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The Sulzer 2-Cycle Marine Engine.

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READ

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CHAIRMAN: MR. JAS. CARNAGHAN (Member of Council).

In a 2-cycle internal combustion engine the charging and exhaust strokes of the 4-cycle engine are eliminated, both these operations being performed while the piston is completing the power or firing stroke and commencing the compression stroke. The piston thus receives an impulse every revolution in the 2-cycle engine instead of every two revolutions as in the 4-cycle engine, giving a more even turning moment on the shaft, and provided that in each engine the piston speed and mean pressure remain the same, the power developed by a given size of cylinder will, in the case of the 2-cycle engine, be twice that developed in the 4-cycle engine.

It is evident that, without further consideration of the differences between the two types of engine, the 2-cycle appears to have enormous advantages, and yet we are faced with the fact that in many directions the 4-cycle engine is predominant. Practically all motor car engines, aero engines, and the great majority of petrol, paraffin and gas engines are 4-cycle engines operating on the well-known Otto cycle. In semi-Diesel and Diesel engines a more even balance is struck, the semi-Diesels being usually 2-stroke engines, while the Diesel engine is either 4-cycle or 2-cycle, but until fairly recently the former type has been more commonly adopted.

It is not unnatural that many engineers not immediately in touch with developments of the internal combustion engine are impressed with the position, occupied in these directions by the 4-cycle engine, and are liable to form incorrect opinions when considering the application of either type to a special field such as marine engineering.

The 2-cycle principle applied to petrol, paraffin or gas engines holds out very little hope of successfully competing with the 4-cycle engine. For various reasons it is impossible to obtain a high mean pressure, or to run at such high mean piston speeds in the 2-stroke petrol, paraffin, or gas engine as in a 4-cycle engine of the same type, with the result that full advantage cannot be taken of the saving in weight, etc., which would otherwise be possible in comparison with a 4-cycle engine. Scavenging is generally imperfect and since a mixture of fuel and air is employed for this purpose a portion of unburnt fuel inevitably escapes with the exhaust, and thereby increases the fuel consumption. Lubrication of the splash type commonly adopted in 4-cycle petrol engines is also unsatisfactory for 2-cycle engines and involves a heavy consumption of lubricating oil. The Diesel cycle of operation lends itself much more favourably to the 2-cycle principle, and for large land and marine engines the difficulties referred to do not arise with properly designed engines.

A comparison of the relative merits of the 2 or 4-cycle principle as applied to Diesel engines is necessarily inconclusive and very possibly quite misleading unless the actual designs of engines incorporating these principles are at the same time studied. It is unfortunately a very easy matter to design a wholly unsatisfactory Diesel engine either of 2-cycle or 4-cycle type, but it is obviously unfair to judge a principle upon applications of this kind. A few years ago an interesting article appeared in *Engineering* giving the experiences of an Italian firm who had constructed a 2-cycle and 4-cycle engine, and after an exhaustive test on both engines they claimed to have proved that the 2-cycle engine was the less satisfactory, and they accordingly determined to build only 4-cycle engines. Considering the actual engines under comparison their decision was unquestionably correct, as this particular 2-cycle engine was of a thoroughly bad design, with valve head scavenging and other features which rendered success impossible. Several other instances can be given where 2-cycle engines produced by inexperienced firms have failed, due to a combination of bad design and constructional defects. Full advantage has been taken by several advocates of the 4-cycle engine to represent such failures as a proof of inherent defects in the 2-cycle principle.

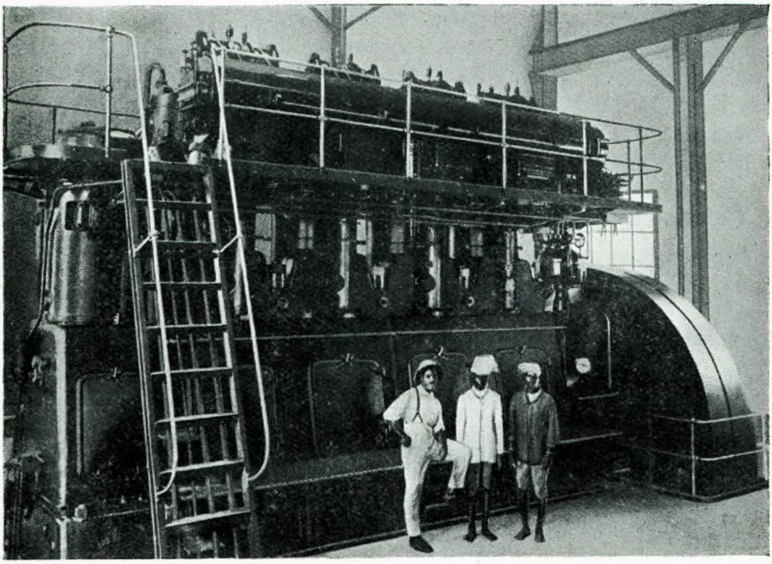
Sulzers have had a unique experience in the construction of Diesel engines owing to the fact that they have built large numbers of both 4-cycle and 2-cycle engines, and in fact still continue to do so. Reliability has perhaps been the dominating idea in their decision to discontinue the building of 4-cycle engines above powers of about 1,000 B.H.P. The main defect of the large 4-cycle engine, which they have found in common with other makers, has been the liability to heat cracks in the cylinder covers, and expensive upkeep of exhaust valves. Other important factors taken into consideration were the excessive weight, space required, and costly manufacture of the large 4-cycle engines.

Fig. 1 shows a 4-cycle engine of 1,000 H.P. made by Sulzers in 1914. The reliability of this was satisfactorily demonstrated in actual service, as during a period of two years the engine ran night and day for a total of 17,147 hours out of a possible 17,520 hours. The wear on the cylinder liners was only 10 to 12/1,000 inch, and the piston rings were in such excellent condition that none of them had to be removed.

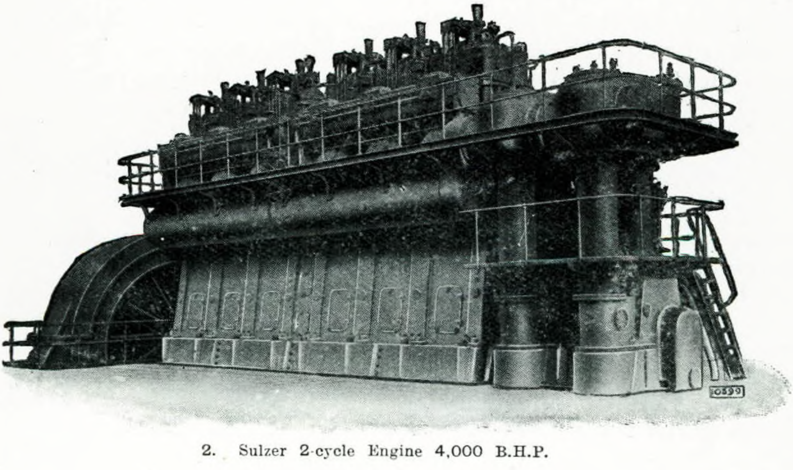
A total of 450,000 H.P. 4-cycle up to and including engines of this size having been manufactured by Sulzers, it will thus be seen that their experience of the 4-cycle engine is fairly complete.

Sulzer 2-cycle.—The Sulzer 2-cycle engine has been built in considerable numbers, and in varying powers up to 4,000 B.H.P., the latter being a six cylinder engine, thus developing a total of 666 B.H.P. or about 900 I.H.P. per cylinder. This engine is the largest Diesel engine in the world, and has been running for the past six years at Messrs. Harland and Wolff's Works, Belfast. The cylinders are 29 inches dia. × 40 inches stroke, and the revolutions are 132 per minute.

Scavenging.—Previous to the construction of this engine Sulzers had already manufactured a large number of 2-cycle



1. Sulzer 4-cycle Engine 1,000 B.H.P.



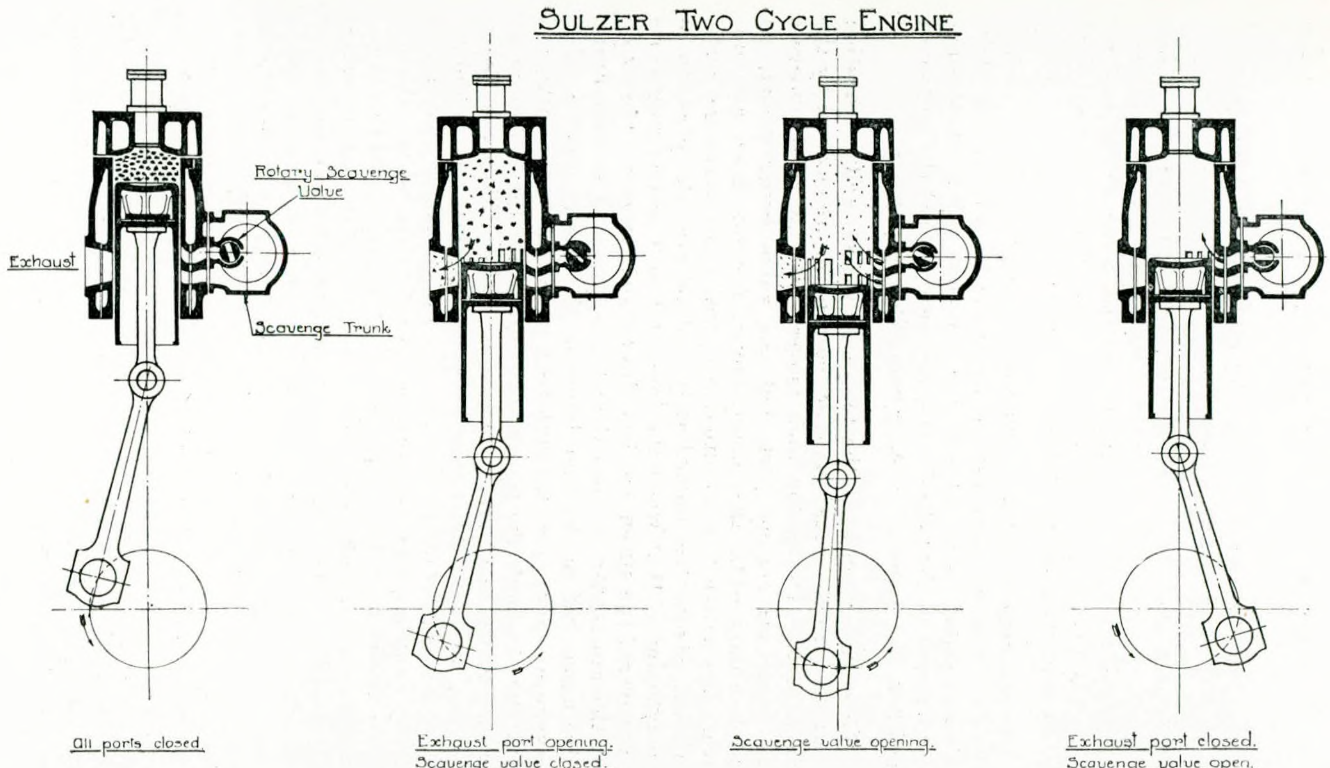
2. Sulzer 2-cycle Engine 4,000 B.H.P.

engines, and had from time to time eliminated various features which had shown signs of weakness.

The most important and distinctive features of the Sulzer design is the method of scavenging and recharging the cylinder, *i.e.*, removing the burnt products of the firing stroke, and replacing these with a charge of fresh air. This is accomplished in the following manner, Fig. 3.

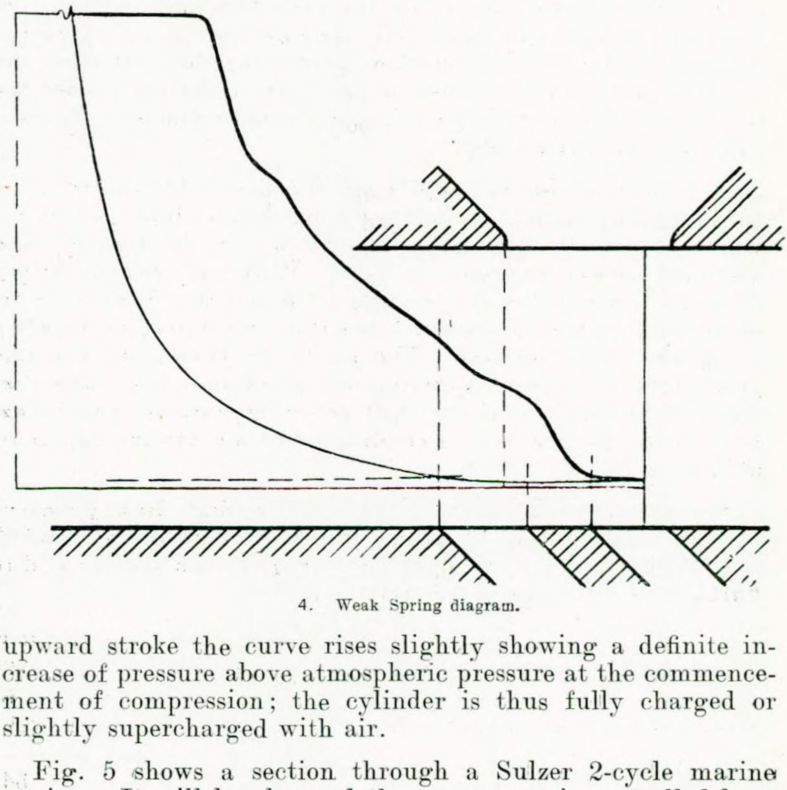
The scavenge air at a pressure of 1·5 to 2 lbs. per sq. inch is supplied by either a reciprocating or electrically driven turbo scavenge pump, and is led to the scavenge trunk which communicates with the cylinder through two ports at the bottom of the liner, the upper port being controlled by a valve.

When the piston is at the top of its stroke, and combustion takes place in the usual manner, both ports are covered by the piston skirt. The piston then moves down and first uncovers the upper scavenging port, but the valve being closed, no action occurs until the piston moves further down and uncovers the exhaust ports situated opposite the scavenge ports. Exhaust commences immediately and relieves the pressure in the cylinder so that when the lower range of scavenging ports is uncovered the air enters the cylinder and commences to sweep out the remainder of the exhaust products. The scavenging valve opens before the completion of the down stroke, so that scavenging takes place through both upper and lower ports, and continues through the upper port until after the piston on its upward stroke has covered the exhaust port. This arrangement not only ensures a most effective means of clearing out the exhaust products, but it also provides for an excess quantity of air by enabling the compression stroke to commence slightly above atmospheric pressure. In other words, supercharging is possible, and actual experience has shown that this important feature of the Sulzer engine has a marked effect on the output and economical working of the engine. Fig. 4 is a weak spring diagram showing the action occurring during scavenging, the exhaust and scavenge ports being indicated above and below the diagram. It will be seen that as soon as the piston uncovers the upper scavenge port there is a slight drop in pressure owing to the gases filling this port as far as the rotary valve which is of course shut. As soon as the piston uncovers the exhaust port the pressure drops so rapidly owing to the large exhaust area that when the lower scavenge port is uncovered atmospheric pressure has been reached. Scavenging now continues through



3. Scavenge System Sulzer 2-cycle Engine.

both ports, the rotary valve having, by this time, opened, and it will be observed that after the exhaust port has closed on the



4. Weak Spring diagram.

upward stroke the curve rises slightly showing a definite increase of pressure above atmospheric pressure at the commencement of compression; the cylinder is thus fully charged or slightly supercharged with air.

Fig. 5 shows a section through a Sulzer 2-cycle marine engine. It will be observed the upper port is controlled by a simple rotary sleeve valve, which rotates at half the crank shaft speed. This valve is situated where the conditions of both temperature and pressure allow it to work under very favourable conditions. The valve is simply a cast iron sleeve which rotates slowly in a cast iron cylindrical valve casing. Ample clearance is provided since only low pressures have to be dealt with, and a slight leakage is of no consequence. Figs. 6, 7, 8.

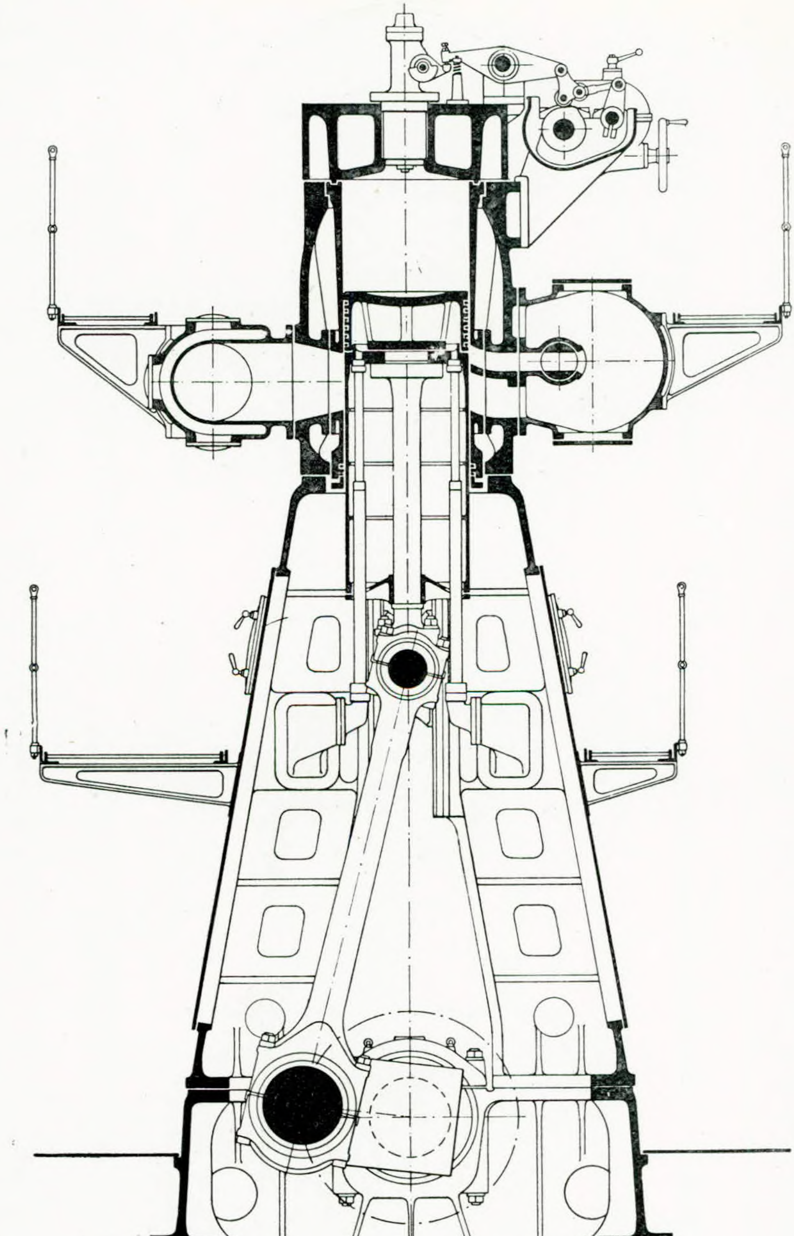
In marine engines of over 1,000 H.P., Sulzers provide electrically driven scavenge pumps in preference to the direct driven scavenge pump, and several important advantages are thereby obtained. See figs. 9, 10, 11.

Direct driven scavenge pumps absorb about 6% of the B.H.P., and by using separate electrically driven pumps with power supplied from the auxiliary engines more power is available, with a given engine, on the propeller shaft without increasing the speed or mean indicated pressure. This increase in power is thus obtained without increasing the stresses on the working parts, and does not involve heavier shafting, since the design of these parts is based upon cylinder dimensions, indicated pressure and speed.

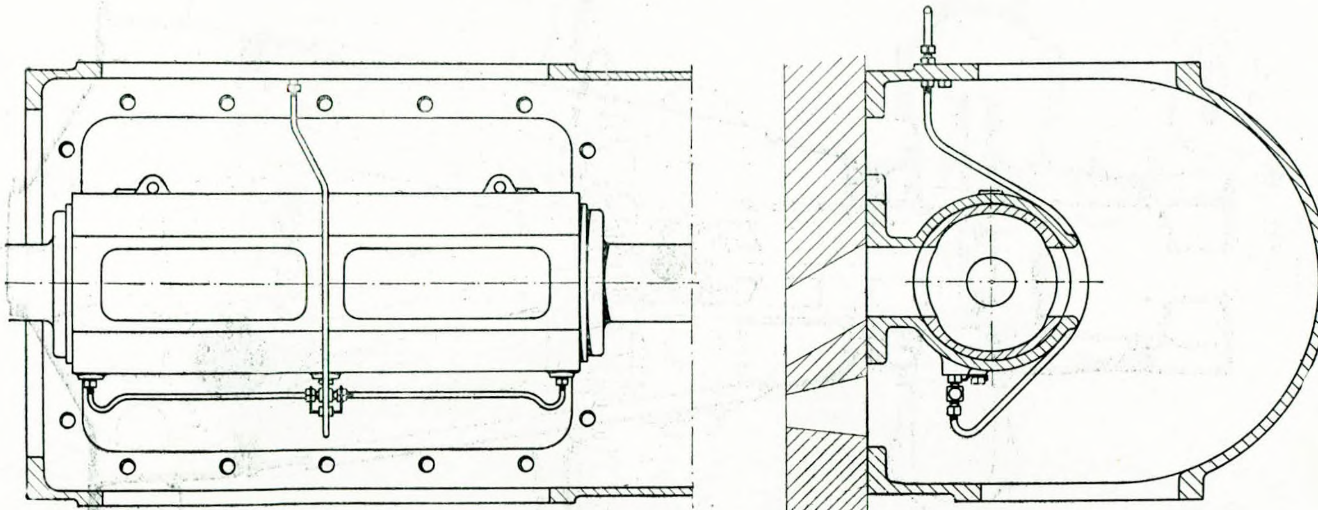
In a marine engine installation the power for the separate turbo scavenge pumps is obtained from the auxiliary generators which require to be of ample size to provide for lighting and auxiliary power purposes in port. With electrically driven deck machinery, it will generally be found that the power required in port will not be much less than is required at sea when using the turbo blowers. The auxiliary Diesel engines and generators have thus approximately a constant load factor instead of having to work at full power in harbour and at extremely low loads at sea—a condition which is always unfavourable to a Diesel engine.

