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NOTES ON TURBOCHARGED TWO-STROKE CYCLE DIESEL ENGINES

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Turbocharging as such is not new with Diesel engines in American railroad service. For many years, four-stroke cycle Diesel engines equipped with turbochargers have been in daily operation. During the last few years, the turbocharger has also been accepted and introduced on a rather large scale for many types of trucks and for off-highway earthmoving equipment. The latter's operational conditions are rough and demanding; nevertheless, the turbochargers operate very satisfactorily at speeds up to 50,000 rpm on Diesel engines of the 150 to 500 hp class and, in many applications, with engine exhaust gas temperatures of 1400F and higher. Maintenance or service intervals of many thousand hours have been demonstrated to be quite realistic.

Now, how much horsepower can be extracted from exhaust gases of reciprocating engines? An engine consuming 100 pounds of air per minute and operating with an exhaust gas temperature of say 900F when back pressured to about 10 psig delivers 100 hp gas energy into the turbine of the turbocharger. Such engine with 100 pounds per minute of air through flow is not very big and, as a two-stroke cycle Diesel, could have 1,000 cu in. displacement and 1,000 rpm producing between 350 and 400 hp. A good exhaust gas turbine has a shaft efficiency upwards of .75 or, in other words, could, under the above conditions, drive a charging compressor consuming 75 to 85 hp.

Fig. 1, containing two plots in one chart with the ordinates in common, gives an idea of the energy contained in the exhaust gas in terms of ft lb per lb frequently called adiabatic head and of power per 100 lb per min gas. The straight line is valid for a constant expansion ratio of 2 across the turbine and shows an increase from approximately 30,000 to 56,000 ft lb per lb with a temperature change from 400 to 1200F, whereas the convex line shows the influence of the expansion ratio on the adiabatic head with a constant gas temperature of 1200F.

Developments within the last decade only have established the turbocharged two-stroke cycle Diesel engine, an observation well illustrated by the plot, Fig. 2, which BBC published. The comparison shown in Fig. 3, also taken from BBC, is just one of the many examples demonstrating the reduction in engine size and weight which are some of the many benefits from turbocharging.

"Fitted with an exhaust turbine, any degree of supercharge can be applied to a two-cycle engine, for the higher the exhaust back pressure, the greater the return, and that with interest from the turbine, until, in the limit, the turbine is developing the whole of the useful power, that of the piston engine being absorbed entirely by the blower." This statement was made by Sir Harry R. Ricardo in 1950 when delivering the "Thomas Hawksley Lecture". The voices of many other well-known authorities could be added. Long time testing as well as some years of practical operation have demonstrated that through turbocharging, up to close to 100 per cent increase in rating, satisfactory operation without sacrifice in reliability or wear can be attained. It has been proven over and over that even power boosts of this magnitude cause only small, if any, increase in the so-called thermal loading. As can be expected, greatly reduced fuel consumption values have been shown and it appears that the two-stroke cycle engine is--in this respect--competing rather critically even with some of the better four-stroke cycle developments.

Why then did it take so long for the turbocharger to be accepted for the two-stroke cycle engine? The two-stroke cycle engine is known to be critically dependent on ample air supply and scavenging in order to function properly, that is to produce power and not to smoke. In contrast, the four-stroke cycle engine can be and has been called a self-breathing engine which means that it is capable of operation under all load conditions without outside help for air supply. The two-stroke cycle engine always depends on air supplying means such as blowers. Scavenge air can only be expected to pass through the cylinder if the required pressure differential is available, that means if the pressure at the inlet to the cylinder is higher than at the outlet therefrom. It can easily be shown that to produce the pressure differential needed for scavenging by way of a turbocharger requires two conditions to exist, namely (a) a sufficiently high amount of energy in the exhaust gas and (b) an efficient utilization of this energy by the turbocharger, which means high efficiency of turbine and compressor. Up to not too long ago, either one or both of these conditions appeared hard to fulfill; particularly so, during periods of low engine loads and the starting cycle where, obviously, these pressure and temperature conditions are much more critical. Using simple equilibrium considerations between turbine power produced and compressor power required, the relationship between turbine inlet and compressor outlet pressures has been plotted in Fig. 4 with the exhaust gas temperature as parameter.[1] A chart of this type, calculated for a given turbocharger with certain turbine and compressor efficiencies, demonstrates clearly that there exists a minimum exhaust gas temperature below which the required pressure differential across the cylinder cannot be obtained or, to express it differently, below which the scavenge (or charge) pressure cannot exceed the back (or discharge) pressure.

