SOME ASPECTS CONCERNING THE SUPERCHARGING OF EXISTING TWO-STROKE MARINE DIESEL ENGINES

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The greater part of the paper is devoted to the basic problems encountered in supercharging two-stroke Diesels. In this connexion, the principal methods of supercharging are described and their relative merits discussed, examples from the basic research work in this direction being cited. The second part of the paper presents results obtained with a number of supercharged two-stroke test engines of Sulzer design. In closing, a summary of the performance characteristics of these engines is given.

Supercharging has become the predominant subject of discussion wherever users and suppliers of marine engines meet. It is by now common knowledge that this problem, particularly for the two-stroke engine, absorbs the greater part of the development efforts of nearly every builder of heavy oil engines. It may therefore be appropriate to discuss here questions solely pertaining to this subject.

Before looking at the various methods of supercharging two-stroke engines, it may be asked why it took this type of high-output engine so much longer than its four-stroke counterpart to appear on the market. One reason for this is the fact that a two-stroke engine, having double the number of working strokes, gives a much higher heat transfer to the cylinder walls and the piston, thus setting a limit to the maximum output. The supercharging of large two-stroke engines only became feasible, therefore, after methods of supercharging had been found, which allowed a moderate increase of the mean effective pressure whilst still keeping the thermal loading within reasonable limits. These methods include the cooling of the scavenge air after its compression, and layouts which result in a reduced specific fuel consumption.

In the two-stroke engine, it is also much more difficult to obtain a sufficiently high output from a free running turbocharger group; the latter may be defined as a turbocharger which is driven exclusively by the energy derived from the engine exhaust gases, so that the turbine output must be capable of balancing the input required by the blower for providing sufficient scavenge air at all operating loads.

Basically, the turbocharger can be regarded as an open-cycle gas turbine, in which the combustion chamber is replaced by the engine. The optimum running conditions for such an installation are obtained by having low pressure losses through the engine and a high temperature of the exhaust gases driving the turbine. Here the two-stroke engine has a great handicap

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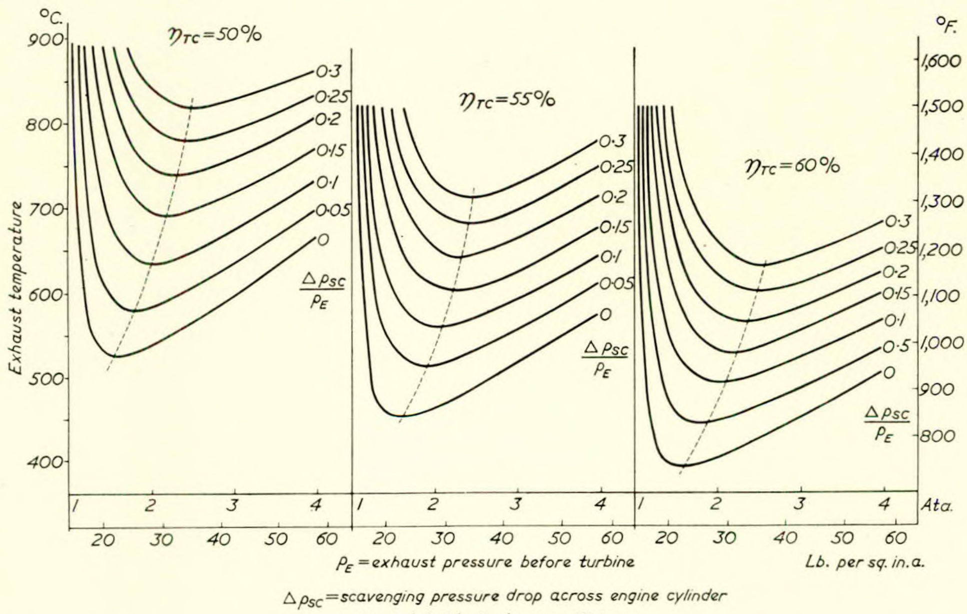


FIG. 23—*Exhaust temperatures required for operation of turbocharger on the constant pressure system*

Assumptions: Ambient pressure 28·9in. h.g.; ambient temperature 90 deg. F.; combined pressure losses through air and exhaust ducts 1·0lb. per sq. in.

in that the pressure drop necessary for scavenging reduces the output of this fictitious gas turbine and the exhaust gas temperature is much lower than in a four-stroke owing to the limited thermal loading of the engine and the additional cooling of the already cooler exhaust gases by dilution with a large excess of scavenge air. Fig. 23 shows the influence of the various parameters—such as supercharging pressure, the ratio of scavenge pressure drop to supercharging pressure (which is a direct measure of scavenge-air flow) and turbocharger efficiency—upon the exhaust temperature required for self-sustaining turbocharger operation. It is quite interesting to note that there is such a thing as an optimum supercharging pressure for a given set of conditions, which allows the engine to run at the lowest possible exhaust temperature. The effect of the turbocharger efficiency is also quite remarkable, since an improvement from 50 per cent to 60 per cent efficiency results in a drop of as much as 280 deg. F. in the required exhaust temperature.

Still another difficulty arises from the fact that the two-stroke engine is not normally self-aspirating and, therefore, starting and operation at low loads are problematic.

As a result of many years of intensive search by a great number of investigators and inventors, a number of practical supercharging schemes have been evolved. The generally accepted arrangements for the provision of the supercharging air are shown in Fig. 24.

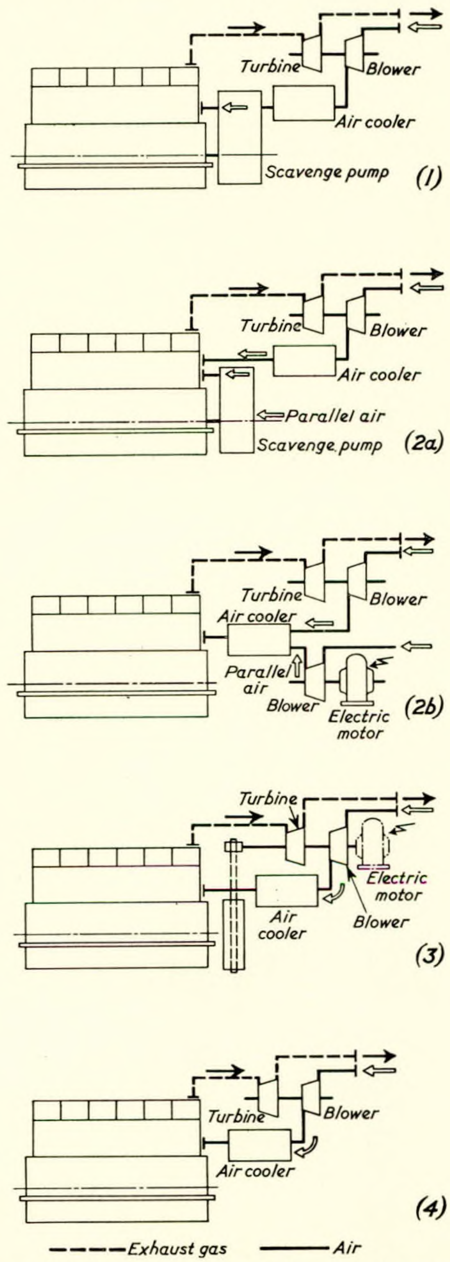


FIG. 24—*Two-stroke supercharging diagram for the arrangements of scavenge pump and turbocharger*

(1) Curtis arrangement of scavenge pump.
(2a) Parallel arrangement of small scavenge pump.
(2b) Parallel arrangements of small additional blower.
(3) Without scavenge pump: turbocharger with supplementary drive
  (a) from engine shaft
  (b) by electric motor.
(4) Without scavenge pump, turbocharger running free.

1. Curtis Arrangement

The scavenge pump of the normal two-stroke machine is retained and the engine supercharged with a free running turbo-

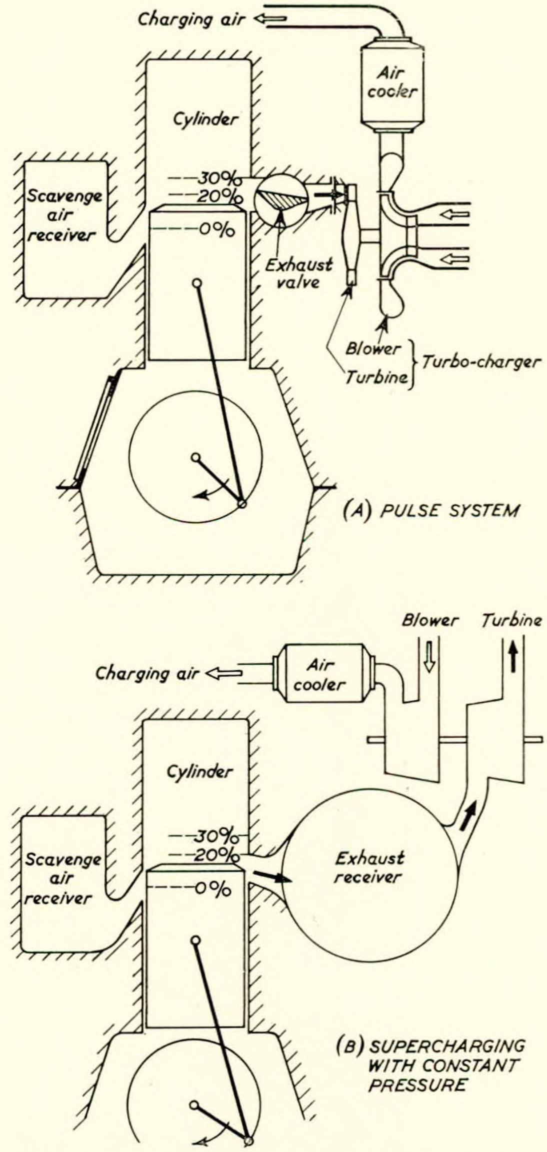


FIG. 25—*Two-stroke supercharging; arrangement for (a) pulse system; (b) constant pressure, on turbine side*

charger; the latter, being in series with the scavenge pump, takes over only part of the total compression of the full quantity of scavenge air.

