown in the author's contribution* last year to the symposium entitled "Advanced Machinery Installations Designed for the Maximum Saving in Weight and Space".

During the past year or so, Brown, Boveri and Co., Ltd., have designed a single flow inlet casing for the turbine end of their turboblower, which enables the three pipes from each half of a six-cylinder engine to be combined into this single inlet, and the arrangement is such that the gases enter tangentially on the periphery of the casing with a minimum of loss of energy, which has enabled the efficiency to be increased. With this arrangement and the more efficient turbine, it will be possible to have two turboblowers only on even the largest of the Doxford engines and develop up to 15,000 h.p. in six cylinders.

This single inlet turbine, however, neecssitates that the exhaust ports shall be opened somewhat earlier than with the previous two-inlet turbine. Referring to Fig. 13, which shows the timings for a two-inlet turbine, the opening and closing of the exhaust ports will be increased by 3 degrees for the single inlet turbine and the period of opening of the air ports will also be increased to preserve the lead and lag periods. This is quite an important advance and will enable large engines to be built of six-cylinder types with only two turboblowers and also enables smaller turboblowers to be used on some sizes of engine.

HEAVY FUEL

Soon after the 1939/45 war, a demand developed for large marine engines to operate on boiler class fuels. Table I shows analyses of typical medium boiler fuels which illustrate that they contain unburnable ash and water, and heavy asphaltenes which are not readily burnable and are prone to carbonize and cause deterioration in the operation of an engine. It was considered that if the water, ash and asphaltenes could be removed from the fuel it could probably be burnt satisfactorily in an engine, though it was feared that there might be increased wear of the liners and piston rings and increased sticking of the piston rings and of the fuel valve nozzles and needles. From the analyses of these medium boiler fuels it was appreciated that they contained more sulphur than was customary in normal Diesel fuels, and it was known that sulphur was a contributory cause of cylinder liner wear. In the early experimental work, arrangements were made to centrifuge the fuel in two stages, first at about 180 deg. F. and secondly at approaching 200 deg. F., with the object of removing any water and ash in the first stage and the heavier asphaltenes in the second stage. There was intermediate heating between the stages, steam heaters being employed. This

* Jackson, P. 1955. "Direct Drive Diesel Machinery (Doxford)". Trans.I.Mar.E., Vol. 67, p. 312.

dual centrifuging of heavy fuels was well known, having been developed by Commander Le Mesurier and first used in a British tanker built in 1928. Fig. 15 shows the fuel system which has been adopted on the Doxford engine for the burning of heavy fuels. Arrangements were made to heat the fuel prior to its entry into the fuel pump, and to heat the fuel pipes between the fuel pump and the fuel valves. With these arrangements, there were no difficulties in burning the heavy boiler fuel, with a perfectly clean exhaust and with a fuel consumption only 5 per cent higher than the fuel consumption with Diesel fuels. During prolonged runs it was observed that under certain conditions some of these heavy boiler fuels deposited carbonaceous sludge around the air ports and around the lower piston scraper rings. Subsequently, upwards of 200 ships have been built or converted to operate on boiler fuels, showing a saving per year of up to £3 per h.p. installed. A few engines, however, have had a difficulty which was not at first appreciated.

There have been cases of corrosion in the crankcase and crosshead bearings: crankpins and journals have been attacked. When this first occurred it was puzzling, but investigations showed that there was water in the lubricating oil and that this contained diluted sulphurous acid. From a fuel oil containing $3\frac{1}{2}$ per cent of sulphur, the sludge around the air ports and collected in the lower piston scraper ring box could contain as much as 8 per cent of sulphur, and some of this sludge was getting past the scraper rings and into the crankcase lubricating oil, where any leakage of water from the piston cooling gear or lubricating oil coolers could mix with it to form a dilute sulphuric acid. This was then circulated around the engine with the lubricating oil and attacked the journals and pins. Precautions were therefore taken to double centrifuge the lubricating oil to remove any water or sulphurous acid. Whenever it became apparent that the lubricating oil was acidic it was treated before further use in the engine. The treatment