For instance, considering a large intermediate liner for cargo and passenger trade of about 10,000 tons D.W. and 6,000 S.H.P., the average auxiliary power requirements at sea and in harbour would be approximately as follows:—

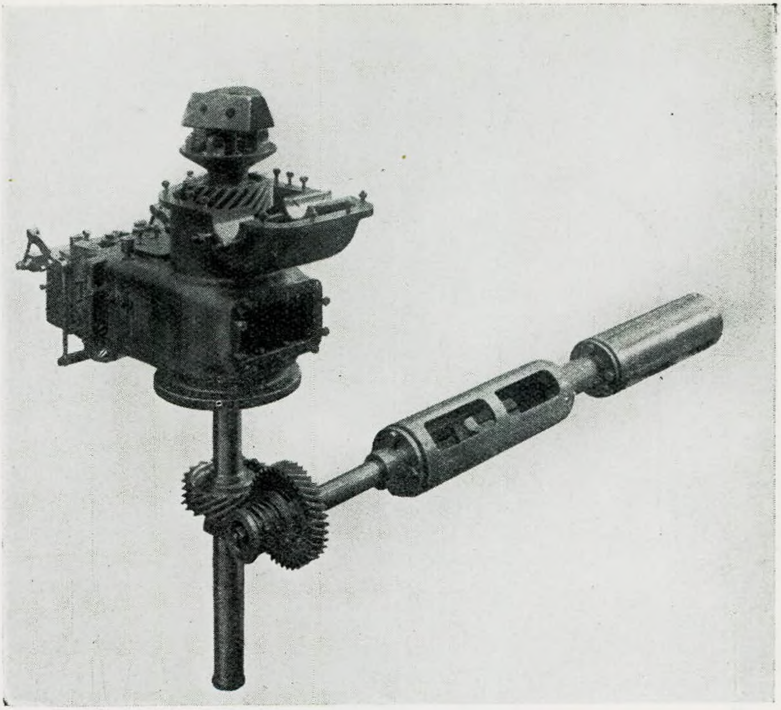
At Sea.		B.H.P.
For turbo blowers and cooling water pumps		380
Lighting		60
Steering and other auxiliaries		100
		540
In Harbour.		
Winches 12 at 25 H.P. = ...	300	
4 at 35 H.P. = ...	140	
	440	
Assuming approx. 2/3 winches in operation		300
Lighting		50
Other auxiliary purposes, pumping, etc.		60
		410



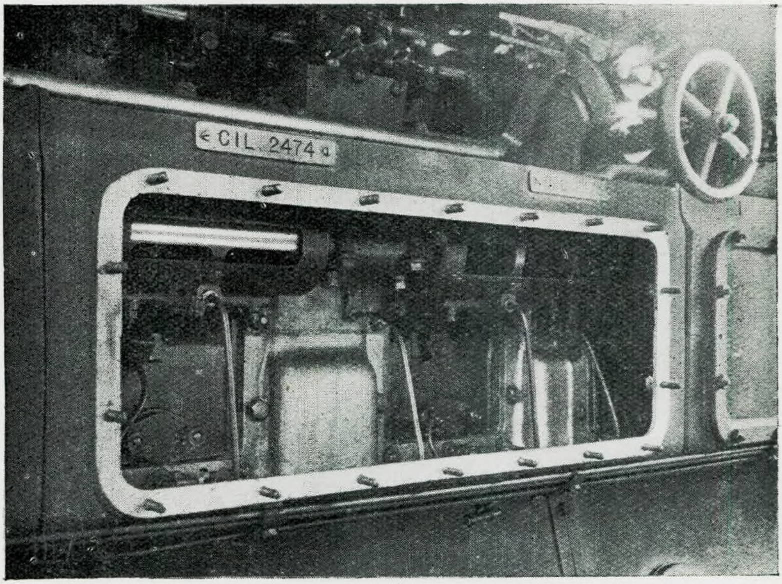
5. Section through Marine Engine.



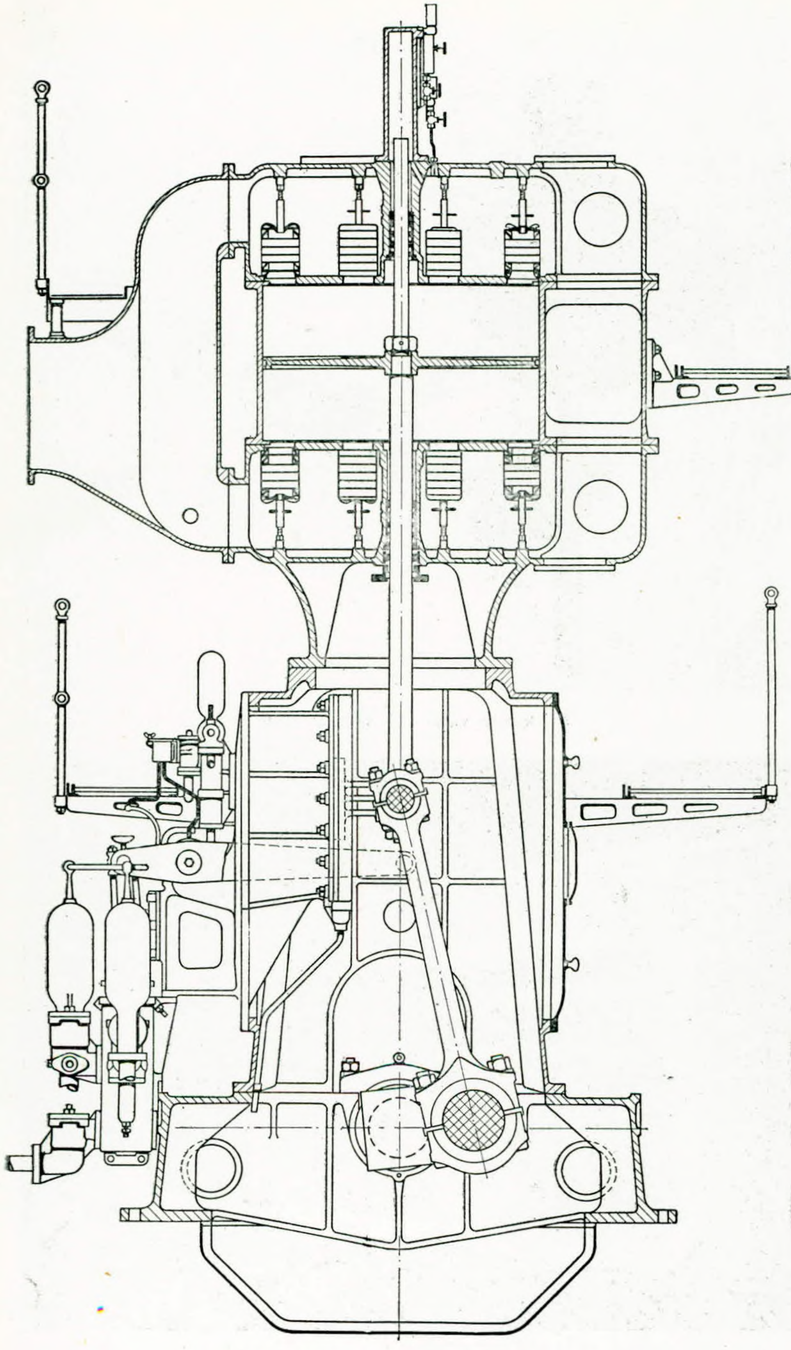
6. Section through Rotary Valve.



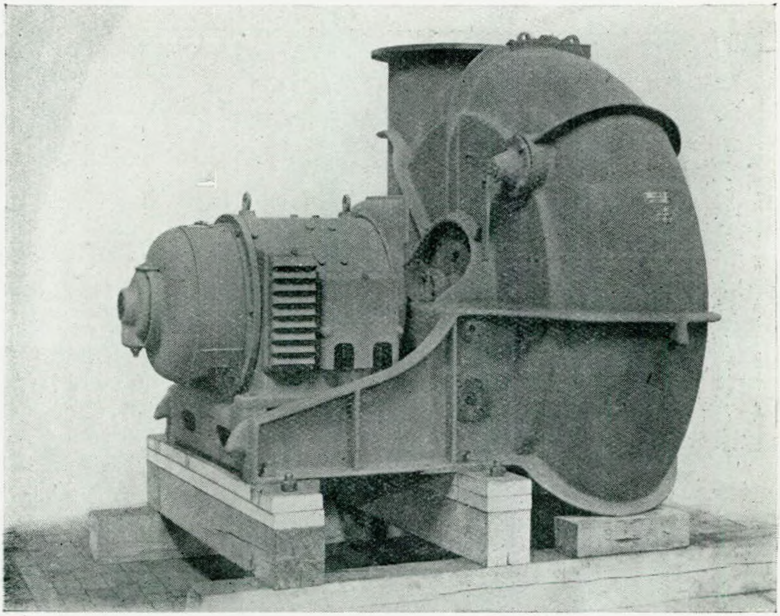
7 Rotary Valve and Vertical Shaft.



8. Scavenge Trunk and Rotary Valve.



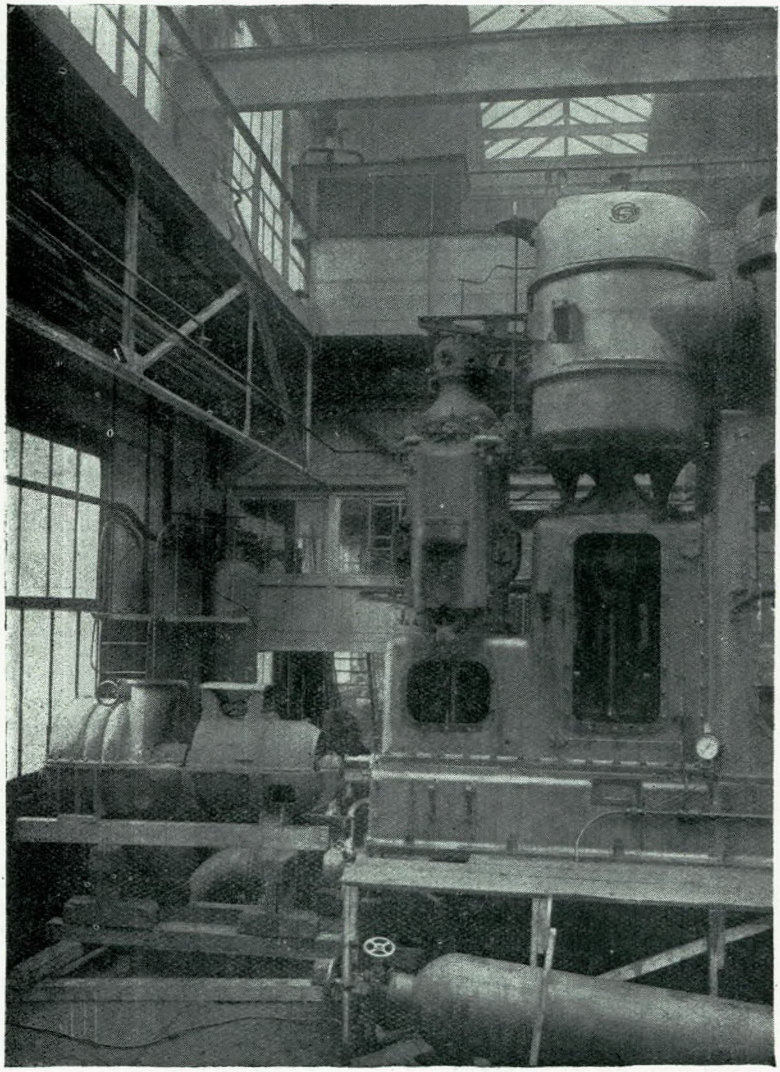
9. Section through Reciprocating Scavenge Pump.



10. Turbo Blower for Motorship "Handicap."

Assuming three auxiliary generators are fitted, one being a complete stand-by, then three engines each of about 275 to 300 H.P. will be required and will under above conditions be generally running at 3/4 to full load. In other cases it will be found that a still more even balance is struck between the normal load at sea and in harbour, and the best possible use is thus made of the auxiliary Diesel engines and generators.

Further advantages of the separate turbo scavenge are that for a given power the size and weight of main engines and of the complete machinery installation are reduced, even taking into account that two complete turbo pumps and motors are supplied, one of which forms a complete stand-by. The turbo blowers can be run at the speed and output which is found most suitable for the load on the main engines. Starting and manœuvring is improved by supplying the slightly warmed scavenge air in sufficient quantity to completely sweep out the cold starting air, and thus commence compression with a temperature which will ensure a sufficiently high final temperature to secure proper combustion.



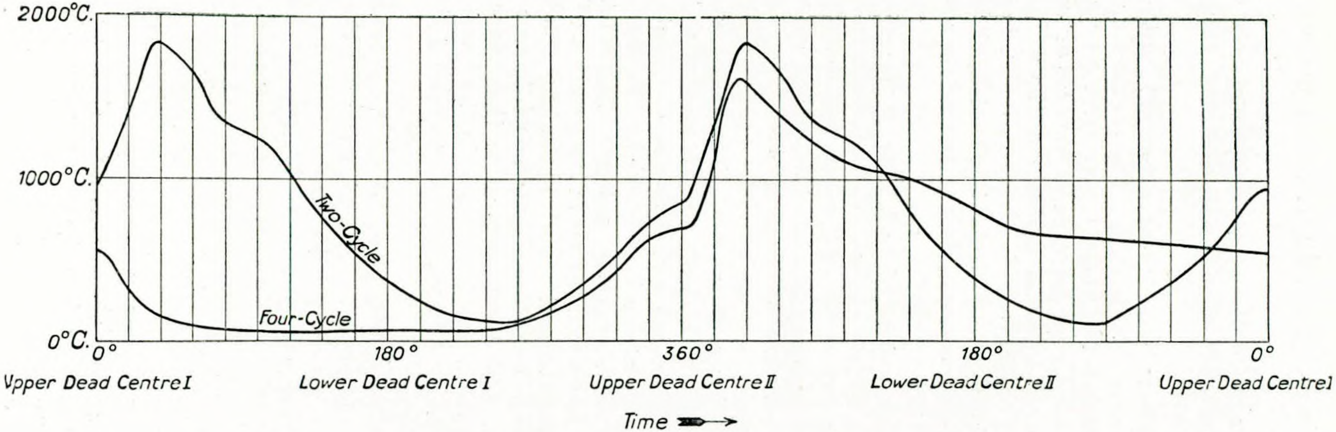
11. Turbo Blower and Reciprocating Scavenge Pump.

If, however, for any reason, such as in a ship fitted with steam deck machinery, it is not desirable to provide electrically driven turbo pumps, then the scavenge pumps may either be driven by steam turbines or else the ordinary reciprocating pump can be employed.

Heat Stresses.—The cylinder cover is the next most important feature to which attention is drawn. The combustion space in a Diesel engine is bounded by the cylinder cover, piston and the upper portion of the cylinder liner, or in the case of an opposed piston engine by two pistons and a portion of the cylinder liner. The temperature of combustion is about 2,500°F. to 3,000°F. and falls during the working stroke to about 1,200°F., these figures varying, of course, according to the relative weight of the fuel and air charge.

Assuming the same weight of fuel is burnt per firing stroke in a 2-cycle and in a 4-cycle engine, the 2-cycle engine thus in a given time burning twice the amount of fuel and developing twice the amount of power as compared with the 4-cycle engine, the maximum temperature obtained in each case will be approximately the same, the 2-cycle being slightly higher owing to a higher compression temperature. The mean temperature of the complete cycle will, however, be approximately 25% higher in the 2-cycle engine owing to the fact that in the 4-cycle engine the temperature is reduced by the lower average temperature obtained during the exhaust and inlet strokes. Roughly it may be taken that the mean temperature of a 4-cycle engine working at approximate full load is about 900°F., while the mean temperature of the 2-cycle engine while burning twice the amount of fuel and developing twice the power is about 1,100°F. Fig. 12 shows the fluctuation in the temperature of gases in a 2-cycle and 4-cycle engine when working at the same indicated mean pressure.

The heat flow through the material of the cylinder liners, piston and cylinder head, gives rise to stresses in the material which become more serious as the size of cylinder increases. The cylinder cover has always proved the greatest difficulty as the casting is usually the most complicated and at the same time is subjected to the maximum temperature and pressure conditions. The cylinder cover and piston are together responsible for over 80% of total heat flow to the cooling medium, the remainder of the heat being carried away through the cylinder liner. Although the water or oil-cooled piston of a large engine carries away at least as much heat as the cover, the casting is much smaller than the cover casting, and freedom for expansion is more easily obtained. The liner of the normal



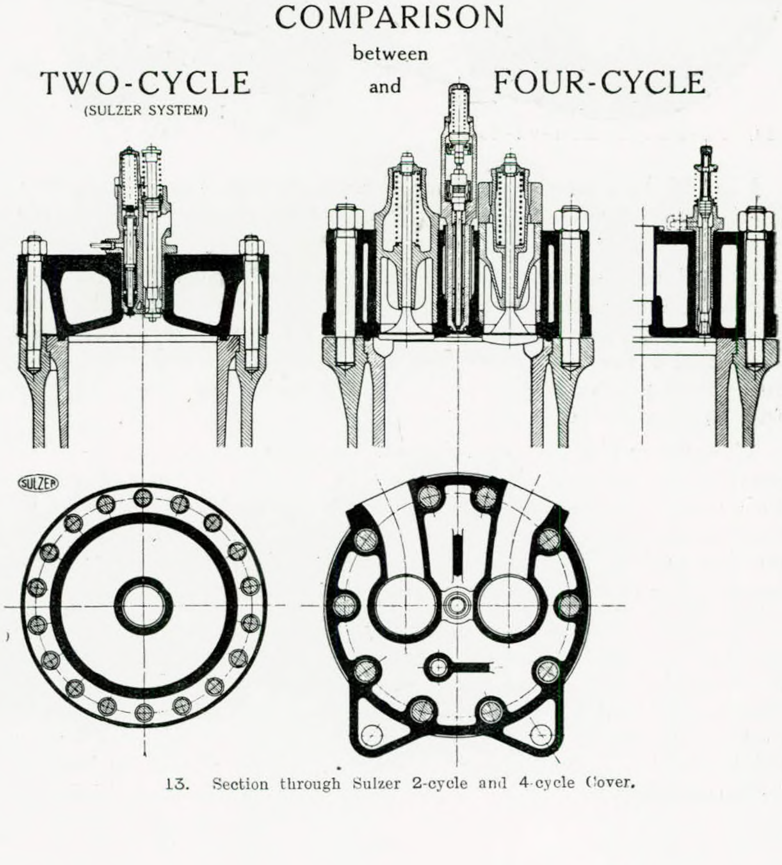
Mean Indicated Pressure: Four-Cycle M.I.P. = 99.7 lb./sq.inch.
Two-Cycle M.I.P. = 98.4 lb./sq.inch.

12 Temp. diagram 2-cycle and 4-cycle.

type of engine is also a less difficult problem than the cover as it is a perfectly symmetrical casting at the region of the combustion chamber, where the maximum temperatures are encountered. This remark, however, does not apply to liners of the opposed piston type of engine where holes have to be cut in the combustion space, as in this case very serious stresses are introduced, and several cases of cracked lines have occurred.

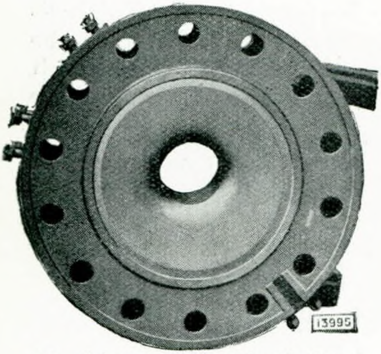
The freedom from failure due to heat stress of the Sulzer 2-cycle engine is primarily attributable to the design of the cylinder cover which was evolved about six years ago as a result of careful investigation of the causes of failure in both 2-cycle and 4-cycle cylinder covers.

Fig. 13 shows section through a Sulzer cover and a 4-cycle cover of normal design.

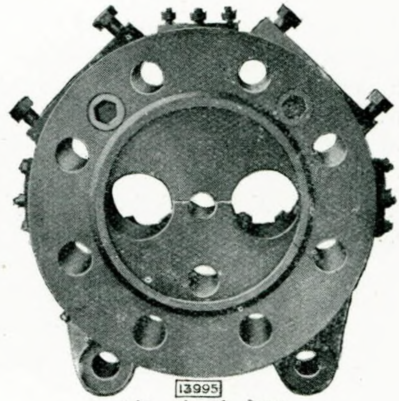


13. Section through Sulzer 2-cycle and 4-cycle Cover.

Figs. 14 and 14a are photos of Sulzer 2-cycle cover and a 4-cycle cover. It will be seen that the 4-cycle cover is cracked between the fuel valve opening and the valve openings on either side, this being a common source of weakness in 4-cycle covers of large engines.



14. Sulzer 2-cycle and 4-cycle Cover.



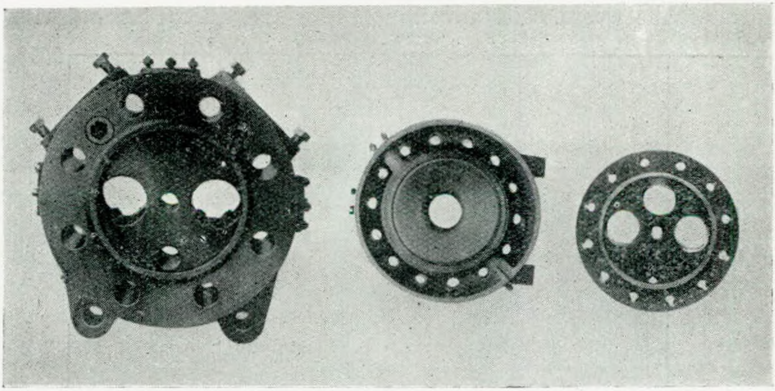
14A 4-cycle Cover.