Advantages: With this arrangement, named after Curtis, who first patented it, the engine behaviour at all loads, including starting and manœuvring, is similar to that of a non-supercharged engine.

Disadvantages: The fuel consumption in comparison with other arrangements is not so good, the load on the scavenge pump being only slightly reduced, especially when the engine is lightly loaded. The size of the scavenge pump is only a little below that required for a non-supercharged engine.

2. Parallel Arrangement with a Small Scavenge Pump

The output deficit of the turbocharger may be made up by a small scavenge pump operating in parallel with the blower (Fig. 24, arrangement 2(a)). In this way, both supply part of the required scavenge air quantity and compress it up to the total scavenge pressure. In Fig. 24, arrangement 2(b), the small engine-driven scavenge pump is replaced by a separately driven blower.

Advantages: Compared with the Curtis arrangement, both the turboblower and the scavenge pump will be reduced in size. The smaller scavenge pump or auxiliary blower requires less energy, which results in a better fuel consumption.

Disadvantages: Starting and manœuvring, where no special provisions are made, are not as good as with the Curtis layout.

3. Turbocharger with Supplementary Drive, Engine Without Scavenge Pump

A further possible means of overcoming the output deficit of the turbocharger is to supply it directly with additional energy, either straight from the crankshaft through step-up gears or by means of an auxiliary electric motor.

4. Engine Without Scavenge Pump, Free-running Turbocharger

All the arrangements mentioned previously require additional equipment and thus complicate operation of the installation. The aim will always be to make possible the set-up shown in Fig. 24, arrangement 4, i.e. to use only a free-running turbocharger and omit the heavy and expensive scavenge pump, thus improving the fuel consumption. This optimum arrangement is only possible when the turbocharger is capable of providing sufficient scavenge air at all loads. In order to achieve this result, all possible means have to be employed. These include extracting the maximum energy in the most efficient turbocharger while avoiding as far as possible all losses of pressure and temperature, and choosing the most favourable supercharging pressure.

On the exhaust side two principal arrangements are possible, as shown in Fig. 25. Either as much as possible of the gas energy in the cylinder can be utilized, exhausted into a large receiver in order to damp out pulsations and drive the turbine in the most efficient way with a nearly constant gas pressure, or as much as possible of the gas energy contained in the cylinder at the opening of the exhaust ports can be transmitted in the most direct manner to the turbine and used there with a somewhat lower turbine efficiency.

Which of the two schemes is more favourable depends on the particular features of each supercharging problem. The constant pressure arrangement is usually to be preferred for high supercharging pressures, since in this case the proportion of energy contained in the gases at constant pressure is so great compared with the blowdown, that this energy must be utilized at the maximum turbine efficiency. This arrangement is also more easily adapted to existing engines. Any workable two-stroke scavenging system may be modified to suit supercharging. It usually requires a somewhat larger exhaust receiver, well insulated against heat losses. The turbo-group may be installed in any convenient place.

On the other hand, the blowdown or exhaust pulse system is more exacting in its requirements on the engine. The port timing has to be unsymmetrical, as is the case for an engine with an oscillating valve in the exhaust duct; the exhaust ports have to be higher up in the cylinder since the expansion during blowdown is slower owing to the back-pressure effect caused by

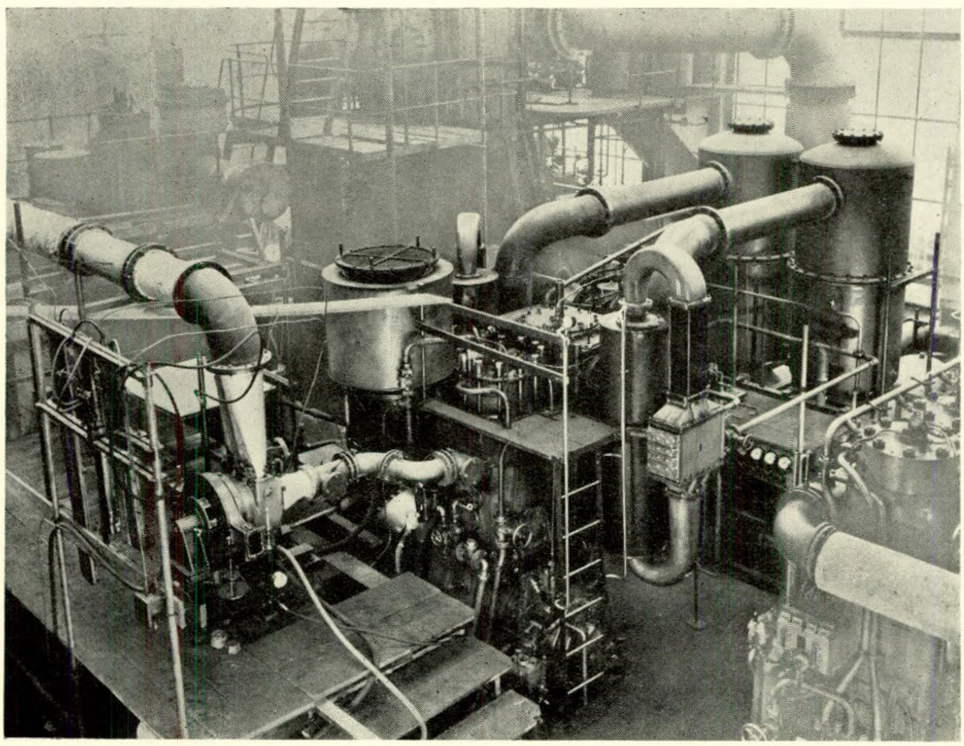


FIG. 26—*Single-cylinder two-stroke test engine of 42 cm. bore*

the turbine nozzle. The higher position of the exhaust ports cuts down somewhat the expansion energy transmitted to the piston, but—and this is a very important point—it supplies a larger proportion of the gas energy to the turbine. To deliver as much of this energy to the turbine as possible, the exhaust passages should be kept as small in volume as possible. Furthermore, interference of the pressure pulses from individual cylinders must be completely avoided. Thus more than one turbine has to be used on engines with more than four cylinders, which complicates the installation.

In order to learn more about the proper size and shape of the exhaust ducting for the pulse system and to get a better idea about the relative merits of pulse as against constant-pressure systems, the author's company have undertaken some research work in this direction. One of the test sets (Fig. 26) designed to answer the above questions consists of a single-cylinder two-stroke engine with a bore of 42 cm., a stroke of 50 cm. and an exhaust turbine connected to an air brake This engine is fitted with cross scavenging and the timing curve for the opening and closing of the ports is symmetrical, no exhaust valve being provided. Fig. 27 indicates the location of the various measuring devices. Measurements of the

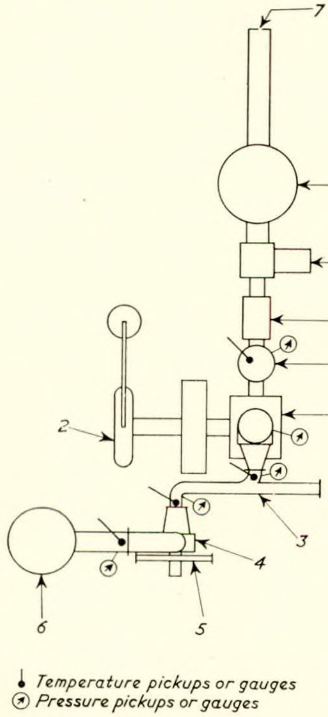


FIG. 27—*Experimental set-up for analysing the pulse system for supercharging two-stroke Diesel engines*
(1) One-cylinder two-stroke Diesel engine; (2) Water brake; (3) Exhaust duct of variable length and shape; (4) Exhaust gas blowdown turbine; (5) Air brake; (6) Exhaust stack; (7) Orifice for measuring scavenging air flow; (8) Surge tank; (9) Scavenging air compressor; (10) Scavenging air cooler; (11) Scavenging air receiver.

fluctuating pressures in the scavenging receiver, the engine cylinder, at the cylinder outlet and before and after the turbine were taken simultaneously by means of quartz pick-ups. The latter were connected to a multi-channel recording oscillograph. As an independent check, a Cox indicator was used. The gas flow through the entire system was measured by means of a calibrated intake orifice. A set of typical pressure curves from one of the test runs is shown in Fig. 28.

On the basis of the results obtained, the potential energy in the exhaust gases for pulse operation was calculated. In this case, the potential energy is composed of a blowdown and

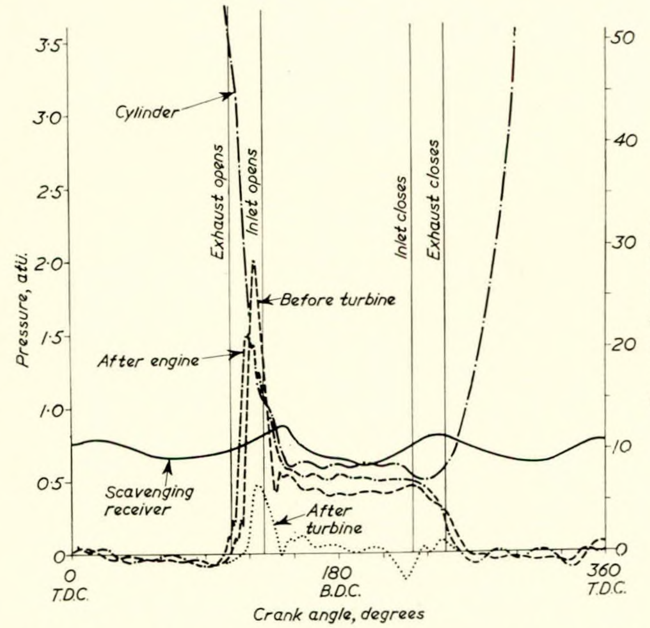


FIG. 28—*Pressure fluctuation at different points of the induction and exhaust system (obtained with pulse system)*

a constant pressure portion, the latter occurring during the scavenging period. For comparison purposes, the potential energy of constant pressure operation at equivalent engine conditions has also been calculated.