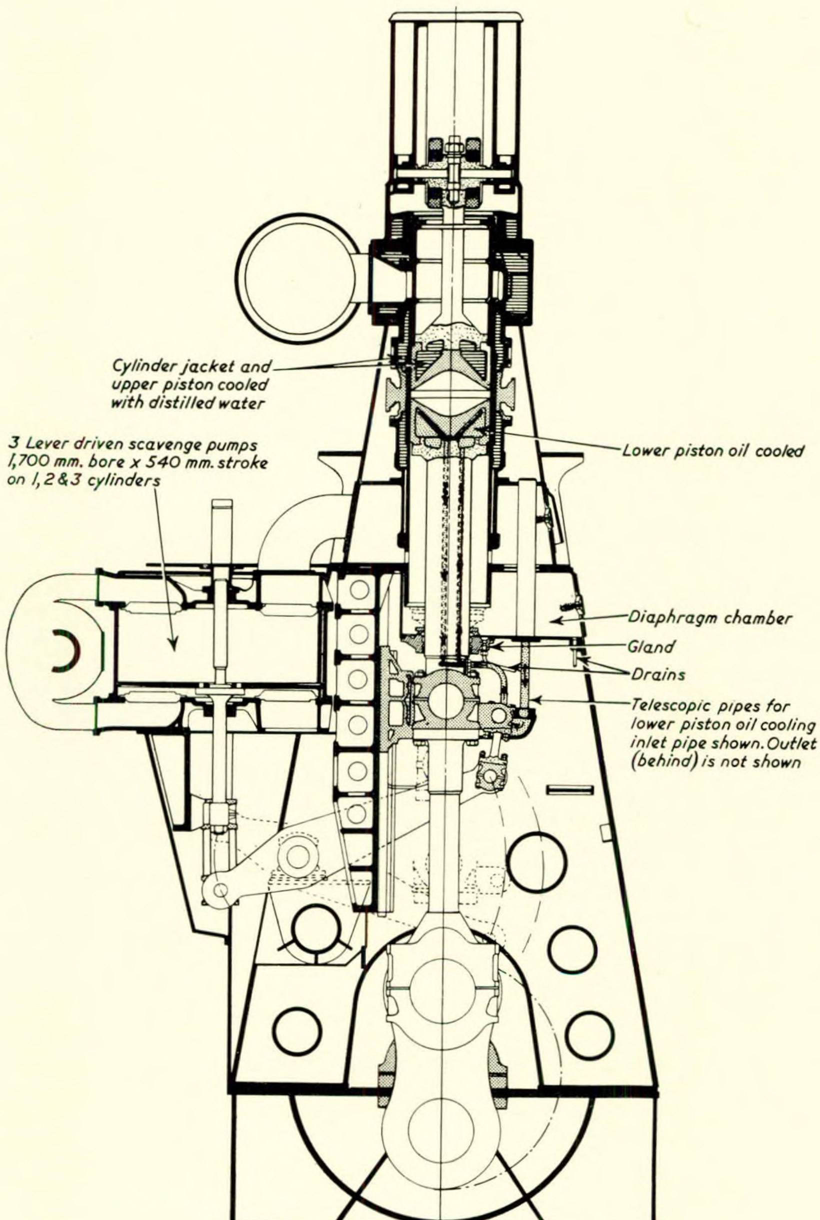


FIG. 16—*Doxford diaphragm engine*

TABLE I.—APPROXIMATE CHARACTERISTICS OF HEAVY FUELS

	Marine Diesel fuel	Admiralty fuel oil	1,500-sec. fuel oil	3,500-sec. fuel oil
Specific gravity at 60 deg. F.	0·88	0·94	0·96	0·98
Flash point, P.M., closed, deg. F.	185	180	200	200
Flash point, open, deg. F.	205	205	230	230
Flash point, fire, deg. F.	235	250	270	280
Conradson carbon, per cent	1·4	7	10	12
Asphaltenes, per cent	0·5	4	6	8
Viscosity, Redwood No. I at 60 deg. F., sec.	110	1,500	8,500	—
Viscosity, Redwood No. I at 100 deg. F., sec.	55	300	1,350	3,300
Viscosity, Redwood No. I at 140 deg. F., sec.	43	120	370	750
Viscosity, Redwood No. I at 180 deg. F., sec.	—	65	150	260
Calorific value, Btu. per lb.: gross	19,150	18,750	18,600	18,400
net	18,000	17,650	17,500	17,300
Calorific value, cals. per gm.: gross	10,630	10,420	10,330	10,220
net	10,000	9,800	9,720	9,610
Ultimate Analysis				
Carbon, per cent	85	84·5	84·5	84·5
Hydrogen, per cent	12	11·5	11	11
Sulphur, per cent	1·4	2	2·5	3
Ash, per cent	Below 0·01	0·04	0·07	0·08
Water, per cent	0·05	0·1	0·2	0·2
Approximate cetane number	38	33	30	26

consisted of water washing the lubricating oil to remove the sulphur and then centrifuging to remove the water, and it has been found that water can be separated from lubricating oil more easily if the oil be heated to 165 deg. F., instead of centrifuging it at its temperature in the engine sump of about 125 deg. F. The sludge is highly abrasive and the combined action of abrasion and corrosion had caused excessive wear and hence defective operation of the lower piston scraper rings.