Fig. 15 is a photo showing three different types of cover: (1) a normal 4-cycle design; (2) Sulzer 2-cycle cover; (3) 2-cycle cover with valve scavenge openings.

It will be observed from Fig. 13 that the Sulzer cover is a perfectly simple and symmetrical casting with a single hole in the centre of the combustion space. In this hole is placed a combined valve box containing the fuel and air starting valves and small passages are in addition provided for a relief valve and the indicator connection.

The under side of the cylinder cover only extends over the combustion space to the pressure joint between liner and cover. This is an important feature to provide against the inequality of expansion which must occur between the highly heated central portion and outer portion of the underside, if extended, as is usually the case, to the outer edge of the cover.

The simplicity of the casting ensures soundness in manufacture and enables a comparatively thin wall of metal to be provided between the combustion chamber and cooling water. The rate of heat flow being dependent not only on the difference in temperature between each side of the wall, but also on its thickness, the same temperature difference enables more heat to be transferred, according as the wall is diminished in thickness without increase of heat stress. Comparisons have not infrequently been made showing the heat flow per unit surface in various types of engines and it has been suggested that because the heat flow is greater in a 2-cycle engine than in a 4-cycle engine troubles due to heat stresses are more liable to occur. This is, of course, a perfectly unjustifiable argument, as it takes no account of wall thickness which is much less in the 2-cycle engine, seeing that for a given output the cylinder dimensions are considerably smaller. For instance, a much higher stress may be obtained in a normal design of 4-cycle cylinder cover than in the Sulzer 2-cycle cover with its thin walls even though in the latter case the heat flow per unit area is much greater than in the 4-cycle cover.

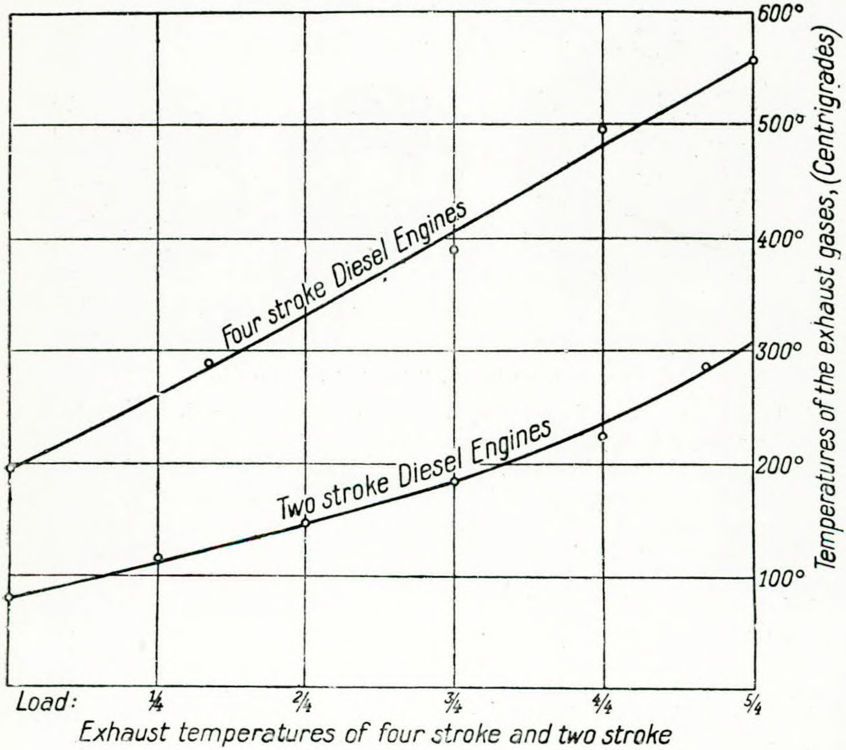


15. 4-cycle, Sulzer 2-cycle, Valve Head Scavenge Covers.

The pistons and liners are, as already stated, much less liable to trouble. The Sulzer liner follows generally adopted practice as regards the combustion chamber portion, while at the bottom of the liner exhaust ports and a double row of scavenge ports are provided. The bars in the exhaust ports are water cooled, the passages being perfectly straight and easy to clean if required. The temperature at the bottom of the cylinder liner being very much less than in the combustion chamber and moreover not being subject to more than the exhaust pressure, there need not be the slightest anxiety in regard to this part of the engine. The mean temperature of the exhaust in a Sulzer 2-cycle engine running at full power is only 450° F., while the actual temperature of metal in the bars of the exhaust ports is only 300° F., that is, little more than the temp. to which many C.I. parts of a steam installation are subjected. It, therefore, seems a little absurd to suggest—as has been done—that the Sulzer design simply transfers the heat stress troubles from the cylinder cover to the liner.

Fig. 16 shows the exhaust temp. of a Sulzer 2-cycle and a 4-cycle engine running at various loads.

An investigation of the actual stresses caused in the walls of the combustion chamber of an I.C. engine is an exceedingly difficult matter, and it is very doubtful whether there is at the present moment sufficient experimental data on which to base reliable calculations. Heat stresses are caused not only on account of the mean temp. drop between the inner and outer walls, but they are considerably affected by the cyclic variations in temperature occurring throughout the various operations of the cycle. In addition, there are, in all probability



Exhaust temperatures of four stroke and two stroke Diesel Engines.

16. Exhaust Temperature, 2-cycle, 4 cycle.

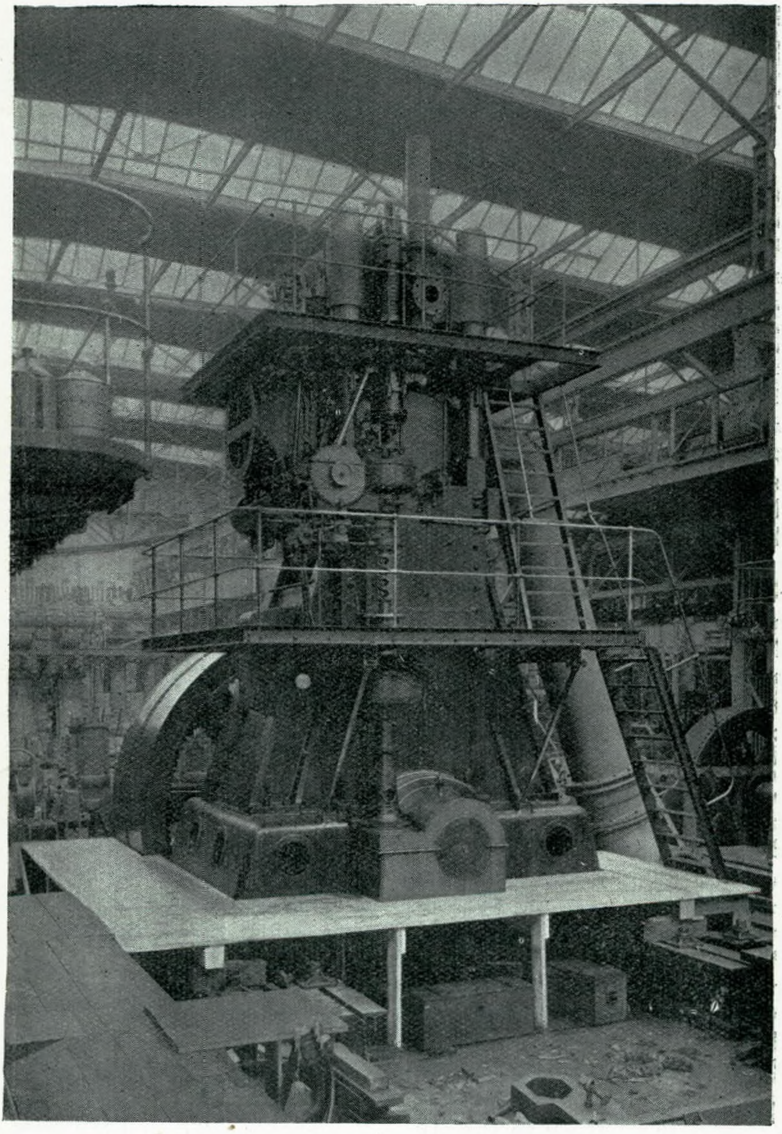
initial casting stresses, particularly in complicated castings such as 4-cycle cover—which cannot be assessed. However elaborate, therefore, theoretical calculations may be, the results are not convincing unless backed up by actual practical experience. Sulzer's experience in this direction has clearly shown that the 2-cycle engine is much less susceptible to heat stress troubles than the 4-cycle engine.

Fig. 17 shows an experimental single cylinder 2-cycle engine constructed by Sulzers in 1914. The cylinder dia. is 39·4 inches and the stroke 43·5 inches and slightly over 2,000 B.H.P. was developed when running at about 150 revolutions per min. Exhaustive tests carried out on this engine during a period of 2 years during which several six hours trials were made at a mean pressure of over 120 lbs. per sq. inch failed to reveal any defects due to heat stress in either the cylinder liner, cover or piston.

Another interesting experiment was carried out on a 2-cycle submarine engine cylinder to discover, by means of running the engine continuously under a very high pressure, the first source of failure. The dimensions of the cylinder were $21\frac{1}{4}$ inches diameter and 19·6 inches stroke, the designed output being 400 B.H.P. at 310 revolutions per min. A continuous full power trial of 10 days duration was carried out on tar oil and this was succeeded by a $3\frac{1}{2}$ hours trial at over 50% overload and at an indicated mean pressure of 124 lbs. per sq. inch. No defect of any kind whatsoever was to be found in cylinders, pistons or valves, and the only apparent evidence of this excessive overload was in the exhaust pipe which was uncooled and become red hot during the overload trial.

Tests such as these, valuable though they are, are still less conclusive than results of actual running in service. It is, therefore, interesting to note that there are at least 1,500 Sulzer 2-cycle cylinders in service, and of these over 400 cylinders are above 20in. dia. The mean indicated pressure is generally about 100 lbs./ins. and the power per cylinder in several cases is over 750 I.H.P. If there was some defect inherent in the design, as for instance, if heat stresses had not been taken properly into account there would surely be a considerable percentage of failures due to cracked liners, pistons or covers. The fact that this has not been the case is the clearest proof that the design is sound, both in theory and practice.

Another point of advantage associated with the Sulzer 2-cycle cover is that owing to the slightly concave shape of the under side. taken in conjunction with a similarly concave shaped piston head a well shaped combustion space is obtained, contributing to efficient burning of the fuel.



17. Single Cylinder Sulzer Engine, 2,000 B.H.P.

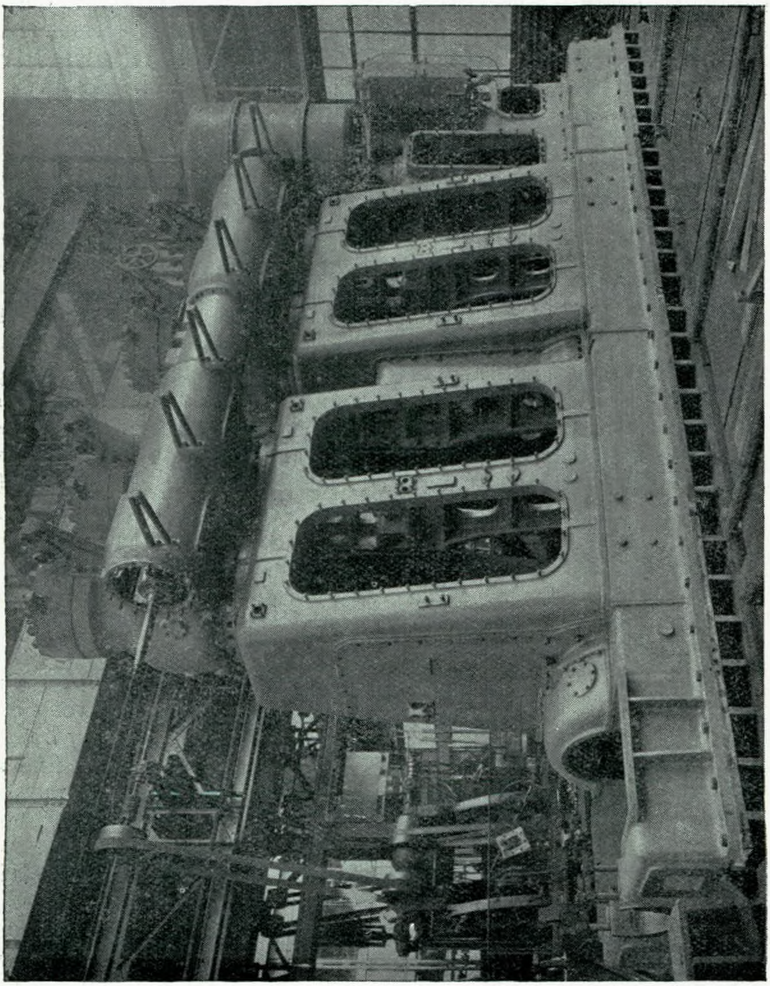
Mechanical Features.—The design of various details in the Sulzer Engine have been evolved as the result of long experience. Bearing pressures and stresses are kept within the limits of what has been found to be amply sufficient for reliable working under the particular conditions of service called for. In the Sulzer marine engine, the working parts, such as the crankshaft, connecting rods, guides, vertical shaft, cam shaft, etc., generally follow normal practice, as may be seen from the various drawings and photographs of the engine. Accessibility and ease in dismantling and overhauling have in particular received special attention. Fig. 17A.

In a 2,000 B.H.P. Sulzer 4-cylinder marine engine, a cylinder cover can be removed and piston wholly withdrawn, in less than one hour from commencing to dismantle.

Forced lubrication is invariably employed, the oil being led into the bottom of the main bearing brasses and thence to the connecting rod bottom ends. The pressure in the system is about 10 to 15 lbs. in marine engines, the oil being supplied from a gravity tank according to the usual practice, commonly adopted with a geared turbine installation. Instead of oil being supplied to the gudgeon pins through a separate pipe fixed to the connecting rod or through a hole in the connecting rod, a separate telescopic pipe is fitted, this being a simpler and more reliable fitting, and furthermore, advantage can be taken of increasing if necessary the pressure of oil in this system to ensure adequate lubrication. Comments have been made on this departure from normal practice, but it is difficult to understand where either difficulties or troubles are to be expected in such an obviously simple fitting.

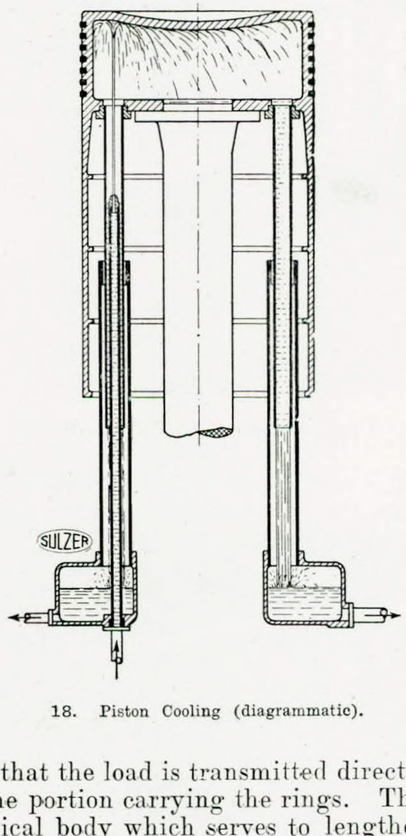
The cylinders are lubricated in the usual manner by oil supplied by a pump with sight feed adjustment. In 4-cycle engines owing to the short piston and the absence of scavenging or exhaust ports in the cylinder liner it is obvious that any burnt lubricating oil or other impurities in the cylinder must find a way out at the bottom of the cylinder liner. For this reason it has been found desirable in most 4-cycle engines to lengthen the piston rod and provide a special chamber between the cylinder and crank-case to avoid contamination of the lubricating oil in the crank chamber by foreign matter coming from the cylinder. No such difficulty arises in the Sulzer 2-cycle engine,as it is found that practically all impurities are blown out the exhaust ports, and as a further precaution a gland ring is fitted at the bottom of the liner, thus effectually isolating the cylinder from the crank chamber.

The consumption of lubricating oil amounts to less than 1% of the fuel oil used and in a ship of 3,000 B.H.P. would amount to about 24 gallons per day. With careful attention, however, the consumption can be still further reduced.



17A. Sulzer Marine Engine under erection.

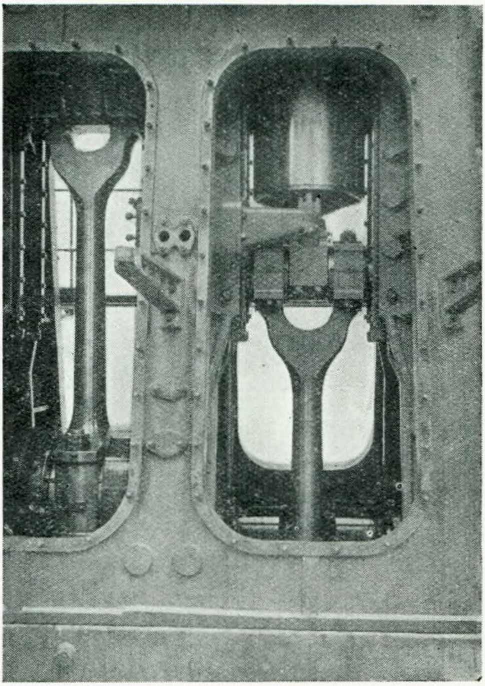
Fig. 18 shows diagrammatically the arrangement adopted for piston cooling. Sea water is generally used, and it will be seen that no glands are necessary, as the water under atmosphere pressure escapes freely through an open drain. The piston is built up in two portions, the piston head being so attached to



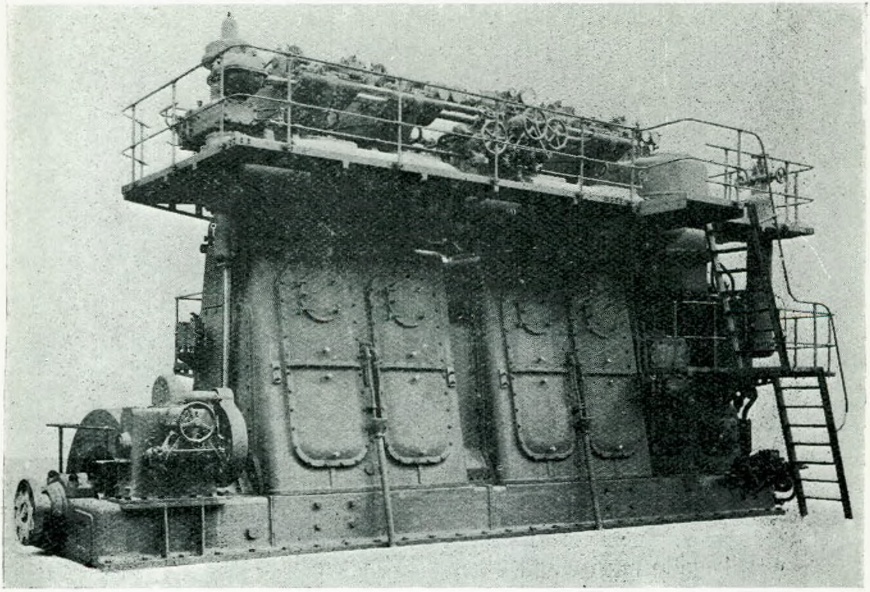
18. Piston Cooling (diagrammatic).

the piston rod that the load is transmitted directly without passing through the portion carrying the rings. The piston skirt is a thin cylindrical body which serves to lengthen the piston as necessary without adding appreciably to the total reciprocating weight. Fig. 19.