Fig. 29 shows an energy balance comparing the two systems. On this must be said that quite a considerable part of the potential energy in the exhaust is lost with pulse operation through throttling effects and turbulence in the piping system between engine and turbine. The so-called "availability

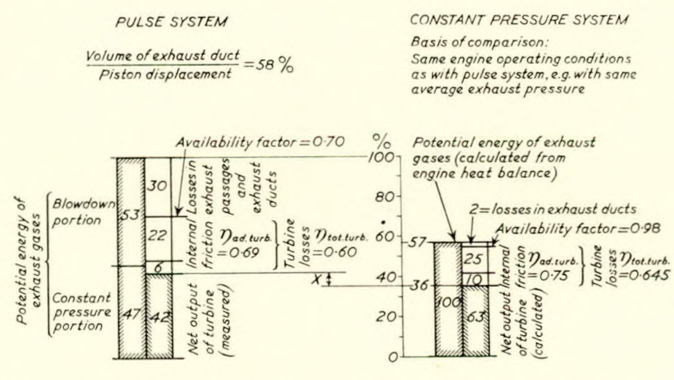


FIG. 29—*Utilization of exhaust gas energy for different exhaust systems*
X = gain of turbine net output with pulse system as compared with turbine net output with constant pressure system (15 per cent)

factor" represents the ratio of the mechanical energy obtained from an ideal exhaust turbine to the potential energy of the exhaust gases. It is, therefore, a direct measure of the losses in the piping system which leads to and from the turbine. Further losses are caused by the turbine itself. These are larger with the pulse system than with the constant pressure system. On the test turbine these losses were carefully determined from the efficiency curve as measured during constant pressure operation. In order to apply these results to intermittent flow conditions, it was necessary to divide the pressure curve versus time into small intervals, each representing a state of practically constant pressure. It was therefore possible, from the foregoing test, to assign a corresponding turbine efficiency to each interval. The qualified average value of these efficiencies was assumed to be representative for the efficiency of the blowdown turbine.

Finally, the bearing friction and windage losses were accurately determined for the turbine by measurements and checked by calculations. These losses are not insignificant, particularly in single-cylinder operation, since the turbine is running idle for 60 per cent of the time.

Now from Fig. 29 it may be seen that for the typical low pressure supercharging arrangement as tested, only 42 per cent of the total potential energy in the exhaust gases was left as useful turbine output when operating on the pulse system. Together with a total turbine efficiency of 60 per cent, this results in an availability factor of 70 per cent. The respective figures for the constant pressure system have been calculated and found to be: available turbine output 63 per cent, total turbine efficiency 64·5 per cent and "availability factor" 98 per cent. However, if the two systems are compared, it is seen that the potential energies contained in the exhaust gases are quite different. Owing to the complete dissipation of the kinetic energy contained in the blowdown portion, as is the case with the constant pressure system, this difference amounts to some 43 per cent. This explains why the net output of the constant pressure turbine falls 15 per cent short of the pulse turbine output.

Another method of determining the "availability factor" consists of measuring the temperature and pressure fluctuations ahead of the turbine (or throttling orifice for that matter) and the pressure behind it. With the knowledge of the flow characteristic of the turbine or throttling orifice, plus the above data, it is possible to compute the power which an ideal turbine would develop. Furthermore, these measurements, in conjunction with a dynamometer-loaded turbine, permit the overall turbine efficiency to be determined directly.

Unfortunately, none of the available temperature measuring devices provides a response fast and accurate enough to follow the rapid temperature fluctuations within the exhaust ducts. Therefore it was necessary to develop a special termocouple with wires measuring as little as 0·05 mm. (0·002 inch) in diameter. Particular care had to be taken during the welding operation to avoid an increase in wire diameter at the junction. Measurements taken before and after the turbine with such thin wire thermocouples are shown in Fig. 30.

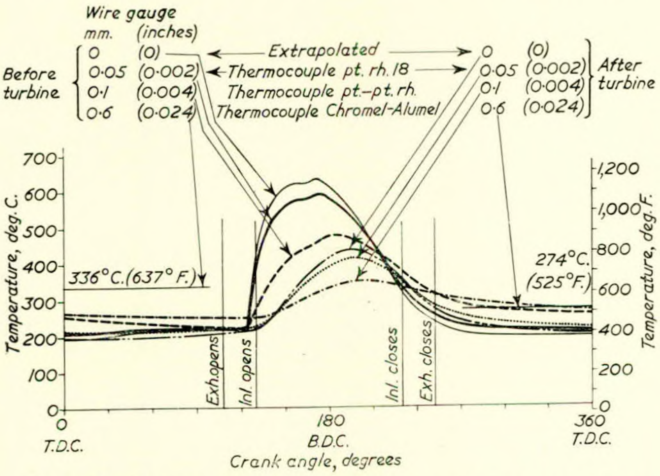


FIG. 30—*Temperature fluctuations of exhaust gases before and after turbine (obtained with pulse system)*

Whilst a wire thickness of 0·1 mm. (0·004 inch) obviously gives too sluggish a response to the temperature changes, there is very little difference between the temperature curve of the 0·05 mm. thermocouple and that of the extrapolated value for a wire thickness of 0 mm. The curves show that the exhaust gas temperature before the turbine rises rapidly from its nearly steady value of about 255 deg. C. to 625 deg. C. as a result of the exhaust pulse and then falls off again. For the future it is planned to carry these tests one step further with check measurements using wire of 0·03 mm. (0·001 inch) diameter. It is also planned to provide this wire with some form of protection against the deleterious influence of the exhaust gases upon the electrical properties of the junction, without introducing undue delay in indication.

It was somewhat surprising to find that the length of the exhaust duct between engine and turbine had no apparent effect on the "availability factor". A possible explanation may be drawn from Fig. 31. Diagram *A* comprises both a pressure-time and temperature-time record, taken with a pipe length of

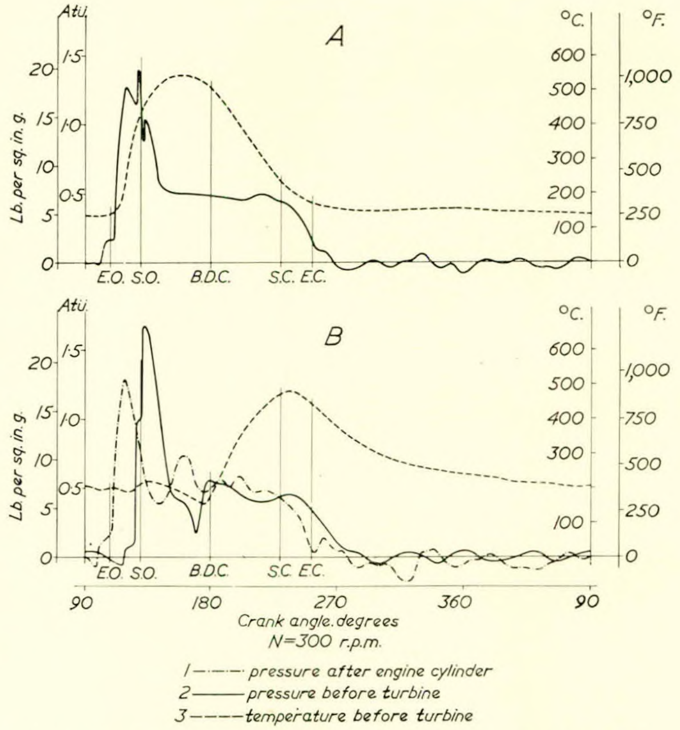


FIG. 31—*Pulse system supercharging tests: pressure and temperature records*

3·5 diameters, whereas *B* shows similar records from a pipe 32 diameters long. As expected, the phase-shift between pressure and temperature peak increases with pipe length. However, even with this very big distance between engine and turbine, the region of elevated temperature remains pretty much within the scavenging period of one cylinder. Although in case *A* the regions of high pressure and high temperature overlap to a certain degree, the resulting gain is nullified by the relatively low temperatures prevailing during the later part of the scavenging period. Of course, the actual situation is considerably more complex than this explanation suggests, since not only the momentary enthalpy differences, but also the instantaneous gas flow must be taken into consideration. However, it still helps to illuminate the interplay of the physical properties involved.

After this outline of the basic problems of two-stroke supercharging, it may be opportune to say something about the recent supercharging experiments with various Sulzer engines. The purpose of the tests described in the following paragraph was to prove conclusively whether each of the existing engine types could be successfully supercharged. It was assumed that only minor modifications would be necessary to obtain power gains of up to 40 per cent.