It was considered that the engines should be modified so that both sludge and water were eliminated from the crankcase. There had previously been a few cases of corrosion on engines operating on Diesel fuel which were traced to the presence of salt water in the crankcase, this having entered through the lubricating oil coolers or through defective operation of the cooling water distillers. It was decided, therefore, that water should be eliminated from the crankcase, and experiments were carried out with oil cooling of the lower pistons. Because of the reduced specific heat of oil relative to water, it was necessary to pump some 2½ times as much oil through the piston heads for the same amount of cooling as was required with water cooling. In addition, to prevent the carbonization of the lubricating oil on the hot surfaces of the piston head, experience had shown that it was desirable to restrict the maximum temperature of the oil to 145 deg. F. Two methods of conveying the oil to the pistons have been tried, one with swinging links of large diameter and the other with telescopic pipes, and both are equally successful, the latter being somewhat lighter, and much simpler.

DIAPHRAGM ENGINE

To prevent the sludge and products of combustion penetrating the crankcase, the engine has been redesigned (Fig. 16) with a piston rod and gland box to separate entirely the open end of the cylinder and piston from the crankcase, and packing rings in the gland box prevent any products of combustion penetrating into the crankcase. The diaphragm space is accessible and can be cleaned out, or the sludge can be taken away through drain pipes. The whole space is illuminated and visible so that the surface of the rod and piston can be seen. It has been noticeable that when the engine is operating with clean combustion and with the fuel oil maintained at a temperature of 185 deg. F. for 3,000-second fuel, and with the jacket water maintained at 140 deg. F. inlet and 165 deg. F. on the outlet, there is little formation of sludge. On the other hand, if the jacket temperature be reduced below 110 deg. F. inlet, or the temperature of the fuel be reduced below 140 deg. F., considerable sludge can be formed in a short time, and this indicates that much of the difficulty with heavy fuel has been due to defective operation, particularly operation at low temperatures. The formation of sludge is also dependent on the fuel injection which can reduce the quantity of sludge from over 20lb. per day to under 2lb. on heavy fuels of 3,000-second viscosity.

