DISSERTATION

ON

INVESTIGATION

OF

ENGINE SYSTEM RESPONSE

TO

VARIATION OF AIR INTAKE PRESSURE

Guide :

Col TV Sadasivan

Done by :

Maj Abhijit Das

JAWAHARLAL NEHRU UNIVERSITY, NEW DELHI

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CERTIFICATE

Certified that Dissertation on Investigation of Engine System Response to Variation of Air Intake Pressure submitted by Maj Abhijit Das during the year 1999 in partial fulfilment of the requirement for the award of the Degree of Master of Technology in Mechanical Engineering by the Jawaharlal Nehru University, New Delhi is a record of student's own work carried out by him under my supervision and guidance.

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INVESTIGATION OF ENGINE SYSTEM RESPONSE TO VARIATION OF AIR INTAKE PRESSURE

BY

MAJOR ABHIJIT DAS

DISSERTATION

SUBMITTED IN PARTIAL FULFILMENT

OF THE REQUIREMENTS

FOR THE AWARD OF THE DEGREE OF

MASTER OF TECHNOLOGY

IN

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GUIDE : COL T.V. SADASIVAN, ME,FIE

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CONTENTS

LIST OF FIGURES & TABLES

CHAPTER I

CHAPTER III

CHAPTER V

NOTATIONS

SYMBOLS	DESCRIPTION
$\dot{m}$	Air flow rate
N	Engine speed
$\rho_m = \rho_2$	Inlet manifold density
V_{SW}	Swept volume of engine
η_{Vol}	Volumetric efficiency
T_1	Ambient temp or temp at compressor input
T_2	Temp at compressor output or engine inlet
T_3	Temp at engine outlet or turbocharger inlet
T_4	Temp of exhaust gas
P_1	Ambient pressure or pressure at compressor input
P_2	Pressure at compressor output or engine inlet
P_3	Pressure at engine outlet or turbocharger inlet
P_4	Pressure of exhaust gas
R	Gas constant
C_d	Coefficient of discharge
A	Mean flow area
ϵ	Charge cooler effectiveness
T_{2c}	Temp at compressor exit
T_2	Temp at cooler exit
T_{Cool}	Temp of coolant charge

{for eqns 7, 8 and 9 under Charge cooling condition}

η_i	Indicated thermal efficiency
η_{mech}	Mechanical efficiency of engine
ISFC	Indicated specific fuel consumption
BSFC	Brake specific fuel consumption
CV	Lower calorific value of fuel
$\dot{W}$	Power output
$\dot{m}_f$	Fuel consumption
$\dot{m}_a$	Air consumption
F	Equivalence ratio
AFR	Air – Fuel ratio
$\left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{st}$	Stiochiometric air fuel ratio
C_p	Constant pressure

SYNOPSIS

Diesel engines have extensively found their use in the automobile and industrial fields all over the world due to their durability and better system responses. Any naturally aspirated engines draws air of the same density as the ambient atmosphere, based on which, the various system responses like BMEP, Torque, SFC, Power output etc. are determined. This is because of the fact that this sir density dictates the maximum weight of fuel that can be effectively burned per working stroke of the cylinder, which, in turn, determines the maximum power output that can be developed by the engine.

By increasing the density of charge air between the air intake and cylinder, the weight of the induced per working stroke is increased and thereby, a greater weight of fuel can be burned, with the consequent improvement in the system response.

In this dissertation the above made statement has been put to test in form of an experiment on a D0026 M 8A 4-stroke diesel engine which is fitted on one of the most extensively used load carriers of the Indian Army i.e, truck " Shaktiman" . The effect on various system responses under different conditions on a number of engines have been discussed and the effect on the Torque and Power developed by the D0026 M 8A diesel engine have been investigated under varying air intake pressures. An analysis of the results obtained has been done and an optimum limit of increasing the air intake pressure without affecting the engine parameters has been worked out.

CHAPTER - I

INTRODUCTION

1. A naturally aspirated engine draws air of the same density as the ambient atmosphere . Taken by and large, the density of the atmosphere that Nature has provided for us at sea level, we may take as much as we please, as a free issue, as long as we are content to accept the density she offers us.

2. This air density determines the maximum weight of fuel that can be effectively burned per working stroke of the cylinder and it also determines the maximum power that can be developed by the engine.

3. By increasing the density of charge air between the air intake and cylinder, the weight of air induced per working stroke is increased and thereby, a greater weight of fuel can be burnt, with a consequent increase in the power output. The increase in density of air between the air intake and the cylinders to achieve a higher power output can be accomplished with the help of a compressor, though, at the same time, it is important to remember that the power expended in driving the compressor will influence the operating efficiency of the engine. Thus it is relatively uneconomical to drive the compressor mechanically form the engine by some chain or gear drive as some additional power will be thereby absorbed and there will be an increase in specific fuel consumption for the extra power obtained.

4. Not with -standing the statement made above, an increase in air intake pressure or pressure charging will have the following advantages:-

(a) A substantial increase in the engine power output for any stated size and piston speed, or conversely, a substantial reduction in engine dimensions and weight for any stated horsepower.

(b) An appreciable reduction in the specific fuel consumption rate at all engine loads.

(c) A reduction in initial engine cost, and

(d) Increased reliability and reduced maintenance costs, resulting form less-exacting condition at the cylinders.

5. The power output at the piston head of any internal combustion engine is directly proportional to the product of the weight of air it can consume in unit time, multiplied by the thermal efficiency at which it is employed . The useful power output at the crankshaft is the same, less the internal frictional losses of the engine and the proportion of power that is consumed by the compressor, as has been stated earlier.

6. For any given capacity of engine, and for any given thermal efficiency, we can double the indicated power either by doubling the speed of rotation or by doubling the density if air. The former is not usually practicable, for it is to be presumed atleast that we are already running engine as fast as prudence permits, and even if we cast prudence to the winds, we should still be bulked, on one hand by inadequate breathing capacity, and on

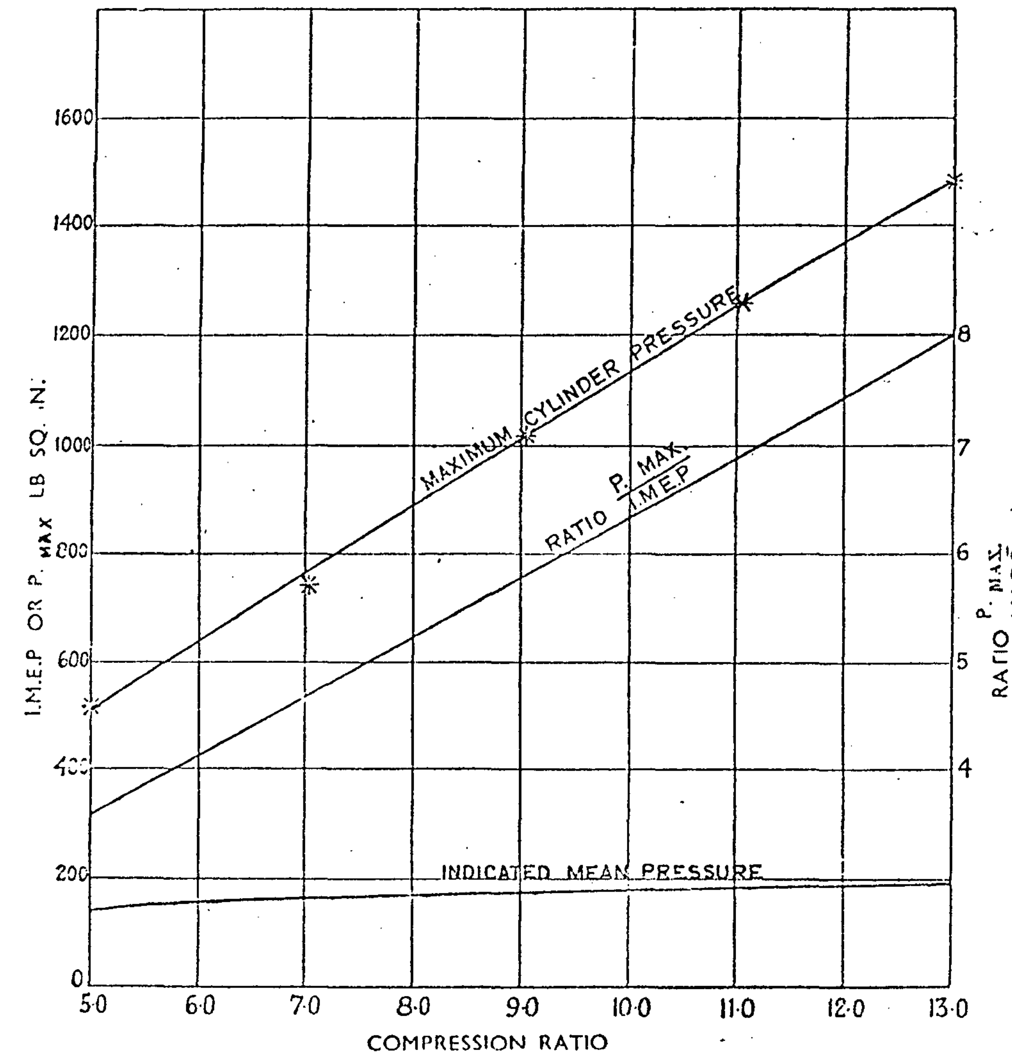


Fig. 1.1.—Relationship between maximum and mean indicated pressure as recorded on the small variable-compression E6 engine when using a non-detonating fuel

the other, by the mounting internal frictional losses, due to excessive dynamic forces.

7. We can increase the thermal efficiency by raising the ratio of compression or expansion but in the case of a spark ignition engine, we are restricted in this direction both by tendency of fuel to detonate and by the very rapidly increasing maximum pressures. In round figures the maximum peak pressure of a spark-ignition engine increase by 120 lb. per sq. in. for each ratio of compression. The curve of thermal efficiency, however, tends to flatten out to such an extent that, taking into account the higher mechanical losses resulting form high peak pressure, there is little to be gained form the use of a compression ratio higher than about 7.5:1 or 8.0:1 if the mean load factor is very low, as in ordinary road-vehicle engines. In the case of racing-car engines, where the load factor is higher and we can afford to go to extreme lengths and take some risks in the way of cutting down mechanical losses, it may pay to use a compression ratio of 9:1 or 10:1.

8. Fig. 1.1 shows the observed relation between maximum peak and indicated mean pressure with optimum ignition timing and maximum-power mixture strength, over the range of compression form 5.0 to 13.0 :1 as measured in the small variable- compression "E6" engine , but the readings in fact differ very little from the theoretical values. From this it will be seen that the ratio of maximum to mean pressure range from 3.6 :1 at a compression ratio of 5.0:1, to 7.9 :1 at a ratio of 13.0:1.

9. By contrast when supercharging, the ratio of maximum to mean cylinder pressure remains virtually constant irrespective of the supercharge pressure.

General Considerations.

10. When an engine is inhaling form, and exhausting into the same atmosphere, the volumetric efficiency is a function of the volume swept by the piston, and is independent, or at most independent , of the capacity of the clearance space. Hence, except for a quite small secondary effect, which is due to direct loss of heat to the cylinder walls during the brief period between the effective end of the exhaust and effective beginning of the suction, the volumetric efficiency in-dependent of the compression ratio. However, when the engine is inhaling from a source at higher pressure than that against which it is exhausting, this does not hold good any longer because the clearance volume as well as the swept volume becomes highly charged to an extent depending upon the difference in the pressure between the entering air and the residual exhaust product. Let us suppose, for example, that the volume of clearance space is 20 percent of the swept volume, and that the air intake pressure is twice that of the surrounding atmosphere, then, neglecting secondary factors, half the clearance volume will be charged with fresh air . Under these conditions, the effective cylinder volume will be increased by 10 percent and the power increase will be not 100 percent but 120 percent. Thus when the thermal efficiencies are same, then the lower the compression

ratio and the larger the clearance volume the greater would be the power response to increase in air intake pressure.

11. When a mechanically driven supercharger is employed the exhaust is, of course, discharged against atmospheric pressure only, and full advantage can be taken of the supercharging of the clearance volume, but when an exhaust turbine is used to drive the supercharger, this imposes a certain amount of hack pressure which may well be equal to or even greater than, the supercharge. By dividing up the exhaust system, so that in any one pipe the exhaust flows just do not overlap, i.e. by using, in the case of a four-cycle engine, a separate short exhaust pipe for each group of three cylinders, it is possible to reduce considerably the terminal exhaust pressure, for some use can then be made of the Kinetic energy in the exhaust both to help the turbine at the beginning, and to make use of it as an exhauster at the end of each exhaust stroke, but the extent to which this can be done in practice is generally restricted by geographical and plumbing limitations

12. When we vary the air intake pressure, it is of vital importance to keep its temperature as low as possible not only from the obvious motive of getting the maximum possible weight of air in the cylinder, but also because

(a) In the case of a spark ignition engine, the higher the initial air temperature, the greater the tendency to detonation or pre-ignition.

(b) In any form of internal combustion engine, the temperature range throughout the whole cycle is a function of the initial temperature. The higher the initial temperature, the greater the losses, due both to direct heat loss to the walls and to dissociation etc. Thus the thermal, as well as the volumetive efficiencies are reduced with every increment of air temperature.

(c) All the mechanical troubles arising from high temperature, such as piston failure, ring sticking, and exhaust-valve troubles, are, of course, accentuated greatly by any increase in the cycle temperature.

13. Let us consider as an example, the effect of a difference of say 100°C in the temperature of pressure charging in spark ignition and a compression ignition engine.

(a) In a spark ignition engine of average compression ratio, the absolute temperature at the end of compression, that is, before the liberation of heat, will be nearly twice the initial temperature and thus the difference at the effective starting point of the cycle, therefore becomes 200° C and the flame temperature will be increased by nearly a like amount. Thus both the direct heat losses and those due to increase of specific heat and dissociation will all be increased very considerably, with a corresponding reduction in thermal efficiency .

(b) In the case of a compression ignition engine where the compression ratio is much higher, a difference of 100° C in the initial temperature means a difference of nearly 300° C at the end of compression and therefore through-out the rest of the cycle. Moreover, the CI engine is more susceptible to direct heat loss. On the other hand while any increase in the compression or flame temperature tends to promote detonation or pre-ignition in the spark ignition engine, this does not apply to the compression ignition engines.

14. It is thus essential on the grounds of temperature that the adiabatic efficiency of the supercharger shall be as high as possible and, wherever possible, effective intercooling between the supercharger and the engine cylinder should be provided; the higher the degree of supercharge, the more important does this become.

Charge Air Cooling

15. The increased weight or density of air introduced into the cylinder by pressure charging enables a greater weight of fuel to be burnt, and this in turn brings about an increase in power output. The increase in airdensity is, however, fractionally offset by the increase of air temperature resulting form adiabatic compression in the turboblower, the amount of which is dependent on compressor efficiency. This reduction of air density due to increased temperature implies a loss of potential power for a stated amount of pressure charging. For example, at a charge air pressure of, say 0.35 bar, the temperature rise is of the order of 33°C--- equivalent to a 10% reduction in charge air density. As the amount of pressure charging is increased the

effect of turboblower temperature rise becomes more pronounced. Thus for a charge air pressure of 0.7 bar, the temperature rise is some 60 °C, which is equivalent to a reduction of 17% in the charge air density.

16. Much of this potential loss can be recovered by the use of charge air coolers. For moderate amounts of pressure charging cooling of the charge air is not worthwhile, but for two-stroke engines especially, it is an advantage to fit charge air coolers which are standard on all makes of two-stroke and most medium-speed four –stroke engines .

17. Charge air cooling has a double effect on engine performance. By increasing the charge air density, it thereby increases the weight of air flowing into the cylinders, and by lowering the air temperature it reduces the maximum cylinder pressure, the exhaust temperature and the engine thermal loading. The increased power is obtained without loss, and in fact , with an improvement in fuel economy. It is important that charge air coolers should be designed for low pressure drop on the air side; otherwise, to obtain the required air pressure the turboblower speed must be increased.

18. The most common type of cooler is the water cooled design with finned tubes in a casing carrying seawater over which the air passes. To ensure satisfactory effectiveness and a minimum pressure drop on the charge air side and on the water side, the coolers are designed for air speeds of around 11m/sec and water speeds in the tubes of 0.75 m/sec. Charge air cooler effectiveness is defined as the ratio of charge air temperature drop to

available temperature drop between the air inlet temperature and cooling water inlet temperature. This ratio is approximately 0.8.

Scavenging

19. It is essential that each cylinder should be adequately scavenged of gas before a fresh charge of air is compressed, otherwise this fresh air charge is contaminated by residual exhaust gases from the previous cycle. Further, the cycle temperature will be unnecessarily high if the air charge is heated by mixing with residual gases and by contact with hot cylinders and pistons.

20. In the exhaust turbocharged engine the necessary scavenging is obtained by providing a satisfactory pressure difference between the air manifold and the exhaust manifold. The air flow through the cylinder during the overlap period has a valuable cooling effect; it helps to increase the volumetric efficiency and to ensure a low cycle temperature. Also, the relatively cooler exhaust allows a higher engine output to be obtained before the exhaust temperature imposes a limitation on the satisfactory operation of the turbine blades.

21. In two-stroke engine the exhaust/scavenge overlap is necessarily limited by the engine design characteristics. In Figure 1.2 a comparison of the exhaust and scavenge events for poppet valve engines and opposed piston engine is given. In the poppet valve engine the camshaft lost motion coupling enables the exhaust pre- opening angle to be 52° ahead and astern. In the opposed piston engine the exhaust pre-opening angle is only 34°

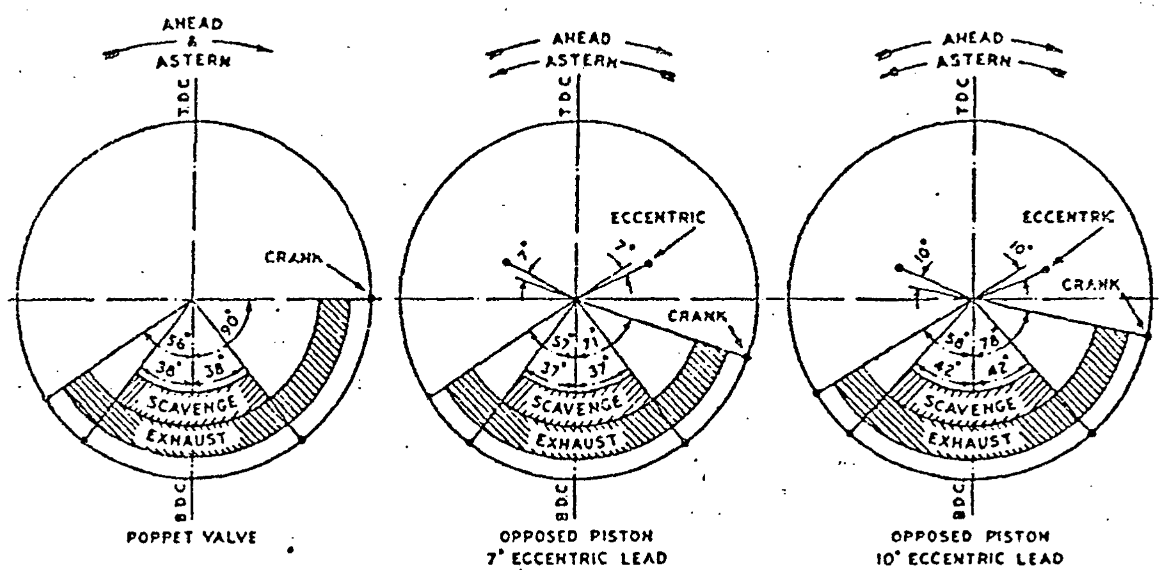


Figure 1·2 Single-acting two-stroke engine timing

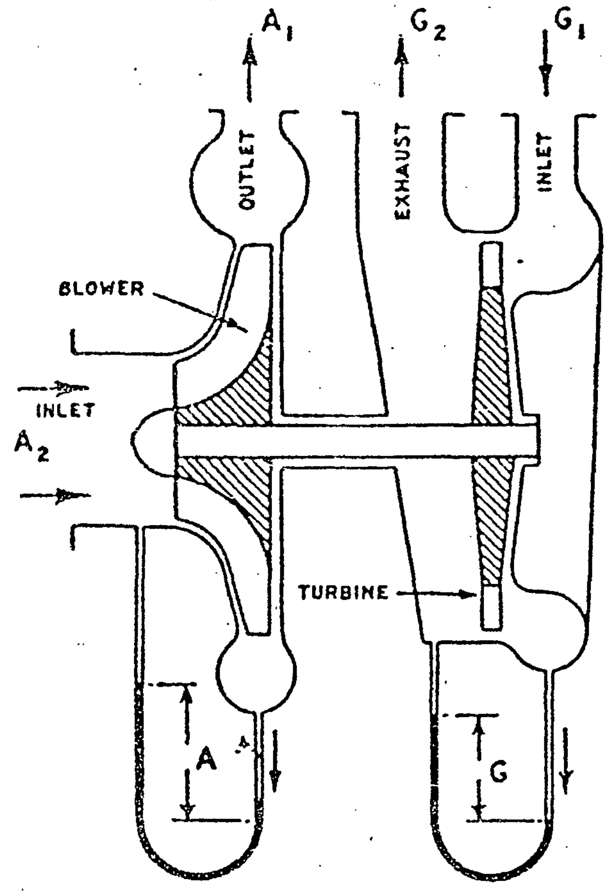


Figure 1.3 Scavenge gradient

ahead and 20° astern. Against this, however, the rate of port opening in opposed piston engines is quicker than in poppet valve engines. It should be emphasised , however, that opposed piston slow-speed engines are no longer in production and poppet valves are used in the majority of new designs.

22. In the four-stroke engine the substantial increase in power per cylinder, obtained from turbocharging, is achieved without increase of cylinder temperature. In the two- stroke engine the augmented cylinder loading, is without significance.

23. A strange fact is that the engine exhaust gas raises the blower air to a pressure level greater than the mean pressure of the exhaust gas itself. This is because of the utilisation of the kinetic energy of the exhaust gas leaving the cylinder, and the energy of the heat drop as the gas passes through the turbine. In Figure 1.3 the blower pressure gradient A exceeds the turbine pressure gradient G by the amount of the scavenge gradient. The design of the engine exhaust pipe system can have an important influence on the performance of turboblowers.

24. The test results of a turbocharged engine of both two-and four- stroke type will show that there is an increase in temperature of the exhaust gas between the cylinder exhaust branch and the turbine inlet branch, the rise being sometimes as much as 95°C. The reason for this apparent anomaly is that the kinetic energy of the hot gas leaving the cylinder is converted, in part, into additional heat energy as it adiabatically compresses the column of

gas ahead of it until, at the turbine inlet, the temperature exceeds that at the cylinder branch. At the turbine some of the heat energy is converted into horsepower, lowering the gas temperature somewhat, with the gas passing out of the turbine to atmosphere or to a waste heat recovery boiler for further conversion to energy. Though the last thing to emerge from the exhaust ports or valves is slug of cold scavenge air which can have a cooling effect on the recording thermometer, adiabatic compression can still be accepted as the chief cause of the temperature rise.