The significance of a clean charge, i.e. the possibility of good scavenging, has a decisive influence on the degree of economical maximum power boost. In this conjunction, a word about the heat load may be in order. Leading manufacturers have reported findings confirming what can also be demonstrated by thermodynamical calculations. With power boosts up to over 50 per cent obtained with a good turbocharging system, the same material temperatures were measured as in the unsupercharged engine. To arrive at such good results, engine development work expecially in relation with the engine exhaust process had to be done.

A very special problem with the two-stroke cycle engine exists for turbocharging when the engine has to be started or has to operate at a low load level.

In order to make up for charge energy missing at low engine loads and, therewith, in starting, different arrangements of air-charge systems are being used on turbocharged two-stroke cycle engines. The more familiar ones are:

The engine retains its blower or scavenging pump and the turbocharger "sits on top". With proper layout and good efficiencies, the participation of the mechanically driven blower decreases with increased engine load and speed. Quite obviously, when of the positive displacement type, the mechanical blower can be unloaded entirely and, in the extreme, becomes an expansion machine with the accompanying delivering of power and reducing of the charge-air temperature.

The turbocharger shaft is mechanically (or hydraulically or pneumatically) connected with the engine crankshaft. This arrangement allows the turbo to deliver power to or to draw power from the engine dependent upon the engine's operational condition, i.e. load and speed.

The EMD engine equipped with AiResearch turbochargers in a Union Pacific locomotive as well as the just announced G M Detroit Series 71 engines,[2] have the turbocharger "sitting on top" of the Diesel engines retaining their mechanically driven blowers which, in these cases, are of the Roots type.

A connection between the engine shaft and the turbocharger shaft can be observed in such engines as the Clark Bros. produced gas engines[3] widely used in oil field pumping stations.

A somewhat different arrangement with compressor and turbine physically separated and geared to the engine individually can have certain merits on a turbocharged two-stroke cycle engine. The energy required by the blower with the blower geared to the crankshaft can always be taken through this drive. The turbine in turn can deliver all its energy to the engine crankshaft. One advantage offered by this arrangement is the exclusion of overspeed of the turbocharger components. Overspeeding can occur with a free-floating turbocharger if, due to foreign material such as pieces of a valve or piston ring, the effective turbine nozzle area is reduced leading to higher engine back pressure, i.e. higher expansion ratio across the turbine. Thus, the power balance of the turbocharger is upset; the turbine can produce excess torque and the turbocharger must normally speed up.

Naturally, the simplicity of an arrangement in which a turbocharger alone satisfies all requirements does challenge the engineers. Within the last seven years approximately, success has crowned the efforts. Such European Diesel engines with Brown Boveri turbochargers are in service predominantly in marine applications. More recently in the U. S. A. a Kahlenberg two-stroke cycle Diesel engine with a Napier turbocharger[4)] was announced. It starts and operates throughout the full speed range without a scavenge pump or any assisting of the turbocharger.

With a turbocharger being the only means of supplying air to the cylinders and being at a standstill when the Diesel engine is to be started, the engine would not start because of not having an air supply. Therefore, in cases where a turbocharger is the only scavenging and charging means, and especially where no mechanical connections between turbocharger and engine exists, other means must be used to start engine and/or the turbocharger. Of different possibilities to solve this problem, one involving certain special provisions in the turbocharger might be mentioned here. When starting the engine, compressed air can be used on the rotor of the turbo to bring it up to the required speed and, therewith, to a self-sustaining powerplant operation. Such compressed air can, for instance, be admitted to the turbine through special passages in the turbine nozzle system or by direct impingement onto the blading of the rotor. The higher the efficiency of the turbocharger unit and the better the scavenging system works, the sooner the engine speed-load condition will be reached at which the complete powerplant can operate without extraneous energy supply.

Returning to pertinent engine characteristics, the following generally known thermodynamical relations should be remembered.

In first approximation, the horsepower output of a Diesel engine like any other combustion engine is proportional to the air-weight flow per time unit available for combustion of fuel in the cylinders. While under load, the "rich-mixture" gasoline engine has a low air/fuel ratio of e.g. 12 to 14, the Diesel engine operates much "leaner" e.g. between 20 and 35, and the gas turbine is normally "leaner" yet e.g. 50 to 75. Also, for a given factor of utilization called thermal efficiency, more fuel burned means more power produced. By doubling the density of the air retained in the cylinder, at least twice as much fuel can be burned which then will, of course, result in twice as much power.