The most extensive tests were carried out on a single-cylinder engine of 58 cm. bore and 76 cm. stroke with a full-load speed of 240 r.p.m. This engine is representative of the modern Sulzer crosshead diaphragm marine engines. A cross-section through the non-supercharged version of this engine type may be seen in Fig. 32, whereas Fig. 33 shows a view of the actual test engine equipped with a Brown Boveri turbo-

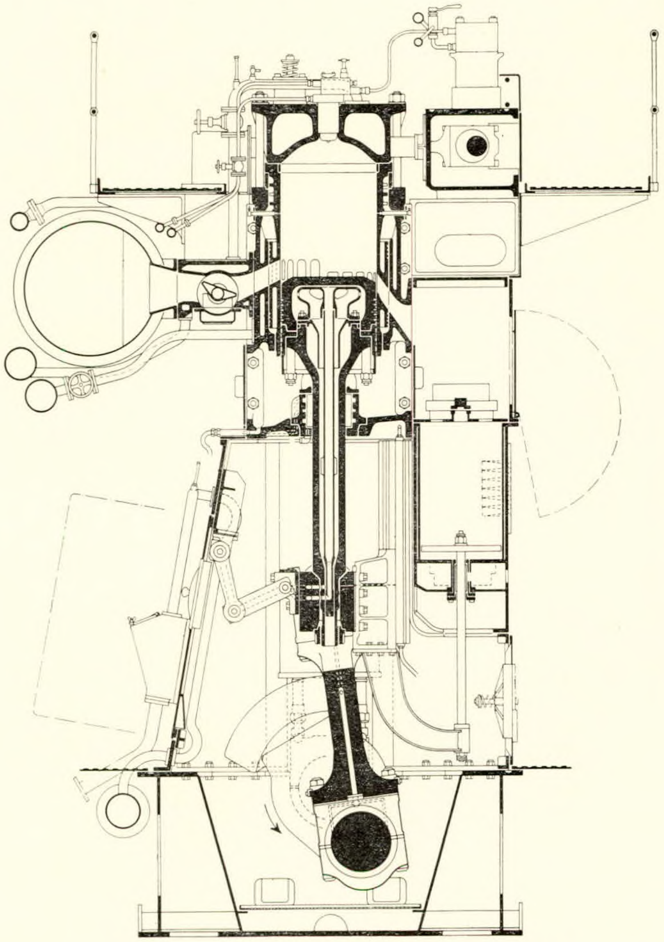


FIG. 32—*Cross-section through non-supercharged crosshead diaphragm marine engine of 58-cm. bore*

charger. Since this kind of engine is fitted with an oscillating exhaust valve, most of the test work was concentrated on the pulse-type supercharging system in order to make the highest possible use of the blowdown energy contained in the exhaust gases.

Quite early in the experiments with this engine, it was possible to demonstrate that it could be started, accelerated and reversed as well as a non-supercharged engine without the aid of a scavenge pump or an auxiliary blower. In addition, the load carrying capacity as well as the fuel consumption was very satisfactory, as may be seen from Fig. 34. It may be noted that a saving of over 0·018lb. per b.h.p./hr. in fuel consumption is obtained with the exhaust temperature unchanged. The absolute values of the fuel consumption are still somewhat high, but this is due to the fact that a single-cylinder high-speed engine having a relatively short stroke is under consideration. The engine was loaded with a clean exhaust up to a mean effective pressure of 107lb. per sq. in. After a continuous run over a period of three weeks (total 442 hours) at a mean effective pressure of 100lb. per sq. in., all parts of the engine were in perfect condition.

What are the reasons which made it possible to achieve

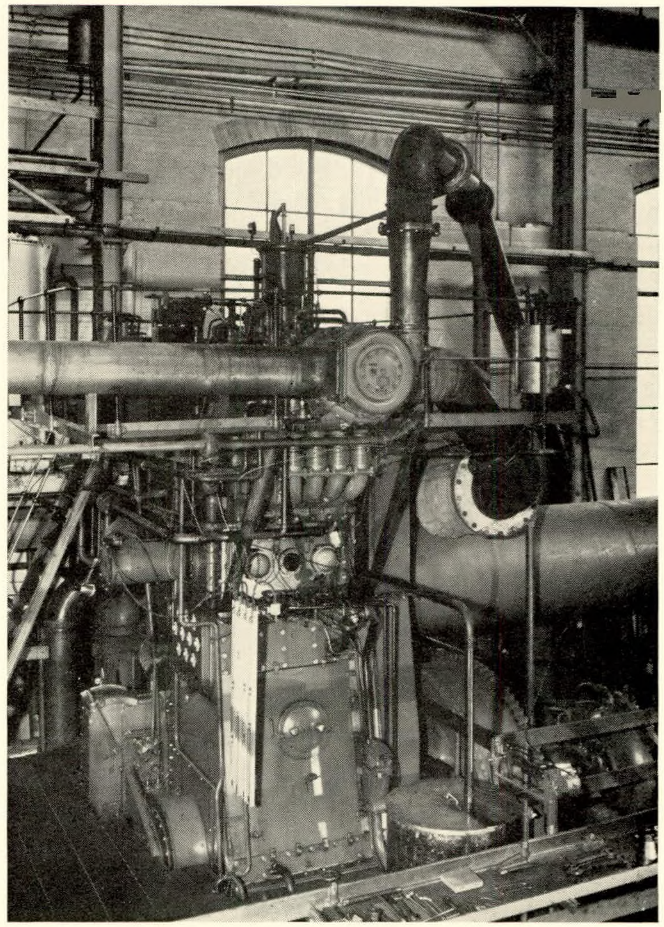


FIG. 33—*Crosshead diaphragm marine engine of 58-cm. bore equipped with Brown Boveri turbocharger*

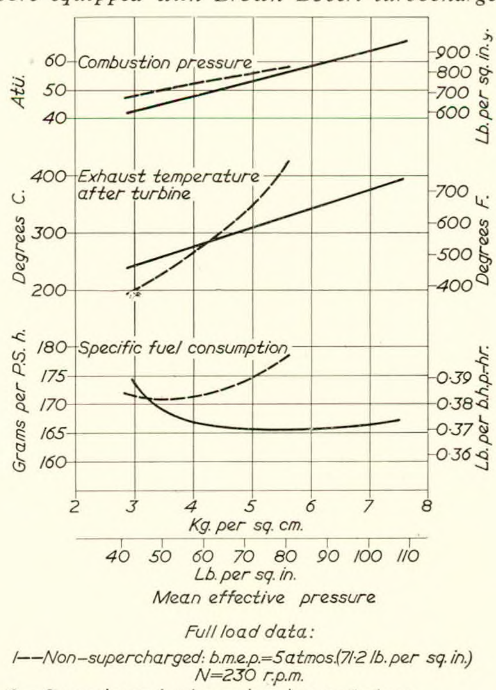


FIG. 34—*Comparison of supercharging test results with the non-supercharged engine performance (single-cylinder crosshead engine with 58-cm. bore)*
Operating condition: according to propeller curve

this result, considering the handicap due to the rather unfavourable conditions of a single-cylinder engine? It must not be forgotten that, as has been demonstrated in Fig. 31, the turbine is driven in such a case mainly by the blowdown of the cylinder pressure between the opening of the exhaust ports and the beginning of scavenging, this lasting for only some 10 per cent of the cycle. It is followed by a short period when the turbine receives some energy during scavenging, but for the remainder of the cycle the turbine is idling, with consequent high windage losses. In a multicylinder engine the several cylinders drive the turbine in succession, thus improving its operation considerably.

Although there are always many factors contributing to the particular performance of a certain engine, it is considered that the efficient utilization of the piston underside as a scavenging aid to be the most important contributory factor.

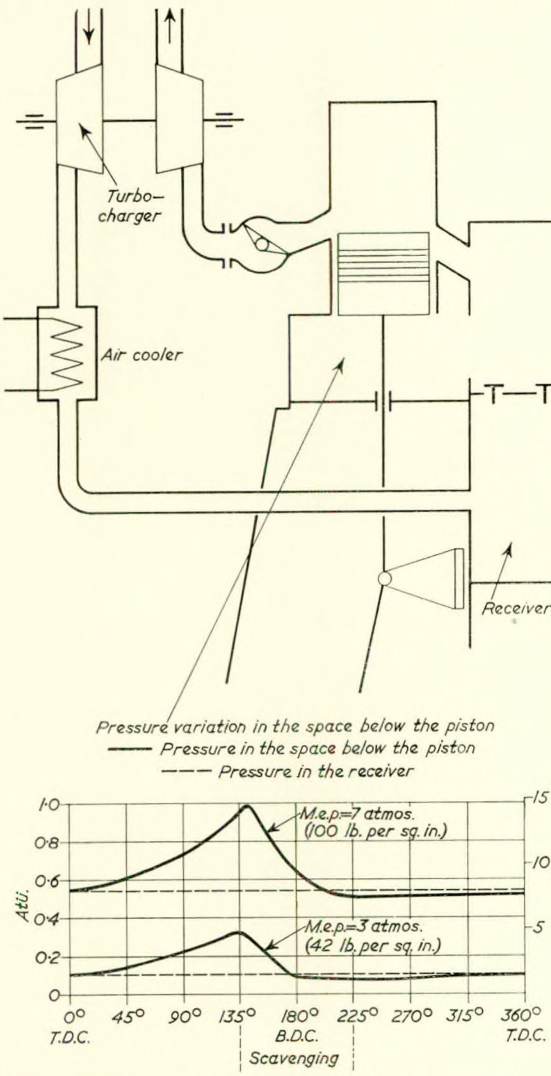


FIG. 35—*Supercharging tests (pulse system) diagram*

As illustrated in Fig. 35, there is a separate scavenge air chamber provided for each cylinder, thus forming together with the air space below the piston a buffer chamber connected through non-return valves of very low flow resistance to the common scavenge air manifold, which is supplied in the normal manner with air from the turboblower.