Variation of air intake pressure in Spark ignition engine

25. Apart form the direct increase in power due to the greater weight of air inhaled per cycle , the effects of variation of air intake pressure upon spark ignition engine are :

(a) The increased density and somewhat increased temperature both tend to speed up the combustion process not only by reducing the delay period, but also by accelerating the rate of speed of inflammation. In this latter respect, the effect is similar to an increase in the degree of turbulence. Since, in most modern high- duty petrol engines, the degree of turbulence is already adequate, the effect of variation of air intake pressure will be rendering the engine more sensitive to mixture ratio and tending to narrow down the range of burning at both the reach and weak ends. For best results under such condition of varying air intake pressure, the normal degree of

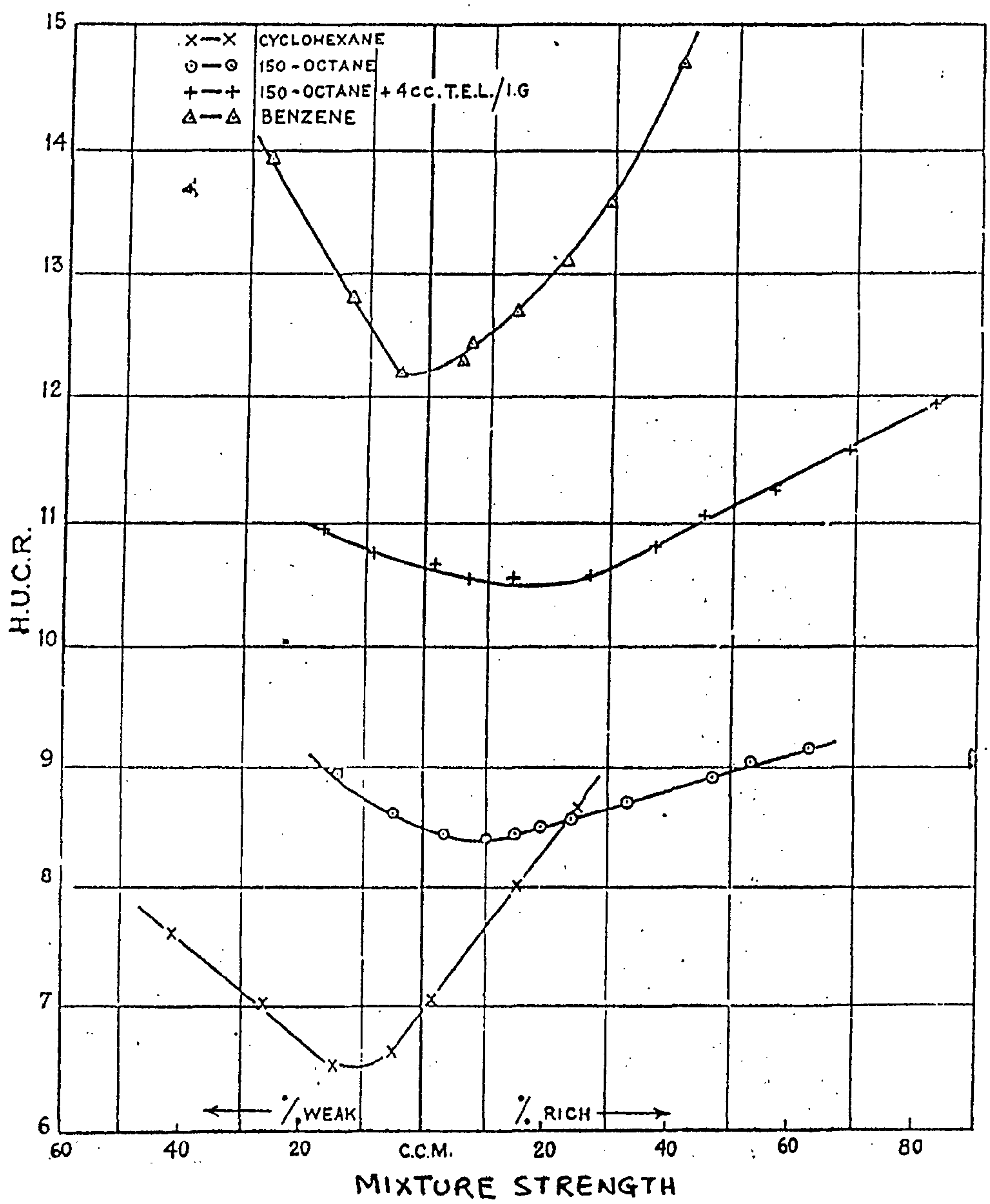


Fig 1.4 - H.U.C.R mixture strength curves on representative fuels

Test conditions:

Engine speed 1500 rpm. Engine Coolant temp 90°C

Inlet air temp 120°C Solex carburettor

Ignition advance to give maximum cyl pr 13° after TDC in all cases turbulence should be some what below rather than above the optimum.

(b) The increased density and temperature increase, of course, the tendency both the detonation and to pre-ignition and thus set a limit to degree of variation of air intake pressure that can be employed in the petrol engine. Here it is difficult to generalise, for some fuels are more temperature and other more pressure-sensitive; thus two fuels rated at the same octane number by the usual technique may respond somewhat differently to the variation of air intake pressure. In the case of all volatile petroleum fuels, however, the tendency both to detonation and to pre-ignition can be reduced enormously by employing a very rich mixture of the order of 50 per cent to 60 per cent above the chemically correct value. The effect of this is two fold:

(i) All such fuels, and more especially those of high octane number, show, under the same temperature conditions, a greatly reduced tendency to detonate when a large excess of fuel is present. This is illustrated in fig 1.4, which shows variation in H.U.C.R. with mixture strength of iso-octane, cyclo-hexane, and benzene.

(ii) The latent heat of evaporation of the excess fuel serves to lower the intake temperature. In the case of fuels of the alcohol group, as used in racing - cars, this latter plays a very important

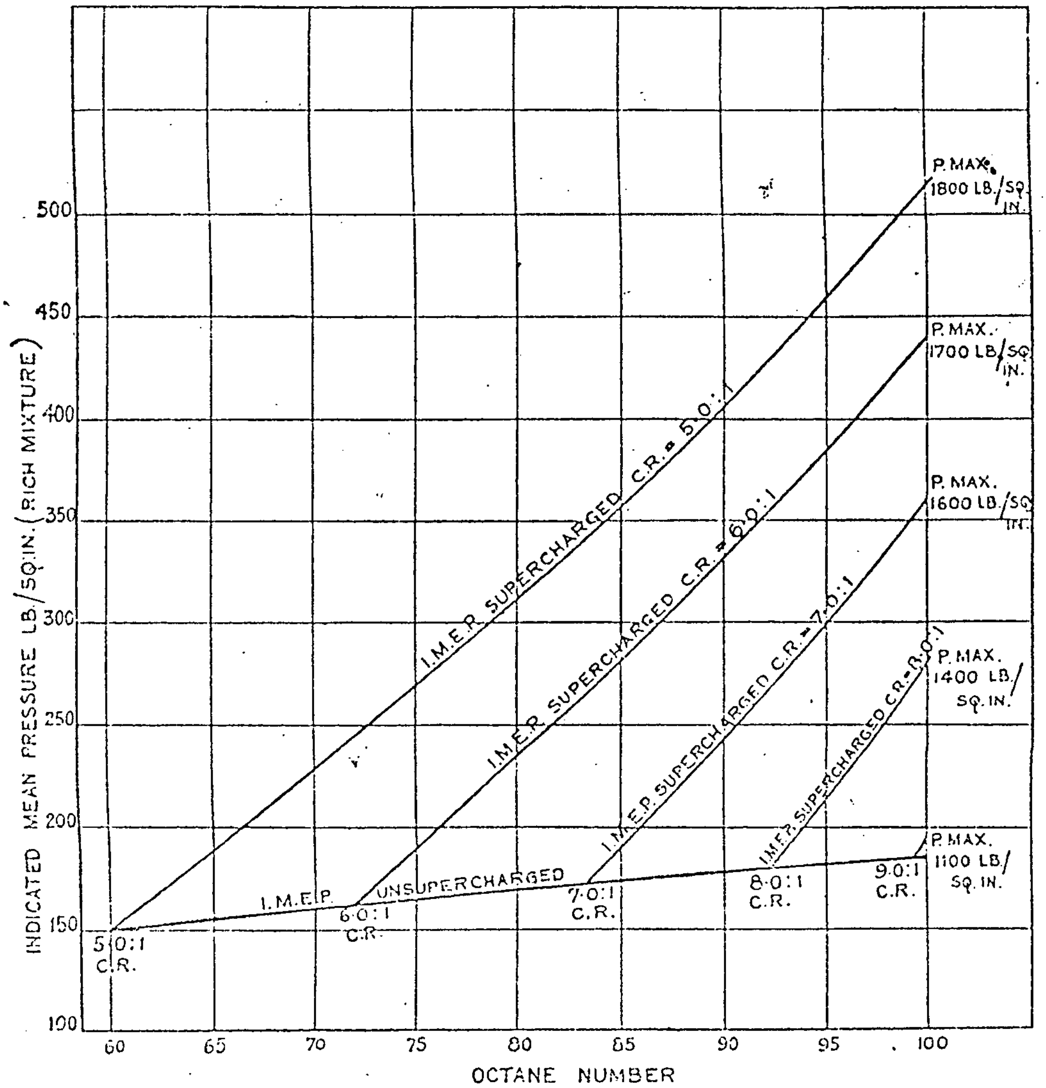


Fig. 1.5 —Graph of I.M.E.P. at different compression ratios over the range of octane numbers 60 to 100 with and without supercharging

part, but in that of hydrocarbon fuels, whose latent heat of eveaporation is relatively low, it cannot be a very important factor.

(c) The heat losses to the cylinder walls during combustion and expansion do not increase in direct proportion with the degree of variation of air intake pressure, e.g. with a increase of 2 atmospheres absolute at the same intake air temperature, the total flow of heat to the cooling medium is increased by only about 70 per cent. This should mean that the thermal efficiency would be slightly improved, but in most cases any gain in thermal efficiency resulting from reduced heat loss is largely offset by inability to run on so weak a mixture strength as when naturally aspirated. By the same token, of course, the temperature of the exhaust is increased, hence the exhaust valve in particular has a more trying time.

26. Fig. 1.5 is a composite graph complied from a very large number of tests carried out in the laboratory on variable-compression and other research units showing the relation between the octane number of the fuel and the indicated mean effective pressure as limited by the incidence of detonation, when:

(a) The compression ratio is raised until the incidence of detonation

(b) at any given ratio, the I.M.E.P. is increased by supercharging to the same limit.

27. It assumes that:

(a) The engine has an individual cylinder capacity of between 1.5 and 2.5 litres; if larger, the M.E.P. is would be reduced ; if smaller, they would be increased all along the line.

(b) The cylinder is liquid-cooled ; if air –cooled the cylinder wall temperatures would be higher and the performance reduced by the earlier incidence of detonation or pre-ignition or both.

(c) Poppet, not sleeve, valves are used. In the case of sleeve-valve engines nearly one ratio higher compression, or about 25 per cent more I.M.E.P. ,when supercharged, could be used throughout.

(d) The temperature of the supercharge is such as would be supplied by a blower having an adiabatic efficiency of 70 per cent. If intercooling be applied, then again the whole scale would be raised.

(e) The revolution speed lies between the limits 2000 and 3000 r.p.m. if lower, the limiting I.M.E.P. would be reduced.

(f) The fuels used throughout the range of octane numbers have normal average characteristics. This latter is perhaps the most dangerous assumption but, throughout all the experiments, care was taken to avoid the use of fuels whose response to temperature, pressure, or mixture strength differed widely from average values.

(g) All tests were carried out with a fuel/air ratio of between 50 and 60 per cent in excess of the chemically correct. Attempts to carry out

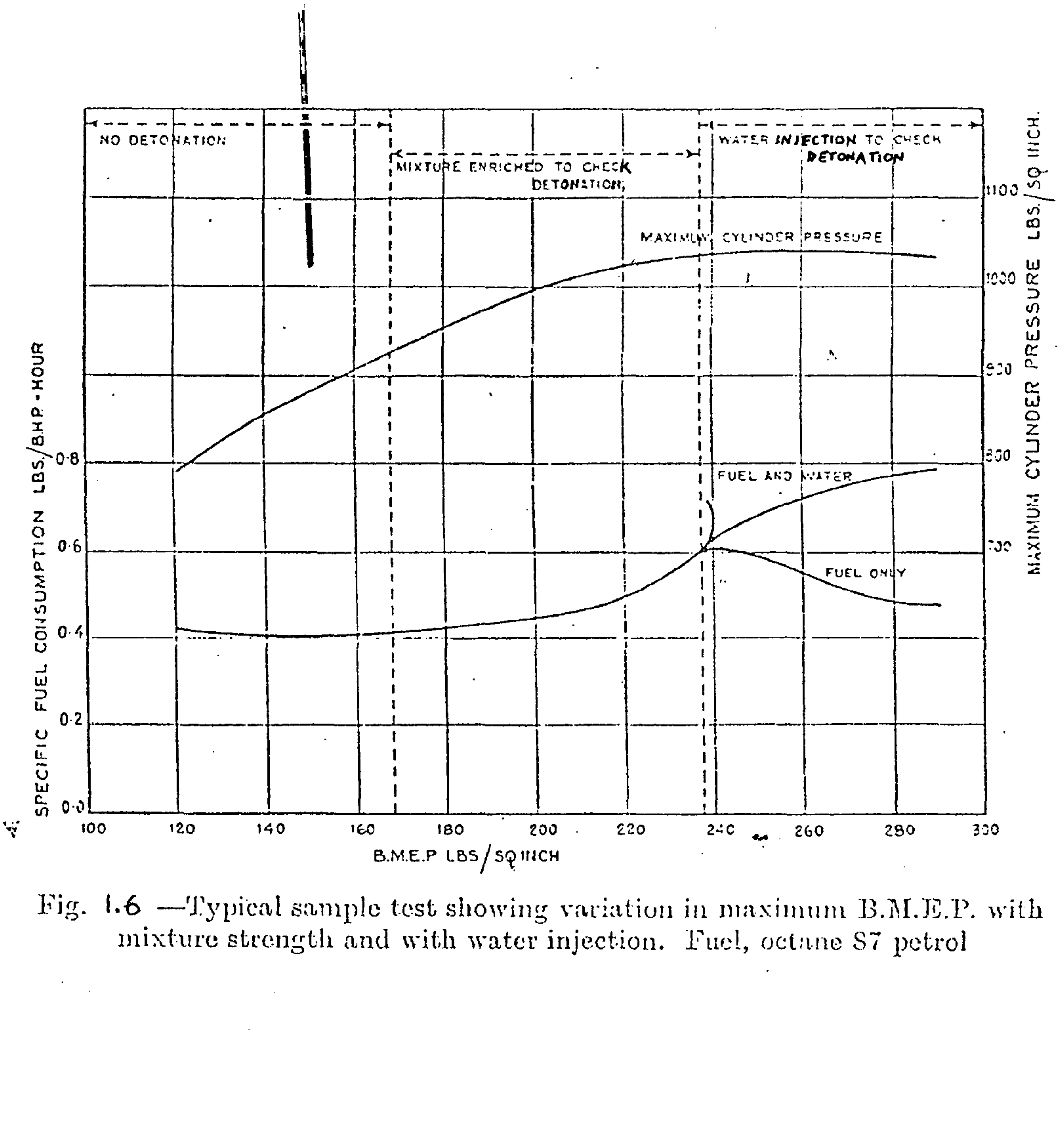


Fig. 1.6 —Typical sample test showing variation in maximum B.M.E.P. with mixture strength and with water injection. Fuel, octane 87 petrol

similar tests with a mixture strength giving minimum fuel consumption yielded such widely varying results, depending upon the chemical nature of the fuels, that no general conclusions could safely be drawn from them.

28. Such a graph may serve as a guide but, of course , must not be taken too literally, since so many variables enter the picture.

29 Fig.1.6 shows the results of an individual test, on a sleeve-valve engine, using 87 octane fuel and running throughout at a speed of 2500 r.p.m.

30 In this case, and with a compression ratio of 7.0:1, the engine was run on an economical mixture, i.e. about 10 per cent weak, and supercharge applied until the first incidence of detonation, which occurred when the B.M.E.P. had reached 168 lb sq. in. The mixture strength was then increased , step by step, and more supercharge applied, until the same intensity of detonation was recorded; this process was continued until a point was reached at which no further enrichment was effective. In fact after about 60 per cent excess fuel, not only did further enrichment have no effect but there was even some indication that it increased the tendency to detonate. A finely pulverized water spray was then delivered into the induction pipe, which served to suppress detonation, in part by the intercooling it provided , and in part by the influence of steam as an anti-detonant, and so allowed of further supercharging. This was continued progressively, admitting just sufficient

water at each stage to ward off detonation, until a B.M.E.P. of 290 lb. per. sq. in. was reached, which was found to be the limit of the dynamometer. At the same time it was noted that, with the addition of water, the influence of steam as an anti-knock allowed of the fuel/air ratio being much reduced. From this curve, fig.(1.6), it will be seen that under these operating conditions the limiting B.M.E.P. that could be reached with 87 octane petrol alone at an economical mixture strength was 168 lb. per. sq. in. (=200 lb I.M.E.P.). By enriching the mixture to the limit of usefulness the B.M.E.P. could be stepped up to 237 lb. Per. sq. in. (= 267 lb. I.M.E.P.). By the introduction of water, it could be further stepped up to 290 lb. per sq. in. or 319 lb. I.M..E.P., and probably more; at the time the fuel/air ratio could be reduced once again; in fact with water injection, no appreciable advantage was found from the use of an over-rich fuel/air mixture. It will be noted that the total specific consumption of liquid, i.e. fuel + water, is not so very much greater than when running on a very rich mixture of fuel alone.

31. The slope of the curve of maximum cylinder pressure is interesting, in that, after the injection of water it no longer rose but even tended very slightly to fall, and the same applied to the gross heat flow to the cooling water which reached a maximum at a B.M.E.P. of about 230 lb.per.sq. in., and thereafter fell off until, at a B.M.E.P, of 290 lb.per.sq.in. it had fallen to the same level as that at 170 lb. without water injection.

32. In this as in other tests, the supercharge air was supplied from an independent source but its temperature was adjusted by pre-heating to that which would be delivered by a supercharger having an adiabatic efficiency of 70 per cent and with no intercooling.

33. In another experimental set-up the engine used was a single-cylinder sleeve- valve research unit of 5 in. bore by 5.5 in. stroke, of very robust design, but the cylinder and cylinder head, which are of light alloy, are extremely light and of aircraft scantling.

34. With water alone, however evaporation of the water was by no means complete even at the highest degree of supercharge , and it was found more effective to add a substantial proportion of some volatile alcohol, such as methanol, in order to increase the volatility. Since for aircraft it was, in any case, desirable to add some sort of anti-freeze, it was considered that methanol would serve the dual purpose of protection against freezing and of increasing the volatility. Methanol, however, is very prone to pre-ignition and, on this score, it is unwise to use too great a concentration; the safe limit appears to be a 50/50 methanol/ water mixture.

35. Comparative tests, as between water injected separately and the same proportion of water dissolved in the fuel by the aid of a mutual solvent such as acetone, showed that the latter was far more effective, due presumably to the fact that, when admitted in solution in the fuel, evaporation was completed much earlier in the cycle and the full effect of its latent heat could

be realized. Again, experiments with the admission of dry steam served to show, independently, the effect of steam as a diluent and anti-knock, as distinct from that of water as a cooling agent, and thus to assess separately its two functions.

36. Where very high supercharge pressure are involved as for example, in the case of military aero-engine, the choice lay between intercooling by means of a heat exchanger or by the injection of water or water and methanol. The former , of course, could be maintained indefinitely but involved a good deal of extra drag and some additional large scale and rather vulnerable plumbing; the latter; because of the heavy consumption of liquid, could be but a temporary expedient only, but served admirably as a means of additional boosting for take off and for emergency use. In practice both methods were adopted the choice depending largely on the type and purpose of the aircraft to which the engine was fitted.

37. So far as spark-ignition engines are concerned, it would seem that, as applied to aircraft; supercharging is essential, not only as a means of restoring ground-level density at high altitude but also as a means of increasing greatly the specific power output of the engine in terms both of weight and of frontal area.

38. The limit of supercharge that can safely be applied is determined by:

(a) The octane number of the fuel.

(b) The ability of the engine to withstand the intensity of pressure and heat flow involved.

39. Both the tendency to detonate and the intensity of heat flow can be reduced by intercooling either by means of a heat exchanger or by water injection, or both ;by the latter means it was found possible to increase the power output by an additional 20 per cent without increasing either the tendency to detonate or the intensity of heat flow to the piston, cylinder walls, or exhaust valves.

40. In the case of engines of fairly large power which are called upon to exert full torque only at high speed, such as in aircraft or marine service, the centrifugal or axial-flow types of supercharger are probably the most suitable, since either can deal efficiently with far larger volumes of air than that can be handled by any form of positive blower of comparable dimensions or weight, but in service involving the exertion of high torque at low speeds, such as apply to all forms of road or rail transport, this type of supercharger is unsuitable, and resort must be had to some form of positive blower. There would seem, however, to be but little argument in favour of supercharging as applied to ordinary spark-ignition road-transport engines

because:

(a) The primary requirement of such engines is high torque at low revolutions, i.e. under the conditions where detonation would be most troublesome and insistent, thus necessitating the use either of a very high-octane and therefore expensive fuel, or some additional complication such as the addition of water or water-methanol mixture.

(b) The weight and space limitation is not nearly so severe as in aircraft , hence a larger unblown engine would seem to be preferable.

(c) At the present day, no form or rotary blower exists which will deliver a high supercharge pressure at low revolution speeds.

Variation of air intake pressure in compression ignition engines.

41. In the case of compression-ignition engines the picture is very different from spark ignition engine because:

(a) The bogies of detonation and pre-ignition are absent.

(b) The greater the density, the shorter the delay period, hence the smoother, more controllable and more complete the combustion.

(c) While it is essential to keep the temperature of the increased air intake as low as possible, yet an increase in temperature, though reducing both the volumetric and thermal efficiency, does not, as in the case of the petrol engine , give rise to detonation or pre- ignition. On the contrary, it tends, though only slightly, to assist the combustion process.

(d) The higher the supercharge pressure the less sensitive does the engine become to either the cetane number or the volatility of the fuel; hence a wider range of fuels can be used , but this applies, in full force, only so long as the increased air intake pressure can be maintained at all speeds and at all loads.

(e) The direct loss of otherwise recoverable heat is more serious in the C.I. than in the petrol engine, and since this loss increases only as about the 0.6^{th} power of the density, the gain in thermal efficiency, due to a reduction in relative heat loss, is greater in the C.I. than in the spark-ignition engine.

(f) There is evidence that in some, if not in most, forms of C. I. combustion chamber, the proportion of oxygen that can be consumed increases slightly with increase of density. Thus the return in power output is somewhat greater than would be expected from the direct increase in density and thermal efficiency. Experiments on a "Comet Mark III" engine, designed for very high pressures, showed that at the point of just visible smoke in the exhaust, and at the same induction air temperature, the proportion of air consumed ranged from 82 per cent with atmospheric induction up to86 per cent with an induction pressure of 3 atmospheres absolute.

(g) As compared with a spark-ignition engine ,the mechanical efficiency of the C.I. engine is considerably lower; hence it benefits more by any increase in the effective mean pressure, more especially so when such increase is not accompanied by a corresponding increase in maximum pressure. In the case of the " Comet Mark III" , for example, the optimum ratio of maximum to brake mean pressure at atmospheric induction was found to be 7.7:1, but with a supercharge of 1.66 atmos. abs., the ratio fell to 6.2:1, a very substantial gain. In the case of a direct-injection open-chamber engine, very much the same relationship as between atmospheric and supercharged conditions was found, but the ratio in both cases was somewhat higher.

(h) Given sufficient intercooling, the gain in both mechanical and thermal efficiency will more than compensate for the power absorbed by the blower at the higher load ranges, when the latter is driven mechanically, and at almost all load ranges when the exhaust energy is used to drive the blower, but so much of course depends both upon the efficiency of the blower and the degree of supercharge and intercooling that it is impossible to generalise as to the point in the load range beyond which the overall efficiency is improved by supercharging.

42. The following data and conclusion have been drawn from a series of experiments carried out in laboratories on a large number of high-speed C.I. engines of both the direct- and indirect injection type and with cylinder diameters ranging from 3 in. to 7 in. In the course of these experiments,

supercharge pressure up to a maximum of 7 atmospheres absolute were explored.

43. The above research may be divided into two categories:

(a) The use of a moderate degree of supercharge in engines of normal design , and at a compression ratio suitable for running under normally aspirated conditions.

(b) The use of a very high supercharge in engine of highly specialised design mostly, but not all, of the two-stroke cycle type intended to serve as the high-pressure elements of a compound system. In this case, and in order to keep the maximum cylinder pressures within practicable limits, a relatively low ratio of compression and a specialized design must be employed.

44. The latter category is probably justified only when full use is to be made of the large amount of potential energy left in the exhaust, and is of immediate practical interest only in connection with compound system, but the results obtained proved useful also as affording extreme points on the performance curves and thus serve to confirm that there is no change or break in the general trend of response to supercharge.

45. The conclusion, and the data from which they are drawn, relate primarily to the first category, though they are modified only in degree when applied to the second.

46. In the first category, it is postulated that the engine shall be of normal four-stroke type, having a compression ratio high enough to permit of normal cold-starting and of running under naturally aspirated conditions on light gas-oil, but modified only in such minor details as were found necessary or desirable.

47. All the data in this category have been obtained from high-speed engines, mostly from single-cylinder research units. It should be pointed out that all tests were run with air supplied from main compressing plant, thus all the data given relate to the gross power, no allowance being made for the work done in driving the compressor. On the other hand, the research units employed have, as compared with multi-cylinder engines, a low mechanical efficiency, hence all fuel consumption figures on a basis of brake horse-power are somewhat higher than normal.

48. The practical upper limit of supercharge is reached when the maximum cylinder pressure are such as to cause:

(a) "Scuffing" of the piston-rings and heavy liner wear.

(b) Overloading of the bearings.

(c) Leakage of the cylinder-head joints, due to springing of the cylinder-head bolts, etc.