The wear of cylinder liners and piston rings is accelerated

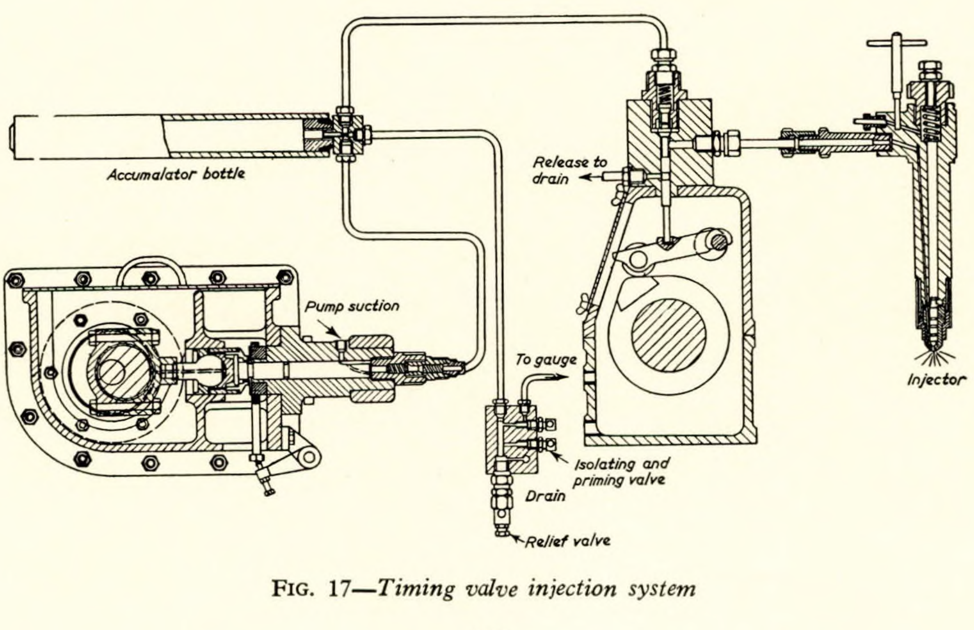


FIG. 17—*Timing valve injection system*

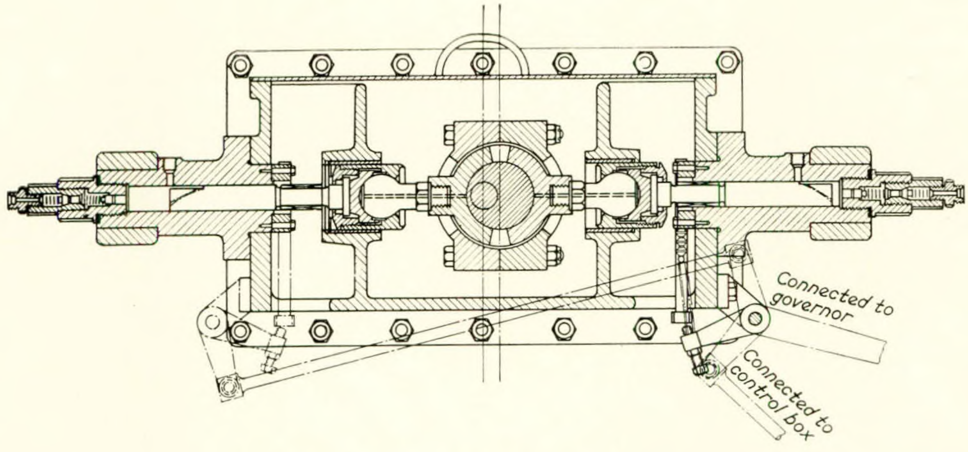


FIG. 18—*Fuel pump for six-cylinder engine*

by operating the engine on heavy fuel at low temperatures, or with late injection. Sulphurous sludge not only acts as an abrasive for scrubbing away the piston rings and cylinder liners, but chemical action also increases the rate of wear.

There are now a large number of diaphragm engines operating continuously at sea on boiler oils of up to 3,500-seconds viscosity and although there are occasional difficulties and a degree of increased wear, the savings in fuel costs outweigh these and it is considered that operation on boiler oils will soon become entirely reliable.

The diaphragm space adequately seals the crankcase from the gases and sludges of combustion, and with oil cooling of the lower pistons any leakage from the cooling system into the lubricating oil is immaterial.