During its upstroke the piston accumulates an air volume corresponding to its effective displacement volume and compresses it during the downstroke. The large buffer chamber prevents the pressure from rising to too high a value. It is thus compressed only to a pressure corresponding to the one prevailing in the cylinder at the time of opening of the scavenge ports. Owing to this pressure peak, the blowback of exhaust gases into the scavenge ports is avoided and scavenging begins with a much stronger impulse as compared with the normal arrangement. Close to bottom dead centre the buffer chamber is discharged down to the pressure of the common scavenge manifold and air begins to flow through the non-return valves. Part of this air is again accumulated by the piston during its upstroke. With this parallel arrangement of the accumulating space below the piston and of the turbocharger, the effective scavenging period is prolonged with unaltered scavenge timing. Furthermore, owing to the pressure peak during the first phase

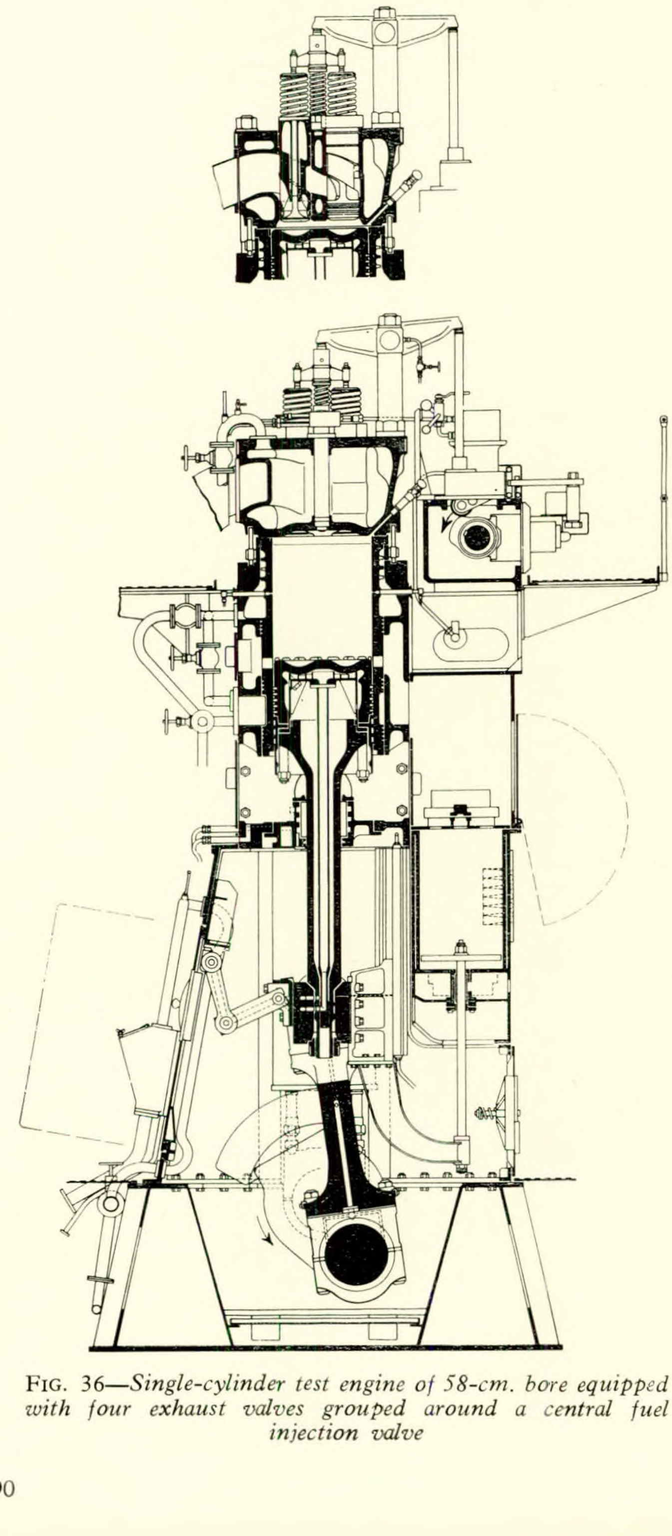


FIG. 36—*Single-cylinder test engine of 58-cm. bore equipped with four exhaust valves grouped around a central fuel injection valve*

of scavenging, with the same mean pressure in the scavenge manifold, a larger air volume can be scavenged through the engine. The increased scavenge air volume also tends to reduce the thermal load on the engine.

The self-scavenging feature is so effective that even with the turbocharger inoperative a b.m.e.p. of 50lb. per sq. in. may be obtained. This means that a ship fitted with a supercharged engine of this type will still be able to run at approximately 70 per cent of her full speed if all the turbochargers should fail.

Furthermore, this engine served as a test rig in an attempt to obtain a conclusive answer to the frequently disputed question of which of the two systems—cross or uniflow scavenging—adapts itself better to the typical requirements of a supercharged engine. For this purpose the engine was rebuilt as shown in Fig. 36, being equipped with four exhaust valves grouped around a central fuel injection valve. Exactly the same turbocharger with the same exhaust arrangement was used.

During the ensuing trials, the modified engine was run under operating conditions as similar as possible to those of the original version, although certain differences could not be completely eliminated. The best results which could be achieved were obtained with practically the same scavenge air flow and are presented in Fig. 37. As expected, the uniflow-

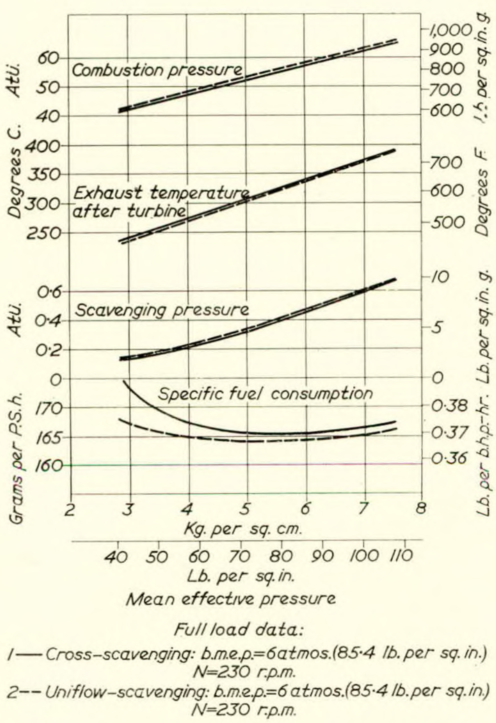


FIG. 37—*Comparison between cross and uniflow scavenging results of supercharging test with pulse system (single-cylinder crosshead engine with 58-cm. bore)*

Operating conditions: according to propeller curve. Supercharging air delivery conditions: turboblower, piston underside active, cooled after blower to 86 deg. F.

scavenged engine yielded a somewhat better performance; however, it may be questioned whether this warrants the additional cost and complications, as compared with the cross-scavenged engine, caused by the exhaust valves, valve gear and the intricate cylinder-head casting. Incidentally, no difference in starting, idling and manœuvring ability could be detected between the two versions. It should be understood, however, that even such carefully controlled experiments do not provide an absolute or general answer to the initial question. It is quite possible, for instance, that the difference between the two systems would be more pronounced for an engine having an extremely large stroke-to-bore ratio.

Another series of supercharging tests was conducted on a standard trunk piston engine, which has a bore of 48 cm., a stroke of 70 cm. and a normal speed of 225 r.p.m. This eight-cylinder engine develops 2,520 b.h.p. without supercharging. A cross-section of it is shown in Fig. 38. For the super-

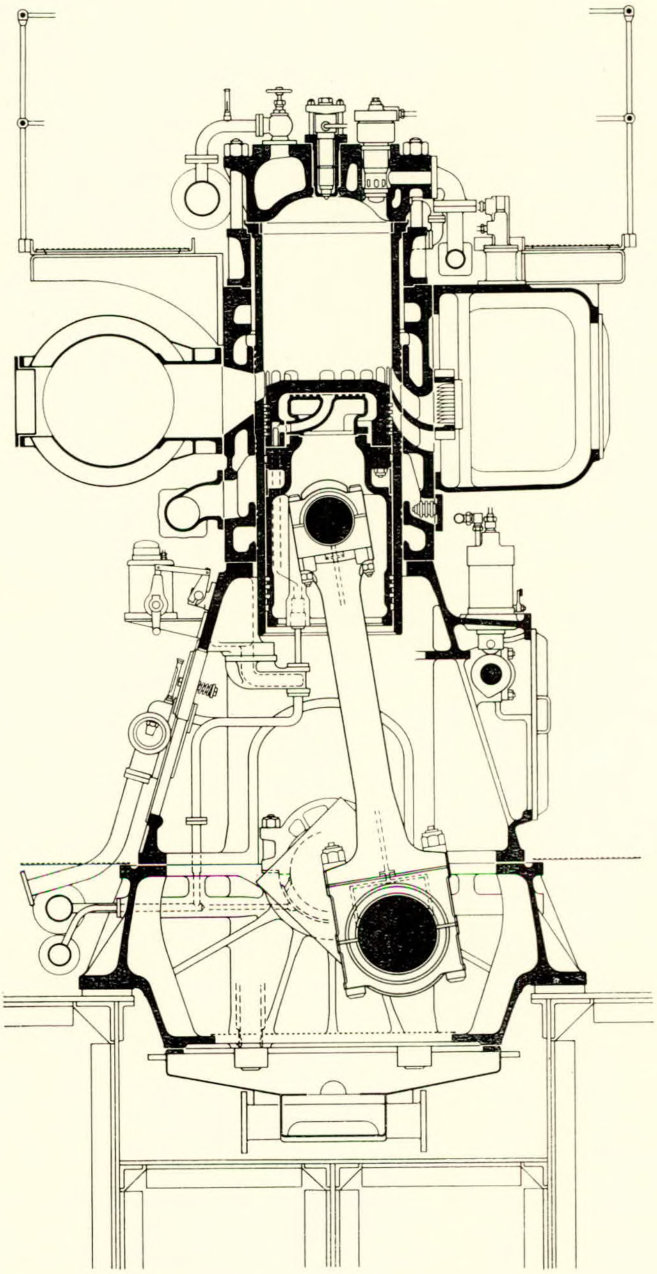


FIG. 38—*Cross-section of a standard trunk piston engine of 48-cm. bore*

charging experiments with the pulse system it was equipped with four Brown Boveri turbochargers. Two types of tests were carried out, one with the arrangement shown in Fig. 24, diagram 1, and the other according to diagram 2(b). Fig. 39 gives some results of the tests conducted under marine power plant conditions. It shows that the reduction in fuel consumption due to supercharging is more pronounced with the parallel arrangement; however, reliability and simplicity favour the Curtis system.