49. With an engine of conventional design, with copper-lead bearings and a surface-hardened crankshaft, maximum cylinder pressures ranging up to 1200 lb. per.sq. in. are usually permissible, but this entails limiting the

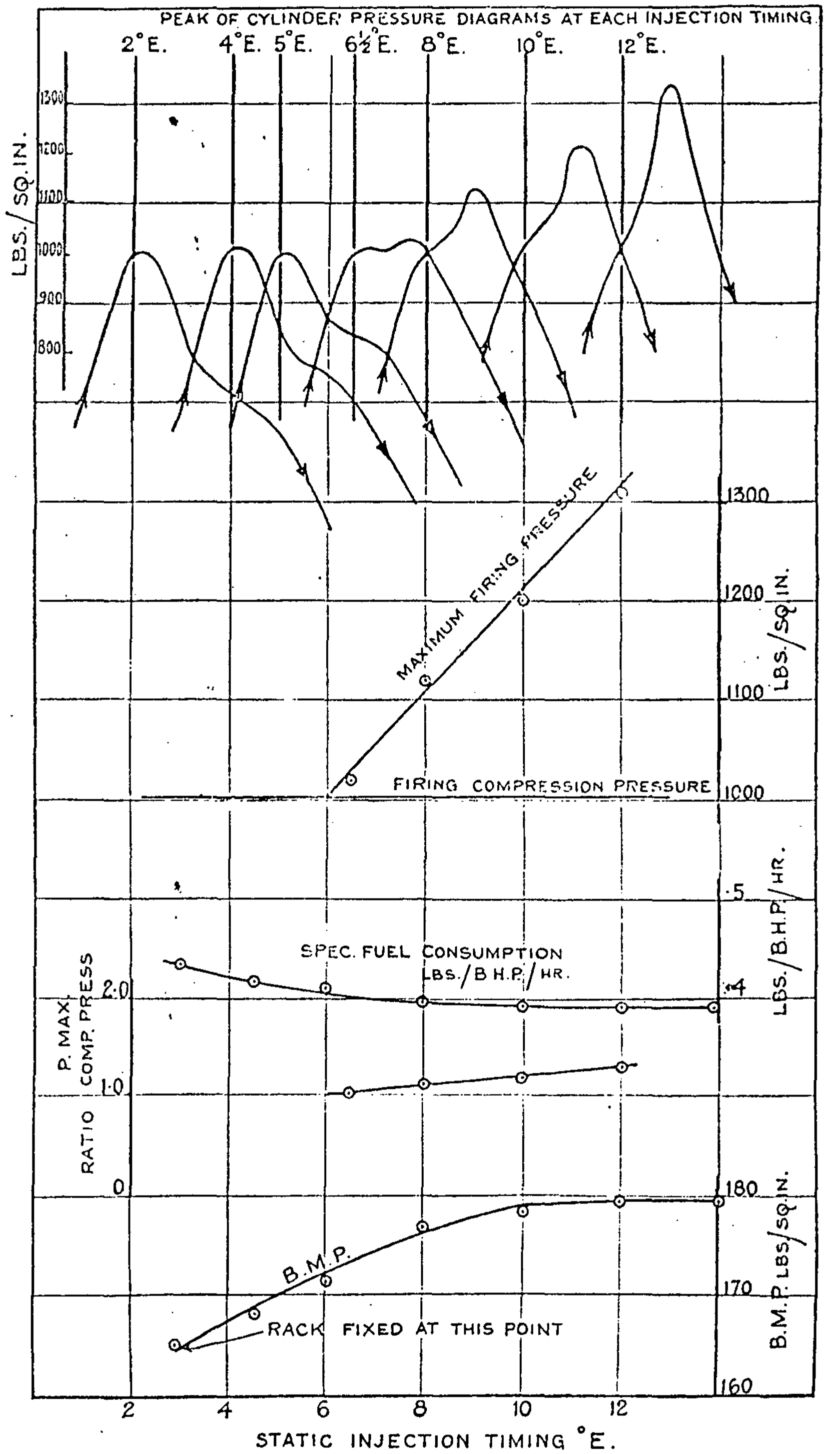


Fig. 1.7—Curves showing the effect of static injection timing on the performance and cylinder-pressure diagram of E18/1 Comet Mark III engine, bore 5 in. × stroke 5 in.

Test conditions:

Engine speed, 1250 r.p.m. 20 in. Hg. boost, 60° C. I.A.T. at 170–180 B.M.P.

supercharging pressure to something not exceeding 2 atmospheres absolute when using a compression ratio of the order of 15.0:1.

Conclusions (General)

50. Supercharging tends very greatly to reduce the ignition delay period and as a result:

(a) The engine runs extremely quietly and smoothly.

(b) The optimum ratio of maximum to compression pressure is considerably lower and under better control.

51. Fig. 1.7 shows the effect on the performance of a "Comet Mark III" unit of varying the static time of start of injection, the true start being, of course, some few degrees later. In this series the quantity of fuel injected per cycle was kept constant throughout, as also all other conditions, the only variable being the time of start of injection. It will be noted:

(a) That with a static timing of 6.5° before top centre the indicator diagram is virtually that of a true constant pressure cycle, i.e. both compression and peak pressures are almost identical at 1000 lb.per.sq. in.

(b) At this timing the brake mean pressure is only 5 percent and the specific fuel consumption 4 per cent, below the absolute optimum.

(c) That for all practical purposes the optimum performance is obtained with an injection timing 9° before top centre, at which setting, the maximum peak pressure is 1160 lb. per.sq.in.

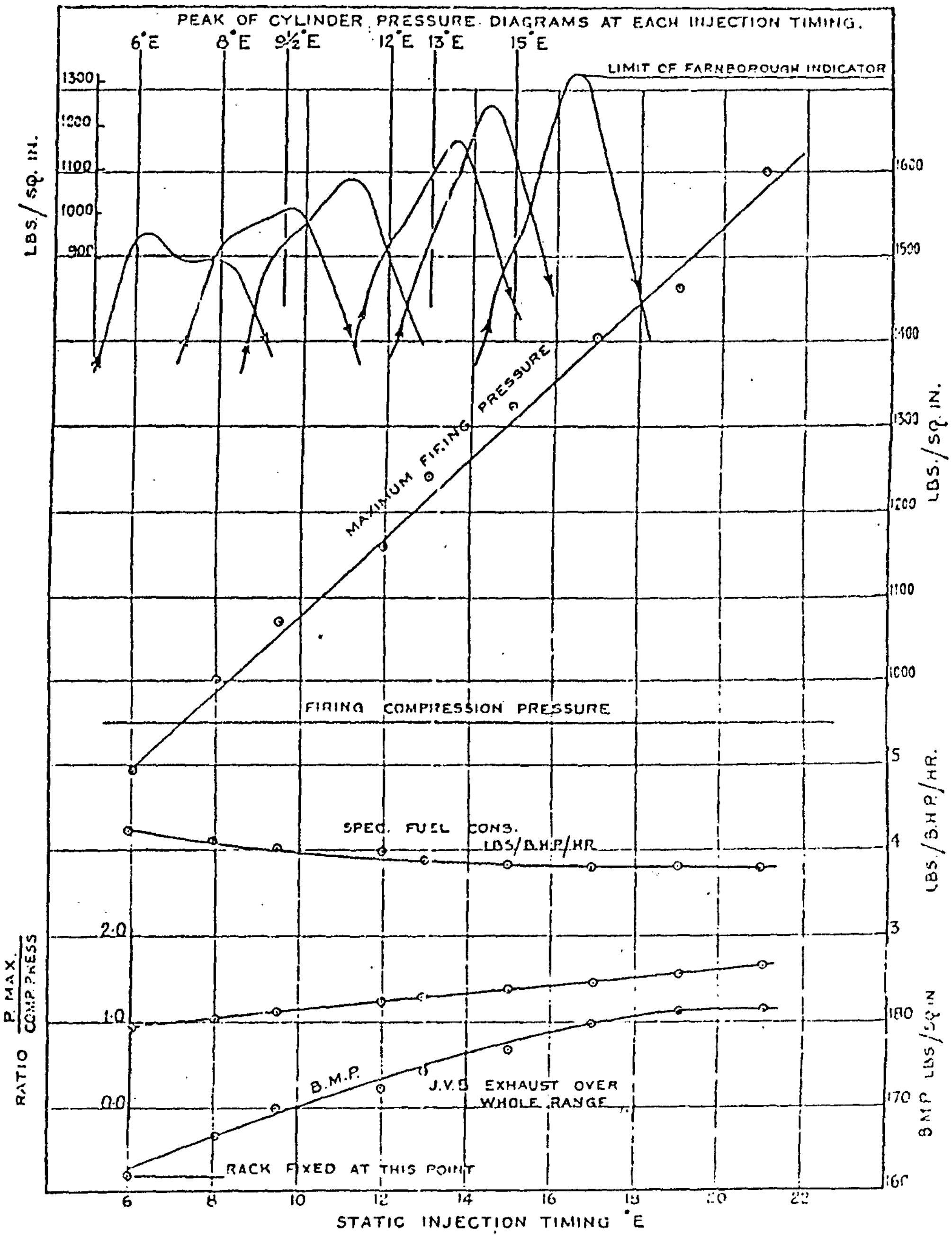


Fig. 1.8 —Curves showing the effect of static injection timing on the performance and cylinder-pressure diagram of E16/10 direct-injection engine, bore 4¾ in. × stroke 5½ in.

Test conditions:

Engine speed, 1250 r.p.m. 20 in. Hg. boost, 60° C. I.A.T. at 170-180 B.M.P.

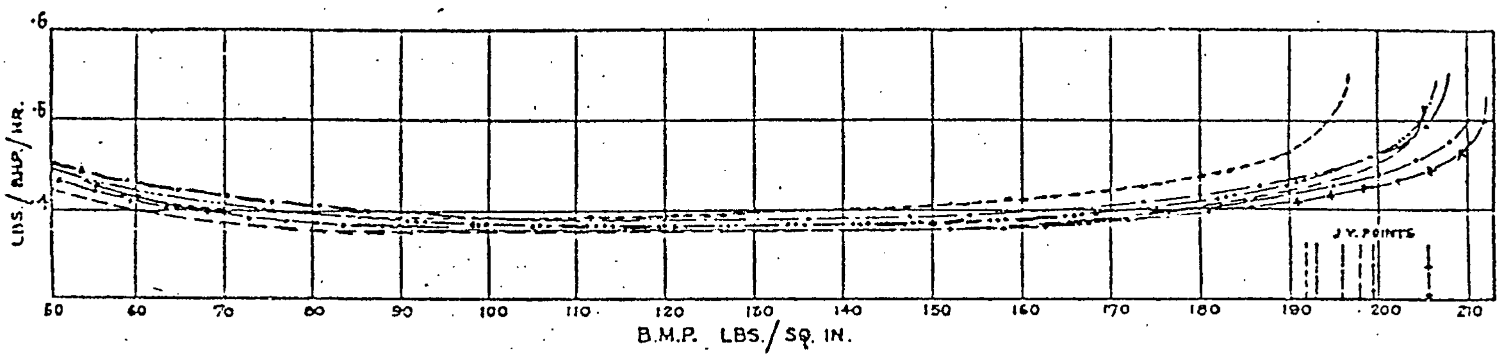


Fig. 1.9—Fuel tests (load-range curves) on E18/1 Comet Mark III engine, bore 5 in. × stroke 5½ in.

Fuels:

		Cetane No	Sp Gr at 15°c
— · — · —	Heavy Diesel	38	0·9145
— — — — —	Marine Diesel	43	0·8730
—··· —··· —··· —	Heavy Diesel	34	0·9155
— + — + — + —	Industrial Diesel	40	0·8930
— o — o — o –	Pool Gas Oil	49	0·8495
- - - - - - - - - -	Heavy boiler fuel oil	–	0·9450

Test Conditions:

Engine Speed 1250 rpm — oil inlet temp 60°c

20 in Hg boost, 30°c — Jacket inlet temp 70°C

Max pr. limited to 1100 lb per sq. in at 150 lb per sq in. B.M.P.

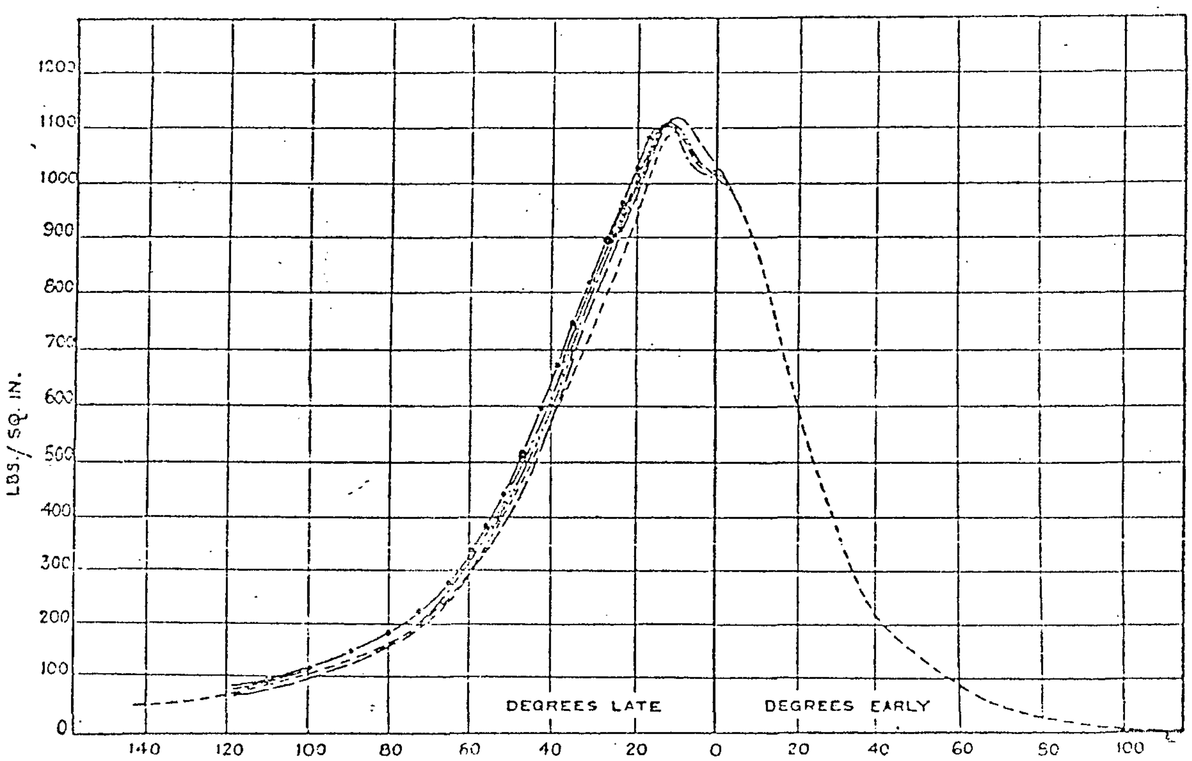


Fig. 1·10.—Fuel tests (cylinder-pressure diagrams) on E18/1 Comet Mark III engine, bore 5 in. × stroke $5\frac{1}{2}$ in.

Fuels:

		Cetane No.	Sp. Gr at 15°C	BMP (lb/in²)
—·—·—	Heavy Diesel	38	0·9145	151·0
– – – – – –	Marine Diesel	43	0·8730	150·0
—···—···—	Heavy Diesel	34	0.9155	155·5
—+—+—	Industrial Diesel	40	0·8930	152·0
—o—o—	Pool gas oil	49	0· 8495	150·4
- - - - - - - -	Heavy boiler fuel oil	—	0· 9450	151·0

Test Conditions:

Engine speed 1250 rpm
20 in. Hg boost, 30°c
oil inlet temp 60°c
Jacket inlet temp 70°c
Max pr. limited to 1100 lb. per sq. in. at 150 lb. per sq. in. BMP

(d) With settings earlier than 9° the gain in performance is negligible and certainly not sufficient to justify the higher peak pressure.

52. Fig. 1.8 shows a similar series of tests carried out under identical conditions on a similar research unit but with a direct-injection open type combustion chamber. By comparison it will be noted:

(a) That the specific consumption at the optimum point is bout 3 per cent lower than the "Comet III" as compared with about 6 per cent lower when running unsupercharged.

(b) That the constant pressure diagram is obtained with an injection advance of about 7° at which point the B.M.E.P. is 10 per cent, and the specific consumption about 8 per cent below the optimum.

(c) That, so far as specific fuel consumption is concerned, the practical optimum is reached with an injection advance of about 13° with a peak pressure of above 1220 lb. per sq in. but that at this timing the B.M.E.P. is still about 4 per cent below the optimum.

53. Fig. 1.9 shows load-range curves under supercharged conditions on various heavy fuels, and fig 1.10 the general shape of the cylinder pressure diagrams for these same fuels. From these figures it will be noted:

(a) That the performance under supercharged conditions is virtually the same for all the fuels tested, despite very wide variations in specific

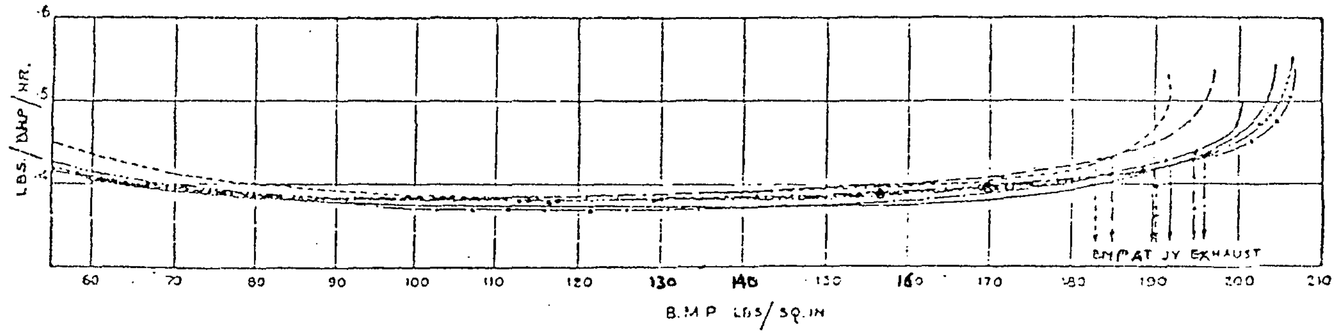


Fig. 1.11 —Fuel tests (load-range curves) on E18/1 Comet Mark III engine, bore 5 in. × stroke $5\frac{1}{2}$ in.

Fuels:

		Cetane No	Sp. Gr. at 15°C
————	Light boiler Oil	41	0·9120
—.—.—	Heavy diesel	38	0·9145
—...—...—	Heavy diesel	34	0.9155
— — — — — —	Marine diesel	43	0·8730
—+—+—+—	Industrial diesel	40	0·8930
—o—o—o—	Pool gas oil	49	0·8495
- - - - - - - - -	Heavy boiler fuel oil	—	0·9450

Test Conditions:

Engine Speed 500 rpm
20 in. Hg. boost, 30°C

Oil inlet temp. 60°C
Jacket inlet temp, 70°C

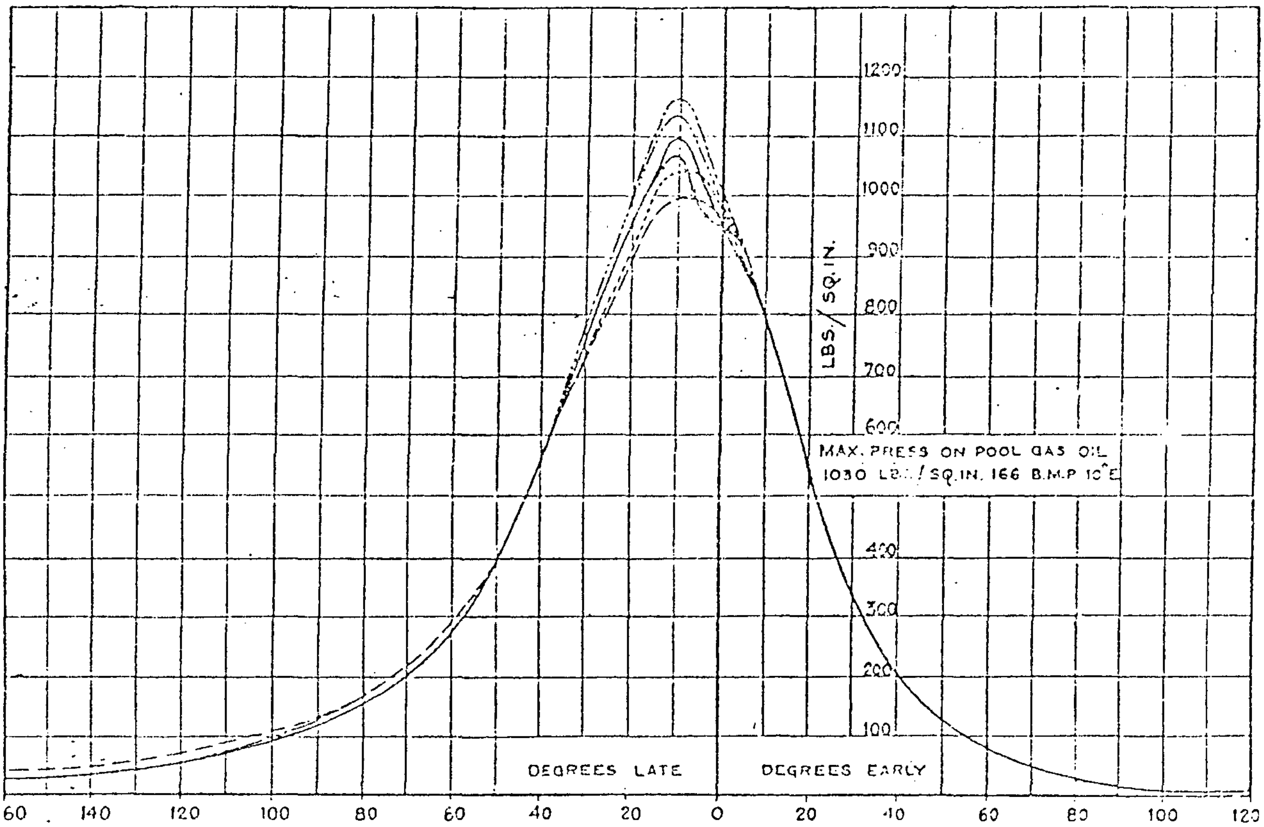


Fig. 1·12.—Fuel tests (cylinder-pressure diagrams) on E18/1 Comet Mark III engine, bore 5 in. × stroke 5½ in.

Fuels:

		Cetane No	Sp Gr at 15°c	BMP (lb/in²)
————	Light boiler oil	41	0·9120	162
—·—·—	Heavy diesel	38	0·9145	160
—···—···—	Heavy diesel	34	0·9155	160
— — — — —	Marine diesel	43	0·8730	160
—+—+—+-	Industrial diesel	40	0·8930	161
- - - - - - - - -	Heavy boiler fuel oil	—	0·9450	161

Test Conditions

Engine speed 500 rpm — oil inlet temp 60°c

20 in. Hg boost, 30°c — Jacket inlet temp 70°c

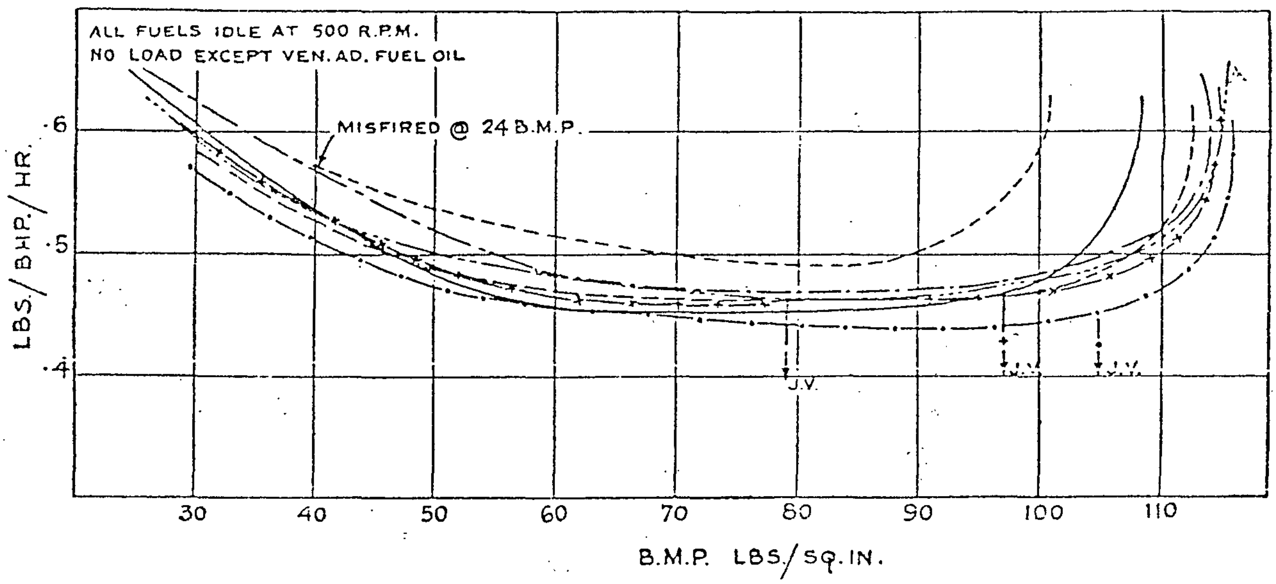


Fig. 1·13 —Fuel tests (load-range curves) on E18/1 Comet Mark III engine, bore 5 in. × stroke $5\frac{1}{2}$ in.