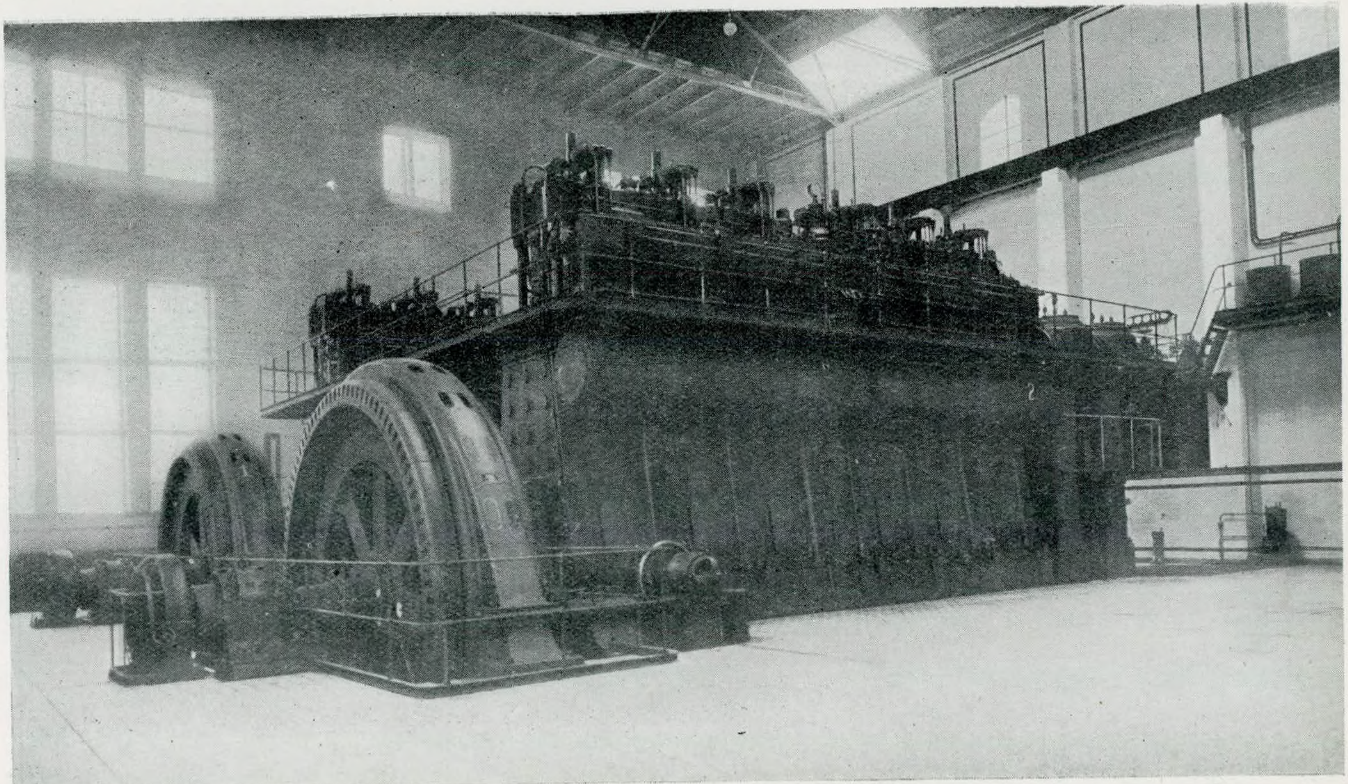
Starting and Reversing Arrangements.—Since a 2-cycle engine receives an impulse every revolution only four cylinders are required to ensure that there is no "Dead point" on starting, whereas a minimum of six cylinders is necessary for this purpose on a 4-cycle engine. Sulzer marine engines are made in units



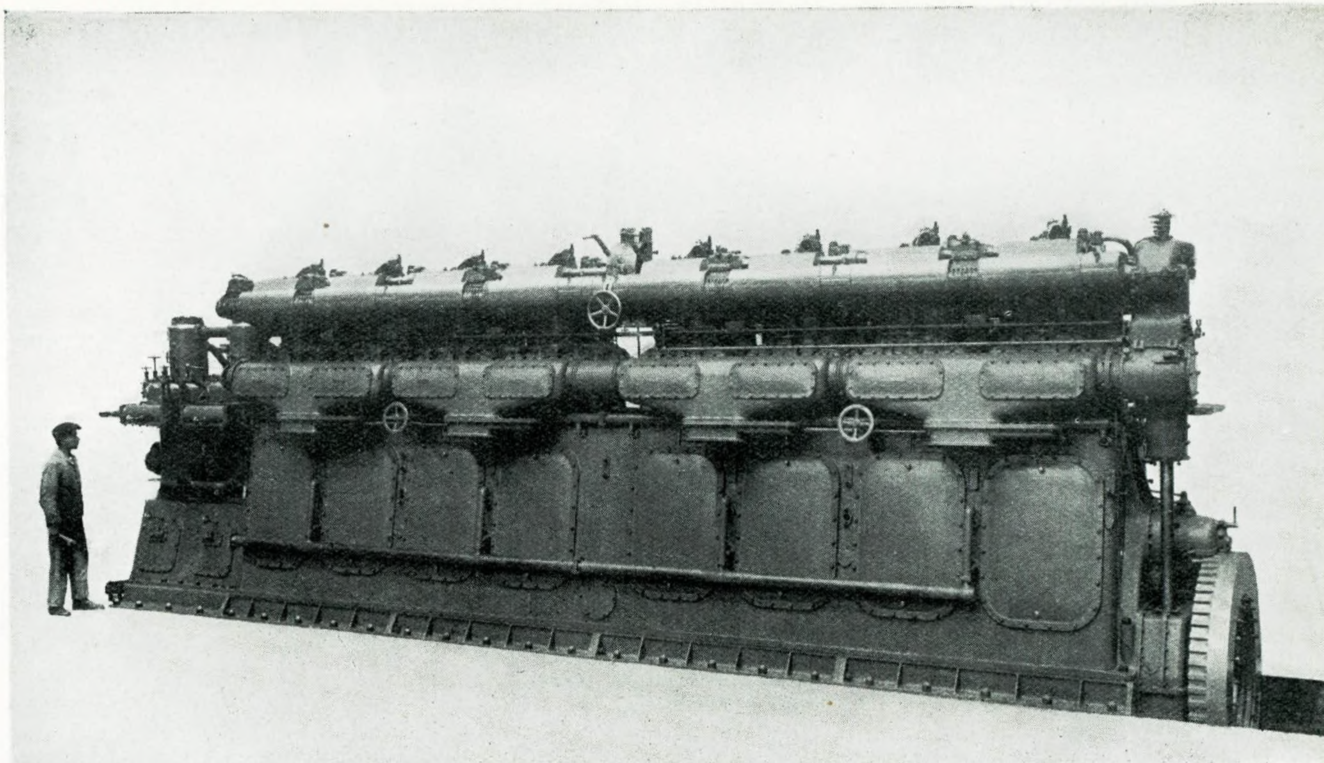
19. Piston and Piston Cooling Bracket.



20. Marine Engine (4 S 60).



21. Two Engines, each 3,000 B.H.P.



22. 3,500 B.H.P. Submarine Engine.

of 4, 6 or 8 cylinders, depending upon the power required and other conditions.

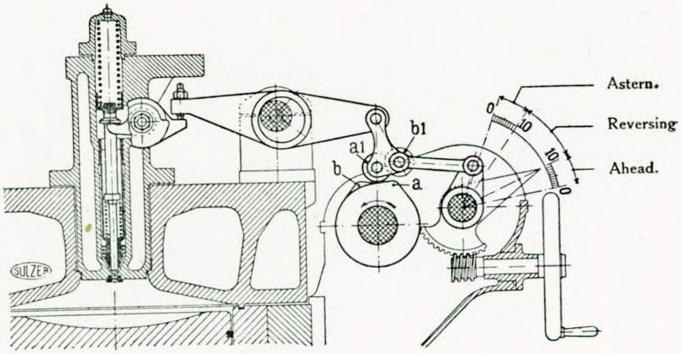
Fig. 20 shows a 4-cylinder 2-cycle marine engine of 1,250 B.H.P. at 100 r.p.m.

Fig. 21 shows two 6-cylinder land engines of 3,000 B.H.P. each.

Fig. 22 shows an 8-cylinder submarine engine of 3,500 B.H.P. at 300 r.p.m.

In a 4-cylinder engine the crankshaft is in two lengths and the pair of cranks on each length are at 180°, while the plane of the cranks on the forward and aft, lengths are at right angles to each other, one crank at least is therefore, at any angular position of the shaft, in a favourable position for starting. Starting is accomplished by admitting compressed air through cam operated air starting valves on all four cylinders. After a revolution or two of the engine, starting air is cut off and fuel admitted to two cylinders, the remaining two cylinders still operating on air. Finally air is cut off and fuel admitted to all cylinders.

Reversing.—For reversing it is only necessary to alter the period or time of opening of the air starting valve, fuel valve, and rotary scavenge valve. The latter valve is driven off the vertical shaft through spiral gearing to the horizontal shaft, and reversing is accomplished by providing a dog clutch with a certain amount of play between the driving faces which is taken up and automatically corrects the timing on reversing the motion of the engine. The fuel and air starting valves are operated by means of the mechanism shown in Fig. 23.

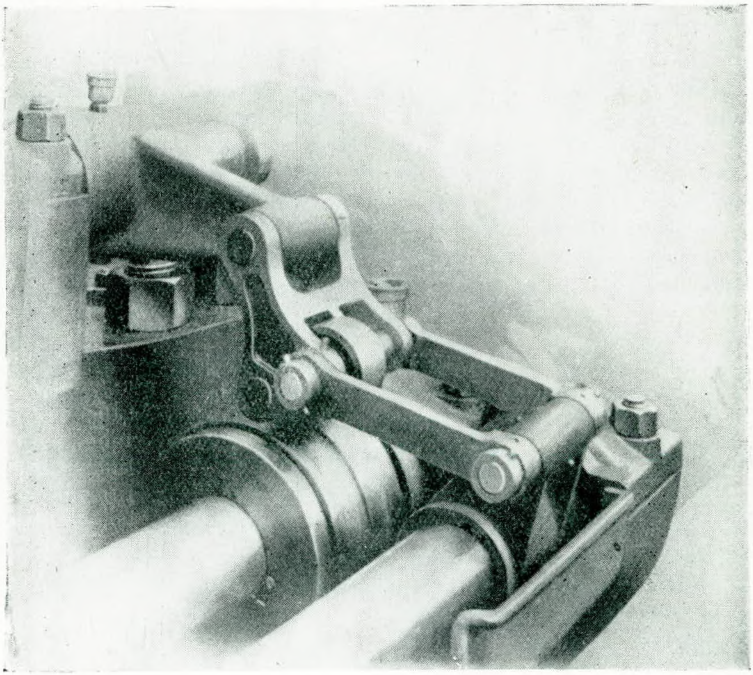


23. Reversing Mechanism.

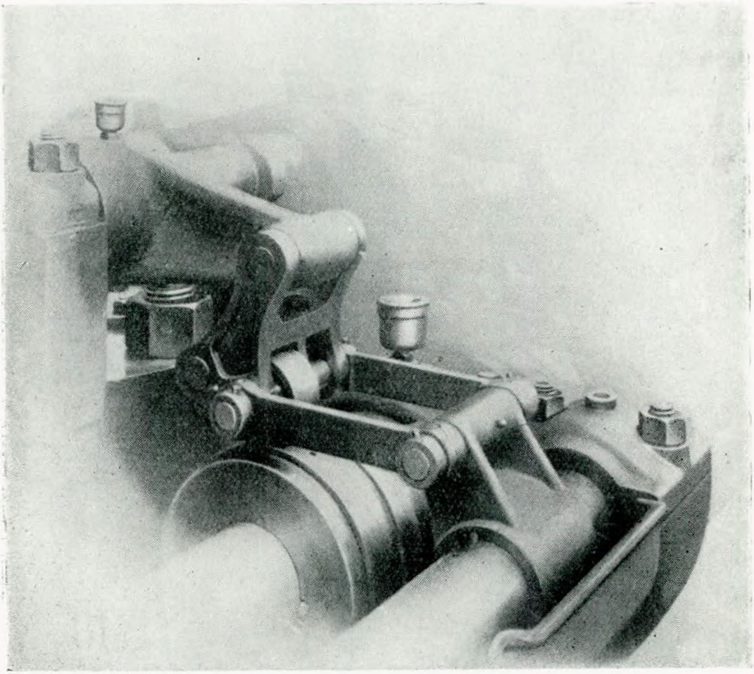
It will be seen that a separate cam "a" and cam roller "a1" is provided for ahead running, while cam "b" and cam roller "b1" are for astern running, and by moving the hand wheel and rotating the reversing shaft either the ahead or astern cams can be brought into action as required. The arrangement is identically the same for both the air starting and fuel valves. Figs. 24, 25.

The alteration in the timing of the engine in the manner described, for either ahead or astern running, requires very little effort and even in engines of the largest size can easily be accomplished by hand, so that no servo motors are ever necessary for this operation.

Starting.—In order to provide for the various operations of starting being carried out in their proper sequence, the rocking levers are mounted on eccentrics on two control shafts, one for each pair of cylinders, which are turned by hand in the case of



24. Reversing Gear (Ahead).

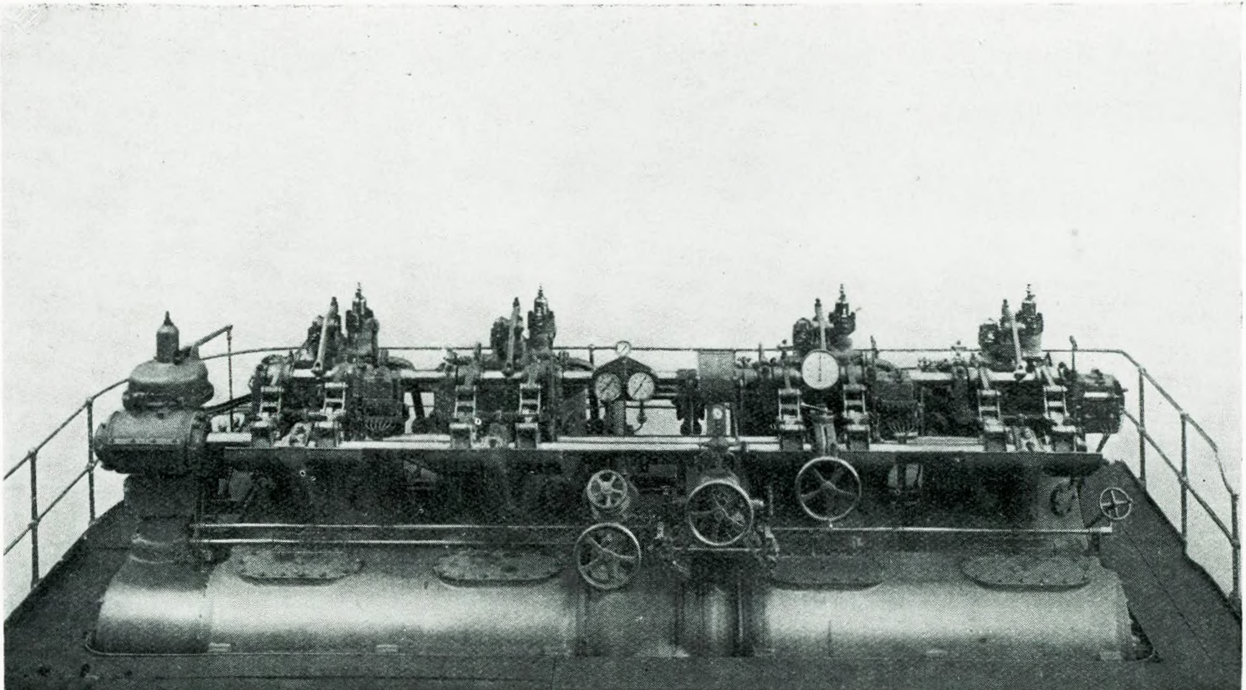


25. Reversing Gear (Astern).

a small engine, or by an air servo motor or starting engine for large engines. The eccentrics are so set that after having put the reversing shaft in the position for ahead or astern running the following operations are carried out. Figs. 26, 27, 28.

(1) In the "stop" position all cam rollers are lifted clear of the cams, and on putting both control shafts into the position for "Running on Air," all air starting cam rollers are brought down on their respective air starting cams, and at the same time the main air starting valve—to which reference will be made later—is opened on each cylinder by means of a cam fitted on the control shaft. In this position the fuel cam rollers are still clear of the cams, but one cylinder being in the correct position air will be admitted through the cam operated advance air starting valve on this cylinder, and the engine will then commence to rotate on air.

(2) The next movement is partially to rotate the aft section of control shaft to "Running on Fuel," which removes the air



26. Starting Platform, 4-cylinder Marine Engine.

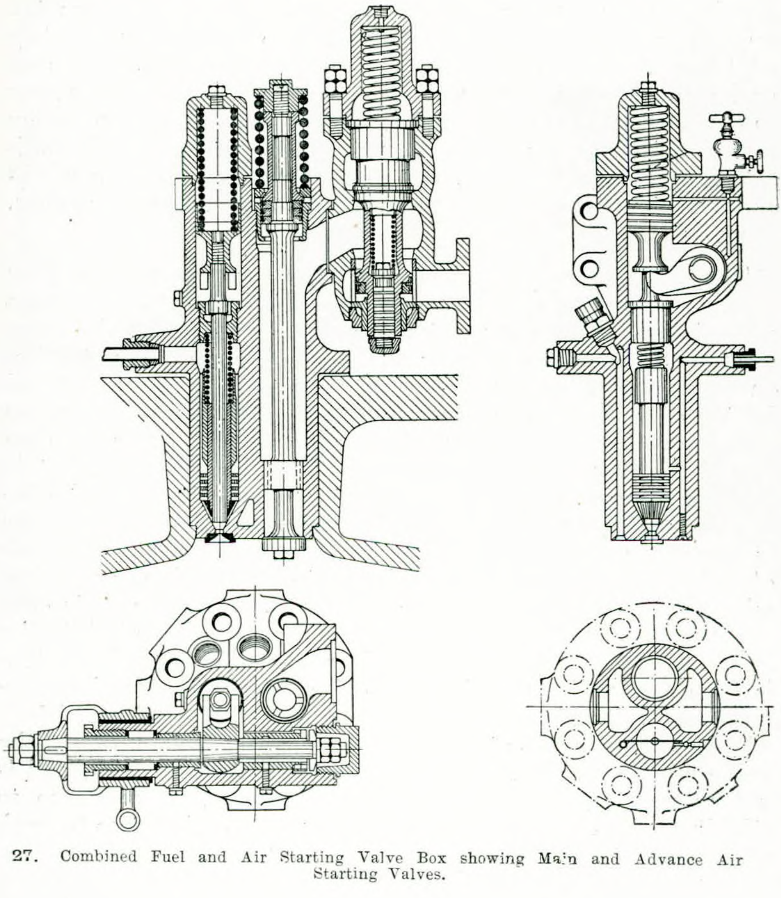
starting cam rollers from the cams and brings the fuel cams into operation on the aft pair of cylinders, thus leaving two cylinders on air and two cylinders on fuel.

(3) Finally a similar movement of the forward section of control shaft brings the fuel cams for the forward two cylinders on to their respective cams and at the same time puts the air starting cams out of action, thus resulting in all cylinders being in operation on fuel. Apart from the simplicity of the mechanism described, an important advantage is gained by the system of levers employed, as it enables the lift and period opening of the fuel needle valve to be reduced by adjusting the hand wheel to give any desired clearance between the roller and cam, and at the same time the exact time of commencement of opening is practically unaltered. This greatly increases the reliability of working under low loads, and enables the quantity of blast air which is dependent on the blast pressure and period of opening, to be adjusted as may be necessary to give the best results according to the particular load and condition under which the engine is running. The adjustment is indicated by a pointer on a graduated sector, the divisions 0—10 showing the range of adjustment for ahead and astern running. (Fig. 23).

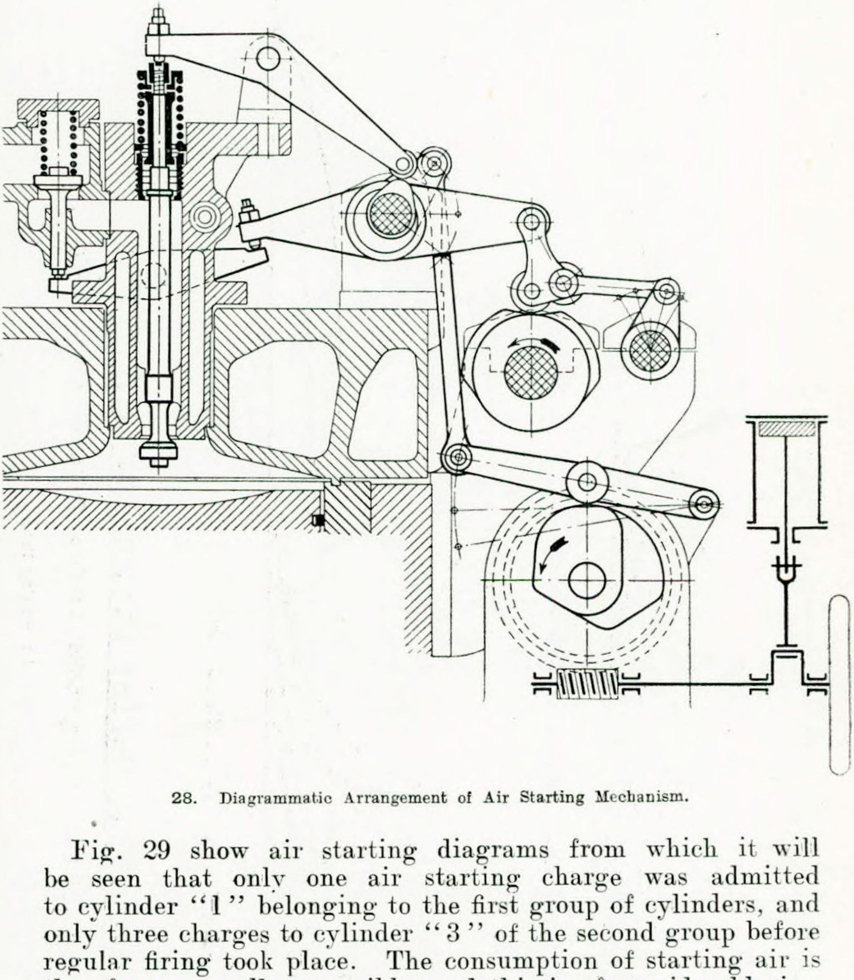
The arrangement of two air starting valves—a main and an advance valve for each cylinder—has the following important advantages. The main valve which opens directly into the combustion chamber remains shut while the engine is running on fuel and thus isolates the advance air starting valve operated by the cam shaft, from the burning gases in the main engine cylinder. Any chance of carbonization or sticking of the cam operated advance valve is thus practically eliminated, but if desired, the advance valve can be removed and cleaned while the engine is running since the main valve in combustion chamber is, under these conditions, shut. The space between these pair of valves is in direct communication when engine is running with either the atmosphere or the scavenge trunk and is consequently not under pressure. Leakage of starting air into the cylinder or of burning gases into the starting air system is thus rendered impossible. During starting operations the main valve is opened by a cam on the control shaft, which it will be remembered is turned by the starting engine, but as the valve remains open during the whole period of starting, and is not required to operate rapidly, no difficulties arise due to sticking. The cam operated advance valve is thus free to work under the best conditions removed from the high cylinder temperatures and absolute reliability in starting is ensured.