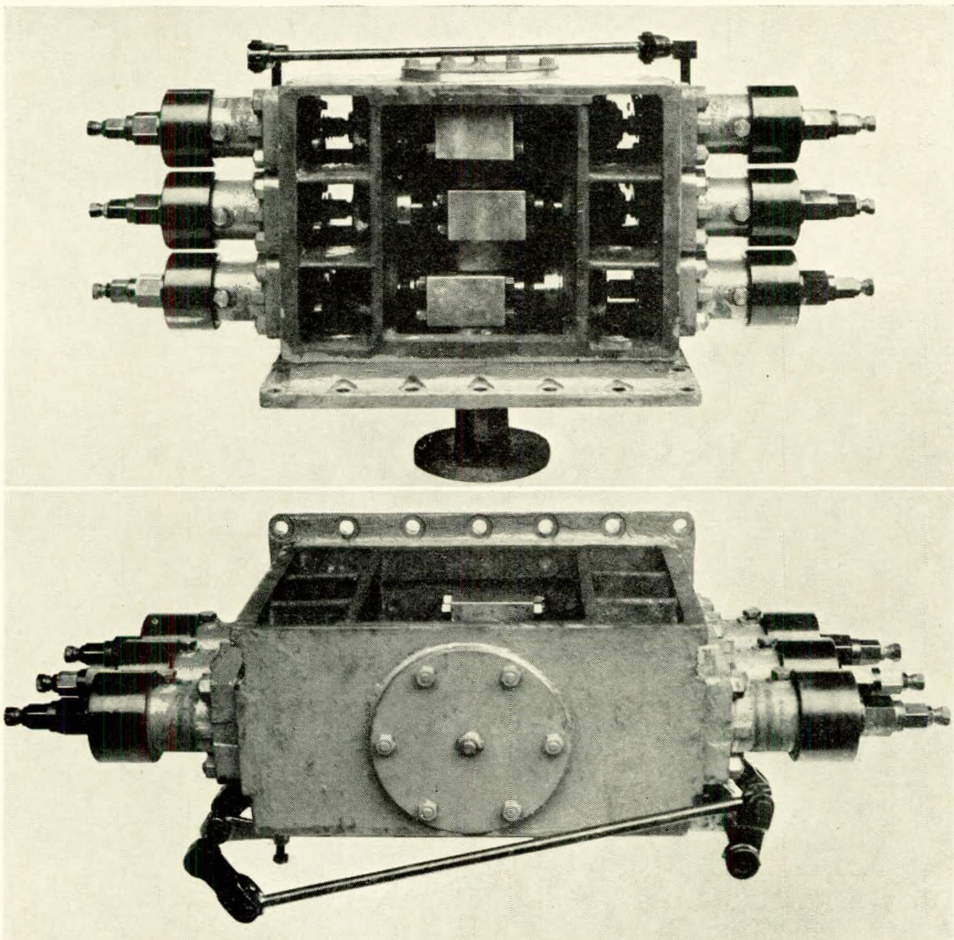


FIG. 19—*Photographic views of fuel pump for six-cylinder engine*

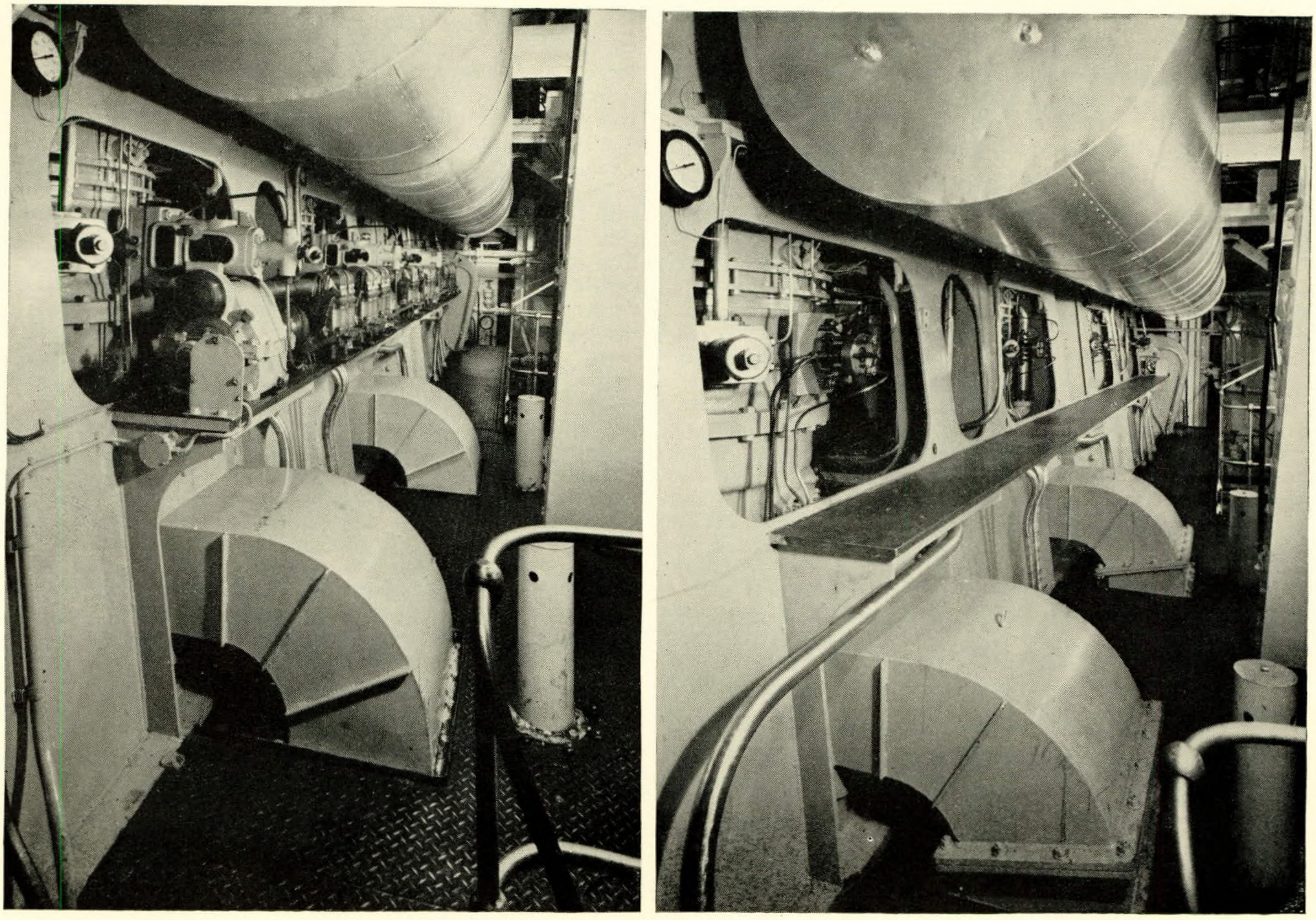


FIG. 20 (a) *and* (b)—*Views of back of engine*
(*a*) *with common rail system, and*
(*b*) *with timing valve injection system*