Naturally, there are some obstacles, of which the mechanical ones will not be dealt with here. Firstly, the compression of the combustion air prior to entering the cylinder unavoidably results in heating. With engine compression ratios between 11:1 and 16:1, an increase in charge-air temperature of 100F would correspond to a "calculated increase[5)] of the air temperature at the end of the compression stroke of about 250F. Aside from the fact that through raising the inlet temperature the thermal efficiency decreases, the distress of the valves, the pistons and all other engine parts exposed to the heat flow would be greatly increased if not compensated for by increased through flow of scavenging air.

Secondly, pushing pre-compressed air into the cylinder without changing the compression ratio of the engine increases the pressure in the cylinder. Roughly speaking, for 10 psi increase in charge-air pressure and in the compression range mentioned above, the "calculated" pressure increase ranges from over 200 to over 400 psi.

A remedy to the temperature problem suggests itself: cooling of the charge air before admitting it to the cylinders. So-called intercoolers and aftercoolers, which are "air-to-air" or "air-to-water" heat exchangers, have therefore been introduced. It takes little, if any, courage to predict that they will be considered standard equipment as turbocharging progresses. Another cooling possibility which is often forgotten should be mentioned. A suitable substance such as a volatile liquid with a high latent heat can be evaporated in the charge air and, thus, reduce the temperature.

The cooling of the charge air results in an increase of air density as long as the pressure losses in the heat exchanger and ducting remain sufficiently low. Thus, an increase in air density is obtained without moving parts. In practical application, the combination of charge-compressor and charge intercooler has much to offer. In some cases, the pressure ratio demanded from turbochargers leads necessarily to highly stressed rotors. Whenever this condition is approached, charge-air cooling can become mandatory. The following little tabulation shows that for a specified density ratio of ambient air to charge air of 2.5 good intercooling results in around 60 per cent lower stress in the compressor impeller[6] while, at the same time, it reduces the thermal pains of the engine cylinder a great deal. Also, without intercooling a boost pressure of 4.4 x 14 = 62 psia would be required; whereas, intercooled to 125F, the boost pressure is only 2.7 x 14 = 38 psia.

TABLE I

for density ratio 2.5 (90 degree day)

all values approximate	P/p	stress level
without intercooling	4.4	160 per cent
with intercooling to 225 deg	3.2	120 per cent
with intercooling to 125 deg	2.7	100 per cent

Fig. 5 serves to demonstrate graphically first the strong influence of the compressor efficiency of a turbocharger on the density increase produced. Secondly, if intercooling is added and if an air temperature of not more than 200 deg can be maintained at the outlet of the intercooler, with airbox pressures higher than about 25 in.Hg, the benefit of even such moderate intercooling shows in terms of higher air density; intercooling back to 100 deg shows increased air density over the entire range.

In all cases where an increase in power output or improved operational conditions for a reciprocating engine are sought, the improvement of the so-called breathing capacity is known to be most significant. With the super- or turbo-charged engine, especially of the two-stroke cycle type, all flow passages should be shaped and made such as to keep all density losses to a minimum. So, whether the engine builder includes a turbocharger in the power-package or whether a turbocharger "kit" is installed later, it pays to observe certain simple rules.

Usually, when "changing" an engine by adding a turbocharger, the increase in air flow requires the provision for additional filter capacity. All air passages to and from the filter should be so dimensioned as to result in minimum pressure loss and a fairly uniform flow-pattern at the compressor inlet. Naturally, the arrangement should prevent all charge-air heating in the inlet system. Then, from compressor outlet to the inlet ports or valves of the engine, the air pressure losses again should be kept low. Remember that the turbocharger's operation is at the low pressure level of the total working cycle of the powerplant. Air velocity and duct shape must result in an even distribution of the charge air to all combustion chambers. If the ducting can be used to help lower the temperature of the charge air, the benefit of a density increase through cooling results. To give some idea about the detrimental influence of pressure loss and heating, citing a few numerical values might be helpful. Three cases will be considered with ambient air conditions being 90F and 29.5 in.Hg and an assumed specified ratio of charge-air density to ambient air density of 1.75. The three cases are (1) negligible pressure losses, (2) 2 in. Hg each pressure loss before compressor inlet and between compressor outlet and engine inlet ports, and (3) 1 in.Hg pressure loss before compressor inlet combined with 25 deg air preheating and negligible pressure loss in the outlet system. The cases cited are fairly representative of conditions often encountered and the results show the magnitude of what in comparison to say 40 in.Hg pressure rise through the compressor is sometimes thought of as small losses.