Another feature of this arrangement is that immediately the main air starting valve on the cylinder is opened by the control shaft, the compression pressure is at once released since the air passes direct from the cylinder into either the atmosphere or the scavenge trunk. As soon as the cam operated advance valve comes into operation the escape to the atmosphere or scavenge trunk is automatically closed and starting air passes by way of the advance valve and main valve direct to the cylinder. Special means are provided in the advance valve for balancing against the air starting pressure so that very little effort is needed to operate the valve.

Fig. 27 shows a section through the combined fuel and air starting valve box.



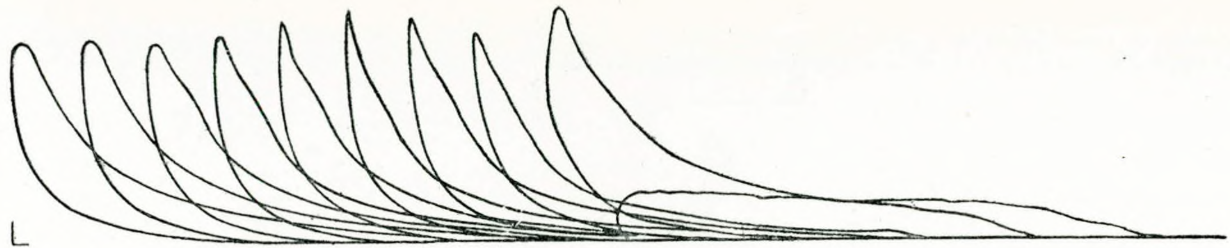
27. Combined Fuel and Air Starting Valve Box showing Main and Advance Air Starting Valves.



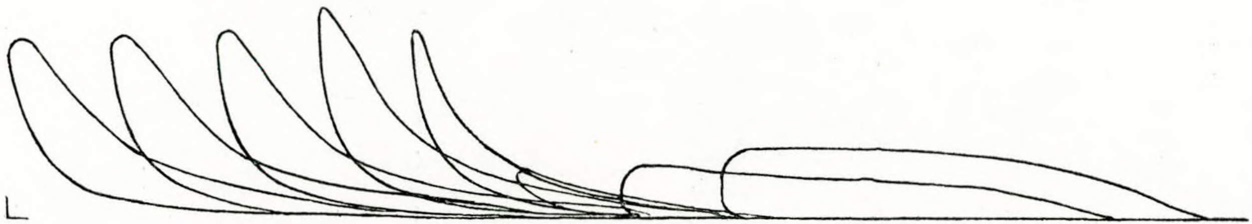
28. Diagrammatic Arrangement of Air Starting Mechanism.

Fig. 29 show air starting diagrams from which it will be seen that only one air starting charge was admitted to cylinder "1" belonging to the first group of cylinders, and only three charges to cylinder "3" of the second group before regular firing took place. The consumption of starting air is therefore as small as possible, and this is of considerable importance in a marine installation where numerous manœuvres have to be performed on a given air bottle capacity without having sufficient time to recharge.

In all Sulzer marine engines the main air compressors are designed of approximately sufficient capacity to supply twice the amount of air actually necessary for blast purposes. Consequently in a twin screw ship the main compressors on either engine are sufficient, in case of the breakdown of one com-



Taken on Cylinder No.1.



Taken on Cylinder No.3.

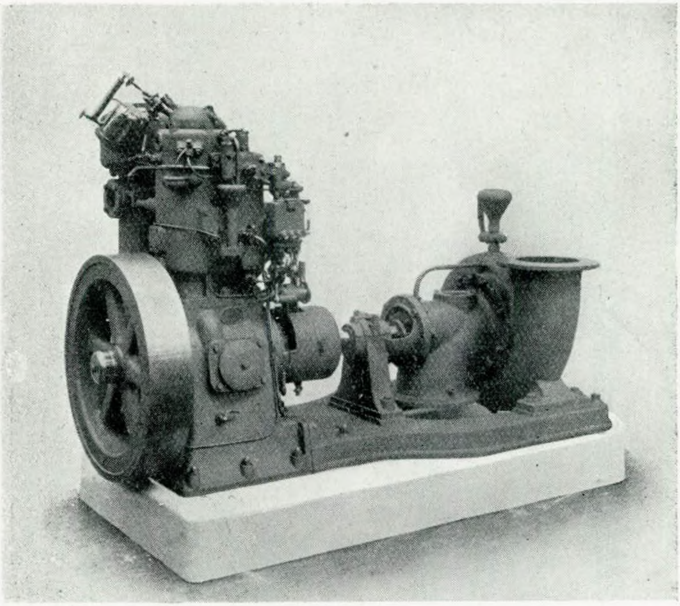
Sulzer Two-Cycle Marine Diesel Engine.

Successive Indicator Diagrams taken while starting.

29. Air Starting Diagrams.

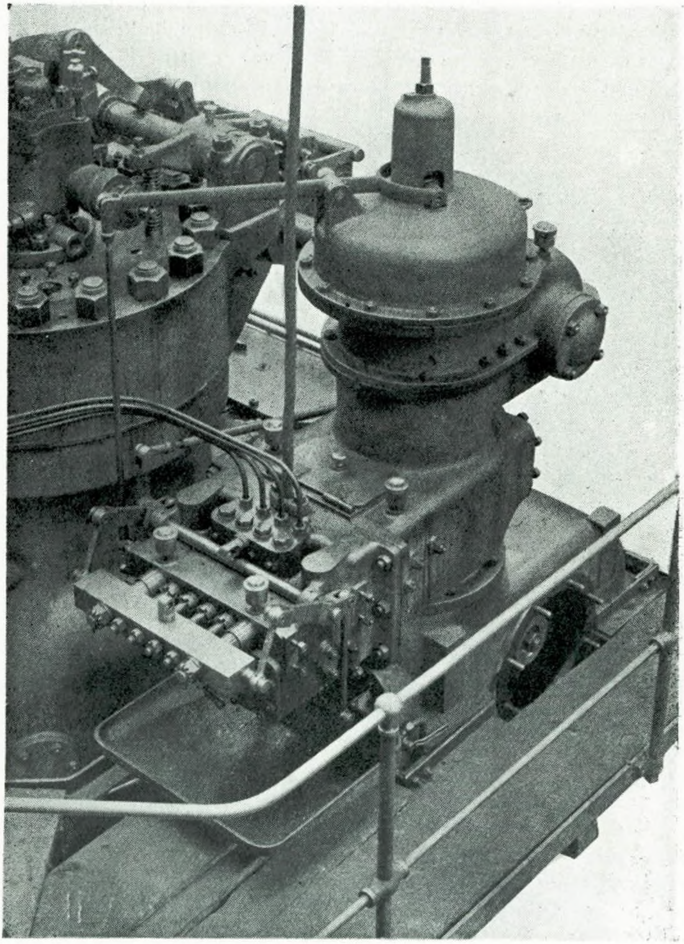
pressor, for running both engines, or when both are running there is plenty of surplus air for recharging the air bottles. Additional electrically driven air compressors are, however, usually fitted as a further safeguard of the air supply. Also to comply with Lloyd's requirements an emergency compressor which can be started without compressed air is fitted so that in the event of all air pressure being accidentally lost recharging can be accomplished.

Fig. 30 shows a type of combined hot-bulb engine and compressor Sulzer provide for this purpose. The engine is started without using compressed air and the set is also suitable for driving an emergency dynamo or can be direct coupled, as shewn in the illustration to a centrifugal pump for cooling water or other auxiliary purposes.



30. Hot Bulb Engine Compressor and Pump.

Fuel Control.—In all operations of starting and reversing, a complete system of interlocking is provided to avoid the possibility of a mistake. The separate fuel pump provided for each cylinder is also controlled from the mechanism operating the control shaft in such a manner that the pump is only delivering fuel when required, *i.e.*, when the fuel valves on cylinder are in operation. When no fuel is required the suction valve is held continuously off its seat, while when running the quantity of oil delivered is varied as necessary by hand adjustment to give the required load, by holding the suction valve open during a greater or less portion of the delivery stroke. The fuel pumps are in addition controlled by a governor of the centrifugal type, which cuts off fuel as necessary from the different cylinders in order to prevent the engine racing. (See Fig. 31).



31. Fuel Pump and Governor 4-cylinder Sulzer Marine Engine.

Air Compressor.—The air compressor for supplying blast air to the fuel valve requires a high degree of reliability, which can only be obtained by the most careful attention to details in the design. In the early days of a Diesel engine the air compressor was a fruitful cause of trouble, and induced several firms to abandon air injection of the fuel for mechanical or solid injection. Sulzers have for many years standardised a design of three stage compressor, and experience covering large numbers of engines in service has shown that all the early defects in design have been eliminated and at least the same degree of reliability is obtainable in the compressor as in the engine itself.

Mean Pressure and Piston Speed.—It is particularly important in the case of a Diesel marine engine to adopt a conservative H.P. rating for any given size of engine. In steam engine practice it is usual to allow a certain mean pressure referred to the L.P. cylinder which taken together with the piston speed represents from wide experience the output the engine will safely develop. The boilers are so designed that under trial conditions with clean fires and engines "linked out" a certain maximum power is obtained, which is practically never required under service conditions at sea. The conditions of steam supply sufficiently guard against the possibility of overloading the engines. In a Diesel engine, however, the power is dependent within limits upon the fuel supplied to the cylinder and the ease with which an overload can be obtained by a simple adjustment of fuel control must be foreseen and guarded against.

The Sulzer 2-cycle Diesel marine engine is rated to work at a mean indicated pressure of about 95 lbs. per sq. inch on continuous service, although a considerably higher pressure is possible without any visible sign of overloading, such as a smoky exhaust. In the trials of the single screw Motor Ship *Augusta*, a converted steamship fitted with a four cylinder engine of 800 B.H.P., the indicated mean pressure at normal continuous load was 98·8 lbs. per sq. inch, while for several hours a mean pressure of 124·6 lbs. was maintained with a perfectly clear exhaust, the engine developing 1,015 B.H.P.

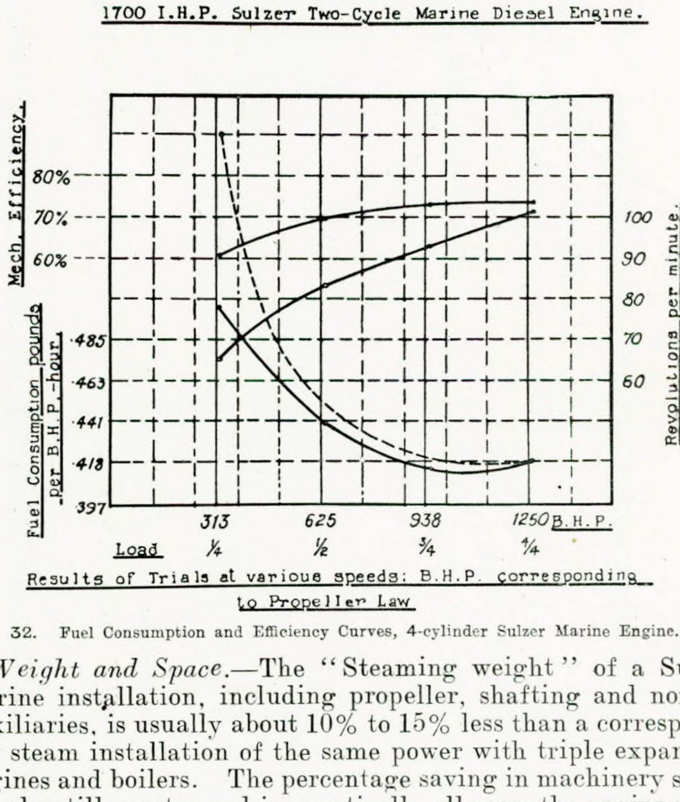
More recently the *Conde de Churruca*, a twin screw tanker, of 2,500 B.H.P., when on her maiden voyage from Newcastle to Morocco and back, worked at a mean indicated pressure varying between 106 and 112 lbs. per sq. inch. During the contract trials of this ship, indicated mean pressures of 120 to 128 lbs.

were obtained. It must be distinctly understood, however, that although these high mean pressures are easily obtained it is not recommended that the rated mean pressure should be continuously exceeded on service. For six hours or so an overload of 10% is, however, justifiable. The piston speed of Sulzer marine engines has been determined with a view to maintaining with a certain stroke bore ratio, a speed of revolution which is generally appropriate for the conditions of service called for. The piston speed is not affected by the question of whether the engine is 2-cycle or 4-cycle. In Sulzer 2-cycle submarine engines, piston speeds of over 1,100 ft./mm. have not caused the slightest trouble, but for mercantile engines the speed is generally 750 to 800 ft./mm. and with a stroke bore ratio of about 1·7/1, this gives a low speed of revolution to the crankshaft, which is generally suitable for propeller efficiency.

Efficiency.—It is sometimes assumed that the mechanical efficiency or ratio of B.H.P./I.H.P. of a 2-cycle engine must necessarily be lower than for a 4-cycle engine owing to the power required for driving the scavenge pump. It must be remembered, however, that the moving parts of a 4-cycle engine are for the same power considerably heavier and both the number and sizes of bearing surfaces greater than in a 2-cycle engine, thus adding to frictional losses. Furthermore, "pumping losses" are incurred during the inlet and exhaust stroke of the 4-cycle engine, owing to the frictional and eddy resistance of the air, and exhaust products passing through the valves and pipes. A badly designed 2-cycle engine with a high scavenging pressure such as is necessary with valve head scavenging is certainly conducive to poor efficiency, but in the Sulzer design with ample scavenging port areas, a pressure of only $1\frac{1}{2}$ to 2 lbs. is sufficient. Indicator diagrams taken from the reciprocating scavenging pumps of a four cylinder Sulzer marine engine developing 1,720 I.H.P. indicated 59·2 I.H.P. and the total mechanical efficiency of the engine was 73·5% or about the same as a normal 4-cycle engine.

Using separate electrically driven scavenge pumps the mechanical efficiency of the main engine neglecting the power supplied to the scavenge pumps is about 78% to 80%. The overall efficiency as shown by fuel consumption tests taking into account the power for the separate scavenging pumps is slightly higher with this system than with direct driven pumps.

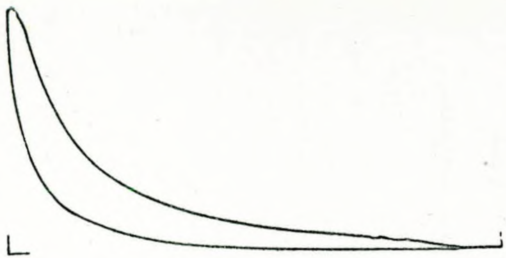
Exhaustive tests have shown that the thermo dynamic efficiency of the Sulzer 2-cycle engine is practically independent of the class of fuel oil used, thus using gas oil, tar oil, mazout or Mexican fuel oil, an engine of 1,250 B.H.P. with direct driven scavenge pumps shows a thermo dynamic efficiency of 32% to 34%. This has been improved upon in large engines using separate turbo scavenging. The fuel consumption using fuel of about 18,000 B.T.U. calorific value is 0·40 to 0·42 lbs. per B.H.P./hour, which is practically the same as a 4-cycle engine. Figs. 32, 33.



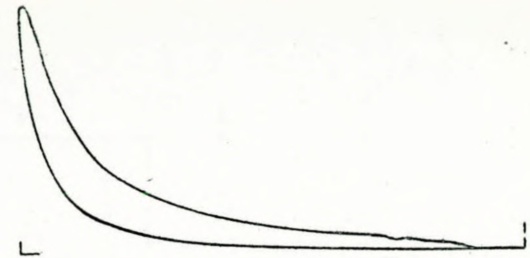
32. Fuel Consumption and Efficiency Curves, 4-cylinder Sulzer Marine Engine.

Weight and Space.—The "Steaming weight" of a Sulzer marine installation, including propeller, shafting and normal auxiliaries, is usually about 10% to 15% less than a corresponding steam installation of the same power with triple expansion engines and boilers. The percentage saving in machinery space may be still greater and in practically all cases the engine room can be even smaller than the minimum size of engine room which, according to Board of Trade rules, it is found necessary to provide in order to obtain the best allowance for nett registered tonnage.

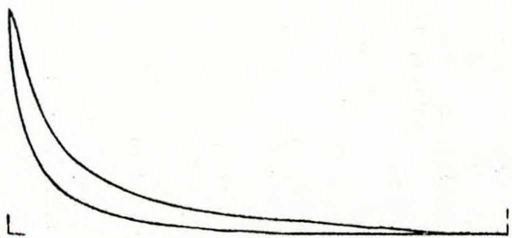
The fact that an engine has a low weight per B.H.P. may suggest light scantlings, special materials, or other means



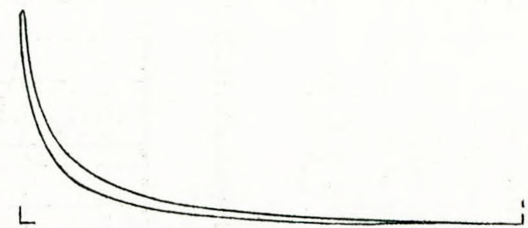
$\frac{3}{4}$ Load, 956 B.H.P., 92.7 r.p m.



$\frac{1}{2}$ Load, 642 B.H.P., 82,8 r.p.m.



$\frac{1}{4}$ Load, 324.5 B.H.P., 65.3 r.p.m.



32 B.H.P., 25 r.p.m.

1700 I.H.P. Sulzer Two-Cycle Marine Diesel Engine.

Indicator Diagrams at various speeds, B.H.P. according to Propeller Law.

33. Indicator Diagrams. 1,250 B.H.P. Sulzer Marine Engine.

which may be employed to reduce the weight of a particular engine in comparison with another engine of the same type. Thus, for example, considering 4-cycle engines, there is a considerable saving in weight in the Werkspoor engine as compared with the Burmeister and Wain engine, due to a totally different method of building up the engine framing and supporting the cylinders. Similarly, submarine engine weights are reduced by using special materials, cast steel instead of cast iron, and so on, and also by running the engines at very high mean pressures and piston speeds.

It will be seen from various illustrations of the Sulzer engine that the engine framing is of robust design. The bedplate, crank case, cylinder columns, cylinder jackets and cylinder covers are all of cast iron, while the main running parts, such as crank shafts, connecting rods, piston rods, cam shafts, etc., are of ordinary mild steel. The relatively light weight is obtained by running on the 2-cycle principle in such a manner that a high mean indicated pressure can be safely maintained; thus, in a cylinder of a given diameter approximately double the horsepower is obtained compared with a 4-cycle engine cylinder of the same dimension running at the same piston speed.

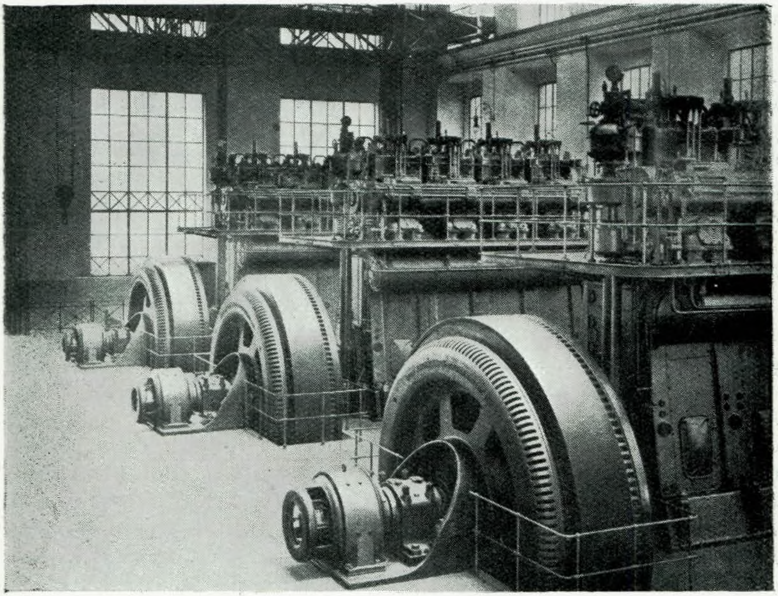
While reduction of weight in a marine engine is not of primary importance from the point of view of the gain in cargo-carrying capacity, it has a very important bearing on the initial cost of an installation seeing that if the same materials are used and if the engine is not more complicated the cost of production must be mainly determined by the weight.

Some Applications of Sulzer 2-Cycle Engines.

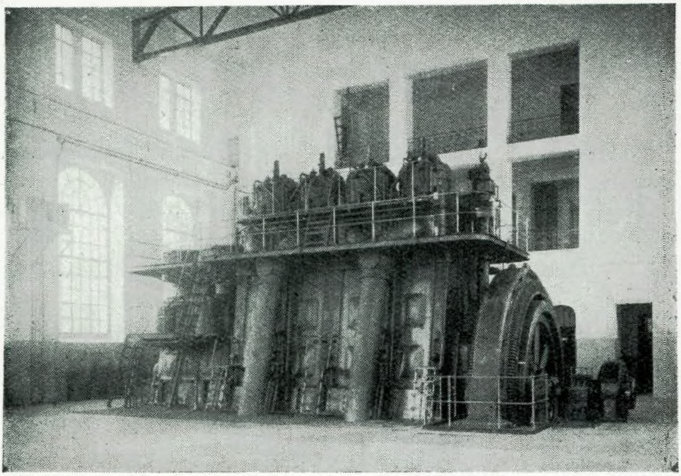
Figs. 34, 35, 36 illustrate some large Sulzer 2-cycle land installations. There are at present 28 land installations of over 2,000 B.H.P. each, and about 50 installations of over 1,000 B.H.P. equipped with Sulzer engines.

Fig. 37 shows a 400 B.H.P. marine engine. About one hundred sets of these engines have been built. No air starting motor is necessary in engines of this size and it will be seen that two hand levers are provided for turning the control shafts for each pair of cylinders.