NEW FUEL INJECTION GEAR

Prior to 1953, the standard injection system in use on the Doxford engine was a common rail system, where a multi-throw pump driven by a small crankshaft charged up a number of accumulator bottles, one for each cylinder, and when the fuel valve was mechanically lifted by a cam on the camshaft the fuel in the bottles stored at a high pressure of 6,000 to 7,000lb. per sq. in. was discharged into the cylinder. This system has given satisfaction over a period of more than thirty years, with a good fuel consumption, and has been very reliable and quiet in operation. There was, however, some need for a simpler injection system and consideration was given to adopting the jerk pump system on the Doxford engine and a design was prepared for an engine cylinder of 1,000 h.p. A jerk pump raises the fuel to the injection pressure and injects it into the cylinder in some 25 degrees of crank angle which imposes a very heavy load on the operating gear and is often quite noisy in operation. A much larger camshaft would be necessary than was used for operating the common rail system of the Doxford engine and a camshaft of 6- to 8-in. diameter, together with a larger driving chain, would have meant a considerable modification to the standard engine. A jerk pump also necessitates heavy manœuvring gear for changing over the camshaft on to astern cams for astern operation, as is required by a marine engine. In view of these factors, it was decided to concentrate on an accumulator type of injection, where a slow-speed pump operated by crank or eccentric could be employed to raise the fuel to the injection pressure and discharge it over 120 degrees or more of crank angle, and where the timing and duration of injection was controlled by separate means and the fuel valve would be of the normal differential needle spring loaded type. The system eventually evolved is shown in Fig. 17, where a crank-driven pump discharges fuel into an accumulator bottle at 6,000 to 7,000lb. per sq. in. and a small timing valve operated by a cam is lifted to permit the fuel to be sprayed by the injector into the engine cylinder at the correct time and for the correct duration according to the load on the engine. A pump plunger, accumulator, and timing valve is required for each cylinder and on the Doxford opposed piston engine there are two fuel valves in the cylinder arranged opposite each other, both being supplied with fuel from the same timing valve. The quantity of fuel to be injected is determined by the fuel pump and there is a separate adjusting screw for regulating the quantity to each cylinder, and the timing and period of injection are controlled by the timing valve, the lever for operating this being mounted on an eccentric shaft. This system is relatively small and compact and operates very quietly. It has been found that the cam toe can be phased symmetrically relative to the top centre of the engine, so that the whole system operates equally in either the ahead or astern direction of running without manœuvring gear. It is also possible to operate on a common rail system when the fluctuations of pressure are reduced and the equalizing of the power between the cylinders is by the timing valve. There is no difference in the performance or fuel consumption of the engine with either method. Fig. 18 shows a view of the fuel pump of the six-cylinder engine which has three plungers on each side of a common crankshaft in *vis-à-vis* arrangement as shown in the photographic view (Fig. 19). Fig. 20 shows a comparison of the back views of the engine and the simplification is noticeable, since only one camshaft is necessary with the timing injection system and the manœuvring gear for change from ahead to astern is entirely eliminated. Many engines are now in operation and on order with the timing valve injection system.

AIR STARTING SYSTEM

Since the new fuel system can operate in either direction of rotation without manœuvring gear, a new starting system was developed, where the direction of running of the engine is determined by the starting lever. When this is pushed forward the engine will start ahead, and when it is pulled backward the engine will start astern. The system is shown in Fig. 21. The air valve in the cylinder is operated pneumatically when supplied with air from a rotary distributor driven by the camshaft, and connexions are provided around this distributor for supplying air to the engine cylinders in correct phase. When the starting lever is pushed forward the air is supplied to the top of the distributor, and the ports on the top section control the timing and order of supply of the air for running ahead, and when the air is supplied to the bottom of the distributor, the bottom section of ports control the timing and order of supply. The equipment required for the new timing valve injection system and new air starting system relative to that required by the common rail injection system and old starting system is shown in Fig. 22, where the simplification is clearly shown.