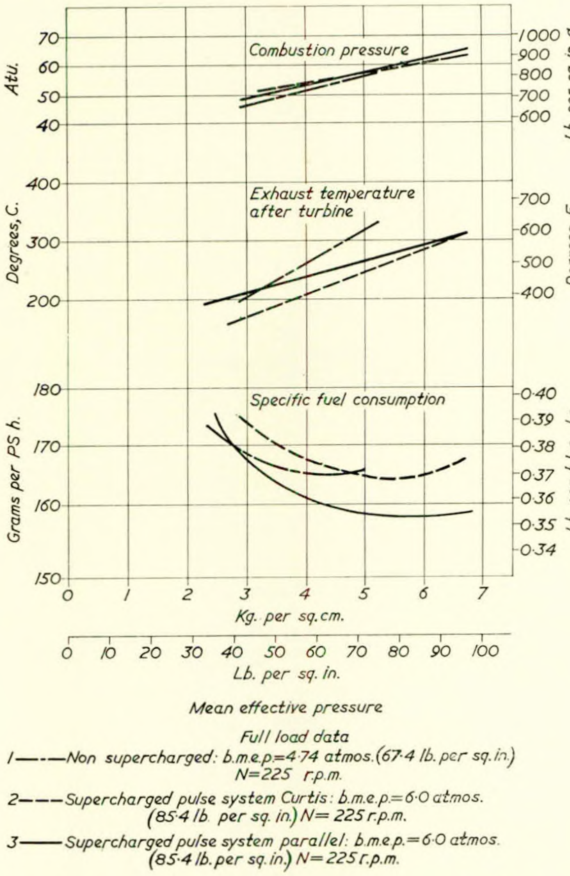


FIG. 39—*Comparison of various supercharging test results with the non-supercharged engine performance of the eight-cylinder trunk piston type with 48-cm. bore*

Operating conditions: according to propeller curve.
Supercharging air delivery conditions:

(2) Seven-eighths of scavenging pumps operative with Curtis arrangement, no aftercooling.
(3) Additional air volume 40 per cent of piston displacement with parallel arrangement; aftercooling to 86 deg. F.

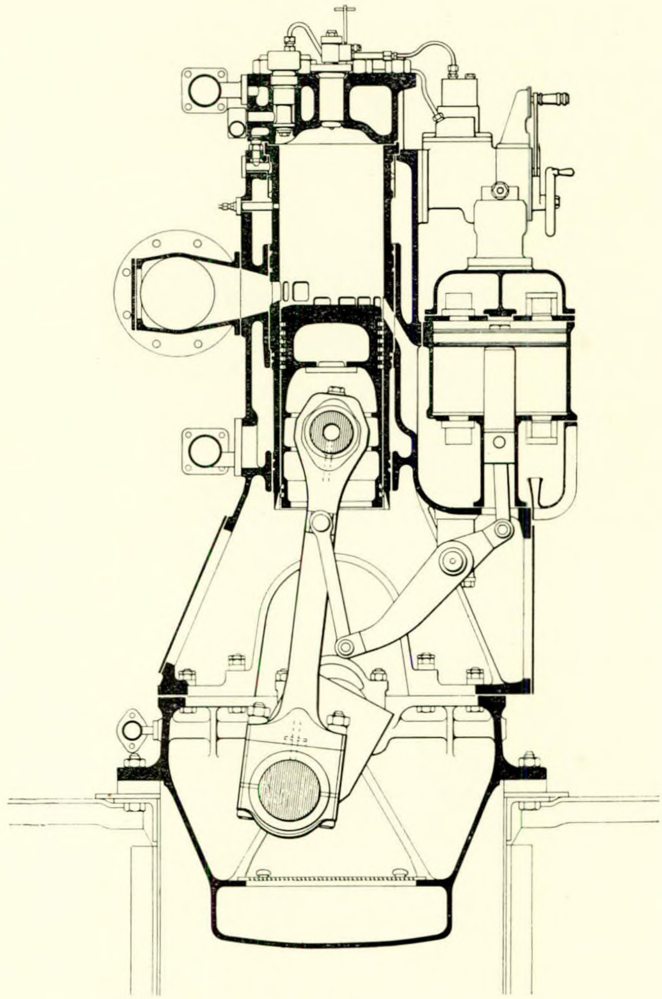


FIG. 40—*Cross-section of a standard four-cylinder engine with a bore of 24 cm., stroke 40 cm., and 400 r.p.m.*

The smallest engine used so far for supercharging experiments is a four-cylinder engine with a bore of 24 cm. and a stroke of 40 cm. running at a speed of 400 r.p.m. For the experiments conducted with the constant pressure system, it was possible to use the engine with only minor alterations affecting the flow of air to and from the scavenge pumps. A cross-section of the unaltered engine is shown in Fig. 40. For the pulse system tests, a rotary exhaust valve was fitted to the engine, which may be seen in Fig. 41.

The results of this experiment are worthy of note, since it proved that even an engine of this size could be run with the turbocharger alone ensuring sufficient air delivery. This resulted in a reduction in fuel consumption of some 0·025lb. per b.h.p.-hr., with a 35 per cent increase in power. Since the trunk piston engine possesses no self-scavenging feature, it is necessary to provide a small auxiliary blower for starting and idling. Fig. 42 gives a comparison of the performance of this engine with various supercharging arrangements. The fact that it was possible to dispense with the aid of the scavenge pump in the case of the pulse system was partly due to the higher efficiencies of the superchargers employed during these tests.

Sometimes one is misled by a diagrammatic presentation which shows fuel consumption versus b.m.e.p., since the part-loaded consumption seems to be higher with the supercharged engine. However, if the same diagram is drawn on a percentage basis as in Fig. 43, the proper balance is quickly restored.

As far as the larger bore engines are concerned, brief mention should be made of some tests which were conducted several years ago on a single-cylinder experimental engine with a bore of 72 cm. This engine proved to be capable of developing a b.m.e.p. of up to 130lb. per sq. in., while endurance tests with a load of 100lb. per sq. in. were successfully completed.

A nine-cylinder engine of the largest type built by his company, with a bore of 76 cm., is about to undergo extensive supercharging trials at their works.

From the foregoing, the conclusion may be drawn that the two-stroke Diesel engine even in its simplest form, i.e. with port-controlled cross scavenging, can be successfully supercharged. Additional equipment, such as rotary or oscillating exhaust valves, improve performance still further and allow the turbocharger to become self-sustaining over the greater part of its operating range. Utilization of the piston underside in the case of crosshead diaphragm engines provides complete independence of any external scavenging aid. The changeover to uniflow scavenging allows another slight improvement in fuel consumption. However, every step towards improved performance entails added complications, which in every instance must be carefully balanced against the prospective gains. The specific supercharging arrangement to be employed depends entirely on the type of engine and operating conditions. The power gains to be expected from low pressure supercharging may go as high as 40 per cent, accompanied by an improve-

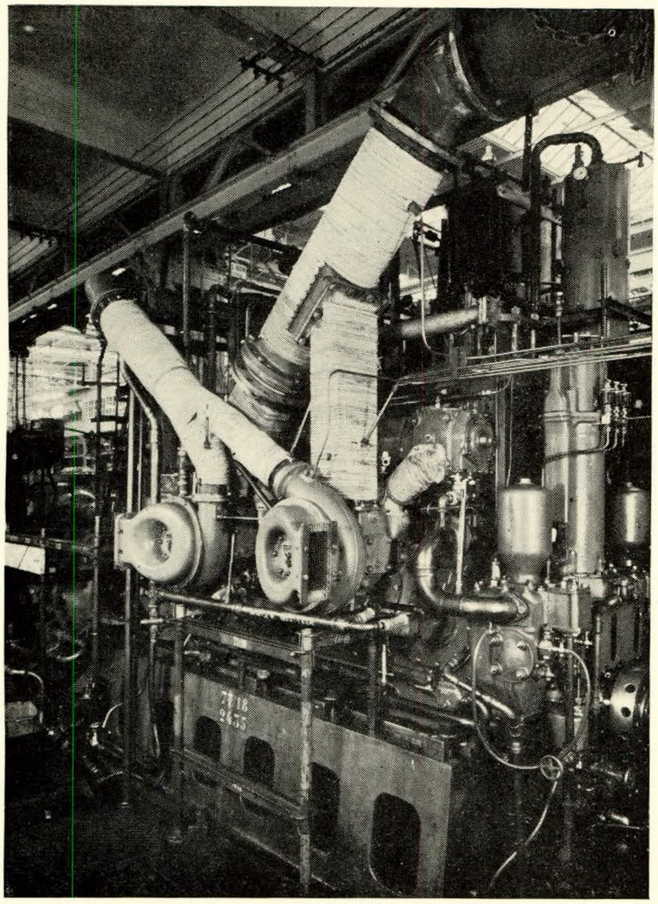


FIG. 41—*Four-cylinder engine as shown in Fig. 40, with rotary exhaust valve*

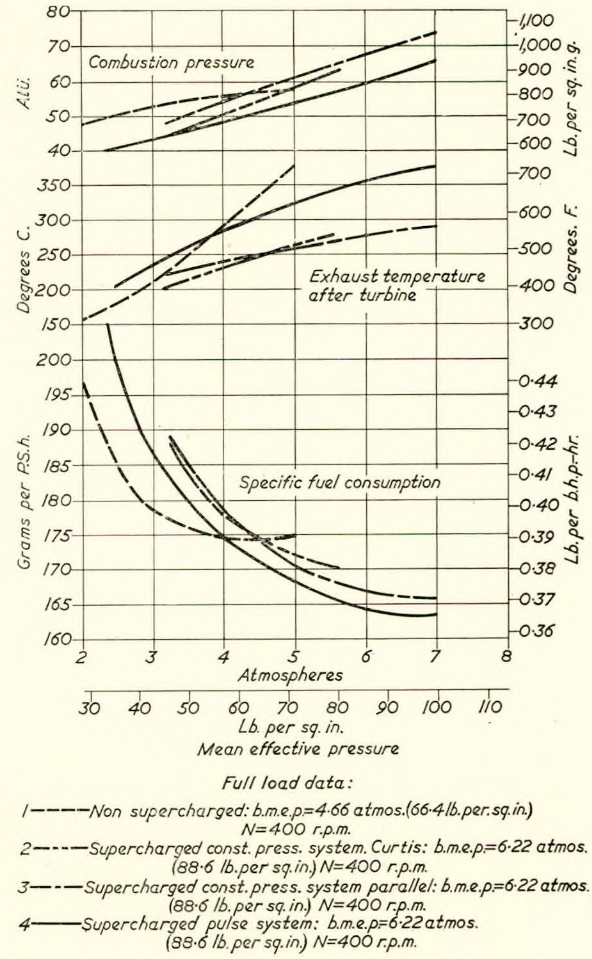


FIG. 42—*Comparison of various supercharging test results with the non-supercharged engine performance of the four-cylinder type with 24-cm. bore*

Operating conditions: according to propeller curve.
Supercharging air delivery conditions:

(2) Three-quarters of scavenging pumps operative with Curtis system, cooled after blower to 68 deg. F.
(3) Three-eighths of scavenging pumps operative with parallel system, cooled after blower to 68 deg. F.
(4) Turbocharger alone, cooled after blower to 68 deg. F.