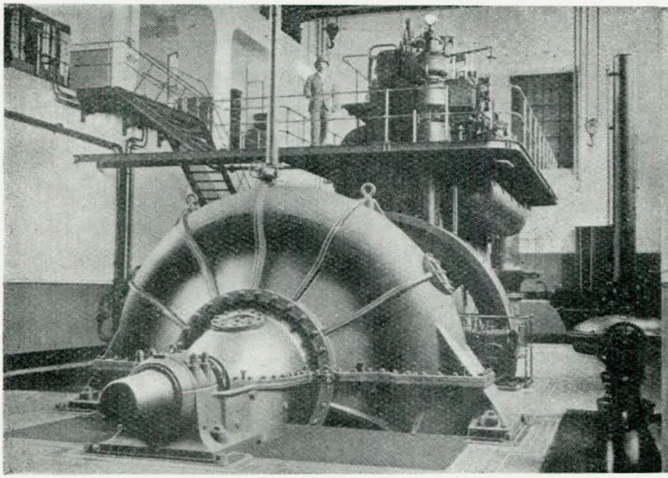
Fig. 38 shows an 800 B.H.P. marine engine which was fitted in the single screw vessel *Augusta*. On sea trials this engine developed over 1,000 B.H.P. It will be observed that single crosshead guides are fitted instead of double guides which are adopted in larger engines.



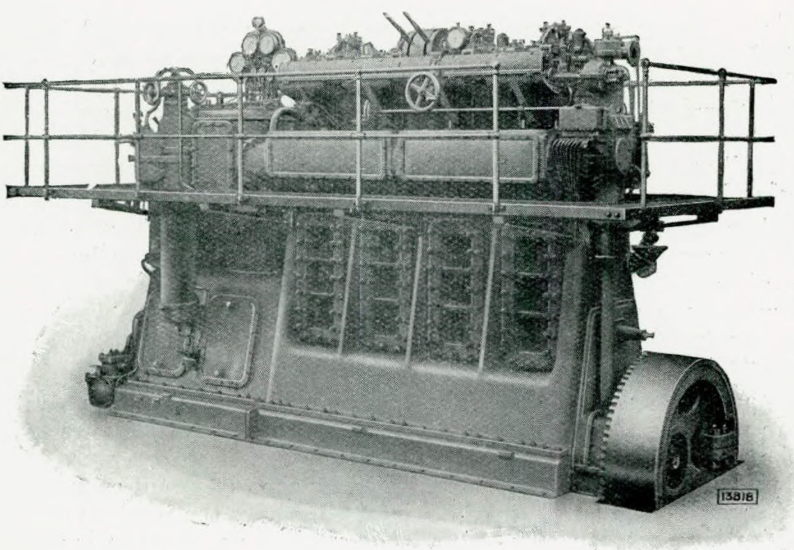
34. Three 1,500 B.H.P. Sulzer 2-cycle Engines.



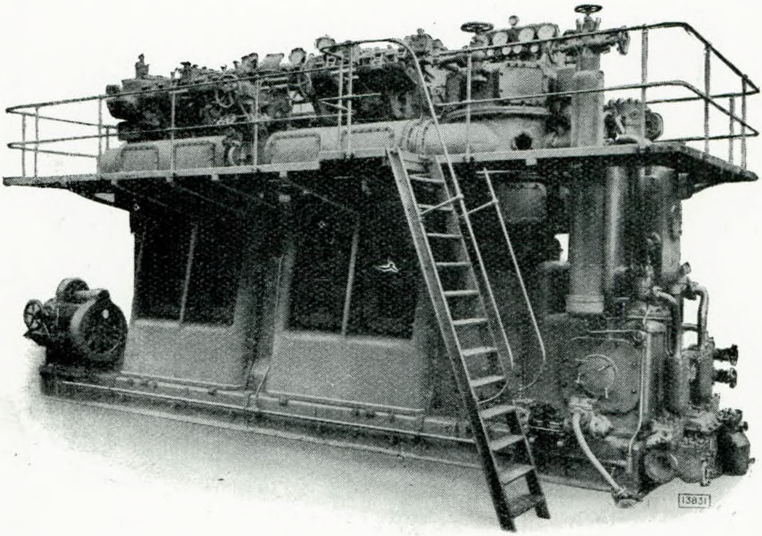
35. 2,300 B.H.P. Sulzer 2-cycle Engine.



36. 2,000 B.H.P Sulzer 2-cycle Engine and Centrifugal Pump



37. Sulzer 2-cycle Marine Engine, 400 B.H.P.

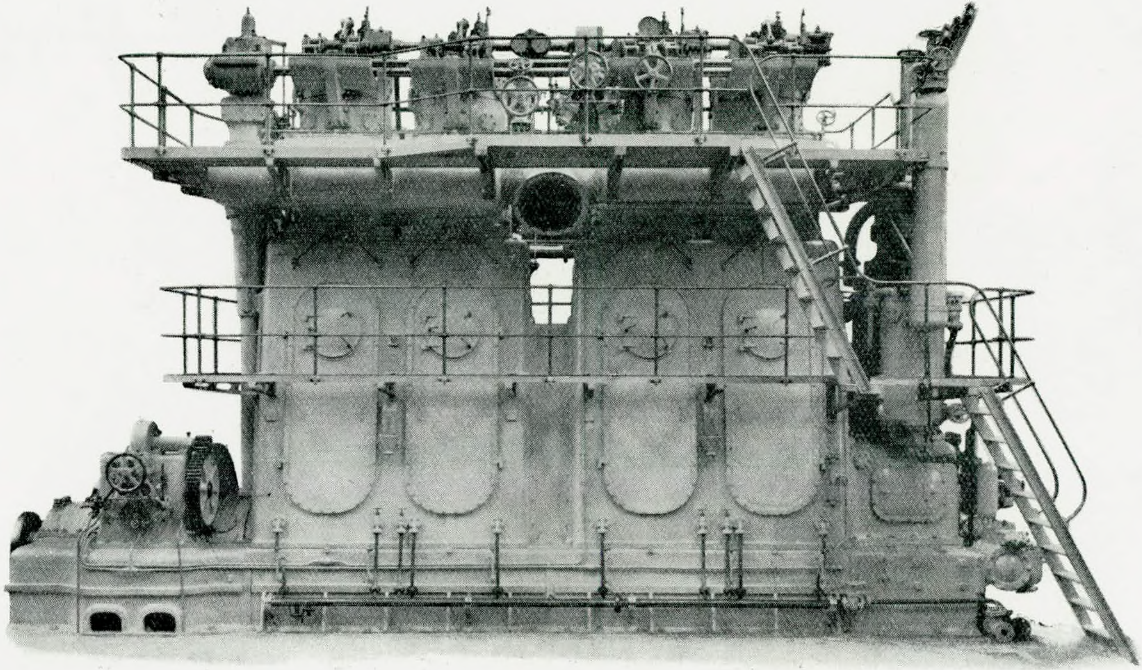


38. Sulzer 2-cycle Marine Engine, 800 B.H.P.

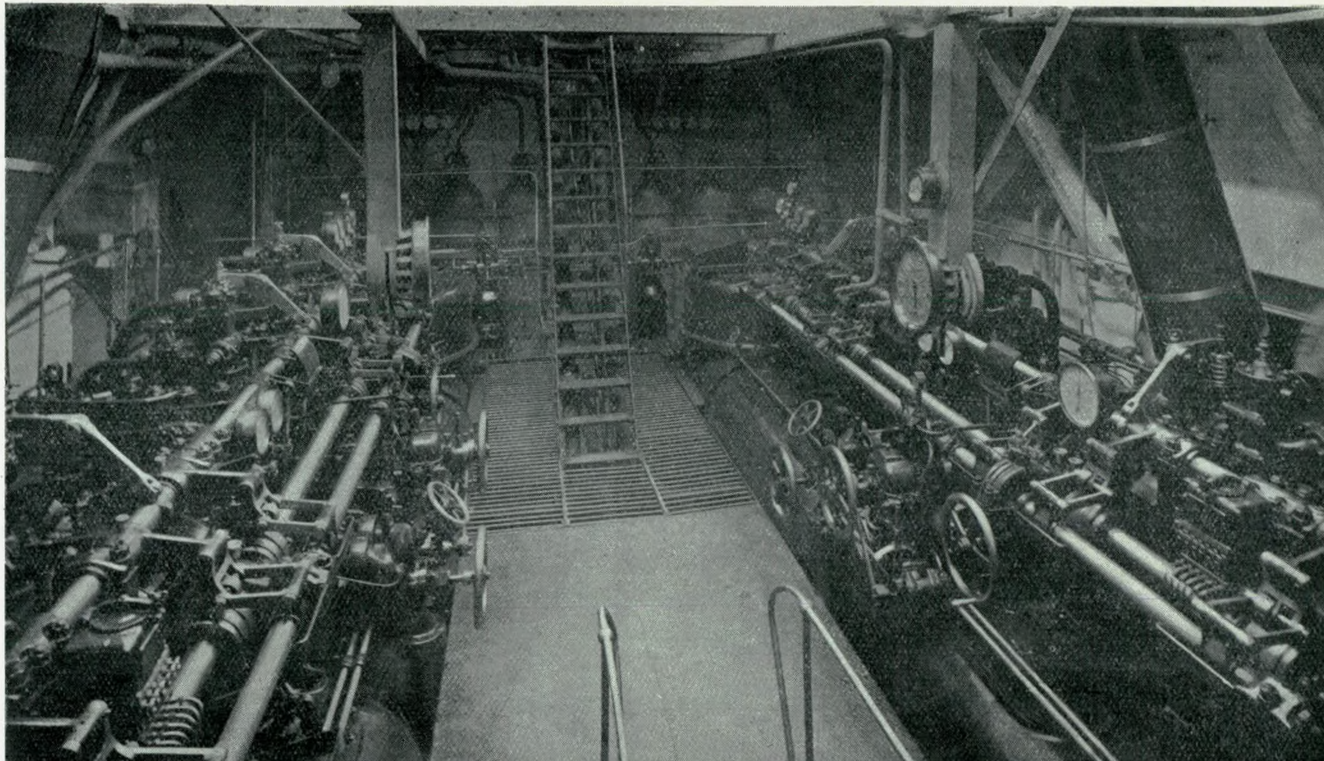
Fig. 38A is a Sulzer 2-cycle marine engine developing 1,500 B.H.P. at 110 revs. per min., the scavenging pumps being in this case electrically driven. Two of these engines are fitted in the Norwegian Motorship *Handicap* and are now in service. This ship was described in the February issue of the "Motor Ship," and she has now reached Bahia Blanca after a very successful voyage of 25 days from Newport.

Fig. 39 shows the starting platform of the motor tanker *Conde de Churruca*, situated on the upper grating. In some larger engines of 2,000 B.H.P. built for the Chargeur Reuni Co. of France, the controls are so arranged that they can be worked either from the top or bottom levels. Figs. 40 to 45 further illustrate the application of the Sulzer engine to marine purposes.

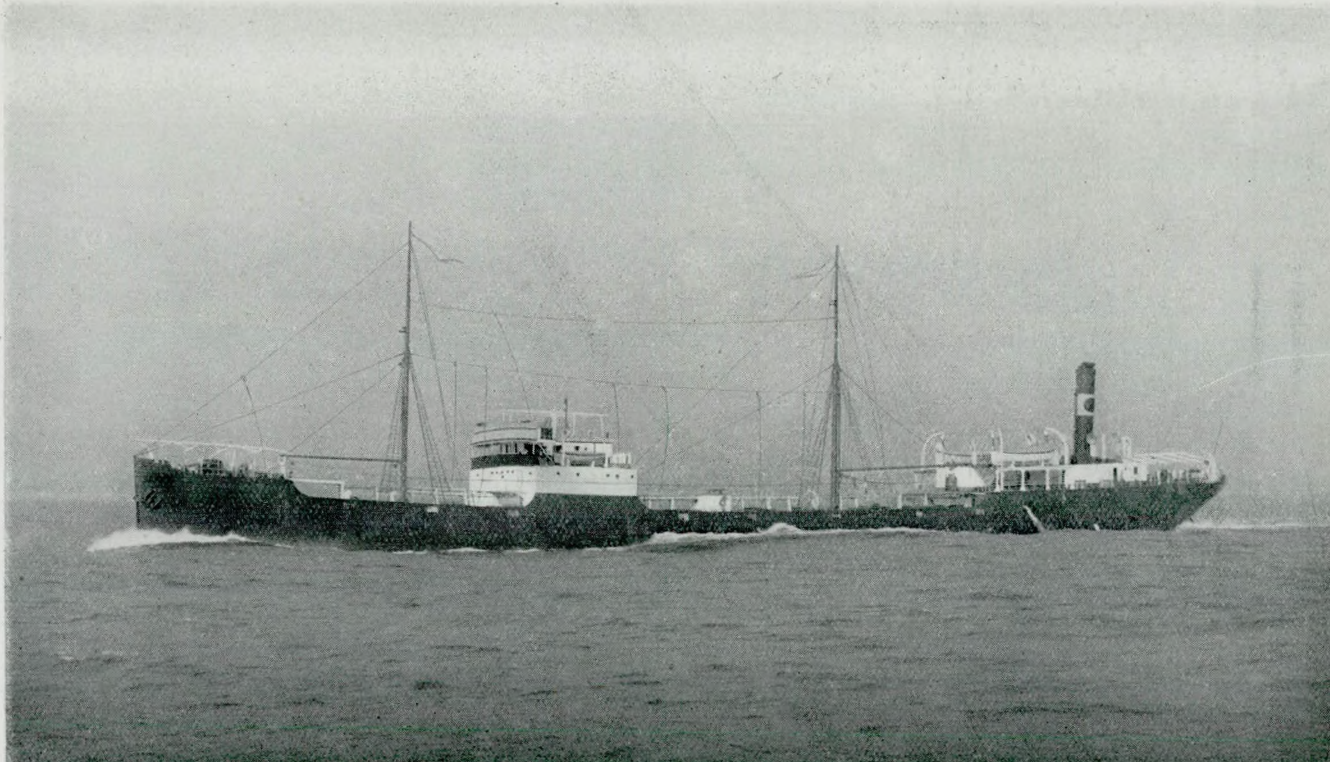
Future Developments.—There are sure signs that shipowners are realising the advantages of the motorship and confidence in the reliability of the Diesel engine, which was at one time in doubt, has now been well established by the successful running of several different types of engines. Perhaps the most serious objection that can now be urged against the motorship is the higher initial cost as compared with the steamship, and this can only be met by striving to reduce the total weight of machinery, to utilise ordinary inexpensive materials and to simplify the



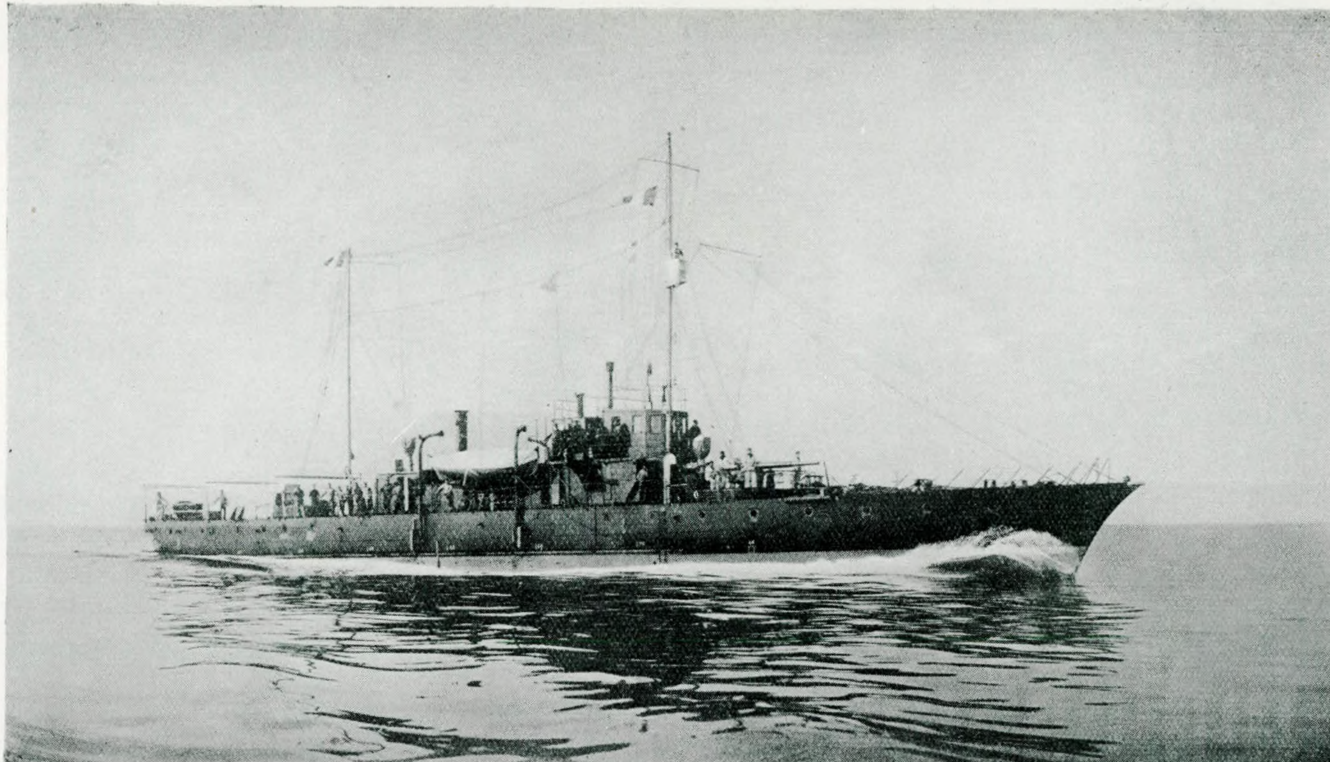
38A. Sulzer 2-cycle (4 ST. 60) Marine Engine, 1,500 B.H.P.



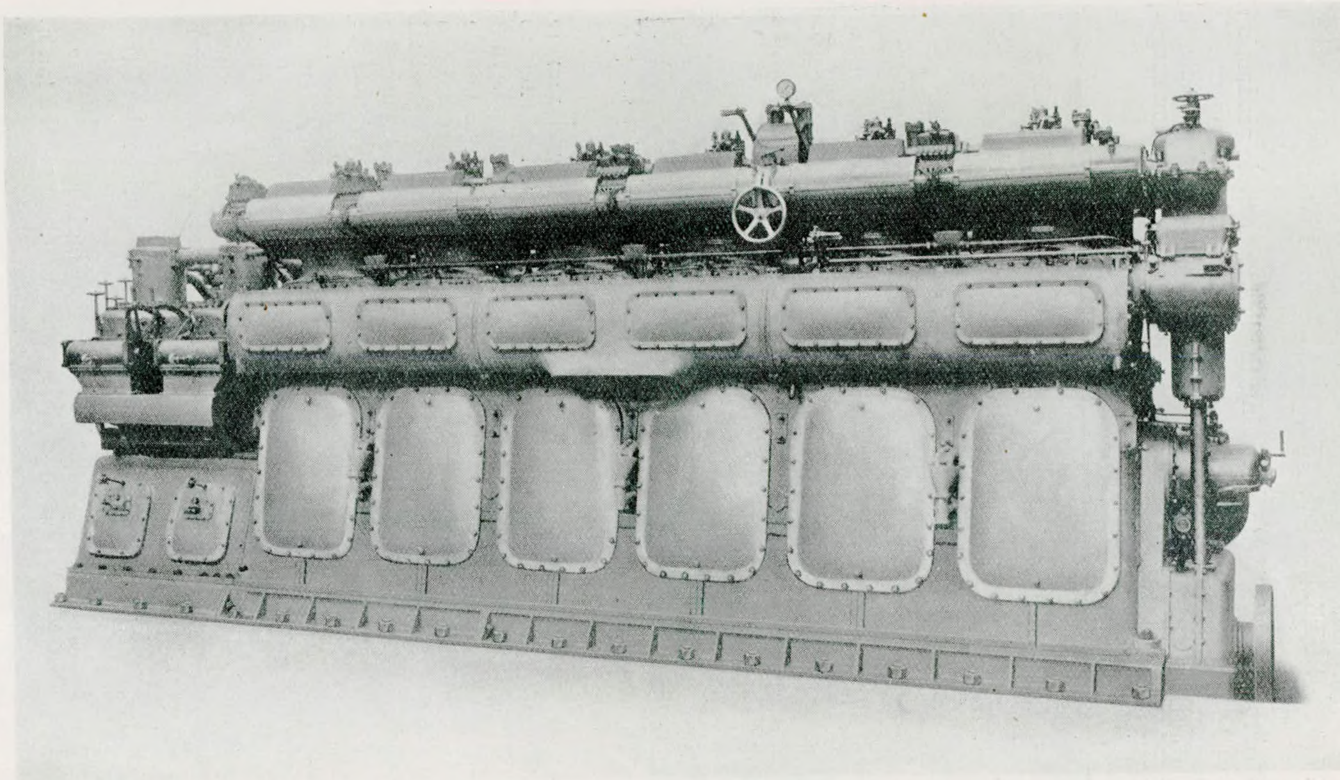
39. Top Platform "Conde de Churruca."



40. "Conde de Churruca."