	Case I	Case II	Case III
Pressure Ratio:	2.41	2.80	2.71
Air Temperature at Compressor Outlet (F):	298	338	365
Impeller Tip Speed (ft per sec):	1200	1320	1320
Power Consumption of the Compressor (hp):	125	149	150

The stresses in the turbo rotor change proportionately to the square of the tip speed, which means at least 20 per cent higher stresses in Cases II and III.

Naturally, basically the same thoughts apply to the exhaust manifolding and ducting from the exhaust ports to the turbine housing and in the gas discharge system attached to the turbine outlet. Since the heat energy left in the exhaust gases is to be utilized in the turbine and since the exhaust temperatures of two-stroke cycle engines are usually painfully low as far as the turbine is concerned, these ducts should be held as small in size as pressure losses and scavenging considerations permit and may be insulated to conserve the gas energy for utilization in the turbine. In the pressurized branches of the turbocharging system, i.e. inlet and exhaust manifolding, leakage of air and gas must be prevented because otherwise even the best turbocharger may be incapable of doing the job expected.

Since the turbine extracts gas energy between the high level of temperature and pressure at the turbine inlet and the low pressure level at the turbine outlet, back pressuring of the turbine discharge cuts down the power which the turbine can produce. Using the 10 psig turbine inlet pressure, i.e. cylinder back pressure, again like in an example cited above, a turbine whose discharge system has only 1 in.

Hg pressure drop or back pressure can deliver 15 per cent more power than the same turbine with 3 in.Hg discharge pressure loss. Therefore, proper and generous dimensioning of the exhaust stack is as significant a factor as is turbocharger efficiency.

Fig. 6 gives an example of how, under rather difficult installation restrictions on a Caterpillar tractor, a well-working ducting arrangement was worked out in conjunction with an AiResearch turbocharger.

All the above matters are hardly new yet they are being violated too often. In an actual case, recently the air at the inlet to the compressor was measured to be up to 60 deg warmer than the ambient air. One remark regarding the gas discharge system might be of some interest. Usually, the gas velocity at the turbine discharge is fairly high, that is between say 300 and 600 ft per sec. Therefore, this gas jet with a properly arranged ejector system can be used to produce air flow e.g. through an engine room.

Aside from the fact that smoking engines are becoming a number one public nuisance and even a danger to human lives, smoke always is the tell-tale of fuel waste. Unsatisfactory combustion will result from too rich a mixture, i.e. too low an air/fuel ratio. This immediately explains the observation that exhaust discoloration disappeared even with notoriously smoking engines when a good turbocharger was added because of its capability of pushing more air into the cylinders under conditions favorable to better burning.

Much can be and has been said about the merit and demerits of the "constant pressure" or "steady-flow" turbine in contrast to the "blow-down" or "pulse system" turbine. Aside from these somewhat misleading appellations, there are numerous reasons why this subject cannot be discussed in this paper except to say that, on more recent engines known to the author, conditions resulted in what might be called a sufficiently simple yet reasonably satisfactory "combination" of the two approaches. Incidentally, even if one can calculate the pulsating energy flow from engine outlet to turbine inlet, it is practically impossible to determine analytically a turbine's capability of utilizing this pulsating energy flow; by nature, the turbine is a steady-flow machine. Naturally, the matter of pressure pulsations in the manifolding system of a two-stroke cycle engine, as well as the subject of exhaust pipe dimensioning and its effect on scavenging and charging process of the engine, cannot be elaborated here.

In closing these more general notes, many of which are applicable to all engines, it may be stressed once more that because of the two-stroke cycle engine's breathing difficulties much more care and perfection are mandatory than is the case with most and especially the older four-stroke cycle engines. Normally, when rolling along in an automobile even at high speeds, the discharge pressure of the cylinders is higher than the charge pressure. If the engine components could talk, the engineer would hear many complaints. With a turbocharger aiding the engine, as proven, down to very low loads, the charge pressure at the cylinders remains higher than the discharge pressure; gone are the reasons for most of the complaints!

And now, one of the more recent events in the forward march of turbocharging will be told.

The undertaking of turbocharging the Diesel engine in a Union Pacific locomotive, as described in the paper "Railroad Experience With a Turbocharged Two Cycled Diesel Engine", has roughly the following background.