Fuels:

		Cetane No	Sp. Gr at 15°C
————	Light boiler oil	41	0·9120
—·—·—	Heavy diesel	38	0·9145
—···—···—	Heavy diesel	34	0·9155
— — — —·—	Marine diesel	40	0·8930
—o—o—o—	Pool gas oil	49	0·8495
- - - - - - - - -	Heavy boiler oil	–	0·9450

Test Conditions:

Engine Speed 500 rpm
Zero boost, 30°.

Oil inlet temp. 60°c
Jacket inlet temp. 70°c

Exhaust – blue grey over load range except for the three fuels where just visible limits are shown.

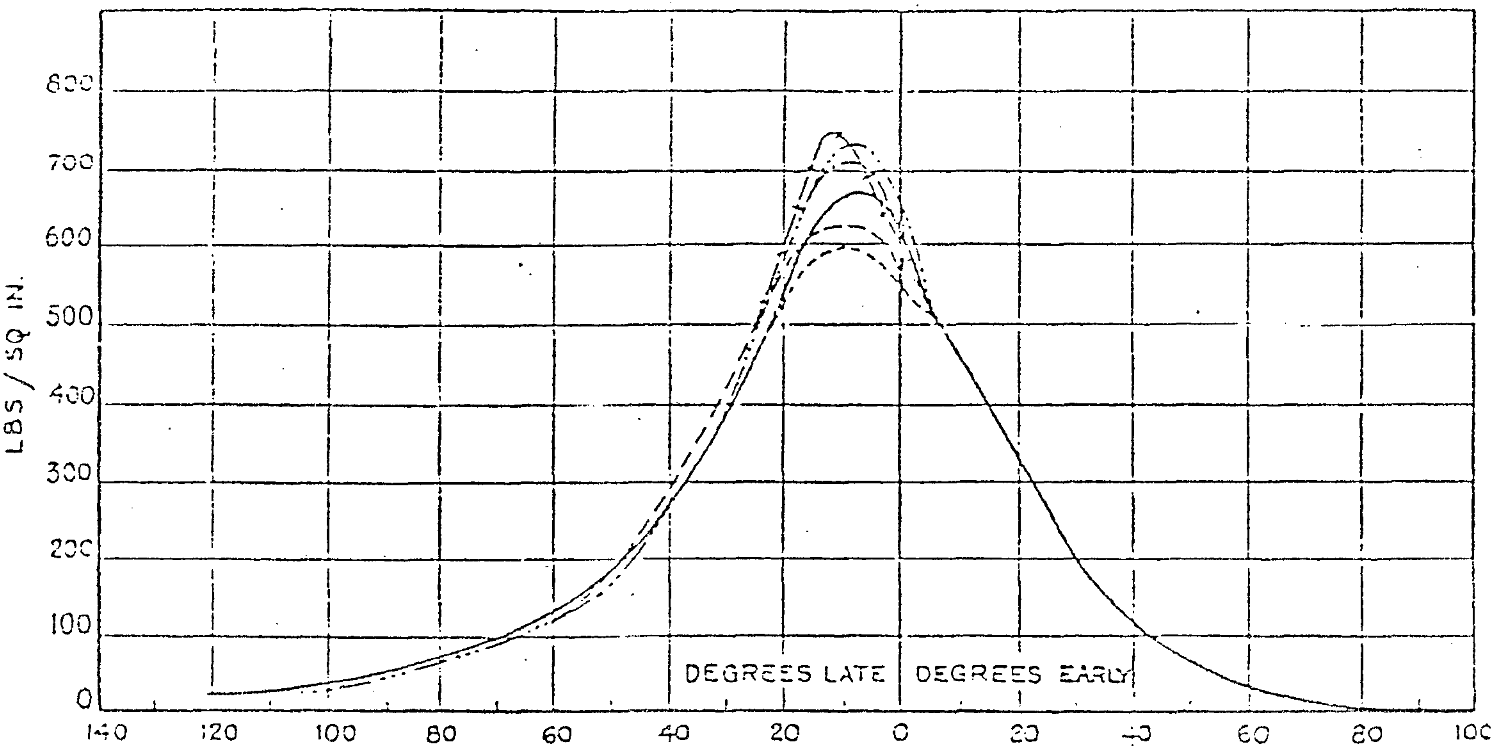


Fig. 1.14.—Fuel tests (cylinder-pressure diagrams) on E18/1 Comet Mark III engine, bore 5 in. × stroke 5½ in.

Fuels:

		Cetane No.	Sp. Gr. at 15° C.	B.M.P. lb./in.²
————————	Light boiler oil	41	0·9120	72·0
— · — · — · —	Heavy Diesel	38	0·9145	73·0
— · · · — · · · —	Heavy Diesel	34	0·9155	68·0
— — — — —	Marine Diesel	43	0·8730	72·7
— + — + — +	Industrial Diesel	40	0·8930	70·5
· · · · · · · · · · · ·	Heavy boiler fuel oil	—	0·9450	71·0

Test conditions:

Engine speed, 500 r.p.m.
Zero boost, 30° C. I.A.T.
Oil inlet temperature, 60° C.
Jacket inlet temperature, 70° C.

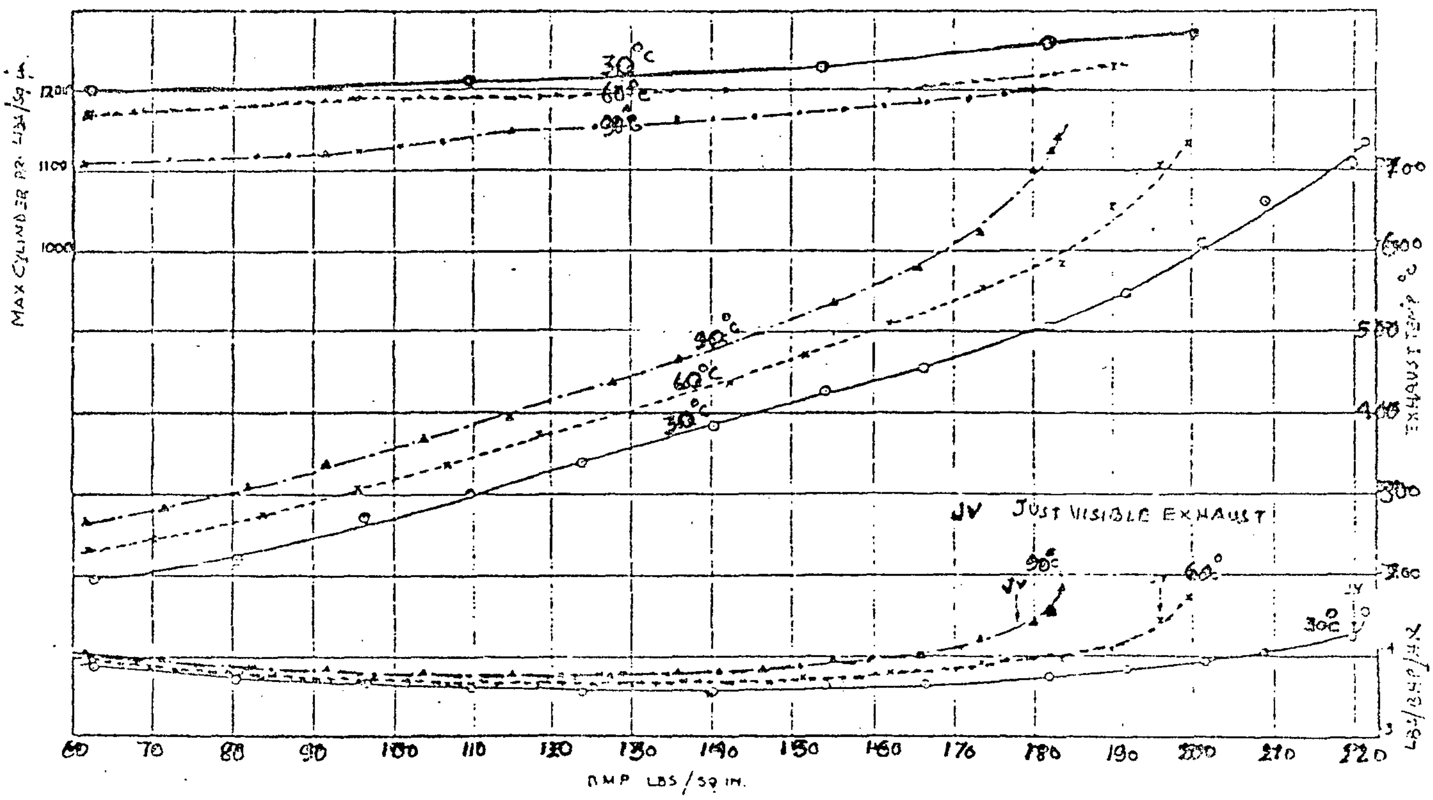


Fig. 1.15 .—Effect of intercooling on consumption, exhaust temperature and maximum cylinder pressure, using E1S/1 single-cylinder 4-stroke Comet Mark III engine, bore 5 in. × stroke $5\frac{1}{2}$ in.

Test Conditions:

Engine Speed 1250 rpm
Air intake temperatures 30°c, 60°c, 90°c at 1·66 atm absolute

oil inlet temp. 60°c
Jacket inlet temp. 70°c

Fuel:

Heavy diesel oil, sp.gr 0·9145 at 15°c.

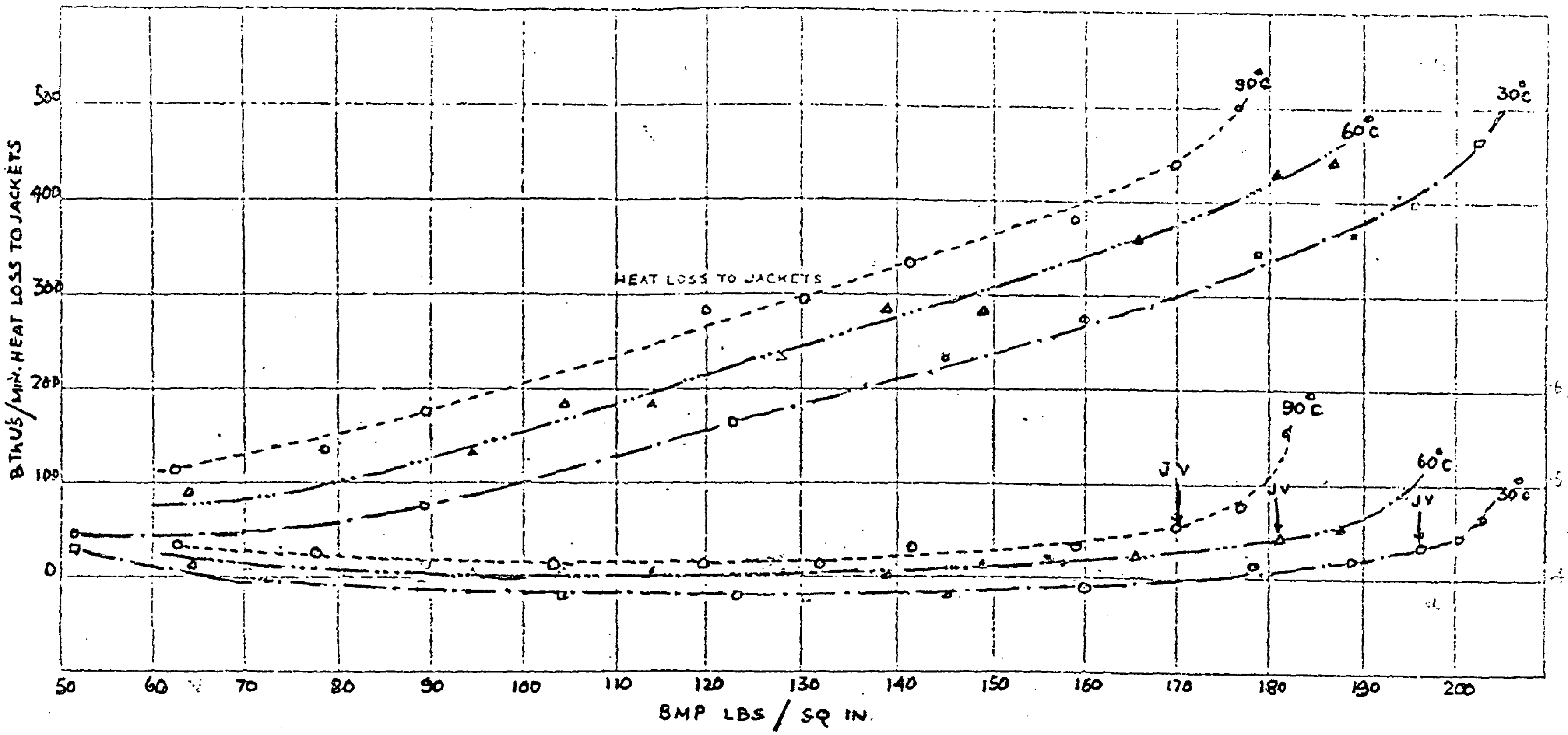


Fig. 1.16 —Effect of air intake temperature on fuel consumption and heat loss to cooling water, using E18/1 single-cylinder 4-stroke Comet Mark III engine, bore 5 in. × stroke $5\frac{1}{2}$ in.

Test Conditions:

Engine Speed 500 rpm
20 in. Hg boost. 30°C, 60°C, 90°C

oil inlet temp. 60°C
Jacket inlet temp 70°C

Fuel:

Heavy Diesel oil. specific Gravity 0.9145 at 15°C gravity, volatility, and cetane number . The point of just visible exhaust varies only 6 per cent over the extreme range.

(b) Although on a basis of lb. per. B.H.P. hour the specific consumption varies; when plotted on that of the lower calorific value of the fuel in each case, the consumptions are identical within the limits of observation, excepting only that of the Admiralty fuel oil (30per cent gas oil and 70 per cent residual) which is a little greater, more especially at the top end of the load range.

54. Fig 1.11 and fig. 1.12 are similar curves to the above but at a lower speed and can be compared with fig 1.13 and fig 1.14 which are taken at the same speed but in the unsupercharged condition.

55. From the figures it will be seen that when running un-supercharged the performance on the same group of fuels differed widely. In most cases the running was very rough and noisy, and in that of the Admiralty fuel oil the exhaust was smoky throughout the entire range, misfiring occurred at light loads and the engine would not run idle.

56. All the evidence confirms that the reduced delay period and improved combustion are a function of pressure rather than of temperature, though both play a part.

57. Fig.1.15 shows the improvement in economy and reduction in exhaust temperature to be obtained by intercooling from 90° C. and 30° C.

58. Fig. 1.16 shows the reduction in the heat loss to the jacket with intercooling.

59. From this it will be noted that the effect of reducing the supercharge air temperature by 60° C. by intercooling is to reduce the heat flow to the pistons, cylinder walls, etc., from 365 to only 240 B.Th.U. per minute or to less than two-third and, as a result, to reduce the specific fuel consumption from 0.43 to 0.39 lb.per B.H.P. hour when operating at a brake mean pressure of 150 lb.per.sq.in. at 500 r.p.m. This, of course is a somewhat extreme case for, at the very low revolution speed of 500 r.p.m., the effects of heat loss are magnified.

60. Fig.1.15 shows that a similar reduction in the supercharge air temperature serves to reduce the exhaust temperature from 515°C.to 410°C. when operating at a brake mean pressure of 150 lb. Per sq. in. at 1250 r.p.m. In this case, owing to the higher revolution speed, the effect of the reduced heat losses on thermal efficiency is, of course, less pronounced, but even so the specific fuel consumption is reduced from 0.385 to 0.366 lb.per. B.H.P. hour; the maximum cylinder pressure is, however, increased by about 5 per cent.

61. At all speeds, and more especially at low revolution speeds. viz. down to 20 per cent of the normal maximum, the running under supercharge is quite remarkably smooth.

62. Throughout the whole speed range the increase in I.M.E.P. is slightly, and in B.M.E.P. at the clean exhaust limit is considerably, greater than the increase in density of the supercharge air. The former is to be accounted for, in part, by the ability, under supercharged conditions to consume a slightly greater proportion of the air retained in the cylinder, but mainly by the higher thermal efficiency at which it is consumed, and the latter by the higher mechanical efficiency,

63. The B.M.E.P. at the clean exhaust limit when running unblown at a speed of 1250 r.p.m. and with 30°C air intake temperature, is 115 lb per sq inch. from fig 1.15 it will be seen that the B.M.E.P. at the same speed and air intake temperature, but when supercharged to a pressure of 20 in. Hg., is 220 lb.per.sq.in. (ratio of intake pressure and density,1.66:1; ratio of B.M.E.P., 1.92:1).

64. At the same intake air temperature the relative heat losses to the cooling water diminish with increased density at about the same rate as with increased, speed under naturally aspirated conditions, i.e, if the power output be doubled either by doubling the density of the air or by doubling the engine speed at atmospheric density, the flow heat to the cylinder walls, pistons, etc., is substantially the same, provided the air temperature is maintained constant.

65. In a high-speed, high –compression engine of normal design, little, if anything, is to be gained by cylinder scavenging, since the inlet and exhaust

valves are in such close proximity that overlap results merely in short-circuiting . When account is taken of the loss of air (on which work has been done in the blower) and the necessary distortion of the combustion space in order to accommodate the valve overlap, the net result is generally negative. No measurable reduction in piston temperature could be observed even with very wide overlaps, involving a considerable loss of air. It would seem that it is only when an exhaust turbo-driven supercharger is employed that scavenging by valve overlap can be justified, and then primarily as an expedient to reduce the exhaust temperature to a figure acceptable to the turbine. Even so, the practice of short-circuiting some air to exhaust, outside the cylinder and under either governor or thermostatic control, so that it occurs only when the exhaust temperature exceeds a safe limits, is probably much to be preferred . Clearly the higher the degree of supercharge, the more costly the loss of air by scavenging or short-circuiting.

66. Reference again to Fig.1.15 will emphasize the important effect of air intercooling upon the exhaust temperature. With air at 90°C., i.e. no intercooling, an exhaust temperature of 600° C. is reached at a B.M.E.P. of 169 lb.per.sq. in.; with intercooling down to 30° C. a B.M.E.P. of just over 200 lb.per.sq.in. can be reached at the same exhaust temperature, or, conversely, at say 170 lb.per.sq.in. B.M.E.P., intercooling from 90° C. to 30° C. lowers the exhaust temperature by 140 °C.

67. After taking all the relevant factors into consideration, it would seem that, for an engine of normal design and proportions, and having a compression ratio high enough to run satisfactorily without supercharge, the optimum degree of supercharge lies. between 1.5 and 2.0atmospheres absolute. Below 1.5 atmospheres it is doubtful whether in the case of a small engine, i.e. below about 100 h.p., the gain will fully justify the extra cost complication, while above 2.0 atmospheres the high maximum cylinder pressure will call for a more robust design of engine. If the supercharge can be available at all times, and under all conditions of load and speed, then a lower ratio of compression can safely be used and a higher supercharge permitted within the same limits of maximum pressure. This appears, in any case, to be an essential condition if difficult fuels are to be used.

Comparison of Direct versus Indirect Injection under Supercharged Conditions

68. Under normally aspirated conditions, the indirect-injection units with compression swirl of the "Comet Mark III" type have a higher fuel consumption of the order of 5 per cent to 10 per cent to higher relative heat losses, but develop from 10 per cent to 15 per cent more power at the clean exhaust limit than the direct-injection type with induction air-swirl due to better air utilization.

69. Under supercharged conditions the relative heat losses of both types are less and the difference in thermal efficiency is much reduced until at

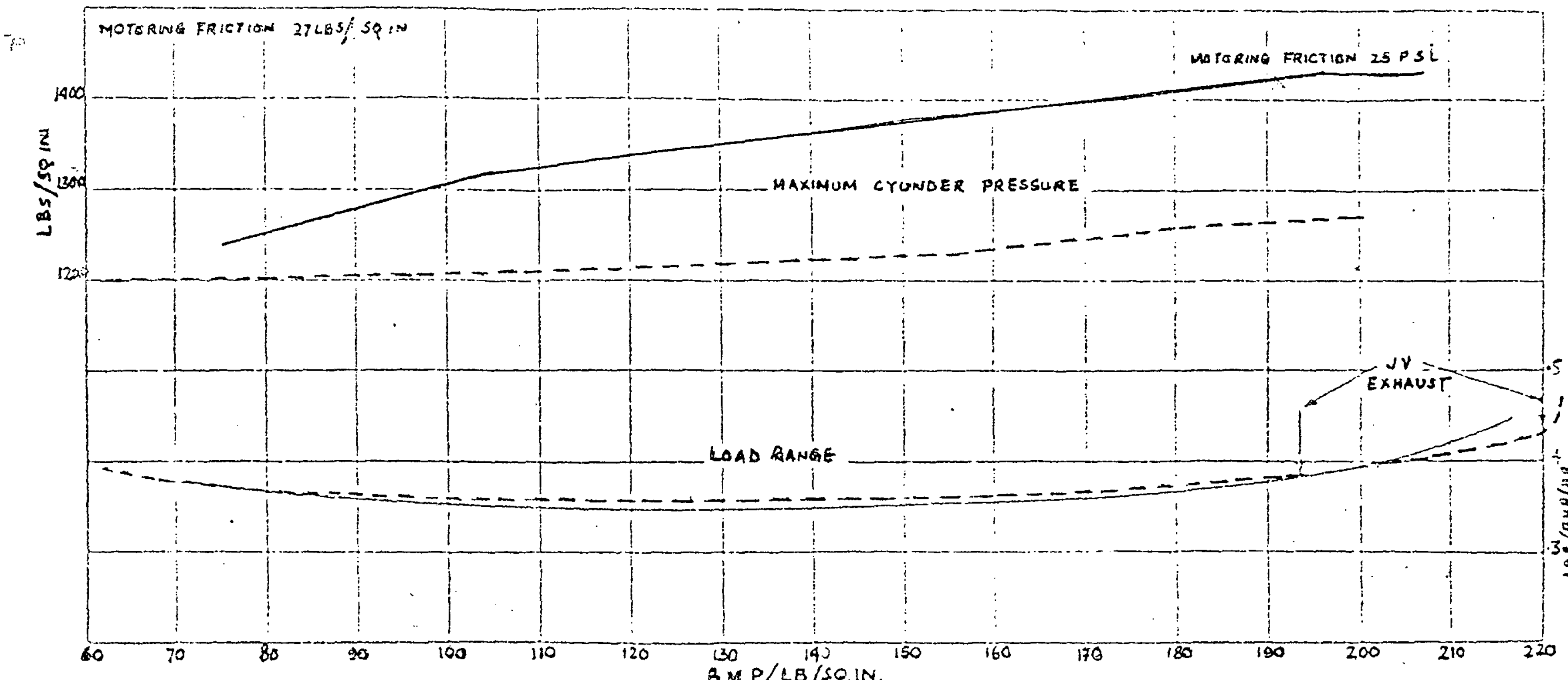


Fig. 1·17 —Comparison of load-range and maximum cylinder-pressure curves of Comet Mark III and direct-injection combustion systems

Test Condition

Engine Speed 1250 rpm
20 in. Hg boost, 30°c

oil inlet temp. 60°c
Jacket inlet temp 70°c

Key:

——————— Direct injection engine

– – – – – – Comet Mark III engine.

2 atmospheres abs. The gap in thermal efficiency is almost closed. Thus fig. 1.17 shows comparative performance curves of the "Comet Mark III" and the Direct Inejction systems under supercharged conditions at 1250 r.p.m.

CHAPTER II

LITERATURE SURVEY

70. An exhaustive literature survey has been done to develop a clear picture on the effect of various types of engines when the air intake pressure is varied, before proceeding with a practical application on a D0026 M8A diesel engine.

71. C.T. Wilbur & D.A. Wright (3) state that by increasing the density of charge between the air intake and cylinder, the weight of air induced per working stroke is increased and thereby a greater weight of fuel can be burnt, and as a result a substantial increase in engine power output is achieved for a stated size and piston speed.

72. Sir Harry R.Ricardo (2) states that for any given capacity of engine and for any given thermal efficiency, theoretically, we can double the indicated power either by doubling the speed of the engine or by doubling the density of air. C.T. Wilbur & D.A. Wright (3) points out that for a given engine it is usually not practicable to double the engine speed as we consider that it is already running at its maximum permissible limits and on the other hand

while doubling the density of air, we are restricted by inadequate breathing capacity.

73. Sam Haddad & Neil Watson (1) observe that throughout the whole speed range of a engine, the increase in BMEP is considerably greater than the increase in the density of supercharge air. Elaborating this with a practical test result, they have shown that in an engine running unsupercharged at 1250 rpm with 30°C air intake temperature, the BMEP was 115 lb per sq inch whereas, when under the same conditions, the engine was supercharged to a pressure of 20 inch Hg, the BMEP shot up to 220 lb per sq inch which is about 1.92 times as compared to a increase in air density of 1.66 times.