FUTURE DEVELOPMENTS

There is very little scope for improving the thermal efficiency of the large marine Diesel engine, apart from the 4-7 per cent advantage which can be gained by supercharging, owing to the high mechanical efficiency of the supercharged engine, and with intercooling of the air down to a low temperature an advantageous position of the thermal cycle can be obtained which will give a further improvement. The rating of large marine Diesel engines is relatively low in comparison with the rating of land engines and particularly in comparison with high duty engines for rail traction and naval purposes. The rating has been limited in the past owing to the properties of the materials which can be used and fabricated for the large parts employed in these large engines, and owing to the high thermal stresses which are imposed on the working parts. Doubtless the rating of these large marine engines will be gradually increased in pace with developments in materials, and with high pressure supercharging there are considerable possibilities for reducing the size and weight of large marine engines. This work is already showing promise on the Continent where four-cycle engines are being supercharged up to mean pressures of 300lb. per sq. in., and doubtless, as more experience is gained, the two-cycle engine will follow in this direction, but probably comparatively slowly. There are also considerable possibilities for simplifying the opposed piston engine from the mechanical point of view, in compactness of arrangement, and forms of construction, and this, together with the higher production and better methods which modern machinery and organization make possible, will enable the cost of marine Diesel engines to be stabilized and, maybe, even reduced. A prototype of a new Doxford engine embodying these principles is now undergoing construction and should result in a much shorter and lighter engine of simplified form.

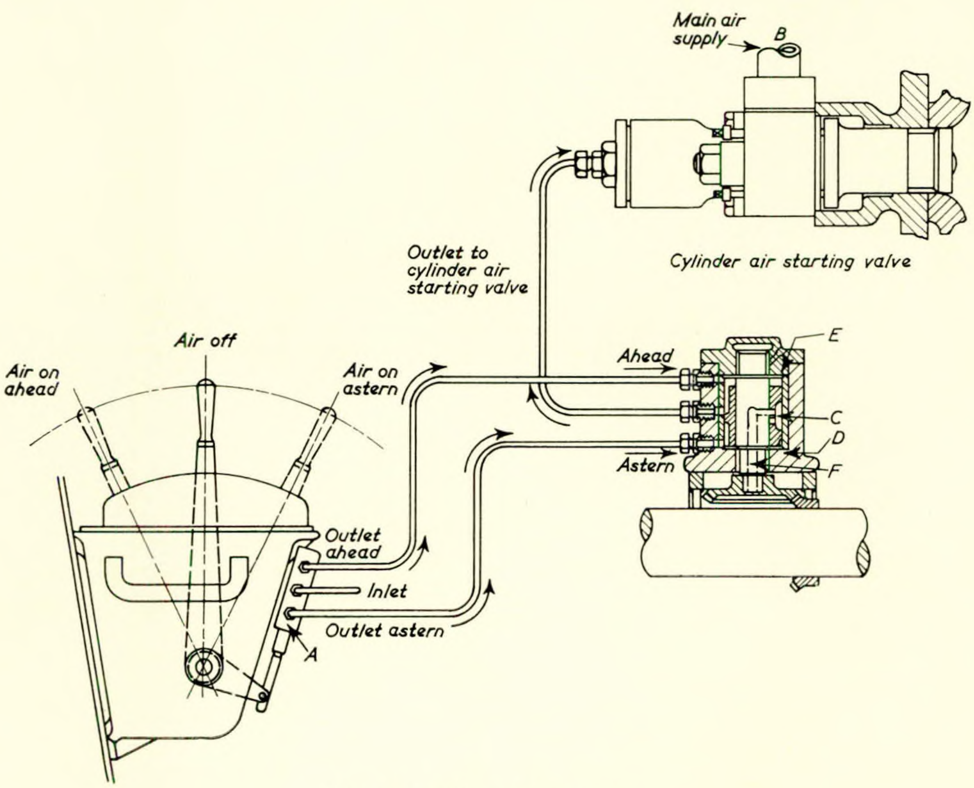
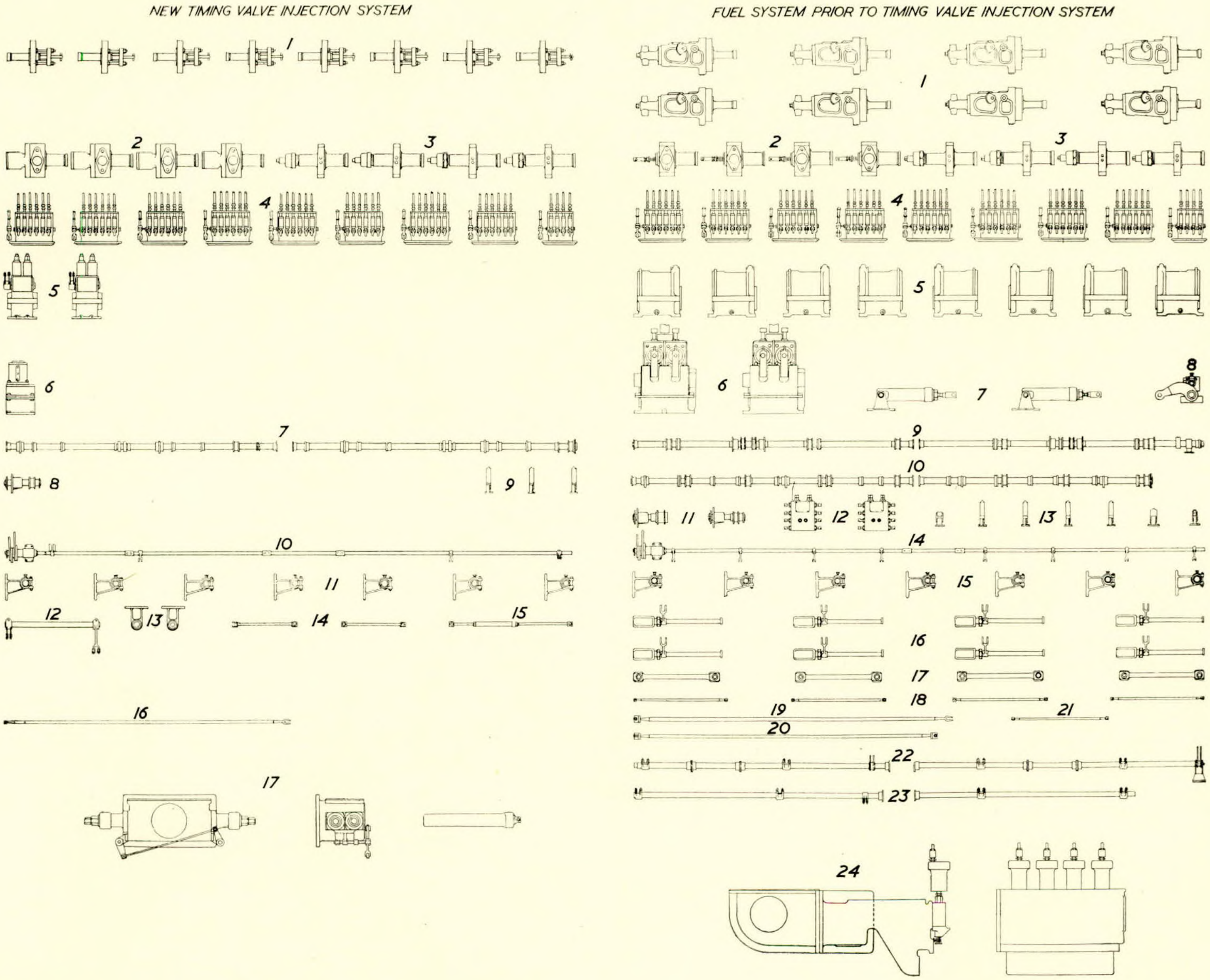