It has been observed by Union Pacific that, at elevations of 4,000 ft and above and at ambient temperatures on the order of 90F and above, the exhaust temperatures of the EMD 16-567C engine were about 200F higher than at rated conditions at sea level. This appeared to indicate that, under these conditions, the engine is operating at marginal air/fuel ratios as would result from the deduction of air density. The observation of heavy exhaust smoke seemed to support this thought.

Union Pacific's operation west of Omaha, Nebraska, includes more than 1,200 miles at elevated sections with close to 500 miles between 6,000 and 8,000 ft elevation. Average ambient air temperatures encountered in these sections during the summer months range from 80F (at elevations of 6-8,000 ft) to 120F (at elevations of 4,000 ft and below). With an average temperature increase of 40F or more from ambient to the engine air inlets inside of the locomotive body, the effective air density may be as low as corresponding to an elevation of 10,000 ft leading to a decrease in engine performance in the order of 10 per cent.

Before entering any installation and testing program, different approaches were evaluated. Naturally, a single turbocharger could be developed for this type engine and could have resulted in great power increases.[7] No immediate solution to the problems at hand could be offered. However, existing AiResearch turbochargers could be adapted easily; their performance characteristics seemed to match the foreseeable requirements quite well; thus, the decision fell in favor of this approach. Four major objectives or limitations should be kept in mind when assessing results obtained so far. (1) Road tests with turbocharging were to be conducted during the hot season in 1956. (2) If at all possible, next to no change in powerplant installation and locomotive configuration should occur. (3) The program was to be held as economical as feasible. (4) Basic performance testing other than on Union Pacific's so-called load box and in rail operation was not anticipated.

Fig. 7 shows an AiResearch turbocharger of the T30 model family which was selected for the locomotive engine. It consists of a centrifugal compressor and a centripetal turbine, uses engine oil for lubrication, and requires no water-cooling. More detailed descriptions have been published.[8] These turbochargers had been developed for heavy-duty operation and demonstrated good performance and very satisfactory operation. The multiple installation, that is one turbocharger per four cylinders, i.e. four units per engine, was a natural choice.

Before proceeding with the description of the installation, a word about operational advantages of multiple installation seems indicated.

When turbocharging, the turbo rpm and, thereby, the boost pressure depend on the engine load condition. With pneumatic connection being the only one between engine and turbocharger, evidently the latter cannot follow changes in engine loading instantaneously; the change in boost pressure is somewhat delayed. This fact must be understood and faced; it requires particular attention with all kinds of vehicle powerplants. One of the various means to obtain quick engine response and good accelerability with a turbocharger suggests itself immediately: keep the moment of inertia of the turbo rotor low! And this requirement is, as can be shown, more easily met when using the units employed in multiple installations. In the case of the T30 units, for instance, the weight of the turbo rotor is only just a few pounds. That, additionally, the multiple installation eases the problem of low manifold losses (and cost) and offers best exhaust energy utilization requires no further explanation. Of course, one of the prerequisites for the multiple unit approach is small yet highly efficient turbomachinery. That "small" and "efficient" do not exclude one another can be considered well proven (although still often doubted!).

By removing the exhaust collectors of the standard EMD 16-567C engine, the space required for the turbochargers was readily available. The turbocharger installation on top of the engine as compared to the four collectors of the standard engine did not impose additional mounting loads on the engine casing. Several exhaust collector configurations were studied taking into account the given inlet port and exhaust valve timing. For time reasons, tests to optimize the collector system could not be conducted. The exhaust gas from the four turbochargers was collected into the two existing exhaust stacks projecting through the locomotive roof, although this arrangement has an undesirable high pressure loss restricting the benefits of turbocharging. The four turbochargers operate in the manner called "free-floating" with no controls in air and gas ducts. For the experimental program, the standard engine air filters were not retained since, with the panel filters in the sides of the locomotive body, intake air contamination was not feared. Inside the locomotive body, air entered the turbochargers through calibrated bell-mouth-shaped inlet ducts. On the discharge side, the compressed air was collected in two parallel manifolds running along the top of the engine. Naturally, the Roots-blowers remained on the engine and, in order to keep the air temperature to the engine as low as possible, charge-air intercoolers were attached to the inlet flange of the Roots-blowers, Fig. 8. These heat exchangers, another product of The Garrett Corporation, were also of an experimental nature being built from aluminum with finned-plate core type using water of the engine cooling system as heat sink with this heat rejection only amounting to a maximum of 6 per cent of the engine heat rejection. The maximum pressure drop on the charge-air side of the heat exchanger was .5 in.Hg; 2 psi pressure drop were observed on the water side.