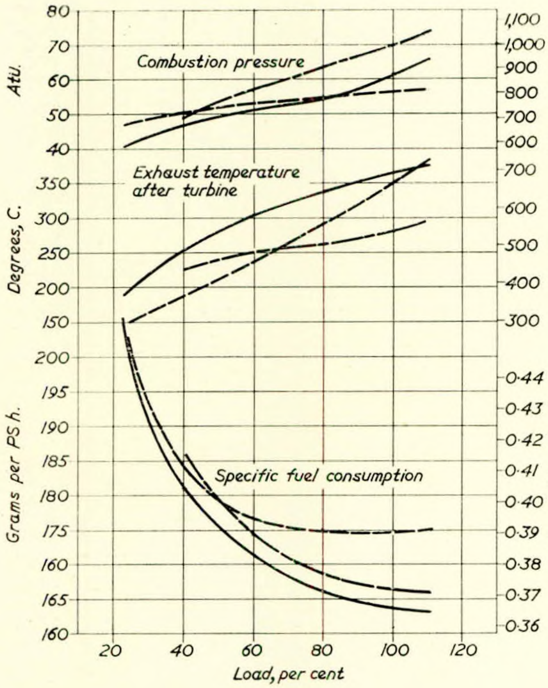
ment in fuel consumption from 1·5 to 6·5 per cent, giving an increase in firing pressures from 10 to 13 per cent, which can be tolerated.

ADDENDUM

On page 392, it has been mentioned briefly that a nine-cylinder engine with a bore of 30 inches and a stroke of 61 inches was about to undergo supercharging trials. The design of this engine is similar to that of the RS58 engine, a cross-section of which is shown in Fig. 32.

Fortunately, a first test series has already been completed and it is possible to present the results obtained. In order to review briefly the method of supercharging employed with this type of engine, reference should be made to Fig. 24, where there is a list of the most commonly used supercharging arrangements for two-stroke engines. These include the Curtis or series arrangement of turboblower and engine-driven scavenge pump; the corresponding parallel arrangement as well as the parallel circuit with auxiliary blower; the supercharging method employing a supplementary drive of the turbocharger; and, finally, the most desirable arrangement, having no other means for provision of the scavenging and supercharging air required by the engine but the exhaust-gas driven turboblower itself. The last-mentioned arrangement, together with the pulse system for the exhaust turbine, is the one employed during the most recent tests with the 9RS76 engine.

At the normally rated output of 11,700 b.h.p., which corresponds to a b.m.e.p. of 100lb. per sq. in. and a speed of 119 r.p.m., the fuel consumption obtained was 0·351lb. per b.h.p./hr., which is lower than the best value ever obtained with a non-supercharged engine of this type. However, even more significant—as can be seen from Fig. 44—is the fact that there is practically no increase in fuel consumption even at 10 per cent overload. Incidentally, this load corresponds to an output of over 17,000 b.h.p. for a twelve-cylinder engine of this size. The complete absence of smoke at this output indicates that the potentialities of this engine have not yet been fully utilized. This conclusion is also borne out by the striking tendency of the exhaust temperature curve to level off at the high power end. As far as the part-load behaviour of this engine is concerned, we may get an even better picture from Fig. 45, which is drawn on a percentage basis rather than absolute values of b.m.e.p. Evidently there is a strong tendency for the fuel consumption curve for the supercharged engine to remain below the one of its non-supercharged counterpart over the total load range. No difficulties whatsoever were encountered during starting and idling, which is quite a remarkable feat for a cross-scavenged engine having no other scavenging assistance than a free-running turboblower.



Full load data

1——— Non-supercharged: b.m.e.p. = 4·66 atmos. (66·4 lb. per sq. in.) N = 400 r.p.m.

3——— Supercharged const. press. system parallel: b.m.e.p. = 6·22 atmos. (88·6 lb. per sq. in.) N = 400 r.p.m.

4——— Supercharged pulse system: b.m.e.p. = 6·22 atmos. (88·6 lb. per sq. in.) N = 400 r.p.m.

FIG. 43 *(left)—Comparison of various supercharging test results with the non-supercharged engine performance for same engine as in Fig. 42*

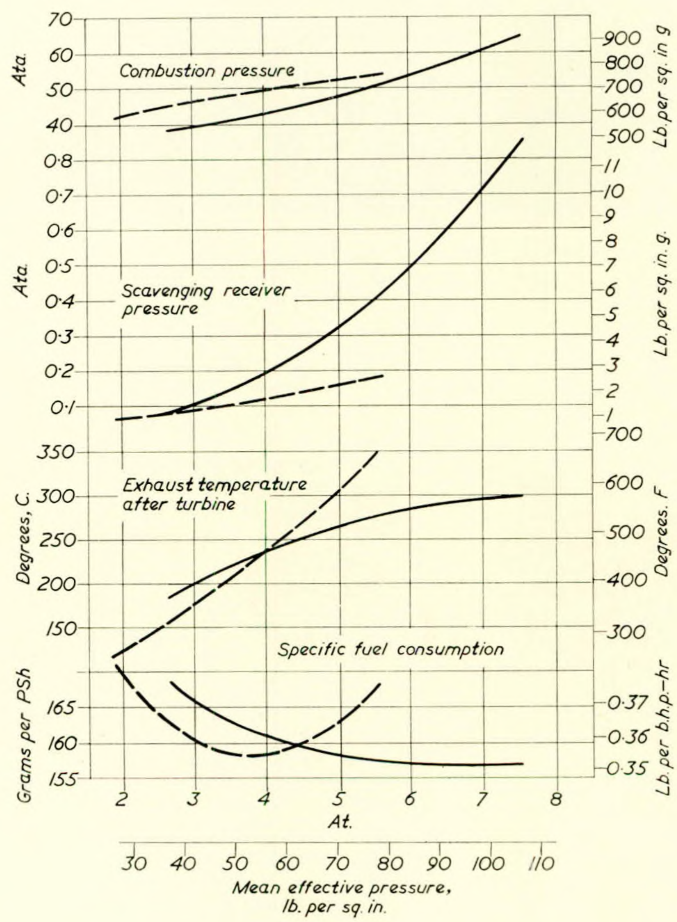
Operating conditions: according to propeller curve.
Supercharging air delivery conditions:

(3) Three-eighths of scavenging pumps operative with parallel system, cooled after blower to 68 deg. F.

(4) Turbocharger alone, cooled after blower to 68 deg. F.

FIG. 44 *(right)—Comparison of supercharging test results with the non-supercharged engine performance of the 9 RS 76 engine*

Operating conditions: according to propeller curve.
Supercharging air delivery conditions: turboblower piston underside inactive; air cooled after blower to 47 deg. C. (117 deg. F.).



Full load data

1——— Non-supercharged (10 cylinder engine) B.M.E.P. = 5 atmos. (71·2 lb. per sq. in.) N = 115 r.p.m.

2——— Supercharged pulse system (9 cylinder engine) B.M.E.P. = 7 atmos. (100 lb. per sq. in.) N = 119 r.p.m.

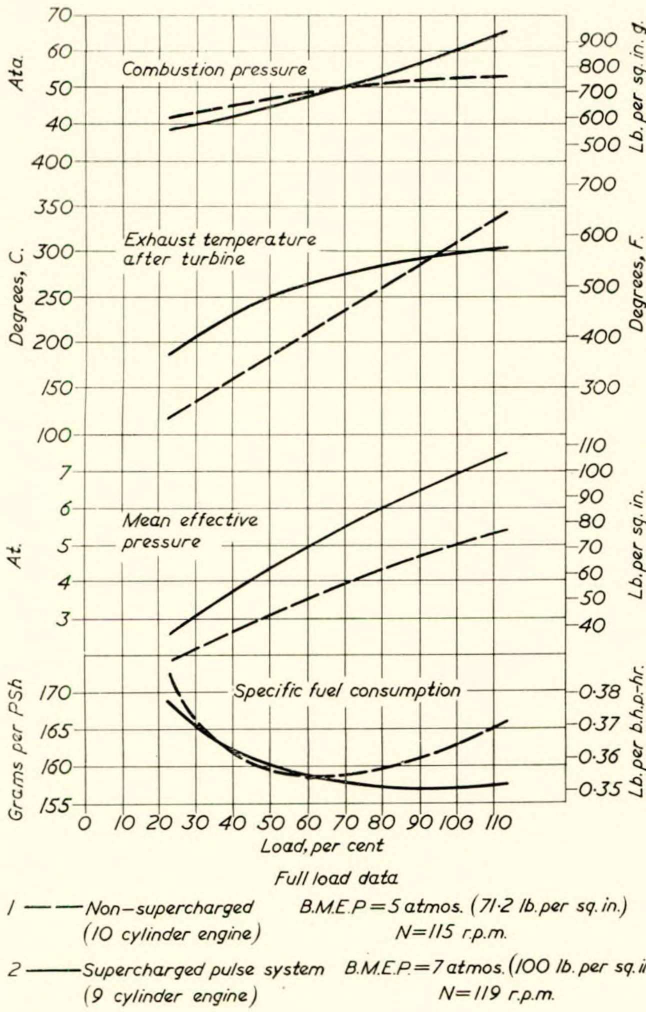


FIG. 45—*Comparison of supercharging test results with the non-supercharged engine performance of the same engine as in Fig. 44*

Operating conditions: according to propeller curve.
Supercharging air delivery conditions: turboblower, piston underside inactive; air cooled after blower to 47 deg. C. (117 deg. F.).