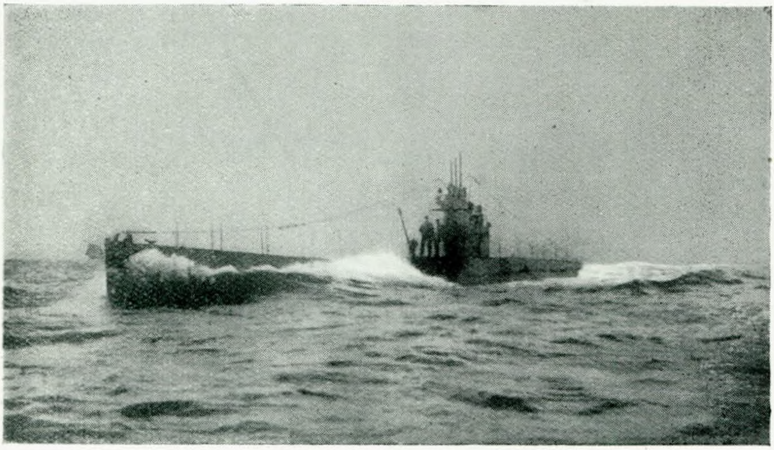


41. French Patrol Boat.



42. Submarine Engine, 2,500 B.H.P.

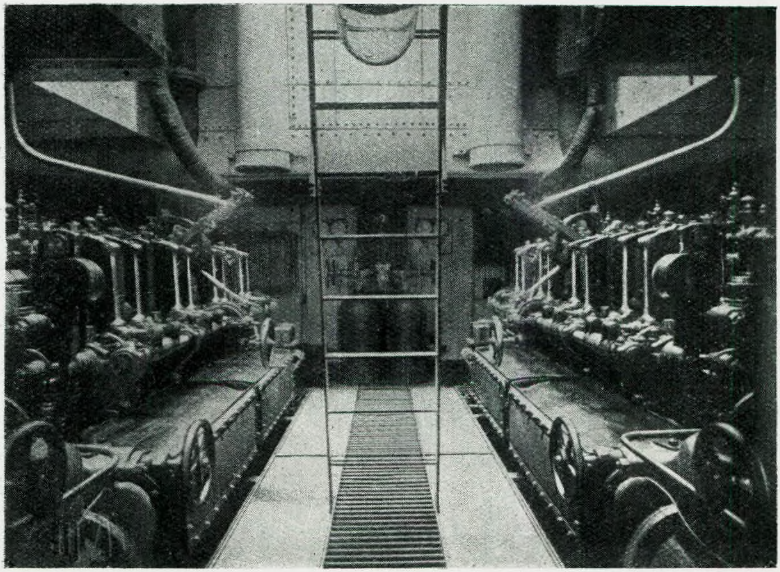
designs with a view to obtaining the lowest possible production costs. Bearing in mind the certain gain in cargo-carrying capacity on a given displacement, it is even now possible to produce a motorship at practically the same price per ton cargo-carrying capacity as a corresponding steamship.



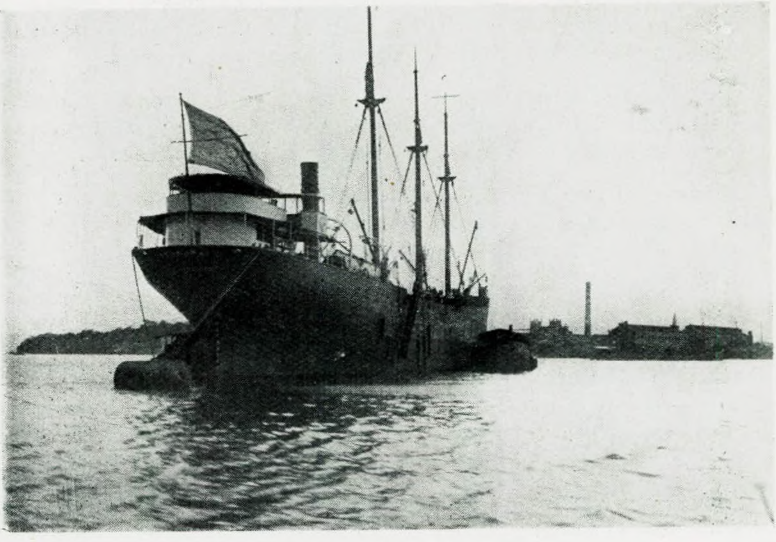
43. U.S.A. Submarine.

With regard to the limits of power of the Diesel marine engine, there appears to be in some directions a tendency to favour double acting engines, presumably on the grounds that by this means larger powers can be developed. The double acting engine has, however, its own problems which remain to be solved by experimental means and by observation of actual results in service whereas there is a vast amount of experience behind the single acting engine. So far the only notable example of a double acting engine is in the Motorship *Assyrian* (late *Fritz*), built by Messrs. Blohm and Voss, and the experiment does not invite a repetition of this type of engine. The first cost must be far higher than for a single acting engine, accessibility for overhaul is extremely bad and there does not appear to be a single advantage of any importance whatsoever as compared with the single acting engine.

At the present moment there is no doubt that all needs of the Mercantile Marine, with the possible exception of fast Atlantic liners, can be immediately fulfilled by the single acting 2-cycle engine. Standard designs of Sulzer marine engines, the main



44. Top E.R. Grating, "Itamaraca."



45. Motorship "Itamaraca."

particulars and details of which are based upon actual practical experience, are already prepared in various sizes up to an eight cylinder engine, which will develop for continuous sea service 7,500 B.H.P. at 82 r.p.m. The diameter of cylinder in this engine is only 35·5 inches, which is 3·9 inches less than the single cylinder experimental engine, and only 5·5 inches larger in diameter than several land engines which have been in operation for the past five or six years. It is inconceivable, therefore, that the much discussed heat stress troubles will arise in an engine of this size. A twin screw vessel of 15,000 B.H.P. or a triple screw of 22,500 B.H.P. is thus an immediate possibility, presenting less experimental features and showing vastly more attractive gains than did the double reduction geared turbine ship of a few years ago. It is not too sanguine to expect that the shipowners and their technical staffs will carefully study the question of the motorship for any new construction and to meet the ever increasing needs of keen competition they will surely be tempted to adopt the Diesel engine, which is the most economical and suitable prime mover at present in existence for marine purposes.

DISCUSSION.

The CHAIRMAN: Unfortunately our Chairman of Council is unable to be present to-night but we have a communication from him which I will ask our Hon. Secretary to read.

Mr. H. A. RUCK-KEENE: We are much indebted to Engr.-Lieut.-Comdr. Le Mesurier for his most interesting and instructive paper on the Sulzer 2-stroke cycle marine engine.

Messrs. Sulzer Frères have had a wide experience with both the 2-stroke and the 4-stroke cycle engines, and therefore the Author's remarks with regard to the two types are all the more valuable.

I have often been asked the question which is raised by the Author as to which typeof engine is most reliable. Up to the present there have been far more 4-stroke cycle Diesel engines fitted into merchant ships, and, consequently, we have had far more experience at sea with the 4-stroke than with the 2-stroke cycle type.

Certain types of 4-stroke cycle Diesel engines have proved themselves to be as reliable as the ordinary reciprocating steam engine, and I see no reason why the 2-stroke cycle engine should not prove itself to be equally reliable.

As the Author states in his paper the 2-stroke cycle engine with the same number and diameters of cylinders and stroke should give nearly twice the indicated horse-power of the 4-stroke cycle engine.

Now the highest indicated horse-power on an 8-cylinder, 4-stroke cycle marine engine is about 3,000. This appears at present to be about the limit of horse-power obtainable with the 4-stroke cycle single acting engine.

To obtain a much higher power than this, as would be required for, say, fast passenger liners, it appears to follow that either a 2-stroke cycle or a double acting Diesel engine would require to be fitted to fulfil the requirements.

The only double acting Diesel engines so far fitted in a merchant ship are, as stated by the Author, those in the Twin Screw *Assyrian ex Fritz*, built by Messrs. Blohm and Voss, of Hamburg, in 1914. These engines are of the 2-stroke cycle type, but they only develop 850 brake horse-power per shaft. Other much larger double acting Diesel engines are, however, in course of construction.

The Author has gone very fully into the question of the scavenge air used in the Sulzer 2-stroke cycle engine, and it is interesting to note that by reducing the pressure of the scavenge air to 1·5 or 2 lbs. per sqaure inch and by using turbo-blowers instead of reciprocating scavenge pumps, a considerable reduction in the power formerly required for producing the scavenge air has been obtained.

The Author points out, I think quite rightly, that the reduction in the number of valves in the cylinder covers of 2-stroke cycle engines as compared with 4-stroke cycle engines enables simpler castings to be used and reduces the liability of the covers cracking through heat stresses.

Air compressors in the early days of marine Diesel engines were the source of some trouble, but the experience which has been gained in the last ten years has led makers to so improve their designs that one hears little about compressor troubles at the present time.

The building of heavy oil engines for marine purposes has been taken up by most of the marine engineering firms in this country and there certainly is a great future for these engines for merchant ships, if only the initial cost of manufacture as compared with steam engines can be reduced.

In conclusion, I should like to thank the Author for his very valuable paper.

The CHAIRMAN: I am sorry that members present are apparently reluctant to open the discussion and therefore by way of encouragement I feel bound to make a few remarks.

I feel certain that the Author would have described many additional details had there been time. As it is, he has told you about a great many interesting features of the Sulzer engines. From a personal experience with a set of these engines which the Author has mentioned in his treatise, I was very much struck by the easy way in which they were handled. I was assured that the engineer who had charge of the port engine of the vessel referred to was a steam trained motor engineer and that he had only on one occasion failed to execute the order given on the telegraph index and that failure was due to the directions being printed in Spanish, a language with which he was not familiar. He immediately discovered his mistake and speedily rectified it.

It is practically impossible for an engineer to make a mistake in the manipulation of the Sulzer marine type engines, described by the Author, owing to a number of cleverly arranged and effective devices which are fitted.

As regards the rotary blower I think Messrs. Sulzer Bros. are working on the right lines in recommending its use. It seems a great waste of space to have a large scavenging pump at the end of the main engines.

With the rotary arrangement Messrs. Sulzer Bros. are really obtaining a relatively longer piston stroke because, with the cylinder filled with fresh air at a pressure slightly above the pressure of the atmosphere, the final pressure will in consequence be higher than in the case of a 4-stroke cycle engine cylinder of similar dimensions, which starts its compression stroke with a partial vacuum.

I notice that with the exception of the Beardmore-Tosi engine, in which fresh water is used for piston cooling, the makers of Diesel engines use sea water for both cylinder and piston cooling purposes. I shall be pleased to learn whether the Author has had any experience of Sulzer-engined vessels running in muddy waters, and, if so, whether any deposit has been found in the cylinder jackets. The use of salt water for piston cooling rather leads the steam engineer to think that scale will be deposited on the hot surface of the piston core, but that is a bogey.

The temperature of the cooling water never becomes high enough to cause such scale formation, being about 150° F. at a maximum.

The crank doors of the Sulzer marine engines look large and heavy. My first impression was that they would be difficult to handle, but I was informed that they are made of aluminium and can be lifted by one man.

It was not my intention to speak on this subject to-night, but I hope these remarks will lead to a good discussion.

Mr. E. A. Evans: The Author laid special stress upon the fact, when showing the diagrams, that the scavenging port remains slightly open after the piston has closed the exhaust port. Further enlightenment on this very important point would be appreciated. The scavenging air appears to be deflected on to the exhaust side of the cylinder wall, and should therefore take a circular path in the cylinder before it is discharged through the exhaust port. Obviously a considerable amount of turbulence must result by the bombardment of the out-going air on the in-coming air. In fact it is possible that a portion at least of the products of combustion will remain in the cylinder by virtue of the circular path of the air, also a proportion of the scavenging air will be deflected directly through the exhaust without performing its function. No doubt provision has been made for this possible short-coming, but it is not obvious to the speaker, probably because he has been more accustomed to associating deflector-headed pistons with 2-stroke engines.

Mention was made in the paper that the scavenging in this type of engine is superior to that in the 4-stroke engine. No doubt this statement is based upon the result of analyses of the gases at various stages of the cycle. Will the Author kindly reveal the results of these analyses and how the gas was sampled; also with demerits of the deflector type of piston.

The system of cooling the piston appears to call for criticism in that the water impinges on one point of the piston head, whereas the opposite side is not cooled because the stream of water is deflected from it by the curvature of the under side of the piston head. Would not this system result in uneven cooling?

The subject of lubrication was very briefly mentioned, no doubt, because the paper was so thoroughly filled with other details, but it is one which is extremely important, especially as there appears to be a considerable amount of confusion about the lubrication of the Sulzer 2-stroke marine engine. One hears in one part of the country that Sulzers do not give any guidance in the type of oil to be used, whereas, elsewhere one finds comparatively thin oil being used and in another place thick oil being used. When one reads Schenker's book on "Fuel and Lubricating Oils for Diesel Engines" one is even more mystified, for he states that with an engine below 30 H.P. per cylinder an oil with a viscosity of 4-5 at 50° C. would be used, whereas engines up to 60 H.P. per cylinder, an oil of double that viscosity should be used, but on engines of largest power, the very thickest oils obtainable should be used, such as those which are suitable in superheated steam cylinders. No doubt the author will deal with this very important point in his reply.

It is very interesting to learn that the 2-stroke engine only runs at a temperature of 25% higher than that of the 4-stroke, evidently ample provision is made for heat radiation. On the first graph showing the temperature at the various points of the stroke, the exhaust temperatures, as far as I could see, were roughly 25% higher in the 2-stroke than in the 4-stroke, whereas on the second illustration the conditions were reversed. Why the difference in temperature?

I was interested to notice on the 2-stroke cover there were 18 bolt holes, whereas on the 4-stroke engine cover there were only 10. Is the slight difference of pressure between the two types of engines sufficient to demand this extra security by the increased number of bolts?

I venture to suggest the Author will revise one expression in the title of the paper, from 2-cycle into 2-stroke, because whether it be 2-stroke or 4-stroke, only one-cycle of operation has been performed.

Mr. H. S. Humphreys: I would like to ask the author how the weight of the 2-cycle Sulzer engine compares with the Cammell-Laird type, taking for example a set of say 3,000-h.p. Also how would the floor space compare in the two types? The author told us that Messrs. Sulzer Bros. had gone into the question of the opposed type and had found difficulties owing to the cutting of the liner, also owing to distortion taking place. I should like to know whether there were any other reasons for failure owing to the cutting of holes in the liner.

Mr. F. O. Beckett: In the illustration shown, the head of the piston seems a very weak design as regards the attachment to the piston rod. When you have a large mass of wrought steel attached to metal of another grade you have got to make extra allowance. With reference to the cylinder cover jointing, may I ask whether it is a type of spigot formed round the liner? The photograph does not make this clear. I see the consumption of lubricating oil is given as 24 galls. per day in a ship of 3,000 b.h.p.; this seems a large amount. Regarding the use of sea water for cooling, I see no objection to this, and if you are only raising the temperature to 160° F. the solids are not depositing themselves, but as the Chairman remarked, there is a possibility of silting up if strainers are not fitted. In many cases where steamships have occasion to sit on sandbanks, special injection has to be provided. I am in favour of the 2-cycle engine, and I fail to understand why engineers do not see the advantages of 2-cycle as against 4-cycle.

Mr. Farenden: The consumption of lubricating oil in a 3,000 b.h.p. engine, namely, 24 gallons per day, seems heavy in comparison with steam engines. I think the consumption in a 12,000 ton ship running to Australia would be prohibitive. I am disappointed that the author has not given a comparison to show what space would be saved by using Diesel engines instead of steam reciprocating engines in a ship of say 7,000-h.p.

Mr. H. Rue: Having come recently into contact with the fitting of Diesel engines in a cargo vessel of about 7,000 tons, I heard from the builder of this vessel that the excess in cost as compared with fitting ordinary steam reciprocating engines and boilers was over £40,000. This would be about a year ago, and the difference would now of course be less. As far as cargo space was concerned, the space saved was as far as I can remember about 3,500 cu. ft. Although the consumption was only about 9 tons per day, in £ s. d. there was no saving in the end, as owing to the cargo she was carrying she could never be fully loaded. It seems to me, therefore, that the steam engine still holds its own.

Mr. A. H. Mather: I should like to congratulate Mr. Le Mesurier on the excellent manner in which he has put his subject before us. The amount of information he has given us is very considerable. Like many marine engineers I have wondered why the 4-cycle engine has become so predominant, and after hearing Mr. Le Mesurier's lecture I am still more mystified to know why the 2-cycle engine has not been more generally adopted. There are so many interesting and important features of the Sulzer engines which Mr. Le Mesurier has touched upon that they would form subject matter for a long discussion. One point which interested me very much was the author's reference to the fact that the lower end of the cylinders is open to the crank chamber. We know what the result was in the 4-stroke cycle engine with cylinder ends open to the crank-chamber, the bearing lubricating oil gradually becoming contaminated with carbonised oil from the cylinder walls and having to be constantly watched to ensure that it did not get into a dangerous condition. Alterations were made in a number of existing engines to prevent this passage of carbonised oil from the cylinders into the crank chamber. I judge that on the top of the stroke the lower edge of the piston does not pass the gland ring, in other words the piston skirt works continually in the gland ring. I am with Mr. Farenden in his remarks regarding the consumption of lubricating oil in comparison with the steam engine, and I think it would be a great advantage if the present heavy consumption could be reduced to a much more reasonable figure.

The CHAIRMAN: I do not know how much time Mr. Le Mesurier can spare to-night to reply to the various speakers. Perhaps he will say how he would prefer to deal with the points raised.

MR. LE MESURIER: The meeting has already lasted so long that I do not think it advisable to reply at length, so I will just briefly refer to one or two points raised, and would like to deal more fully with the remainder in writing.

Mr. Humphreys asked for some particulars regarding a 3,000-h.p. Sulzer engine, as compared with a Cammell Laird engine of the same power, but as I have no knowledge of a Cammell Laird engine which has been constructed of this power it is impossible to make a comparison. The weight of a Sulzer engine of this power, however, would be rather less than 300 tons which, as far as I can gather from information published on certain Cammell Laird engines, will be less than the weight of a Cammell Laird engine of the same power.

In regard to space, the Cammell Laird engine will be higher and probably wider, but the overall length will be less. The saving in overall length does not appear to me, however, to be of much importance as a Sulzer engine can easily be accommodated in less than the space allowed by the 13% Board of Trade rule. Mr. Farenden also referred to the question of space and, as an example, the Sulzer machinery of a ship to develop about the same power (12,500 S.H.P. continuous service) as the ss. *Narkunda* of the P. & O. Company will be accommodated in an engine room 79 feet long.

Referring to Mr. Evans' query as to the reason for the scavenging port remaining open after piston has closed the exhaust port, the advantage of this is, that the cylinder is fully charged, or even slightly supercharged, with air, whereas in a 4-cycle engine the flow of air from outside through the valves into the cylinder implies that the pressure in the cylinder must be less than atmospheric pressure. Furthermore, in the 4-cycle engine the compression space is necessarily left filled with the products of combustion whereas in the Sulzer engine these are effectually swept out during the scavenging process. The nett result is that the Sulzer engine has a larger and purer supply of air into which the fuel is injected and this is conducive to good combustion.

Mr. Rue has evidently obtained a very unfavourable quotation for the Diesel installation to which he refers. No doubt the prices of different makers must vary pretty considerably, but it is quite certain that a Sulzer engine installation as compared with a steam installation could not possibly cost anything like the figure Mr. Rue mentions in excess of the cost of a steam installation.

Other questions raised I would like to deal with in correspondence and in the meantime I wish to thank you all for your attention and for your valued criticisms and remarks.

Mr. A. H. Mather: I should like to propose that we as members of this Institute accord a very hearty vote of thanks to Mr. Le Mesurier. His paper will form a most valuable addition to the Transactions of the Institute.

Mr. P. R. Bray: I have much pleasure in seconding the vote of thanks. We are not only indebted to Mr. Le Mesurier for his paper, but also for the excellent way in which he has explained the details to us to-night. We also thank Mr. Whiteside for working the lantern and placing the views before us.

Mr. A. C. Yeates (Contributed by Correspondence): *Sulzer 2-Cycle Diesel Engines.*—With such an interesting subject, and one which opens up such vast fields for research, perhaps more so than with any other type of Diesel engine, it seems to me more than a pity that the Author should have confined his attention to only one make of engine.

The 2-cycle Diesel engine up to 1915 had a somewhat hectic career, chiefly due to the reasons mentioned by the Author, but since then the development of the port scavenging Diesel engine has continued, and the various problems in connection with the type are being solved and overcome in different ways by several firms.

The very low temperature of the exhaust in the Sulzer engine as compared with that of the 4-cycle engine is no doubt due to the abnormal amount of scavenging air which passes out with the exhaust gases. It certainly has the effect of keeping the bars of the exhaust ports at a reasonable temperature, as pointed out by the Author, but has not this been rather overdone at the expense of efficiency? This air has to be provided, which means larger scavenge pumps than are really necessary, and a consequent reduction in mechanical efficiency.

Apart from this, I cannot understand why the mechanical efficiency of the Sulzer engine is so low. Even with separately driven scavenge pumps a mechanical efficiency of only 79% is obtained. In the case of the Nobel engine for instance, in which the scavenging ports are exceptionally well arranged, a mechanical efficiency of 80% is obtained, and that with direct driven pumps.

Re the fact that only one direct driven scavenge pump is used for four cylinders, could the Author say how the pump crank is set in relation to the cranks of the working cylinders? It would seem to me that with this arrangement there will be a pressure fluctuation which will cause one cylinder to be starved, while another will get more than its fair share of scavenging air.

The Author lays great stress on the ample area of the scavenging ports, and I would like to ask him what the relative areas of exhaust and scavenging ports are, including the auxiliary ports, and further, would like him to explain the extraordinary arrangement of the ports as shown in Fig. 6 (section of rotary valve).

With regard to the mechanical features of the engines, do Sulzer's still use the "through bolts" in all cases, or do they rely on the water jackets taking the stress due to the impulse? From the photographs of the later engines it would seem that the former method had been abandoned.

The design of the cylinder cover is an excellent feature of the Sulzer engine but has been used for many years in gas engines, and it would be interesting to know of what material they are made, presumably they are of cast steel.