74. Neil Watson (7) also states that in general, an engine of normal design and proportion, having a compression ratio high enough to run satisfactorily without super charging, the optimum degree of supercharge life between 1.5 and 2 atmosphere absolute, while, beyond 2 atmospheres the high maximum cylinder pressure will be a limiting factor. D.W. Tryhorn (11) after a preliminary test on a Ruston 4 VRH engine pressure charged with a Bicera compressor arrived to a conclusion that the optimum degree of supercharge is around 1.5 for a engine with compression ratio of around 15:1 .

75. Sam Haddad & Neil Watson (1) emphasises that the principle objective of increasing the air intake pressure is to increasing the power output for a given size of engine and other objectives like increase in efficiency, reduction in exhaust emission etc. are subsidiary and that the power output of an engine can be increased by increasing the air intake pressure. A mention has also been made by them that the increase in air intake pressure is limited by the allowable maximum cylinder pressure.

76. Sam Haddad and Nel Watson (1) also discusse the various factors limiting the engine performance, wherein, he states that the thermal loading of the engine is related with exhaust valve temperature and thus the engine performance is limited by this factor. Since the increase in the exhaust valve temperature is directly related to the variation of air intake pressure, the increase in the air intake pressure beyond a certain limit is not feasible. Regarding mechanical loading, Heywood JB (9), states that since it is usually characterised by maximum cylinder pressure, which in turn is related to the boost pressure, the air intake pressure also can not be increased beyond a certain limit .

77. Sam Haddad & Neil Watson (1), Sir Harry R Ricardo (2) , CT Wilbur and D A Wright (3), C.C. Pounder (4) , all have conclusively shown with a number of test results that the engine performance, especially in terms of

BMEP and power developed have shown a notable improvement when the air intake pressure was increased.

78. Experiments on the Ruston 4 VRH engine by D.W Tryhorn (11) and the project report on turbocharging of BMP II engine by VRDE, Ahmednagar (14) have also established that there is a distinct improvement in the engine performance in terms of torque and power developed when the air intake pressure was increased. The performance curves so obtained during the above mentioned tests are given as appendices `A' and `B' respectively.

Conclusion of Literature Survey

79. A review of the literature survey indicates that in general, all engines having a compression ratio high enough to perform satisfactorily under naturally aspirated condition, show a distinct improvement in their system response when subjected to a increased air intake pressure. Keeping this at the back of mind, the author has subjected a D0026 M8A diesel engine (fitted in truck 'Shaktiman', an extensively used load carrier of the Indian Army) to a varying air intake pressure and plotted its performance results in Chapter IV after which an analysis of the same has been done in Chapter V.

CHAPTER - III

EFFECT OF VARIATION OF AIR INTAKE PRESSURE

General

80. The principle objective of increasing the air intake pressure is to increase the power output of a given size of engine. Subsidiary objectives may be to improve efficiency and/or reduce exhaust emissions and combustion induced noise.

81. Supercharging means increasing the density of the air entering the cylinders in order that more fuel may be burnt. Density is increased by a compressor. Many types of compressor are available but only positive rotodynamic compressors can deliver a sufficiently large quantity of air from a small compressor to be practical. Usually the positive displacement rotary compressor will be limited in rotational speed, relative to a rotodynamic compressor and therefore will be larger. Both machines can be driven mechanically by the engine but the rotodynamic compressor is suitable for a turbocharger, where the driving power comes from exhaust gas energy via a turbine.

82. The density of the air delivered to the cylinders depends on the ambient air density, the pressure ratio and efficiency of the compressor and any heating (for example in the inlet manifold) or cooling (for example from a charge air cooler) of the air.

83. As far as the engine is concerned, supercharging influences the engine through three parameters only :

(a) The inlet manifold pressure
(b) The inlet manifold temperature

$\Big\}$ hence density

(c) The exhaust manifold pressure, if turbocharged; or the power requirement to drive the supercharger, if mechanically supercharged.

84. Usually an engine manufacture is concerned with what a particular supercharger can do for his engine, but it is more instructive to reverse this process and ask the question 'what does the engine require from the supercharger ?'.

85. In this way, target can be set for an 'ideal' supercharger to meet the requirements of a particular engine and its application. We will use a simplified analysis and try to answer this question. The starting point will be how the three parameters listed above influence the airflow through an engine and therefore how much fuel can be burnt and its power output.

Airflow Through An Engine

86. **Four-stroke engine.** Consider first the simple case of a four-stroke engine with no valve overlap, or at least a sufficiently small overlap period for scavenge flow to be negligible. The airflow rate can be expressed as:

$$\dot{m} = \frac{N}{2} \rho_m V_{sw} \eta_{vol} \qquad (1)$$

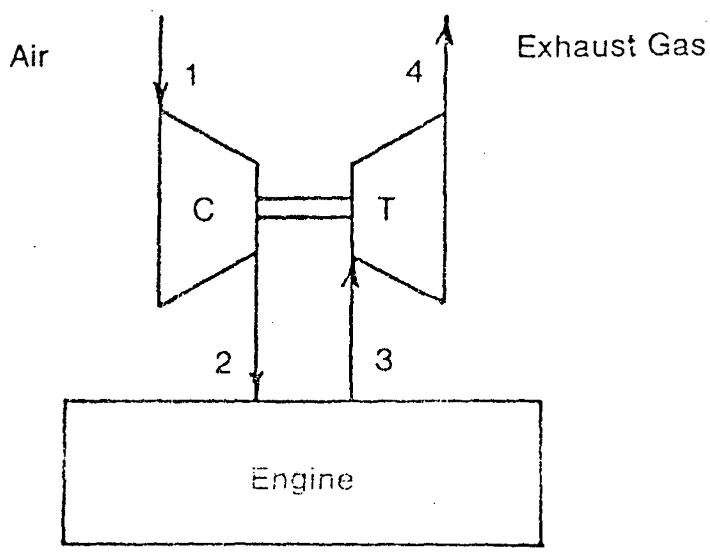


Fig3.1 – Schematic of engine and turbocharger.

where

N = engine speed

ρ_m = inlet manifold density

V_{sw} = swept volume of engine

η_{vol} = volumetric efficiency

87. The inlet manifold density will depend on ambient air conditions and compressor pressure ratio and efficiency. Using the notation of Fig. 3.1, we get :

$$T_2 = \frac{T_1}{\eta_c}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_1 \qquad (2)$$

where η_c = isentropic efficiency of the compressor. Treating air as a perfect gas, and assuming no charge air cooling, the inlet manifold density is given by :

$$\rho_m = \rho_2 = \frac{P_2}{P_1} \qquad (3)$$

$$\rho_2 = \frac{P_2}{RT_1}\left[\frac{1}{\eta_c}\left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} + 1\right] \qquad (4)$$

88. Equation (1) and (4) may be combined to calculate the airflow through the engine. Note that the exhaust manifold pressure of the turborcharged engine does not influence the airflow, as zero scavenge flow has been assumed for simplicity.

89. Fig.3.2 shows a typical volumetric efficiency curve of a truck engine. For simplicity, a constant volumetric efficiency of 0.9 will be assumed and a constant compressor isentropic efficiency of 0.7. Fig.2.3 shows the effect of compressor pressure ratio (P2/P1) and engine speed (N) on the airflow characteristics of a four-stroke engine with ambient conditions of 1 bar,23°C.

90. Obviously airflow increases with engine speed, since the engine behaves as a positive displacement pump. In addition, airflow increases with pressure ratio which is the objective of supercharging.

91. If the engine has significant valve overlap, then the airflow will be influenced by the difference between inlet and exhaust manifold pressure. In the case of a mechanically supercharged engine there will be a positive pressure difference between inlet and exhaust manifolds, creating a positive scavenge airflow (Fig. 3.3). The scavenge airflow will increase as the boost pressure (P2) rise since exhaust manifold will be close to atmospheric, hence the pressure between manifold increases . In a turbocharged engine, the scavenge flow will depend on the relationship between compressor boost pressure and turbine inlet pressure. This can be directly related to turbine inlet temperature and turbocharger overall efficiency.

92. **Two-stroke engine** : In the case of a two-stroke engine, the airflow through the cylinders is largely a function of the pressure drop between inlet and exhaust manifolds and inlet manifold density. The flow can be modelled as compressible flow through a mean equivalent flow area of the inlet and

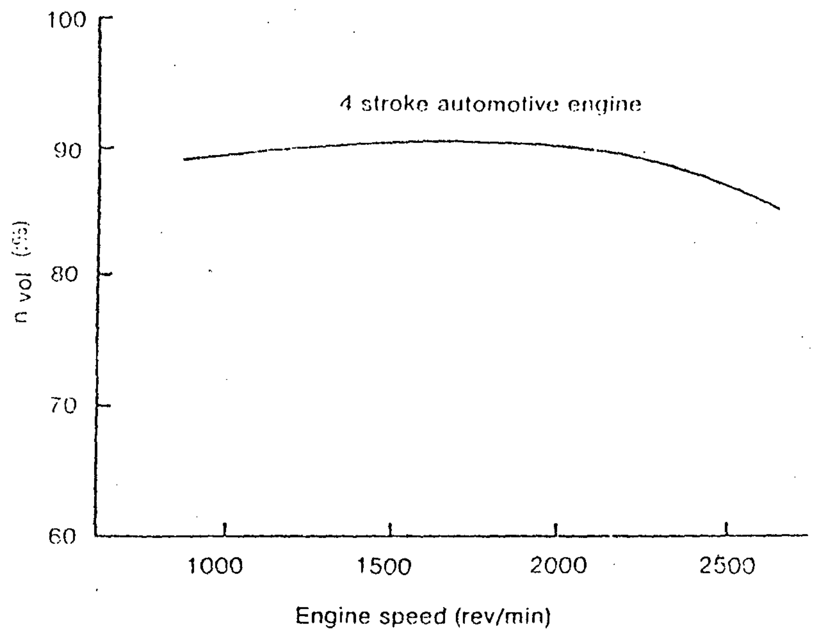


Fig3.2 – Volumetric efficiency curve.

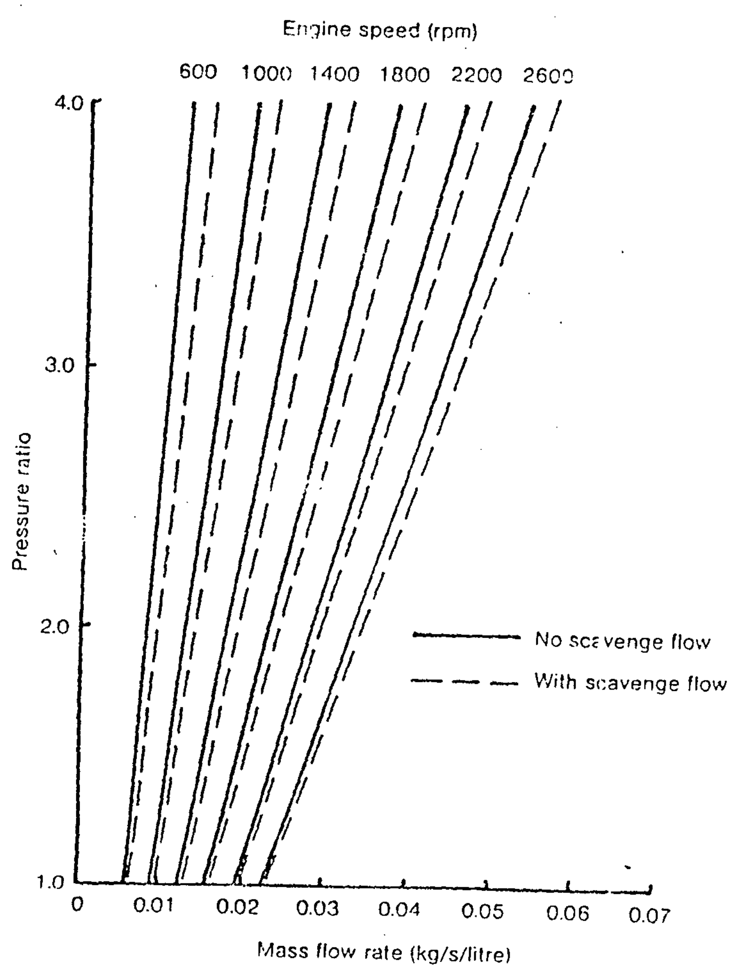


Fig3.3 – Engine airflow rate as a function of speed and compressor pressure ratio (four-stroke, no charge cooling).

exhaust ports (or valves) in series. Obviously the actual flow area is time dependent, but a mean equivalent area can be used for simplified calculations. Sometimes this is related to a 'reference' flow area using a discharge coefficient (fig 3.4). Convenient reference area are maximum port area or piston area although in the latter case, the discharge coefficient does not have its usual meaning.

93. For compressible flow, using the notation of Fig.3.1 the airflow rate is given by:

$$\dot{m} = C_d A \sqrt{\left\{ \left(\frac{2\nu}{\nu - 1} \right) P_2 \rho_2 \left[\left(\frac{P_3}{P_2} \right)^{2/\nu} - \left(\frac{P_3}{P_2} \right)^{\frac{\nu+1}{\nu}} \right] \right\}} \qquad (5)$$

94. If the pressure drop between inlet and exhaust manifold is small, as is likely to be the case, then the simplification of incompressible flow can be made. Equation (5) then becomes.

$$\dot{m} = C_d A \sqrt{\left[2\rho_2 \left(P_2 - P_3 \right) \right]} \qquad (6)$$

95. In the case of a mechanically driven supercharger, the exhaust manifold pressure (P_3) is close to ambient, hence the mass flow rate is a function of boost pressure and temperature only. The airflow characteristic is a single line on Fig.3.5. For a turbocharged engine, the expansion ratio of turbine (P_3/P_4) is mass flow dependent, thus P_3 varies with engine speed and load in manner determined by the flow characteristic of the turbine. Fig.3.5 shows how the airflow characteristic of the engine vary with the exhaust manifold pressure P_3. In practice, P_3 may not be constant at steady engine

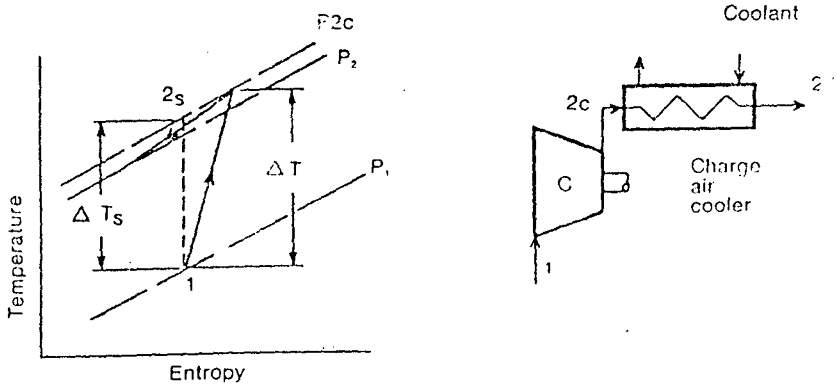


Fig3.6 – Charge air cooling, with a pressure loss.

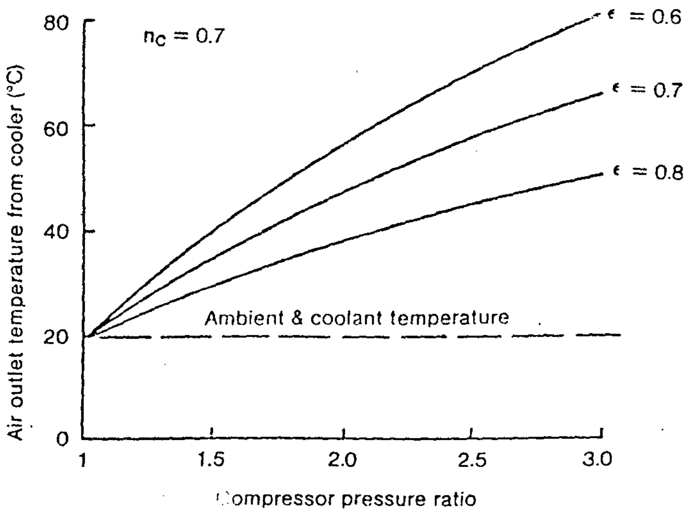


Fig3.7 – Intercooler effectiveness and inlet manifold temperature.

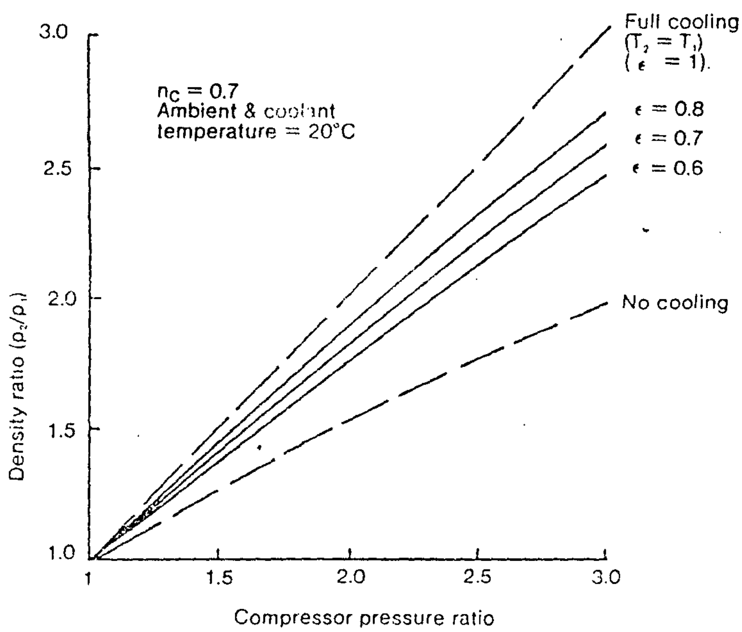


Fig3.8 – Effect of charge cooling on inlet air density.

conditions (depending on the turbocharging system selected) but this complication will be ignored here.

Charge cooling

96. Charge air cooling used on many supercharged engine to increase the effectiveness of the supercharging system. Reducing air temperature by charge cooling increases air density, for given compressor pressure ratio, and hence increases mass flow rate.

97. The effectiveness of the charge cooler is given by the actual heat transfer divided by the maximum possible heat transfer, which for a perfect gas, reduces to :

$$\epsilon = \frac{T_{2c} - T_2}{T_{2c} - T_{cool}} \qquad (7)$$

where subscript 2_c denotes compressor exit; '2' denotes charge cooler exit (inlet manifold conditions); and 'cool' denotes the coolant (Fig.3.6). Thus

$$T_2 = T_{2c}\,(1-\epsilon) + \epsilon\, T_{cool} \qquad (8)$$

Therefore, Eq (2) becomes,

$$T_3 = T_1\left[\frac{1}{\eta_c}\left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} + 1\right]\left[1-\epsilon\right] + \epsilon\, T_{cool} \qquad (9)$$

98. Equation (9) is plotted for a compressor efficiency of 0.7 and T_1 and $T_{cool} = 20°$ in Fig.3.7. As important is the density ratio ($\rho_2/\rho_{ambient}$), plotted in Fig. 3.8 showing the benefit of increased effectiveness of charge air cooling.

99. Here our concern is the effect of charge air cooling on the airflow through the engine. Fig 3.9 is a companion figure to Fig.3.3. but for a charge cooled engine. Since charge cooling increases the inlet manifold density more as pressure ratio increases, then equation (1) shows that the mass flow rate through the engine increases more rapidly with boost. Thus the pressure ratio versus mass flow rate lines at constant speed in fig.3.9 are less steep than those of Fig.3.3.

Engine Performance Limitations

100. Now that basic airflow characteristics of four and two-stroke engine have been established, we need to link the airflow to fuel delivery and power output. In doing this account must be taken of various engine performance limitations imposed by the engine rather than the supercahrging system.

101. The key limitations are mechanical loading and resultant stress, thermal loading and stress, and the minimum air/fuel ratio for efficient combustion and a smoke free exhaust . For a simplified analysis it is convenient to consider the thermal loading limit in terms of exhaust valve temperature, since it is usually the exhaust valve temperature that is the main consideration (on any particular engine design, piston crown, cylinder head, piston ring or liner temperatures may be a problem but we are not concerned with an individual design here). Mechanical loading is usually characterized by maximum cylinder pressure and is related to boost pressure, engine compression ratio, injection timing etc. Since we are considering the engine

$P_1 = 1$ bar T_1 1 300K $n_{vol} = 0.9$ $n_C = 0.7$ cooler $= 0.8$

$T_{cooler} = 300$ K

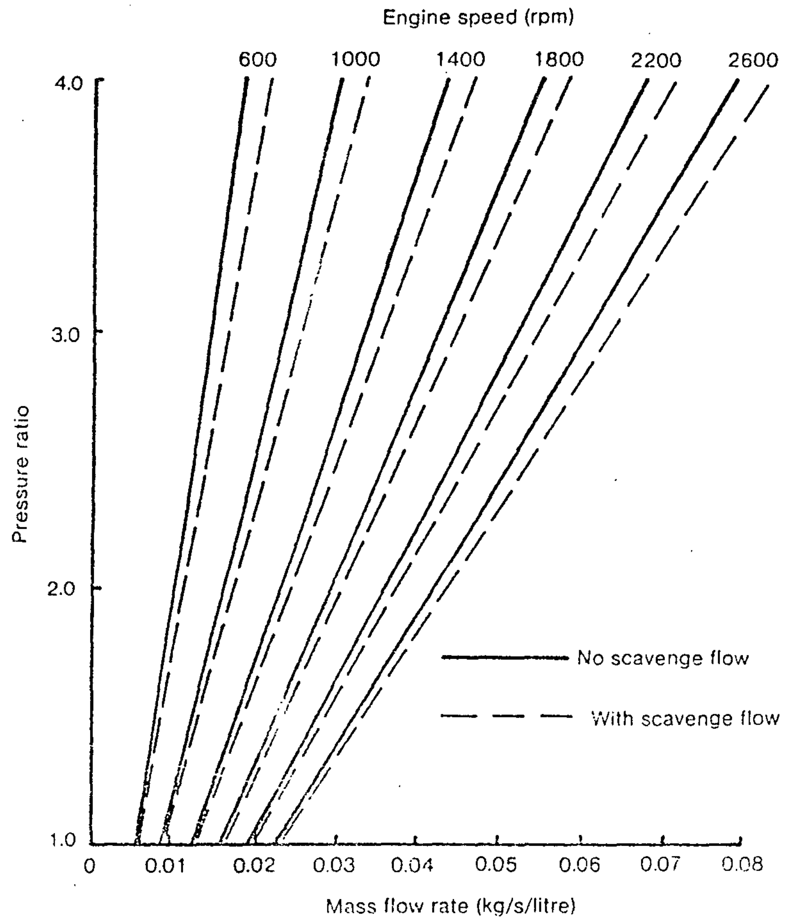


Fig3.9 – Engine airflow rate as a function of speed and compressor pressure ratio (four-stroke with charge air cooling).

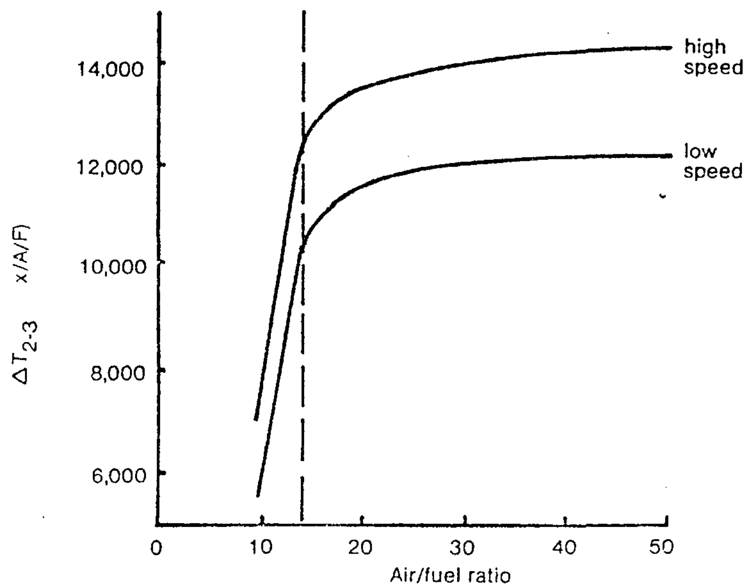


Fig 3.10 – Temperature rise from inlet to exhaust manifolds as a function of air/fuel ratio and speed (four-stroke truck engine).

requirements from the supercharging system at any desired pressure ratio, we need not specifically consider mechanical loading. The exhaust smoke level is determined by combustion system development and air/fuel ratio. We will assume a well developed combustion system, hence need only consider the minimum air/fuel ratio for smoke free exhaust. However, this may not be the air/fuel ratio for most efficient engine operation at any particular speed and load, so the 'optimum' air/fuel ratio must also be considered.