The analysis of the performance to be expected through turbocharging as well as a choice of the best nozzle areas for the turbines were severely handicapped because of the lack of precise knowledge of the engine characteristics. Nonetheless, the results of such analytical work were considered sufficiently encouraging to undertake the building and testing of one locomotive unit initially.

The manifolding for the first locomotive installation, exclusively anticipated for stationary testing, and little, if any, road testing, was built by the Union Pacific shops in Omaha in a very short time. There were no provisions for flexible members such as bellows or sliding joints. Duct areas were guessed at more than calculated and certain obvious objectionable features of this ducting had to be accepted for time and manpower reasons. Initial testing in the Los Angeles yards of Union Pacific commenced only a few weeks after this approach had been decided upon. Having practically no established engine data, it was rather difficult to predict, as AiResearch usually does, the best suited nozzle area for the four turbines. Therefore, three nozzle ring sizes were made available which, in order to correlate their dimensions, we shall call 90 per cent, 100 per cent, and 120 per cent. Exchanging nozzle rings in AiResearch turbochargers is a very simple and quick matter not imposing disassembly difficulties of the turbocharger itself or of much of the installation. Calculations indicated that whereas the standard engine under sea level conditions had an air supply of between 400 and 450 lb per min at full load, the components of the turbocharger system should be capable of 550 to 600 lb per min. During the testing program, concern regarding air heating was substantiated. In one test series, for instance, at an ambient air temperature of 54F, air inlet temperatures to the turbochargers as high as 110 deg were measured.

The only means to determine engine power output was a resistor type load box absorbing the output of the engine's main generator with the locomotive stationary. Naturally, for the first run, the 120 per cent area nozzles were installed and for simplicity reasons the charge-air intercoolers were omitted. These very first tests indicated a satisfactory operational compatibility of the added system with the engine which ran smoothly and demonstrated satisfactory acceleration characteristics. Although the power boost with this large nozzle area was small, the initial runs resulted in improved exhaust smoke conditions upon engine acceleration. Naturally, the additional pre-compression of the air resulted in an air temperature in the airbox approximately 40 deg above normal yet the density increase of the air to the cylinders must have amounted to around 16 per cent.

As expected, a smaller nozzle area was indicated and both the 100 per cent and 90 per cent rings were compared resulting in the decision to install the 100 per cent size. At the same time, the intercoolers were included in the system.

As stated repeatedly, the normal obstacles in such performance work outside of a real laboratory setup slightly impaired the accuracy of the testing. Nonetheless, the results shown in Fig. 9 demonstrate the average performance measured. Power measurements up to over 2,300 hp were easily obtained under sea level conditions with 1,800 hp resulting from the same rack setting as 1,750 hp on a new standard power unit. During the altitude testing, the normally reduced rating of 1,600 hp or less at 4,000 ft elevation with .96 rack setting was measured to have improved to 1,730 hp through turbocharging. The altitude tests had not been completed at the time of this writing since conditions made it impossible to set the engine up at an elevation over 8,000 ft as originally planned.

Shortly after brief load box testing, the locomotive unit was assigned to pusher service over El Cajon pass where it was used for several hundred hours of duty. With the described experimental type ducting, occasional cracks on the hot side were no surprise. Naturally, the operating personnel observed the decrease in exhaust noise that usually accompanies turbocharging.

During the entire time of testing and rail operation, no mechanical or functional trouble occurred in the turbocharger units. After close to 2,000 hours total running time, they were fully disassembled and thoroughly inspected dimensionally. Everything was found to be in an excellent condition with no indication of distress or wear. A few small knicks on two turbine blades resulting from some engine piece going through were found. In radial turbine wheels, such damages, sometimes not just small in size, cause little, if any, concern, are usually not even discovered in operation, and are just smoothed out at overhaul periods. The trouble-free operation of the turbochargers could be expected because, in the application described, the operational speed of the turbochargers always stays below approximately 75 per cent of their rated speed for continuous duty, and the temperature of the exhaust gas never got closer than within about 300 deg of the rated maximum temperature for these turbochargers.

REFERENCES

1) t_{air} = 90F; η_{tot} = .55; pressures at turbine exit and compressor inlet equal ambient pressure.