Nevertheless, it is anticipated that the performance of this engine will be further improved in the course of the extensive tests which are scheduled to be carried out in the very near future. This conviction is supported by the promising results achieved with scavenging tests which have been conducted with the aid of a non-stationary scavenging model. New porting arrangements lead to a considerable improvement in scavenging efficiency.

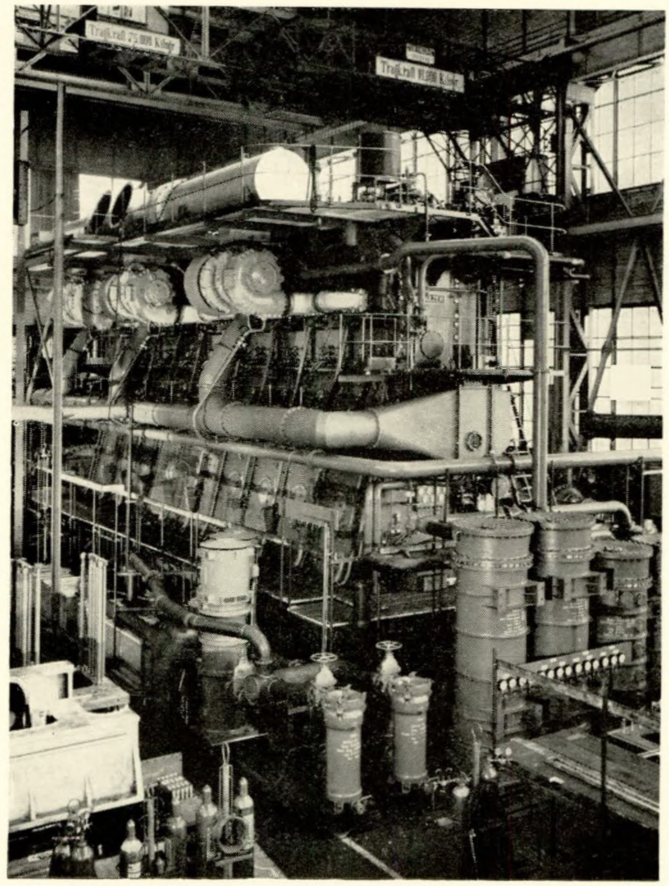
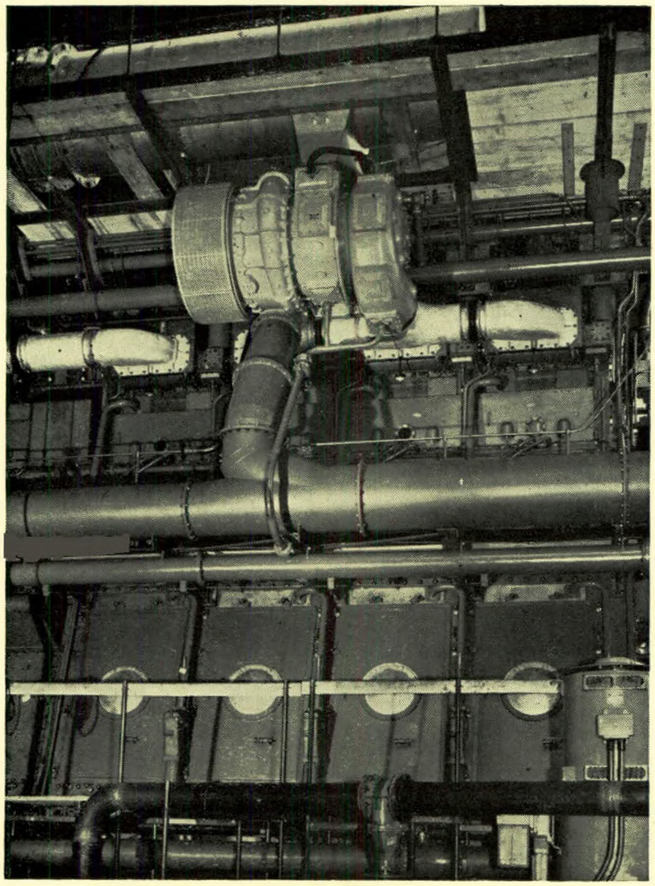
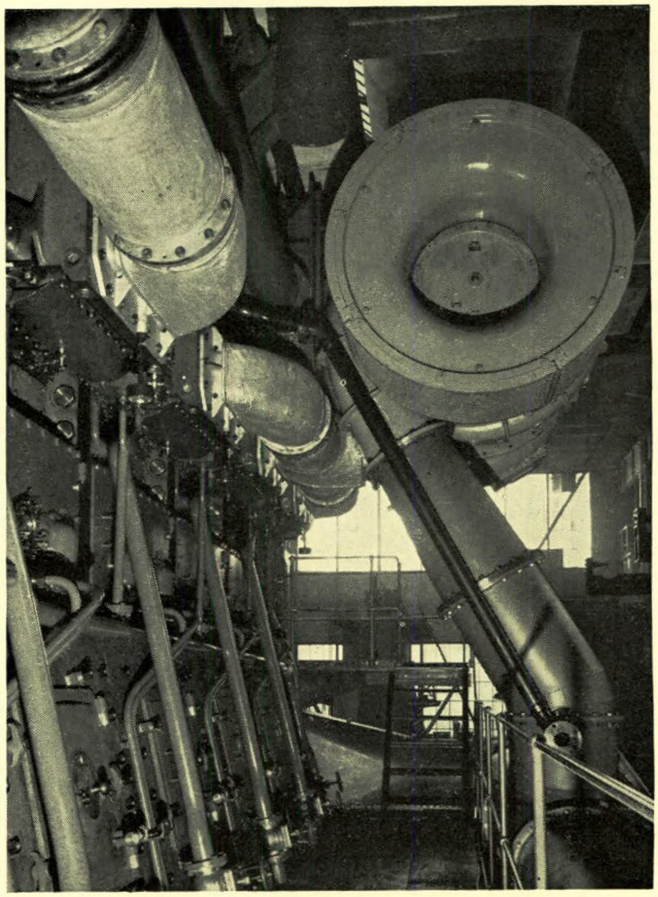


FIG. 46—*General view of the supercharged 9 RS 76 engine on the test bed*

Figs. 46 to 49 give an impression of the appearance of the 9RS76 engine on the test floor. Particular attention should be paid to the mounting of the three turbochargers, each receiving the exhaust gases from three successive cylinders. Although they are arranged in such a way as to receive the gases from each cylinder in the most direct manner, they do not obstruct any vital part of the engine nor reduce the accessibility to any engine component which requires periodic attention. Obviously it is much more difficult to achieve the same feat without compromising in the case of a uniflow scavenged engine where the exhaust is controlled by poppet valves in the cylinder head. This again brings up the question whether or not the slight gain in performance, which was demonstrated in the previously mentioned tests comparing cross-and uniflow-scavenging on the 1RS58 23-in. engine (Fig. 37), warrants the additional complications and servicing difficulties encountered with the uniflow-scavenged engine type.

At any rate it is felt that the merits of the cross-scavenged engine justify a concentrated effort towards the perfection of a reliable and economical supercharged version of the heavy two-stroke marine oil engine of this type.





FIGS. 47 *and* 48—*Close-up of the turbocharger arrangement in the 9 RS 76 engine*

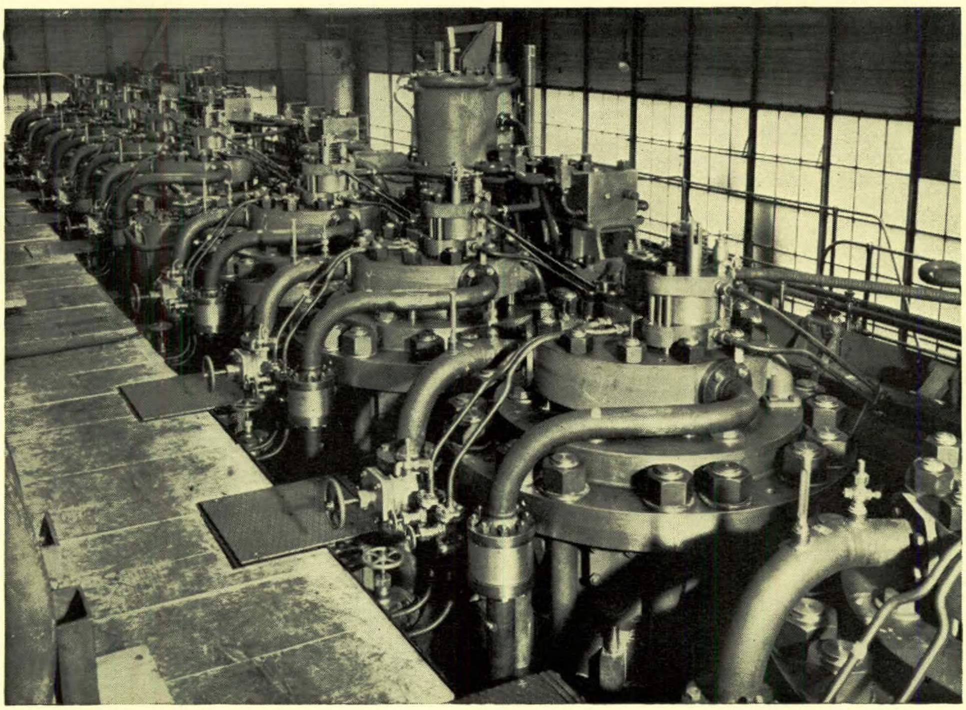


FIG. 49—*View from the top gallery of the supercharged 9 RS 76 engine*