102. Compressor efficiency and charge cooler effectiveness influence the inlet manifold temperature and therefore engine thermal loading directly as well as through airflow rat... However this factor will be ignored in the current analysis. In addition the exhaust manifold pressure is strongly influenced by overall turbocharger efficiency and in turn influences engine efficiency through piston pumping work. This will not be considered since its influence is relatively small, except on very highly rated engines.

103. In conclusion, for a simplified analysis, the most important limitations are exhaust (or turbine inlet) temperature and air/fuel ratio. In practice these are strongly linked, particularly on a four-stroke engine with little scavenge airflow. Fig.3.10 shows the temperature rise ($T_3 - T_2$), with reference to fig.3.1 across a turbocharged four-stroke truck engine as a function of air/fuel ratio, but plotted as temperature rise X air/fuel ratio versus air/fuel ratio. It can be seen that the result is almost constant except near

stoichiometric air/ fuel ratio when dissociation becomes significant. For a large, slower speed engine, with a quiescent combustion chamber, the curve would become non-liner below a greater air/fuel ratio.

Engine Power And Airflow

104. The power requirements of an engine need to be linked to airflow, within the limitation given above, to establish the requirements of the supercharging system . Obviously this link is unique to a particular engine and depends primarily on the combustion system but is also influenced by many other factors such as mechanical efficiency etc..

105. It is convenient to start with the indicated specific fuel consumption (ISFC) or indicated thermal efficiency (η_i) , where :

$$ISFC = \frac{1}{\eta_i \times CV} \qquad (10)$$

(where CV is the lower calorific value of the fuel) and

$$BSFC = ISFC/\eta_{mech} \qquad (11)$$

where BSFC and η_{mech} are the brake engine specific fuel consumption and mechanical efficiency of the engine, respectively.

106. Also the fuel consumption (m_f) relates brake specific fuel consumption to power output, therefore

$$BSFC = \left(\frac{m_f}{\dot{W}}\right) \text{, where } \dot{W} = \text{power output.} \qquad (12)$$

107. The air/ fuel ratio is conveniently considered in terms of the stoichiometric air/ fuel ratio divided by the equivalence ratio (F)

$$\frac{\dot{m}_a}{\dot{m}_f} = \left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{st} \Big/ F, \quad \text{where st} \Rightarrow \text{stiochiometric} \qquad (13)$$

108. Figure 3.11 shows typical variation of indicated specific fuel consumption with equivalence ratio.

109. From equation (10) to (13), power can be linked to airflow as follows:

$$\dot{m}_a = \dot{W} \times \text{ISFC} \times (\dot{m}_a/\dot{m}_f)_{st}/(F \times \eta_m) \qquad (14)$$

110: Equation (14) enables the air requirement for any particular power requirement to be established. Equation (1) and (4) or (2) and (9) for a charge cooled engine link the airflow requirement to the supercharger.

Engine Power And Supercharger Requirements

111. The requirements placed on a supercharger vary substantially with engine type and its application. Three types of engine will be considered, as representative of widely different types, namely a four-stroke truck diesel engine, a four-stroke medium speed marine diesel engine and a four-stroke passenger car diesel engine. In addition the effects of different engine rating and charge air cooling are considered.

112. **Four-stroke truck diesel engine :** A typical direct injection engine will be considered working over a speed range of 600 to 2600 rpm. This speed range is larger than normal but allows the data presented to be used over any part of the range.

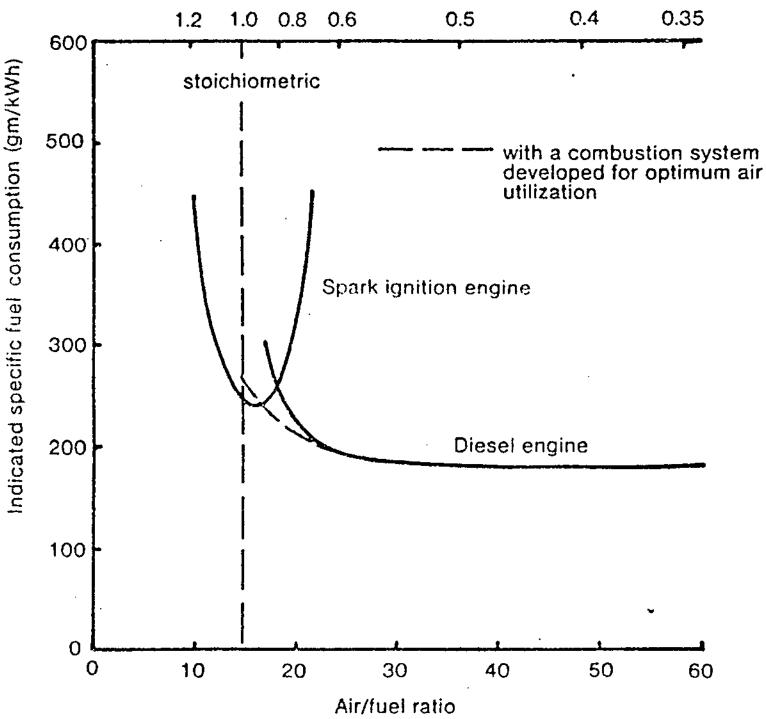


Fig3.11 – Effect of air/fuel ratio on indicated specific fuel consumption (adapted from Zinner [1]).

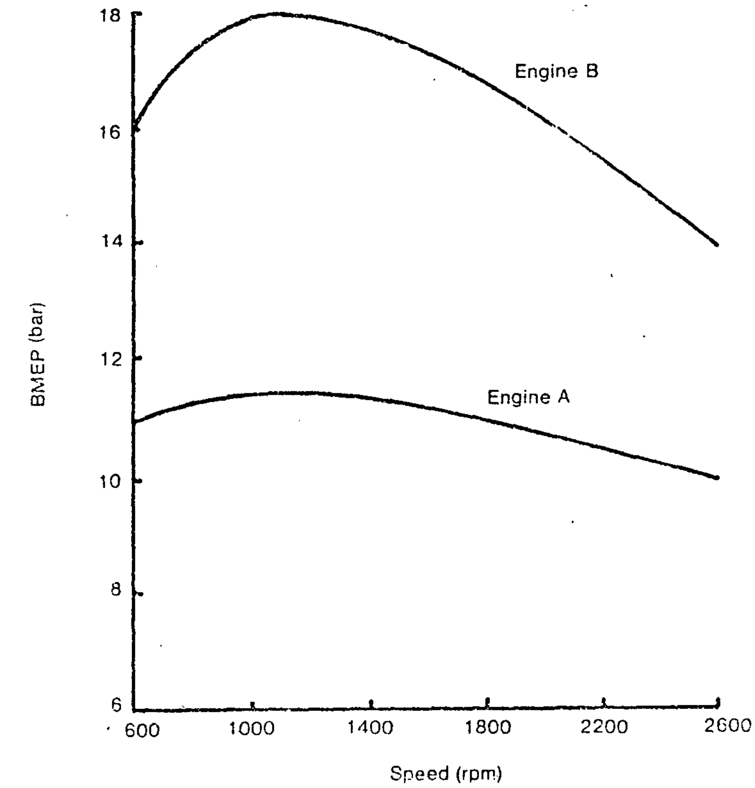


Fig3.12 – Target BMEP (torque) curves for four-stroke truck diesel engines.

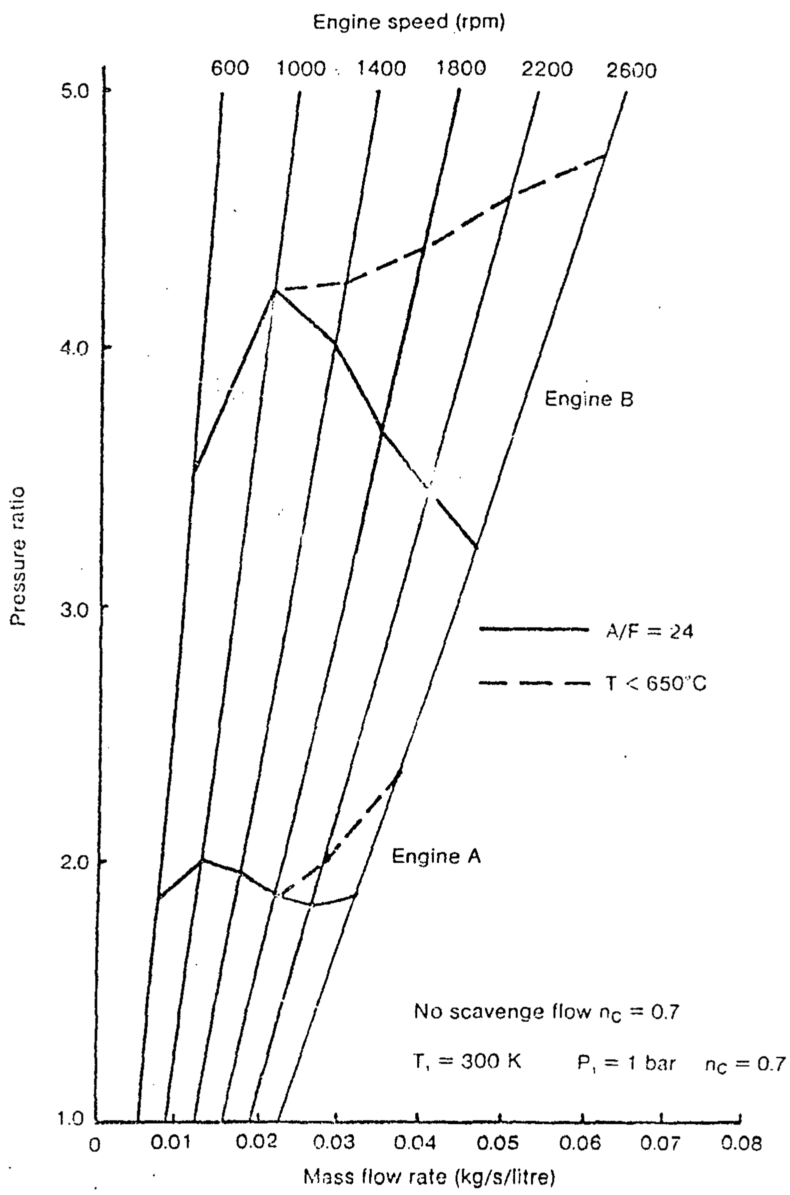


Fig3.13 – Compressor requirements for a four-stroke truck diesel engine without charge air cooling.

113. Figure 3.12 shows the desired torque curves of two engines, 'A' being typical of a current turbocharged truck engine whereas 'B' represents a much more highly rated engine. The airflow requirements of these engines can be calculated from equation (14), once the air/fuel ratio has been established. Experience shows that a normal swirl-type direct injection diesel engine requires an air/fuel ratio of around 24:1 for optimum fuel consumption for a specified BMEP. Figure 3.11 shows that the benefit of higher air/fuel ratios is marginal. The fact that many engines run with much lower full-load air/fuel ratios is largely due to the limitations of existing supercharges and maximum cylinder pressure constraints that must be ignored herein.

114. The airflow required from equation (14) must match that calculated form equation (1). This defines the supercharge pressure ratio required as shown on Fig. 3.13, which is the pressure ratio versus mass flow rate characteristic for a four-stroke engine per litre engine capacity with a volumetric efficiency of 0.9 and a compressor isentropic efficiency of 0.7. Superimposed are the airflow requirements of engines A and B (assuming no charge air cooling), with an air/fuel ratio of 24:1. In practice, the exhaust temperature rises with engine speed at constant air/fuel ratio and this may impose a requirement for additional air at high speeds. This is illustrated by the lines shown in Fig.3.13 for a constant exhaust temperature of 650 ° C.

115. For engine A, a pressure ratio rising form 1.85 at 600 rpm to 600 rpm to 2.0 at 1000 rpm to 2.33 at 2600 rpm is required. The key feature being the very wide mass flow range. Engine B has an even wider flow range requirement and needs a pressure ratio rising from 3.5 at 600 rpm to 4.2 at 1000 rpm to 4.75 at 2600 rpm. Although a requirement for a 70% compressor efficiency over an 80 % flow range at pressure ratios from 1.85 to 2.33 is difficult, but feasible, it become impossible if the pressure ratio exceeds 4:1 with current technology.

116. Intercooling reduces the pressure ratio required to achieve the desired mass flow rate. Thus Fig.3.14 is a comp anion figure to fig.3.13 but assumes air-air charge cooling with an effectiveness of 0.8 (equation (7)). The target performance of engine B now looks more reasonable. Note that since intercooling reduces inlet air temperature, it also reduces exhaust gas temperature for a constant air/fuel ratio. Thus the difference between the constant air/fuel ratio and constant exhaust temperature curves in Fig.3.14 is substantially less than in fig.3.13.

117. Figures 3.13 and 3.14 illustrate several aspects of supercharger requirements for truck engines. These can be summarized as follows:

(a) Charge air cooling (intercooling) is essential for high BMEP engines.

(b) High compressor efficiency over a wide mass flow range is more important than a very high peak value over a small range.

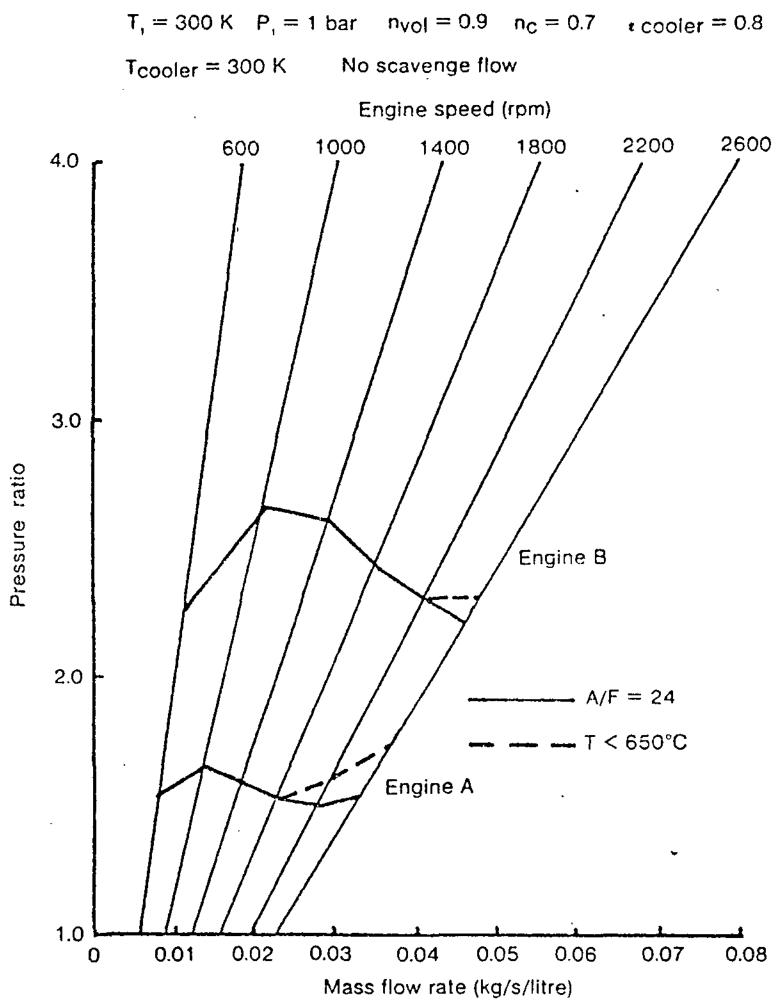


Fig 3-14 – Compressor requirements for four-stroke truck diesel engines with charge air cooling.

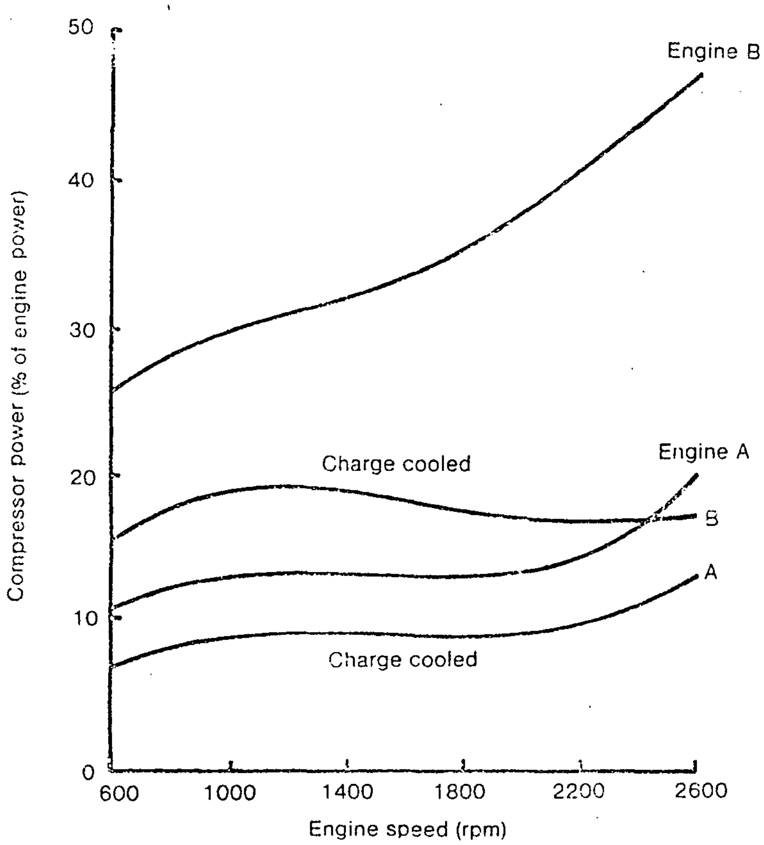


Fig.3.15 – Compressor power requirements for four-stroke truck diesel engines.

(c) A very large surge to choke flow range is required, particularly on a wide speed range engine.

(d) Without charge air cooling, pressure ratio should rise with engine speed to avoid excessive exhaust valve (or turbocharger turbine) temperatures.

(e) With charge air cooling a relatively flat pressure ratio versus mass flow rate and speed characteristic is desirable.

118. The power required to drive the compressor must come form the crankshaft of the engine or an exhaust driven turbine. Obviously the latter system is potentially the more efficient but a mechanically driven supercharger is attractive if its power requirement is small. Figure 3.15 shows the power required to provide the airflow of Figs. 3.13 and 3.14, expressed as a fraction of engine power (with no allowance for mechanically driving the supercharger). With the exception of the non-intercooled engine B (which we have said was impractical), the supercharges required between 9 and and 20 % of the engine power output (less supercharger). In view of the fact that most truck engines spend much of their operational time at high load where the supercharger is needed, this rules out a mechanical supercharger on fuel economy grounds. Thus both engines A and B would be turbocharged.

119. In the case of turbocharged engine, the overall turbocharger efficiency required to achieve the pressure ratios of Figs. 3.13 and 3.14 is of interest.

On the basis of a constant pressure turbine it is easy to calculate an energy balance between compressor and turbine. For a diesel engine, in which fuel is added in the cylinders this becomes:

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{\nu-1}{\nu}}-1\right]_c T_1 C_{p_1}\dot{m}_1=\left[1-\left(\frac{P_4}{P_3}\right)^{\frac{\nu-1}{\nu}}\right]_t T_3 C_{p_3}\left(1+\frac{1}{AFR}\right)\eta_{TC} \tag{15}$$

where AFR = air/ fuel ratio and η_{TC} is the overall turbocharger efficiency. Thus,

$$\eta_{TC} = \eta_c \times \eta_t \times \eta_{mech\,(tc)} \tag{16}$$

120. Normally the turbocharger mechanical efficiency is lumped together with the turbine isentropic efficiency since it is more convenient to measure it this way. Figure 3.16 shows the overall turbocharger efficiency (and turbine times mechanical efficiency, assuming 70 % compressor efficiency) required for engines A and B, assuming equal inlet and exhaust manifold pressure. Even the otherwise impractical case of engine B without a charge air cooler requires a turbocharger efficiency of less than 50 % which is not a difficult target. The product of turbine and mechanical efficiency should easily be 60-70 %, even in a high specific speed unit with a relatively large exit loss.

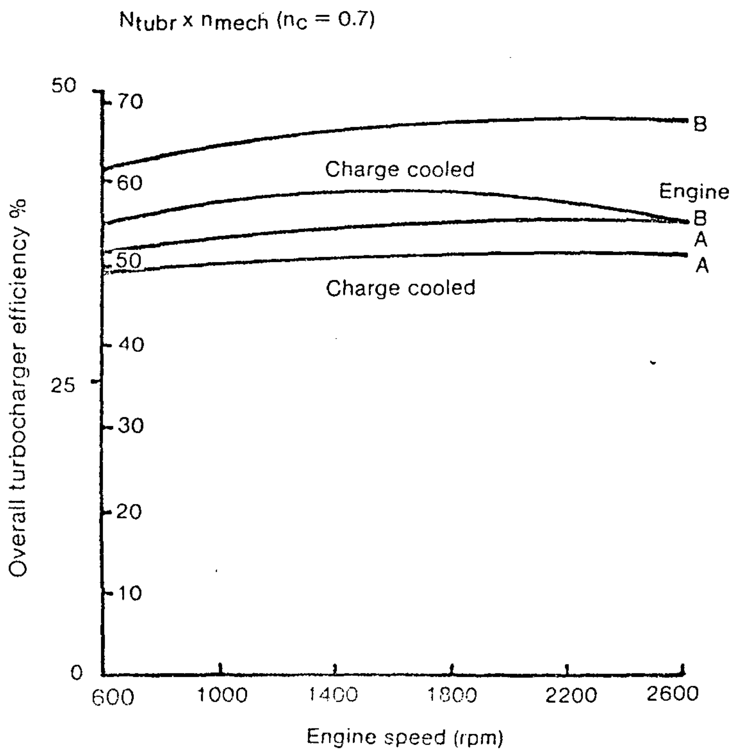


Fig.3.16 – Turbocharger and turbine efficiency required for equal inlet and exhaust manifold pressures ($T < 650°C$).

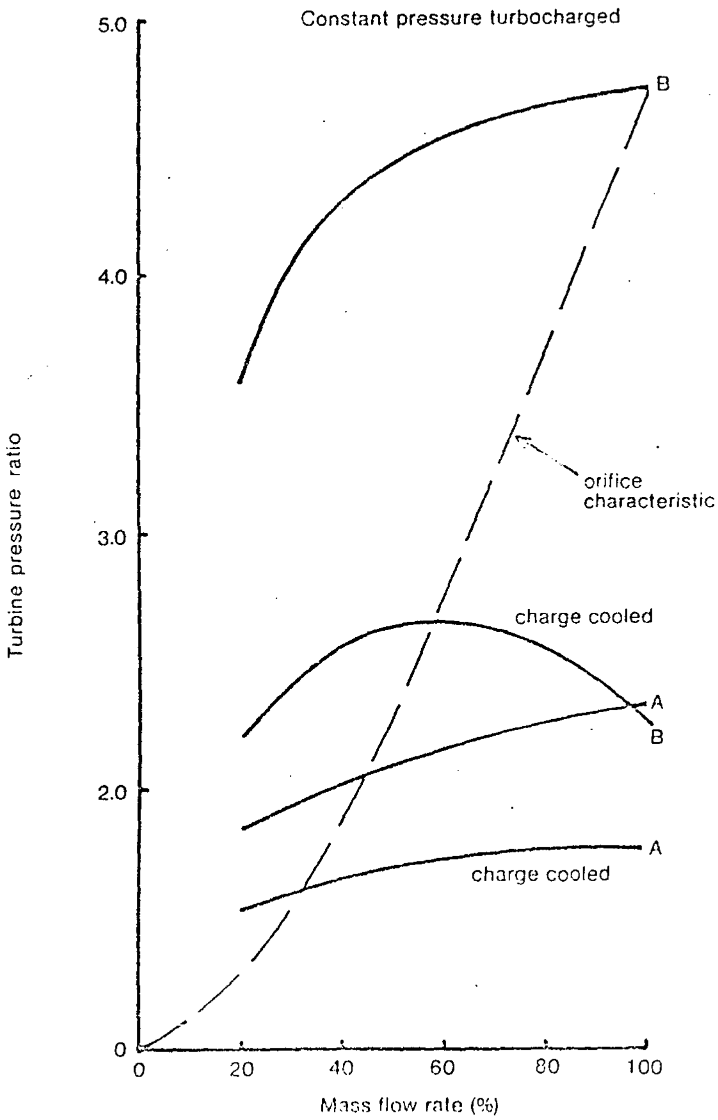


Fig 3.17 – Turbine fl . characteristics required for four-stroke truck engines, with equal inlet and exhaust manifold pressures ($T < 650°C$).