For details see: "Superchargers and Their Comparative Performance" By W. T. von der Nuell, SAE Quarterly Transactions October, 1952

2) Diesel Progress, June, 1956, p. 34

3) Diesel Power, Sept. 1953, p. 36

4) Diesel Engine Catalog, Vol. 21 (1956-57) p. 212

5) Calculations implied here were oversimplified for reasons of demonstration.

6) Certain simplifying assumptions within the boundaries of approximate geometrical similarity were made.

7) "High Supercharging Development of a GM 16-278A Two-Stroke Cycle Engine" by W. G. Payne and W. S. Lang, SAE Transactions, Vol. 63, 1955

8) Diesel Engine Catalog, Vol. 21

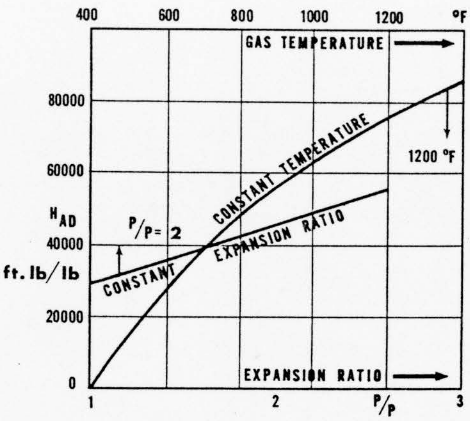

FIG. 1

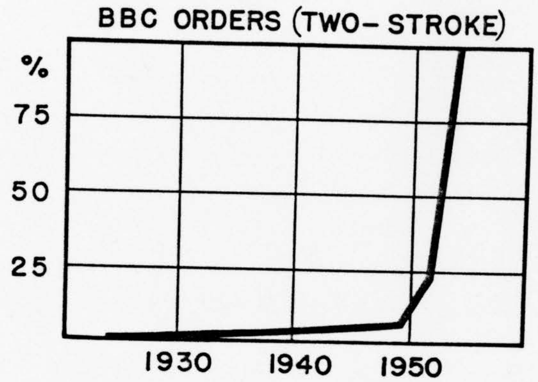

FIG. 2

BBC: TURBOCHARGER AND TWO-STROKE DIESEL ENGINE

		WITH TURBOCHARGER	WITHOUT TURBOCHARGER
ENGINE OUTPUT	H.P	11 250	11 000
SPEED OF ENGINE	R.P.M	115	115
NUMBER OF CYL.'S		9	12
WEIGHT OF ENGINE	t	450	600
LENGTH OF ENGINE	m	17.4	21.3