FIG. 21—*Air starting system*



(1) Fuel valves complete; (2) Non-return air starting valves; (3) Cylinder relief valves; (4) Cylinder lubricators; (5) Timing valves; (6) Air starting distributor; (7) Front camshaft and couplings; (8) Stub shaft and bearing; (9) Camshaft bearings; (10) Main control shaft; (11) Main control shaft bearings; (12) Control relay shaft; (13) Control relay shaft bearings; (14) Timing valve control links; (15) Connecting link relay to main control shaft; (16) Main control link; (17) Fuel pump and oil bottle.

(1) Fuel valve complete; (2) Non-return air starting valves; (3) Cylinder relief valves; (4) Cylinder lubricators; (5) Fuel cam casings; (6) Air starting valve blocks; (7) Air starting relay cylinders; (8) Reversing gear bracket; (9) Front camshaft and couplings; (10) Back camshaft and coupling; (11) Stub shaft and bearing; (12) Filter block; (13) Camshaft bearings; (14) Governor shaft; (15) Governor shaft bearings; (16) Fuel valve eccentric control shafts; (17) Front fuel valve governor links; (18) Back fuel valve governor links; (19) Main control link; (20) Reversing gear link; (21) Cross link between back and front manœuvring shafts; (22) Front manœuvring shaft; (23) Back manœuvring shaft; (24) Fuel pump.

FIG. 22—*Comparison of equipment for timing valve and common rail systems*

The difference in the number and size of the parts required for the two fuel systems for a four-cylinder engine are shown in the accompanying sketches and the advantages of the new fuel system over the older system are as follows:—

(1) First cost and maintenance of working parts is very much less due to the greatly reduced number of parts required.
(2) The front camshaft only being required and no reversing gear means that controls are simplified.
(3) Fuel valves are smaller and can be changed more quickly than the older type of valves.
(4) One small air distributor replaces two large air starting valve blocks and two air relay cylinders.
(5) Continuous operation can be either common rail or isolated.
(6) Individual adjustments to balance power developed by each cylinder can be made whilst the engine is running on either system.