121. Next the pressure ratio versus mass flow rate characteristics required for the turbine must be established. Since the inlet manifold pressure should at least be no less the exhaust manifold pressure, the compressor flow characteristics must virtually be duplicated by the turbine (Fig. 3.17). The requirement is a pressure ratio that is relatively insensitive to mass flow rate. Unfortunately the flow characteristic of turbines are often rather like that of a compressible flow through an orifice of fixed area. Such a flow characteristic is superimposed on Fig 3.17, size to match the maximum power point of engine B without intercooling (similar curves could be drawn for the other engine). The orifice choke at a pressure ratio of 1.8 (approx.); thereafter the volumetric flow is constant and mass flow increases linearly with inlet density at higher pressure ratios.

122. Clearly the orifice type flow characteristic is quite unsuitable for a truck engine turbocharger turbine. Turbine expansion ratio and therefore power is insufficient to achieve the desired engine performance ! We can now add several requirements to those listed in para 107 above, namely:

(a) The power required to drive the compressor is too large for a mechanically driven supercharger to be used, without significant loss in fuel economy.

(b) The overall efficiency required from the turbocharger is not high-less than 50% (but see (f) below).

(c) The benefit of higher turbocharger efficiencies through increased airflow and better engine cycle efficiency are relatively small (for a four-stroke engine)- see Fig.3.11.

(d) The orifice like flow characteristics of a normal turbine are quite unsuitable for this application, resulting insufficient turbine turbine power at low engine speeds.

(e) The turbine must be designed to minimize the variation of pressure ratio with mass flow rate.

(f) Very high turbine efficiency at low flow rates will party offset the potential turbine power deficit.

123. It can be concluded that the turbine power deficit implied by Fig. 3.17 at low mass flow rates is so large that the combination of several measures are required to solve the problem. These include using the 'pulse' turbocharging system, designing for a turbine characteristic whose effective 'orifice area' increases with pressure ratio or mass flow, limiting the speed range of the engine (and hence mass flow range) , sizing the turbine to 'overboost' the engine at full speed and finally designing for high efficiency at low flows. The alternative will be an engine whose torque curve rises with speed and is unsuitable for a vehicle, and/or greatly excessive boost pressure at high speeds.

124. **<u>Four-stroke medium -speed marine diesel engine:</u>** The requirements for this type of engine are totally different from the truck engine

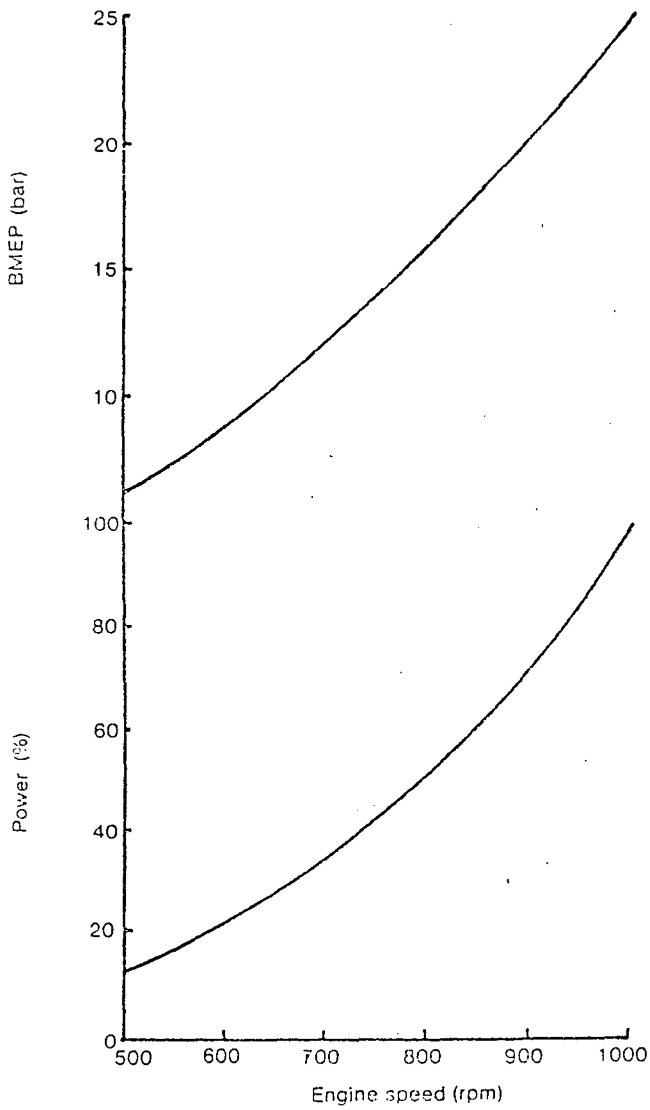


Fig 3.18 – Target performance for a marine four-stroke medium-speed engine (propeller law).

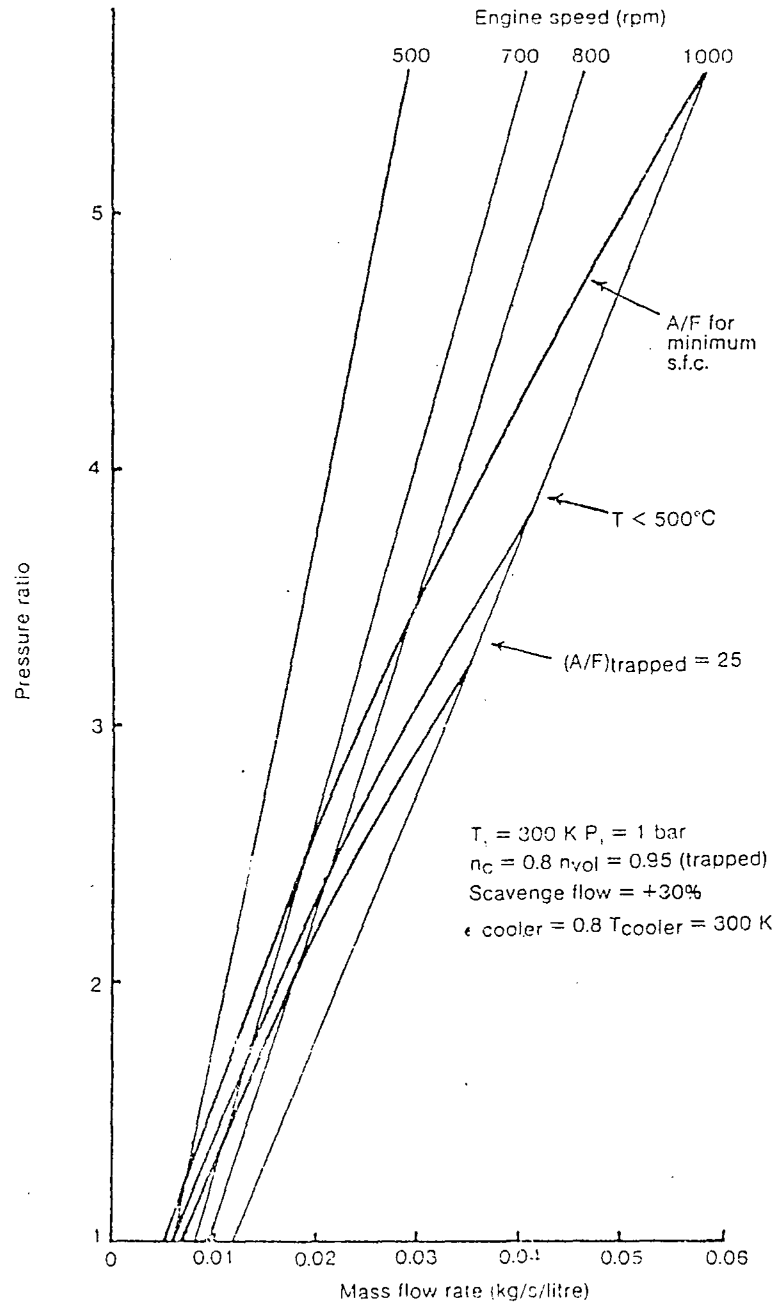


Fig 3.19 – Compressor requirements for marine medium-speed, four-stroke engine (propeller law).

discussed above and are dominated by the characteristics of the ship's propeller. Propeller power increases approximately as the cube of speed (the well known 'propeller law'). Thus the torque or BMEP required increases with speed squared. Fig 3.18 shows the power curve required for a medium –speed engine having a rating of 25 bar BMEP at full speed (1000 rpm) and load.

125. The airflow required for this engine can be calculated from equation (14) but the equivalence ratio (or air/ fuel ratio) should be a 'trapped in cylinder' value and an additional allowance made for scavenge flow. Three cases can be considered. First , a trapped air/fuel ratio sufficient only for acceptable exhaust smoke; typically this will be around 25:1 for a quiescent combustion chamber engine of this type (1000 rpm, 0.2 to 0.3 m bore). Secondly, a trapped air/fuel ratio sufficient for an exhaust temperature of not more than 500°C, for operation on heavy fuel (e.g up to 3500 s Redwood 1). The air/fuel ratio will vary along the propeller law but is likely to exceed 25:1 (trapped) at all speeds. Thirdly , an air/fuel ratio for minimum specific fuel consumption (at specified load). This is likely to be a trapped/ air fuel ratio of around 40:1 (Fig 3.11), but may be limited to a lower value owing to excessive cylinder pressure..

126. Figure 3.19 shows the airflow requirements for the compressor based on the three air/fuel ratio cases above, which a compressor isentropic efficiency of 80%, an intercooler effectiveness of 0.8, a 95% volumetric

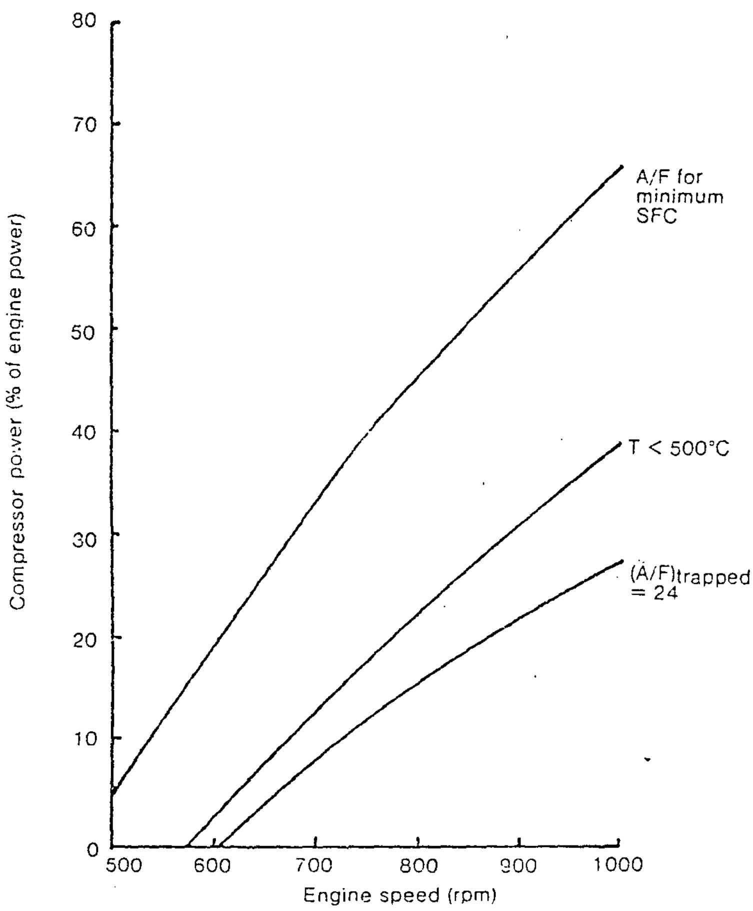


Fig 3.20 -- Compressor power requirement for marine four-stroke medium-speed engine (propeller law).

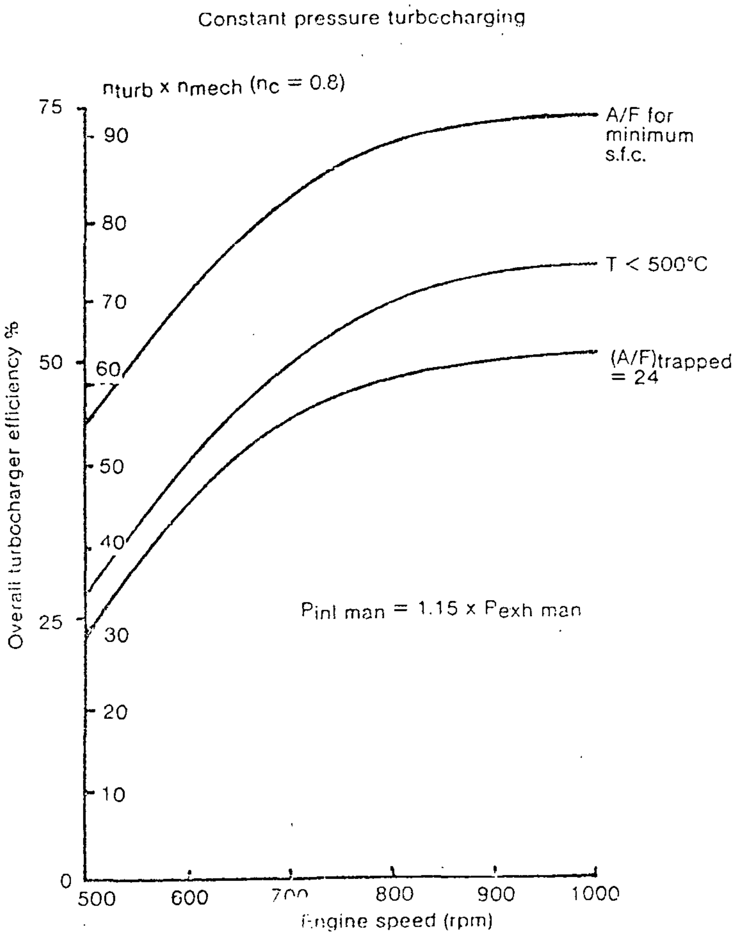


Fig 3.21 – Turbocharger and turbine efficiency required for marine four-stroke diesel engine (propeller law).

efficiency for trapped air plus 30% scavenge air. Obviously the required compressor characteristics are quite different from those for the truck engine. Quite a narrow flow range is adequate, with pressure ratio continually increasing with mass flow rate. It can also be seen that the additional pressure ratio required to achieve a 500°C exhaust temperature over that required for a smoke limit is not unreasonably large. However, for optimum fuel consumption a much higher pressure ratio is required. In this application, engine efficiency improvements as small a 1% are well worth some capital investment, hence there is a strong incentive to achieve high pressure ratio and/ or high compressor efficiency, but the former may be limited by the allowable maximum cylinder pressure.

127. The compressor power required to achieve the pressure ratios shown in Fig. 3.19 are represented in Fig. 3.20 . For optimum fuel consumption is rises to 67% of engine power at the rated condition. Even with sufficient air for burning residual fuel, compressor power rises to 40 % of engine power. Obviously mechanically driven supercharges cannot be considered for this highly rated engine. The overall turbocharger efficiency required must be based on an inlet manifold pressure around 1.15 times the exhaust manifold pressure, for good scavenging with constant pressure turbocharger. The efficiency required is plotted in fig 3.21. Current large turbochargers achieve maximum efficiency of 63 - 65% , which is sufficient for this engine running to an exhaust temperature limit of 500°C temperature, but not for optimum

fuel consumption. Thus if ratings of 4 stroke marine engines are to rise above current values of upto,20 bar BMEP to 25 bar, values of up to 20 bar BMEP to 25 bar, compressor and turbine efficiencies over 85% are required for optimum fuel consumption.

128. Figure 3.22 shows the turbine flow characteristics required for a 500° C exhaust temperature, with inlet manifold pressure 1.15 times exhaust manifold pressure. The requirements is an almost liner increase in mass flow rate with pressure, which suits the characteristic of a choked orifice quite well. It can be seen that the orifice flow characteristic is much closer to the required curve than in the case of the truck engine. The efficiency required to achieve the necessary boost with this turbine expansion ratio curve (Fig. 3.21) falls with reducing engine speed, hence higher efficiencies are likely to be available to compensate for the shortfall in expansion ratio shown in Fig.3.22 at less than rated speed. Thus this application is well suited to an orifice like flow characteristics from the turbine.

129. We can thus conclude that for the marine four-stroke medium speed diesel engine:

(a) A relatively narrow compressor flow range is adequate.

(b) The compressor power requirement is large relative to engine power, typically 40-60% at 25 bar BMEP. Thus turbocharging is essential.

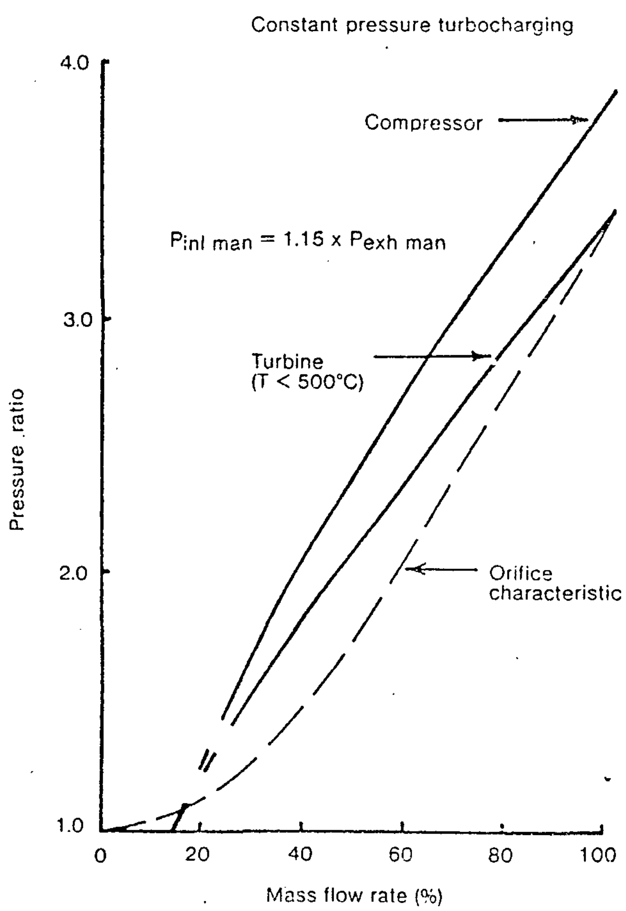


Fig.3.22 – Turbine flow characteristics for marine four-stroke medium-speed engine (propeller law).

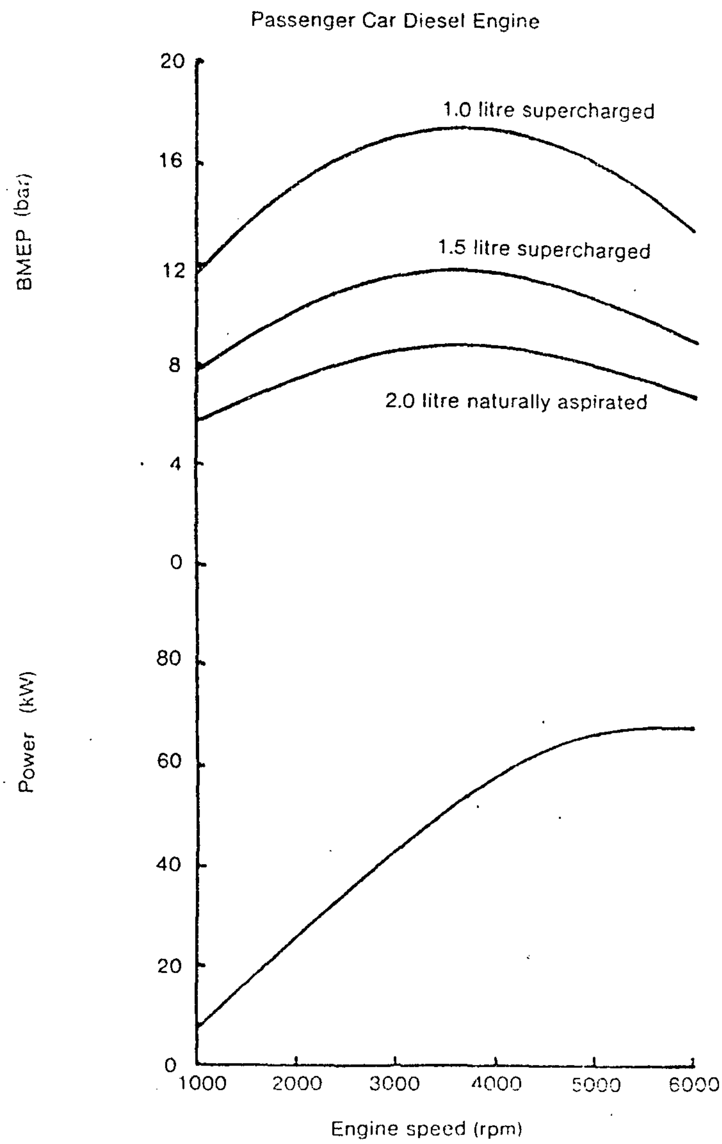


Fig 3.23 – Performance target of 2 litre (naturally aspirated) and equivalent 1.5 and 1.0 litre supercharged diesel engines

(c) For optimum fuel consumption very high compressor pressure ratio and/or overall turbocharger efficiency is required.

(d) Overall turbocharger efficiencies over 60% are required for residual fuel operation at high BMEP.

(e) Turbine expansion ratio should rise with mass flow rate. The flow characteristics of an orifice are quite suitable for this application.

130. **Four-stroke diesel passenger car engine:** The supercharger requirements for this application are dominated by the very wide speed range of the engine (typically 100-5000 rpm).

131. Consider first the desired torque curve. Figure 3.23 shows the power curve for an advanced European 2 litre passenger car diesel engine in naturally aspirated from and its BMEP (or torque) curve. 1.5 and 1 litre mechanically supercharged or turbocharged engine might be considered to replace the naturally aspirated 2 litre. Their BMEP requirements to achieve the same power curve are also shown in fig.3.23. Assuming a minimum air/fuel ratio (usually) smoke limited to 17-18:1 with indirect injection), equation (14) can be used to calculate the required airflow rate. Equation (4) can again be used to plot mass flow as a function of engine speed and compressor pressure ratio. Figure 3.24 shows the results, assuming that the volumetric efficiency varies with speed in the same manner as the naturally aspirated engine. Matching the required airflows at all speeds gives the required boost pressure curve. Not surprisingly, since we are trying to match

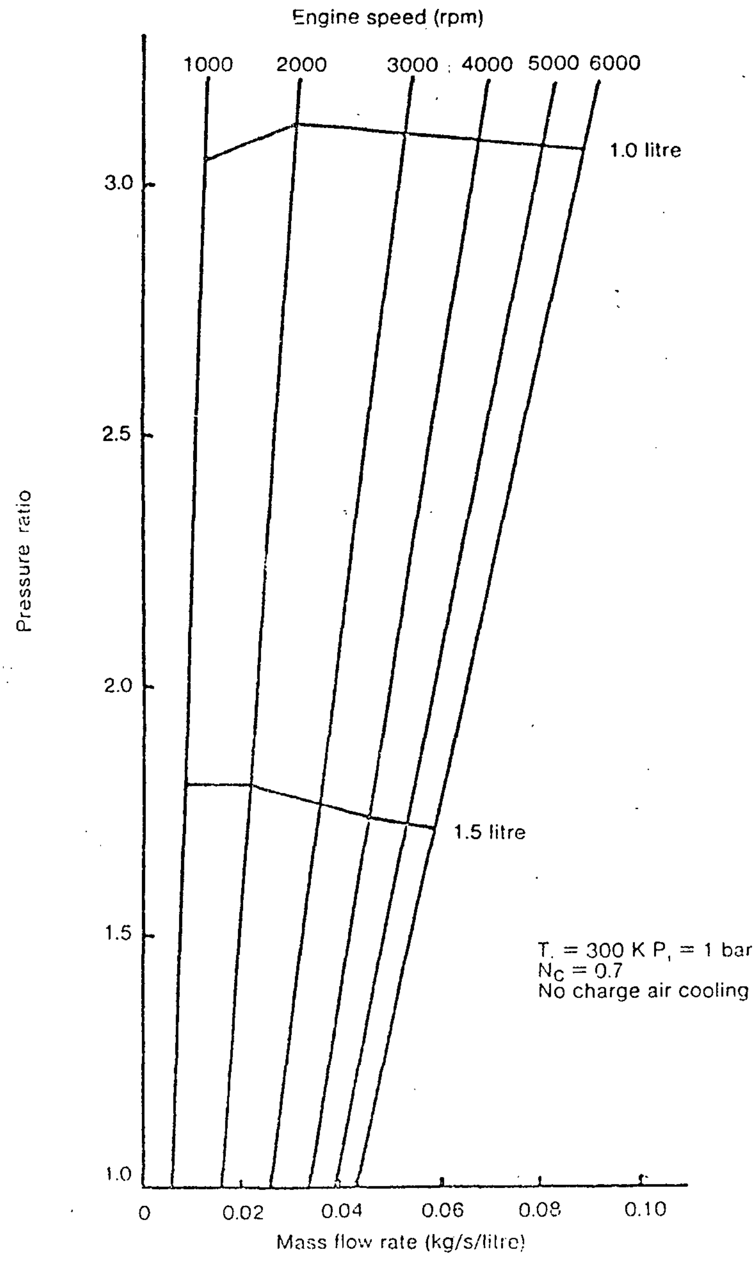


Fig.324 – Compressor requirements for supercharged diesel engine equivalent to 2 litre naturally aspirated.