FIG. 3

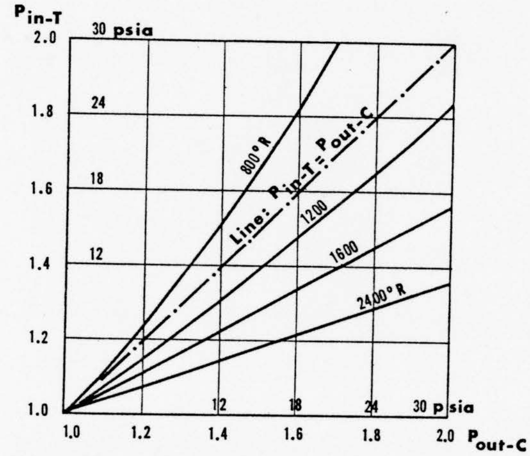

FIG. 4

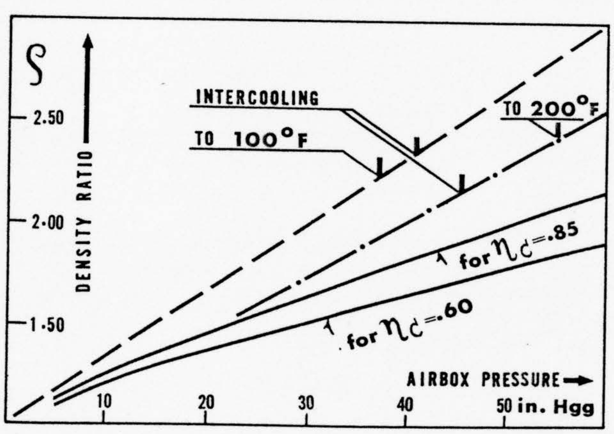

FIG. 5

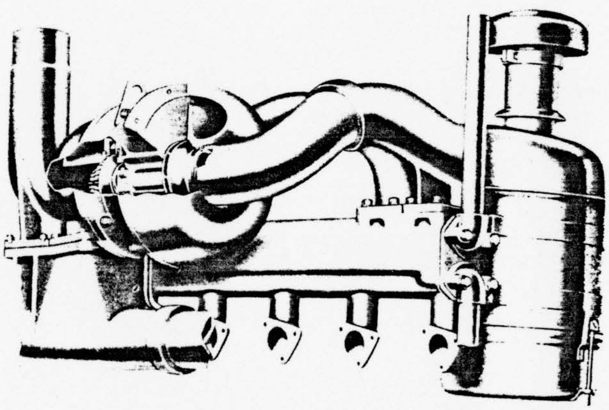

FIG. 6

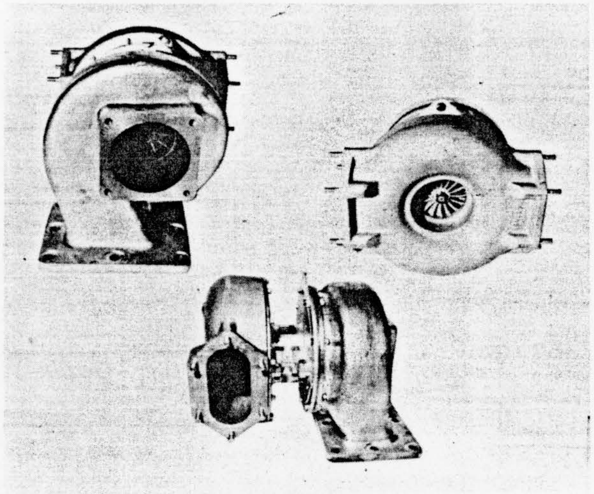

FIG. 7

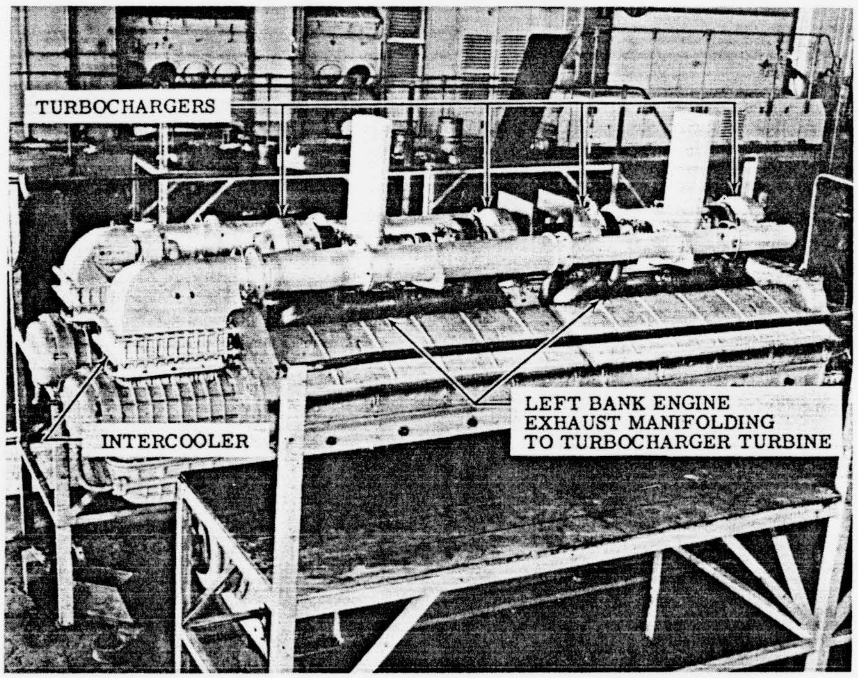

FIG. 8

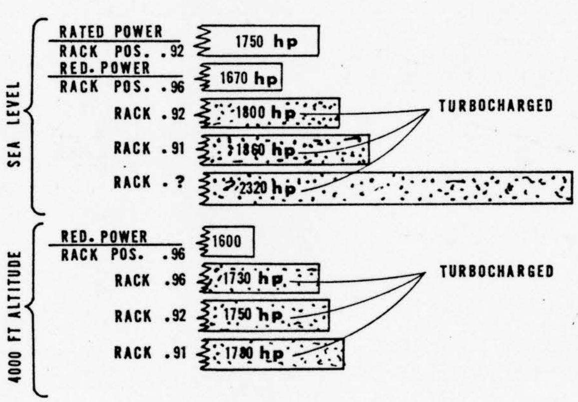

FIG. 9