By Correspondence from Mr. Le Mesurier.

The remarks made by the Chairman were of particular interest as he had visited the Motorship *Conde de Churruca* and inspected her Sulzer engine installation. He was evidently enabled by his visit to obtain a clear idea of the more essential features of the Sulzer engine. The Author would have been very pleased, if time had permitted, to describe some more of the details of the engine, but he had to confine himself to a description of the more important parts.

The Author wishes to thank Mr. Ruck-Keene for his remarks and agrees with his general summary of the Diesel marine engine situation. The fact that there are many Sulzer engines at work in which the power per cylinder far exceeds that of any 4-cycle engine, shows that for marine purposes the 2-cycle engine can also develop much higher powers than are possible with the 4-cycle engine. In point of fact, Sulzers are ready at any time to build a single engine up to 7,500 S.H.P. and unless still higher powers are demanded, Sulzer's see no reason why they should adopt a more complicated and costly engine to achieve the same results as they know can be obtained with a simple single acting 2-cycle engine.

Both the Chairman and Mr. Beckett have referred to the question of piston cooling, and while it is quite possible, and in some cases perhaps advisible, to adopt fresh water cooling so far no ill effects have been observed in various Sulzer engines using salt water cooling. There can be no question of the precipitation of salt as the temperature need never be above about 120° F. The Author readily admits, however, that when a ship is operating in notoriously sandy waters the danger will arise of sand being deposited in the water spaces and in a particular ship building at Messrs. Alexander Stephens for the British India, where she has to navigate the River Hooghly, it has been deemed advisable to instal fresh water cooling for the pistons. Messsrs. Alexander Stephens are well advanced with the construction to Sulzer designs of the engines for this ship and in the course of the present year she is likely to be in service.

Mr. Humphreys referred to the question of difficulties with the liners in engines of the opposed piston type. It has been Messrs. Sulzer's experience that cutting holes of any description, even of small diameter, seriously increases the stresses, consequently in Sulzer engines the liner in the region of the combustion chamber where it is exposed to high temperatures and pressures is perfectly symmetrical and unpierced by any holes whatsoever. In earlier designs, trouble was caused by drilling an indicator hole at this part of the liner.

The Author wishes to point out to Mr. F. O. Beckett that the attachment of the steel piston rod to the piston is quite adequate as it must be remembered that in any case the load is always downwards and there is, therefore, no tendency for the piston head to break away from the piston rod. The cylinder cover jointing is made with a spigot on the cover entering a recess in the cylinder liner; a thin copper joint at the bottom of the recess completes the joint.

The consumption of lubricating oil which is remarked on by Mr. Beckett, Mr. Evans, Mr. Farenden and Mr. Mather, is certainly much greater than in a steam engine, but possibly a shipowner will be content to put up with some extravagance in the use of lubricating oil when in a ship of this size he is saving perhaps more than 20 tons of fuel oil per day. In a recent voyage of the *Conde de Churruca* the lubricating oil consumption was, however, considerably less than that stated, and possibly in the example given the consumption might easily be reduced to 12 or 15 gallons per day. The lubrication of the rotary valve referred to by Mr. Evans only requires a very small amount of oil, say, five to seven drops per minute for each valve.

It is quite true that Sulzer's do not specify exactly what oil should be used, for the reason that in a Sulzer engine it is not necessary to adopt any particular brand of oil. The general recommendations given in Mr. Schenker's book, if carried out, are sufficient to ensure that the lubricating oil will be suitable for a Sulzer engine.

If Mr. Evans will carefully study the diagrams referred to on temperature variation and temperature of exhaust, he will find that these are quite consistent. It is quite true that under the particular conditions named the average temperature of the 2-cycle engine is only approximately 25% higher than the 4-cycle, while the exhaust temperature is as stated and as shown on the curves in Fig. 16.

It is impossible to give Mr. Evans reasons for the design of the scavenging arrangements adopted by Sulzers. It is sufficient, perhaps, to say that the design is based not on a few isolated experiments, but on a vast amount of experience with every conceivable arrangement, including valve head scavenging. Actual results are, however, the most important and whether there are exhaust products or not in the cylinder, the fact remains that they cannot cause any ill-effects, seeing that the fuel economy of the Sulzer 2-cycle engine is good, while the mean pressure is even higher than is advisable in a 4-cycle engine.

I quite agree with Mr. Evans that the correct expression is "2-stroke cycle," but to abbreviate this conveniently the term "2-cycle" seems to me preferable, and is more commonly adopted than "2-stroke."

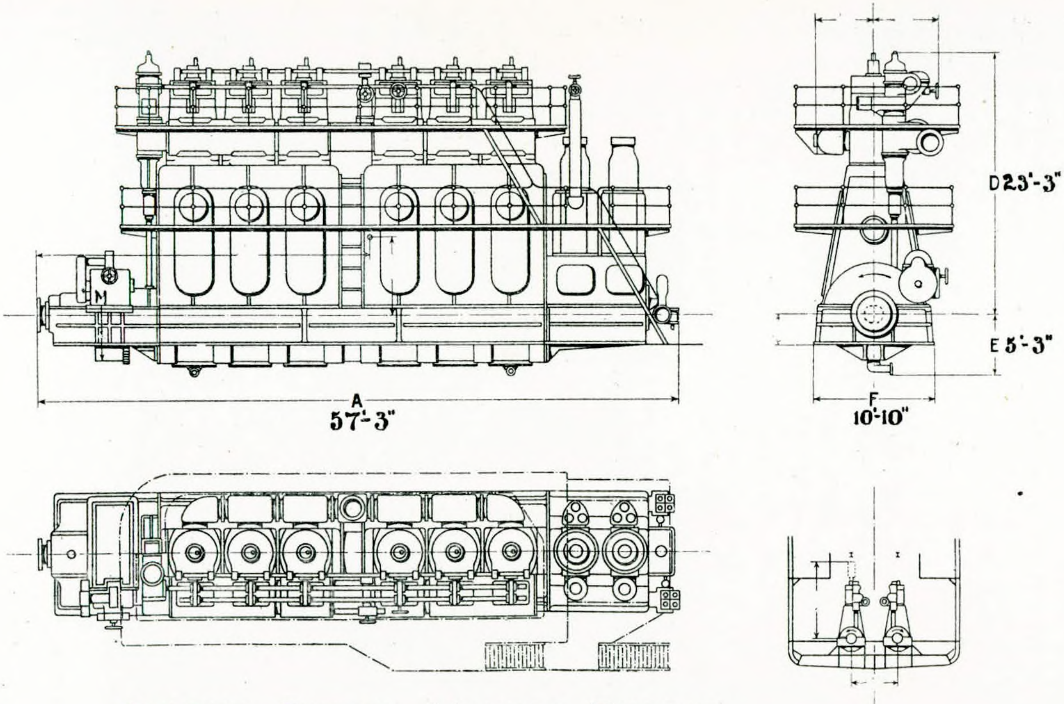
With regard to the cylinder cover, the reason that there are only ten bolts in the 4-cycle cover is that it is impossible, owing to the presence of the inlet and exhaust passages, to obtain a better distribution of bolts. Although there are more bolts shown for the 2-cycle cover, they are smaller in diameter and their distribution is naturally advantageous in securing a good joint.

To comply with Mr. Farenden's request and because it was certainly an omission from the paper, the Author shows in Fig. 46 a Sulzer marine engine of 3,800 B.H.P. from which a good idea can be obtained as to the space occupied by two such engines which would correspond to a steamship of over 8,000 I.H.P. The engine shown is of the separate turbo scavenging pump type. developing 3,800 B.H.P. at 90 revs. per minute; the weight of engine complete with fly-wheel, silencer, thrust-block and thrust-shaft, is about 408 tons.

The Author thanks Mr. Mather for his kind remarks, and with regard to the question of lubrication, Mr. Mather is quite correct in his assumption that the lower edge of the piston skirt works continually in the gland rings, and thus the cylinder is effectively isolated from the crank chamber.

The Author agrees with Mr. Yeates that the subject of 2-cycle Diesel engines is a wide one and for that reason it is quite impossible to deal adequately with all types of engines; the paper was, therefore, confined to the Sulzer 2-cycle engine as representing the engine with the maximum amount of experience behind it.

The low temperature of exhaust is, of course, primarily due to the large supply of scavenging air, but that it is not at the expense of efficiency is clearly shown by the fact that the Sulzer



SULZER DIESEL MARINE ENGINE,
3,800 B.H.P. Fig. 46.

2-cycle engine consumes only ·40 to ·42 lbs. of fuel per B.H.P. per hour. The Author is not aware of any other 2-cycle engine which obtains on the average a better fuel consumption than this.

The figure stated for mechanical efficiency in the Nobel engine, if correct, is certainly remarkable—it would be still more remarkable if given as 90% or 100%—but it is hardly fair to compare the performance of a single engine on the testbed with average performances of hundreds of Sulzer engines in service.

It is difficult to believe that the Nobel engine has a higher mechanical efficiency than the Sulzer engine, seeing that the design is so similar that it would appear to come into conflict with the Sulzer patent scavenging process described in this paper.

No difficulty arises in regard to pressure fluctuations in the scavenge receiver as this is of ample volume to counteract such an effect. The cranks are arranged to secure the best balancing of the reciprocating and rotating masses.

The design of scavenging ports has been based on a large amount of experiments and results of actual practice. The arrangement of ports which Mr. Yeates criticises merely shows the ports in the jacket and the liner ports are, of course, quite different.

Sulzers have abandoned "through bolts" in their Diesel marine engines, and the stress is taken through the jackets, this arrangement having been found by experience to be simpler and superior to the old method of "through bolting."

The material of the cylinder covers is cast iron, and the Author suggests that if Mr. Yeates examines the design more closely he will find several important features which are adopted exclusively in the Sulzer engine.

With regard to the *Conde de Churruca*, mention was made at the meeting of certain vibration which had been experienced in the thrust blocks during a recent voyage. Investigation proved that this was due to the critical speed of the propeller shafting almost exactly corresponding with the engine impulses at normal running speed—about 95 revolutions per minute. The critical speed being dependent simply upon the rotating masses at each end of the shaft (*i.e.*, the propeller and rotating masses of engine), and also the length and diameter of shafting, it was found necessary to raise the critical speed by increasing the diameter of one length of intermediate shaft. This has now been done, bringing the critical speed to 124 revolutions per minute, and there is now no chance of torsional oscillations occurring in the shafting. A full power run from Newcastle to London on 22nd February confirmed that vibrations had completely disappeared.

Another alteration of quite minor importance was also effected in the *Conde de Churruca* by adding an additional water-cooled silencer to avoid any chance of a stray spark escaping with the exhaust. This precaution was deemed essential as the ship has to carry benzine, and it is obviously easily and effectively provided for, where required, by suitable exhaust arrangements.

The Author thinks it well to remark on both these points, as the mere mention of a motorship returning to a building for alterations gives rise to the most fantastic rumours as to all sorts of troubles which are usually alleged to be due to engine defects.

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Notes.

A NEW STEEL.—The following is reprinted from the "Engineer and Iron Trades Advertiser" of February 28th, with due acknowledgments:—

During the reading of a paper before the Newcastle Foremen and Draughtsmen's Association, Mr. H. H. Ashdown, of Messrs. Armstrong, Whitworth, announced the discovery of a new non-temper brittle steel by his firm. Temper brittle disease, said Mr. Ashdown, was a problem which had for the past 15 years been receiving the most serious attention of metallurgists the world over. During the great war, the greatest difficulty met with by the Air Board was this temper brittle disease in the alloy of steels employed in the manufacture of crankshafts. This disease was the cause of the scrapping of a tremendous amount of valuable material, and, incidentally, of serious delay in the manufacture of aeroplane engines. The lecturer stated that the new "Vibrac" steel overshadowed any of the modern alloy steels in its general mechanical, and physical properties. The general alloy steels were inconsistent, and often, even under the most careful treatment, unreliable on their impact properties. "Vibrac" steel had already been produced in great quantities, and under works conditions and under any normal treatment it was impossible to make it temper brittle. This new steel now placed the firm in an ideal position for the manufacture of all highly-stressed parts in machinery, such as were required by aeroplanes, automobiles, railways, etc. Mr. Ashdown dealt very explicitly with many of the difficulties met with in works practice in regard to the treatment and machining of steels. Steel manufacturers now realised that in order that their customers should obtain the best results from material, they must impart the best information possible concerning its treatment and general manipulation. In conclusion, the lecturer ventured a few words of warning regarding foreign competition, and illustrated excellent examples of forgings, castings, and general machinery with which he was associated during his service in Japan. His final remarks were—" For us to regain our markets, we must search every avenue for better, quicker, and cheaper production."

The following abstract of a paper read February 11th before the Mining Institute of Scotland by William Brasenale on " Pipe Friction and Pump Efficiency " is reproduced by the courtesy of " The Iron and Coal Trades Review " :—

The experiments described were carried out to the instruction of Mr. Wallace Thornycroft, and it is with his and Mr. Walter G. Gray's permission that the results are used. Originally the intention was simply to find the cost of pumping worked out in terms of actual horse-power, for three types of pumps:—(1) Turbine; (2) three-throw ram; (3) differential ram. It was seen that while collecting data for these cost figures, other information could be collected. The conditions were excellent, *i.e.*, cool, dry, and clean pump rooms, large lodgments giving clean water, electrically and mechanically well looked after. The ages of the pumps vary from 8 to 12 years, and they were in as good—possibly better—condition as the average pumps seen at a modern colliery. In a word, these were accurate tests of pumps under *practical working conditions*.

TURBINE PUMPS.

Tests made December 5th, 1918.

Type.—Three turbine pumps coupled up in parallel; three cells; specified for 300 g.p.m.

Motors.—50 h.p. inter-pole shunt-wound.

Pipe Column.—248 ft. of 7 in. dia. steel pipes.

Frictional Losses.—Calculated from D'Arcy's formula.

(1) To find cost of pumping worked out in terms of actual horse-power of water pumped, *i.e.*, 1,000 lbs. of water raised 33 ft. in one minute.

Conditions of Test.—The recording wattmeter was coupled up to the positive main leading to these pumps.

This break in the cable into which the recording instrument was connected was made at the back of the main switchboard in the power station. Nos. 1, 2 and 3 pumps were run together for a certain time; then Nos. 1 and 2 pumps were run together, and finally No. 2 pump was run solo.

An abstract of the test figures is given in Table I.

TABLE I.—*Abstract of Pump Tests Figures.*

Type of Pump.	Galls. per min. pumped.	Head in feet.	Velocity and fractional head.	H.P. in water.	E.H.P.	Cost per h.p. of water raised.	Eff per cent. including all losses.	Eff. per cent. of pump and motor excluding pipe friction.
Turbine, Nos. 1, 2 and 3	755	166	30·5	37·98	125·3	0·0373d.	30·3	35·8
Turbine, Nos. 1 and 2 ...	660	166	18·4	33·20	97·7	0·0333d.	33·94	37·6
Turbine, No. 2 pump ...	350	166	6·2	17·60	45·52	0·0293d.	38·7	40·0
Three-throw ram, 6¼in. by 9in. dook pump ...	147·5	275	—	12·29	24·61	0·0225d.	49·9	Including power transmission.
7in. by 5in. by 10 in. differential ram ...	135	398	—	16·28	29·49	0·0205d.	55·2	Excluding power transmission.

(2) To find the E.H.P. absorbed in power transmission and in frictional losses in motor and pump.

Conditions of Test.—An ammeter was coupled up at the back of the main switchboard in the power station to the positive main leading to the pumps. The cover of the delivery valve in the pipe column at the pumps was unscrewed, and the water between this valve and the clack-valve run off. The stop-valve cover was left off. The cocks on the top of pump were opened, the result being that after the first five minutes' running, the pump was doing no work, the water remaining in the pump acting as a lubricant. During the test runs, readings were taken at the ammeter every two minutes, and the voltage was kept steady at 500 on the main switchboard. Speeds of the pumps were also taken. Results were as follows:—

Nos. 1 *and* 2 *turbine pumps running together.*—Taking average current over last ten minutes:— = 30·0 amps.

Speeds:—No. 1 average r.p.m. = 1,541·6 No. 2 average r.p.m. = 1,659·6.

30·0 amps × 500 volts = 15,000 watts. $\frac{15,000}{746} = 20$ e.h.p.

Total e.h.p. when Nos. 1 and 2 were pumping	= 97·8
Total e.h.p. absorbed in power transmission, motor and pump	= 20·0
Total e.h.p. used in water from pump to outfall ...	= 77·8
Actual water h.p. put out when Nos. 1 and 2 were pumping	= 33·2
E.H.P. lost in overcoming frictional head and velocity head	= 44·6

No. 2 *turbine pump running solo.*—(In this case the whole period may be taken because surplus water had been drawn out of pump during last test.) Average current = 9·93 amps. Speed = 1,673·3 r.p.m. 9·93 amps. × 500 volts = 4,965 watts. $\frac{4,965}{746} = 6·65$ e.h.p.

Total e.h.p. when No. 3 was pumping solo	= 45·52
Total e.h.p. absorbed in power transmission motor and pump	= 6·65
Total e.h.p. used in water from pump to outfall ...	= 38·87
Actual water h.p. put out when No. 2 was pumping ...	= 17·60
E.H.P. lost in overcoming frictional head and velocity head	= 21·27

Nos. 1, 2 *and* 3 *running together.*—Taking average current over last ten minutes:— = 35·5 amps.

Speeds:—No. 1 = 1,564, No. 2 = 1,652·5, No. 3 = 1,737·5 av. r.p.m.

35·5 amps. × 500 volts = 17,750 watts.

$\frac{17,750}{746}$ = 23,90 e.h.p.

Total e.h.p. when Nos. 1, 2 and 3 were pumping ...	=125·30
Total e.h.p. absorbed in power transmission, motor and pump	= 23·90
Total e.h.p. used in water from pump to outfall ...	=101·40
Actual water h.p. put out when Nos. 1, 2 and 3 were pumping	= 37·98
E.H.P. lost in overcoming frictional head and velocity head	= 63·43

THREE-THROW PUMP.

Test made March 10*th*, 1919.

Motor.—30 h.p. shunt-wound.

Conditions of Test.—A water-measuring tank was rigged up at outfall of this pump, and wattmeter was coupled up at back of switchboard in power station. Readings were taken at the tank every five minutes.

Because of the position of the wattmeter in the circuit, *i.e.*, at generators, the power used in transmission was recorded as well as the power used in overcoming frictional resistances of motors, pumps and pipes. The head recorded in Table I. is the actual head, and does not include frictional head or velocity head.

DIFFERENTIAL RAM PUMP.

Test made April 1*st*, 1919.

Motor.—30 h.p. shunt-wound.

Conditions of Test.—The recording wattmeter was coupled up at the back of the switchboard in the pump room. The amount of water pumped was measured at the water-measuring tank on surface, measurements being taken every three minutes. Because of the position of the wattmeter, *i.e.*, at the pump, the power used in transmission was not recorded. Abstract of figures given in Table I.

Election of Members

Members elected at the meeting of the Council held on March 13th, 1922:—

Members.

Matthew Nimmo Caird, 777, Commercial Road, Limehouse, E.
Sidney Edward Cheesewright, Dept. of Navigation, Sydney, N.S.W.
Alexander Chalmers Cockburn, 29, Ampthill Road, Fulwood Park, Liverpool.
John Stuart Devlin, 216, Anniesland Road, Glasgow.
Arnold William Gibson, 22, Gladstone Street, Hull.
Arthur Hardie, 12, St. John's Road, Penge, S.E.20.
John William Harper, 23, Burbage Road, Herne Hill, S.E.24.
James Jones (Engr.-Lieut, R.N.), H.M.S. *Vectis*, 2nd Flotilla, Atlantic Fleet, c/o G.P.O.
John Benjamin Jones, c/o Messrs. Lim Soo Hean & Co., Rangoon.
Thomas Herbert Minshall, 34, Victoria Street, S.W.1.
Alfred J. Prince, 22, Westridge Road, Southampton.
Roland John Skidmore, Dept. of Navigation, Sydney, N.S.W.
Johnstone Russell Thompson, 15, Adolphus Street West, Seaham Harbour.
James Thomson, c/o Messrs. C. A. Hunter & Co., Ltd., Works Bungalow, Colombo.
Thomas Stansfield Worthington, 40, Hospital Street, Montreal, Quebec, Canada.
Albert Edward Morley, Richmond House, Rodborough, Stroud, Glos.

Associate-Members.

Reginald Henry Chanter, 48, Micheldever Road, Lee, S.E.12.
Archibald Elliott, 48, Marlboro' Flats, Walton Street, Chelsea, S.W.3.
Herbert Goode, 288, Burdett Road, E.14.

Associate.

William Henry Hodgson, 175, North Road, Wallsend-on-Tyne.

Graduates.

Fred Alfred Bates, Dovedale, Rectory Road, Grays, Essex.
James Blyth Bews, 10, Northumberland Villas, Wallsend-on-Tyne.
William Jefferson Bradley, 84, Grange Road, West Hartlepool.
Arthur Byron Milbanke, 1, Pelaw Cottage, Pelaw Main-on-Tyne.

Edgar Conway de Waterford, 27 Baker Street, Lloyd Square, W.C.1.
George Chadwick, 24, St. Mary's Road, Peckham, S.E.15.
Edward Shaw, 88, Inverness Place, Roath, Cardiff.

Student-Graduate.

George Waters, 28, Oaklands, Gosforth, Newcastle-on-Tyne.

Companion.

John Moffett Peterson, 5, Quayside, Newcastle-on-Tyne.

Transfers.

From Associate-Member to Member.

Leonard C. Cox, 20, South Hill Road, Gravesend.
J. W. O'Brien (Commd.-Engr., R.N.), H.M.S. *Stork*, Chatham.

From Associate to Associate-Member.

T. P. Palmer, 105, Pepys Road, New Cross, S.E.14.