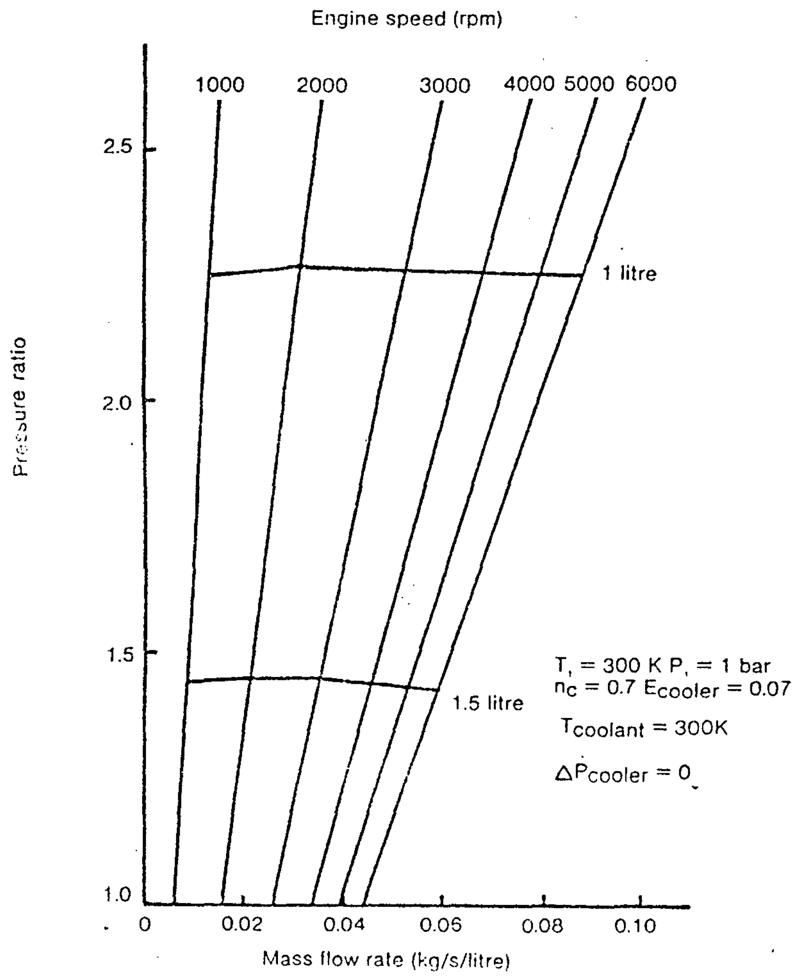


Fig.3.25 – Compressor requirements for charge cooled diesel engine equivalent to 2 litre naturally aspirated.

the torque curve of the naturally aspirated engine, the required compressor pressure ratio is almost constant, at about 1.75 for the 1.5 litre engine and 3.1 for the 1 litre engine. In both cases an extremely wide compressor flow range is required – virtually constant pressure ratio from 15 to 100% mass flow.

132. If a charge air cooler (air-air) is used, then the required pressure ratio reduces. Figure 3.25 is a companion to figure 3.24, an intercooled engine with cooler effectiveness of 0.7, but negligible pressure loss (in practice a small charge cooler will have a significant pressure loss). The charge cooler reduces the required pressure ratio from 1.75 to 1.45 (1.5 litre engine) and 3.1 to 2.25 (1 litre engine). The 1.5 litre engine looks a practical proposition, especially when charge cooled but the 1 litre still needs a high pressure compressor.

133. The power required to drive the compressor (assuming 70 % isentropic efficiency) is shown in Fig 3.26, as a function of engine power. If a mechanically driven supercharger is used, then the IMEP of the engine must be increased to provide the power, hence fuel and airflow must increase also. This has been ignored for simplicity in fig 3.26, hence the power requirement for supercharger is underestimated slightly. However, fig 3.26 reveals that the 1.5 litre engine, particularly in charge cooled from, could easily use a mechanically driven supercharger. If it were de-clutched or even by - passed when not required, the loss of engine efficiency would be

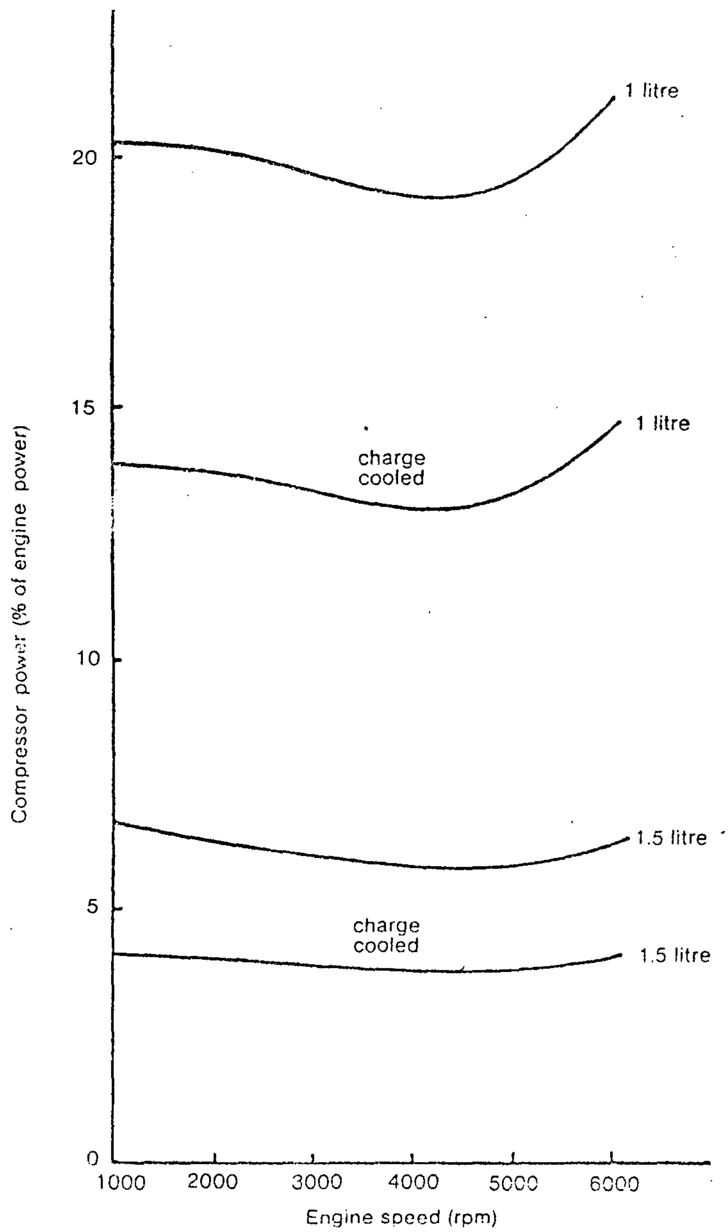


Fig.3.26 – Compressor power requirement for diesel engine equivalent to 2 litre naturally aspirated.

small except at full load. Since passenger car engines are rarely operated at full 'throttle' for any duration, this loss might be offset by the reduced friction and weight of the smaller engine, provided that the engine and vehicle were designed accordingly. Thus the 1.5 litre engine could be supercharged or turbocharged. In contrast, the power required to drive the compressor of the 1 litre engine is 14% (charge cooled) or 20% (not charge cooled) of engine power, which is excessive for a mechanically driven compressor. This is particularly true since the smaller engine will require boost for significantly more of its operating time. This engine must be turbocharged

134. The overall turbocharger efficiency required is similar to that of the truck engine (fig 3.16) but tends to fall with engine speed as turbine inlet temperature rises with almost constant air/fuel ratio. A modern turbocharger should be able to achieve these figures, even in the case of small units for automotive applications. The flow characteristics required from the turbine however, are a problem.

135. Since the compressor boost a should be almost constant, irrespective of engine speed or mass flow rate, so should the turbine expansion ratio. Orifice like flow characteristics are quite unsuitable . In this respect the passenger car engine is even more of a problem than the truck engine and it becomes impossible to achieve sufficiently large expansion ratio at low flow rates without becoming too high at high flow rates. One solution is to use a

very small turbine , equivalent to very small orifice, combined with a waste-gate that opens to bypass exhaust gas around the turbine at high speeds (recognised by high boost pressure). Even so it is difficult achieve the required boost pressure at very low engine speeds without a significant loss of engine efficiency at high speed owing to high exhaust manifold pressure.

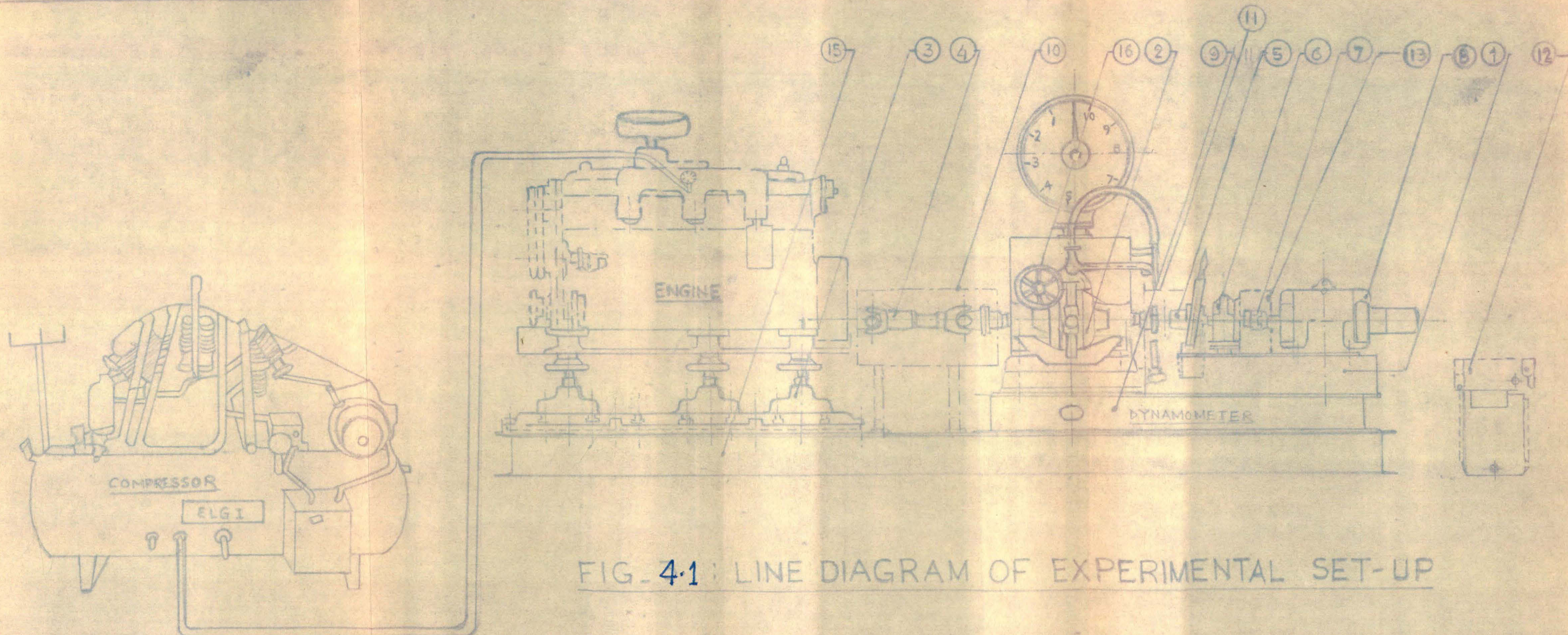


FIG. 4·1 : LINE DIAGRAM OF EXPERIMENTAL SET-UP

CHAPTER IV

PERFORMANCE TESTS AND EXPERIMENTAL SETUP

General

136. The work reported is on **D0026 M8A 4 stroke Diesel engine** fitted on truck **`Shaktiman'** which is one of the most extensively used load carriers in the Indian Army.

137. The object of the work was to investigate the response of this engine to the variation of air intake pressure.

138. To investigate the response of the D0026 M8A 4 stroke "Shaktiman" diesel engine, the experimental set-up was established at the Engine Test House of 505 Army Base Workshop. Delhi Cantt. This arrangement has been shown in fig 4.1 with the line diagram. The description of the engine, the compressor and the dynamometer used for the experiment are given in succeeding paragraphs.

139. **The engine :** The description of the engine is as under :-

(a)	Type	-	D0026 M 8A (Diesel)
(b)	Bore	-	100 mm
(c)	Stroke	-	125 mm
(d)	Operating Cycle	-	4-Stroke Diesel with direct fuel injection in to a hemispherical combustion chamber arranged at the centre of the piston crown (M-Type combustion).
(e)	No of cylinders	-	6 in line

(f)	Piston displacement	-	5880 cc
(g)	Compression pressure	-	26-30 Kg/cm^2
(h)	Compression ratio	-	20 : 1
(j)	Maximum output	-	82.06 KW (110 HP) ± 5% at 2500 rpm
(k)	Maximum Torque	-	343 Nm ± 5 % at 1600 rpm
(l)	Lubrication system	-	Forced feed lubrication by oil pump
(m)	Cooling System	-	Water cooled forced circulation pressurised to 91 kpa (0.8atm) thermostat controlled.

140. **The Compressor :** The description of the compressor (Fig 4.1) is as under :-

(a)	Make	-	ELGI
(b)	Model	-	TC 1000
(c)	Type	-	Reciprocating
(d)	Capacity	-	42 m 3/h
(e)	Working press	-	750
(f)	Tank	-	300 ltrs

141. **The Dynamometer:** The description of the dynamometer (fig 4.1) used is as under :-

(a)	Make	-	SAJ – FROUDE
(b)	Model	-	AWM 300 PB
(c)	Type	-	Hydraulic
(d)	Capacity of power Absorption	-	90 KW at 1000 RPM and 300 KW at 3000 to 5000 RPM

142. The Hydraulic Dynamometer is designed to load and measure power of all types of prime moves, i.e engine like 2 stroke and 4 stroke automotive diesel, petrol, marine aircraft and stationery engines.

143. Typical application of dynamometer include the following :-

(a) Production line tests in factories for final inspection and adjustments.

(b) Development tests on engines and engine components.

(c) Research tests for fundamental work on the process occurring in prime mover.

(d) Rating tests according to any designed standard..

(e) Type tests and endurance test on developed components.

(f) Educational tests for instruction in the performance certificate of engines.

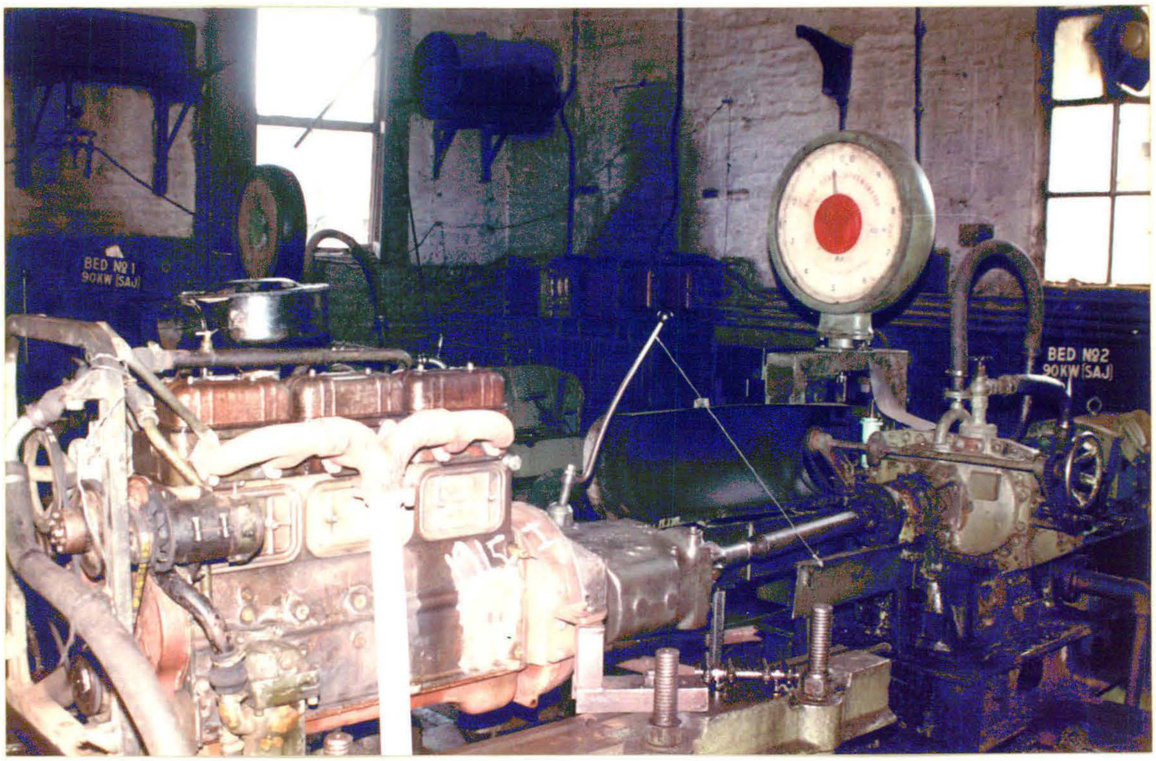
(g) Transmission tests on gear boxes, fluid couplings torque converters, variable speed drives etc.

144. The equipment (Refer fig 4.1) consists of Hydraulic Dynamometer (2) of 300 KW capacity with 22 KW – 1445 RPM running in motor (8). The 1445 RPM of running in motor is reduced to 519 RPM with the help of reduction helical gear boxes unit (6) . This running motor is coupled to gear box by flexible coupling (7) and in turn whole arrangement is coupled to dynamometer through hook jaw clutch (5). Dynamometer, hook jaw clutch, reduction gear box, running in motor is perfectly aligned on a common base

(9). Starter rotor type starter (12) is provided to start running in motor. Cardan shaft (4) is provided to couple different types of engines to the dynamometer. Adjustable six column universal engine mounting of different capacity engines at the time to testing. As safety measure metallic guards (10) are provided on carden shaft (4), hook jaw clutch (5) and flexible coupling (7). RPM indicator is provided to measure RPM at dynamometer shaft and temperature gauge to measure temperature of dynamometer water. Common base (15) is also provided with the dynamometer.

145. The main shaft is carried by bearings fixed in the casing (not in external support). The casing in turn is carried by anti friction trunions , so that it is free to swivel about the same axis as the main shaft. Prime mover is coupled to main shaft either directly or by a carden shaft . Power is transmitted through main shaft to the rotor revolving inside the casing. Water is circulated in the casing to provide the hydraulic resistance and simultaneously to carry away the heat developed by absorption of power.

146. In each face of the rotor are formed pockets of semielliptical cross section divided one from another by means of oblique vanes. The internal faces of the casing are provided with rings which are pocketed in the same way. Thus pockets in rotor and rings together form elliptical receptacles around which the water causes at high speed.



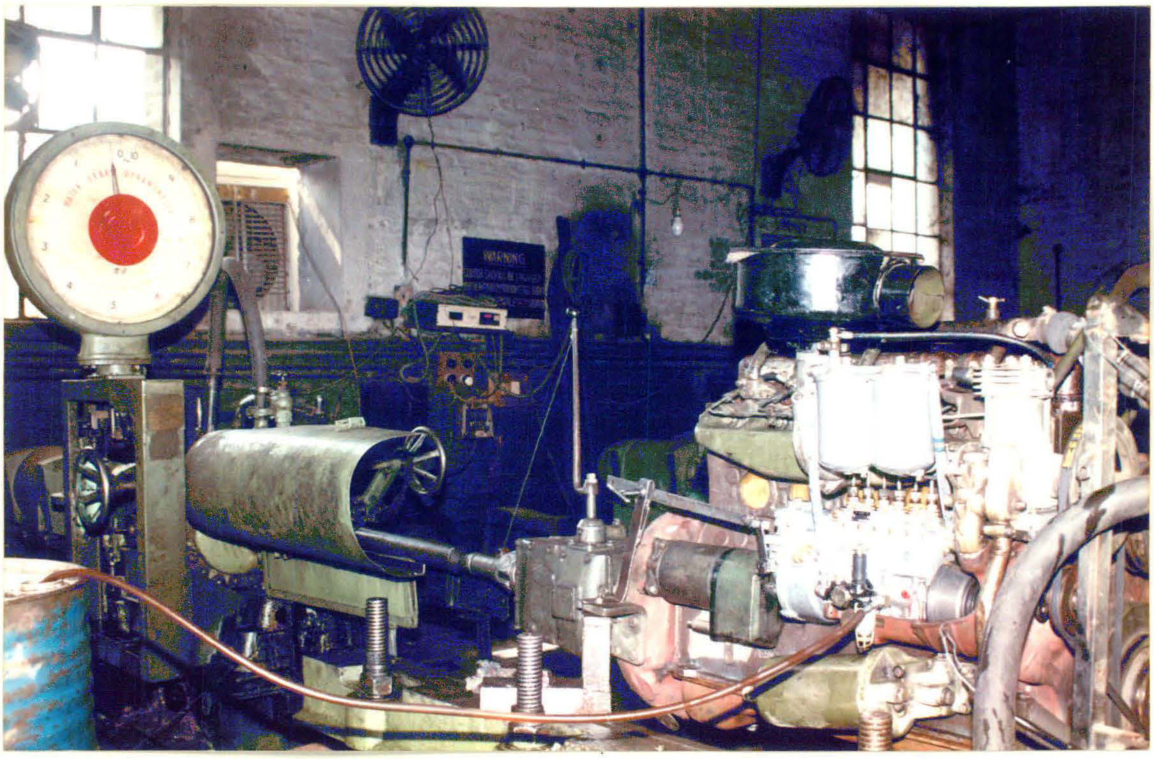


Plate Nos 1 & 2: A D0026 M8A engine coupled to a AWM300PB SAJ-FROUDE Dynamometer.

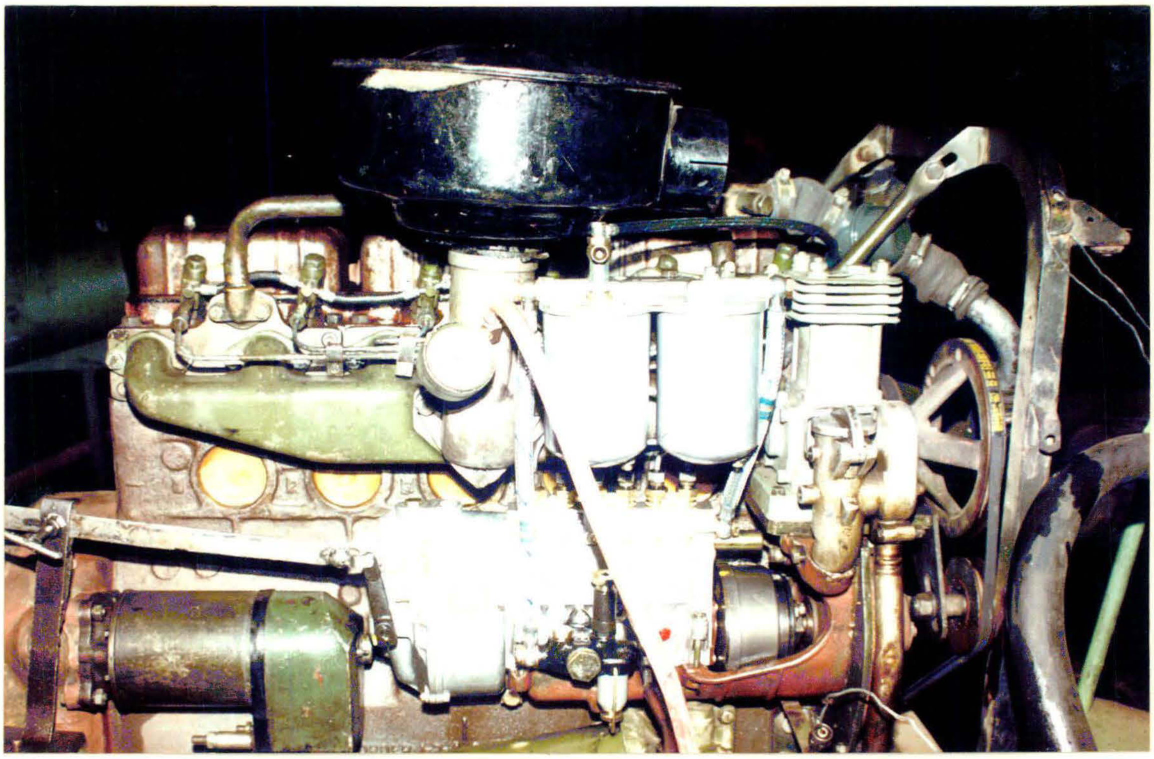
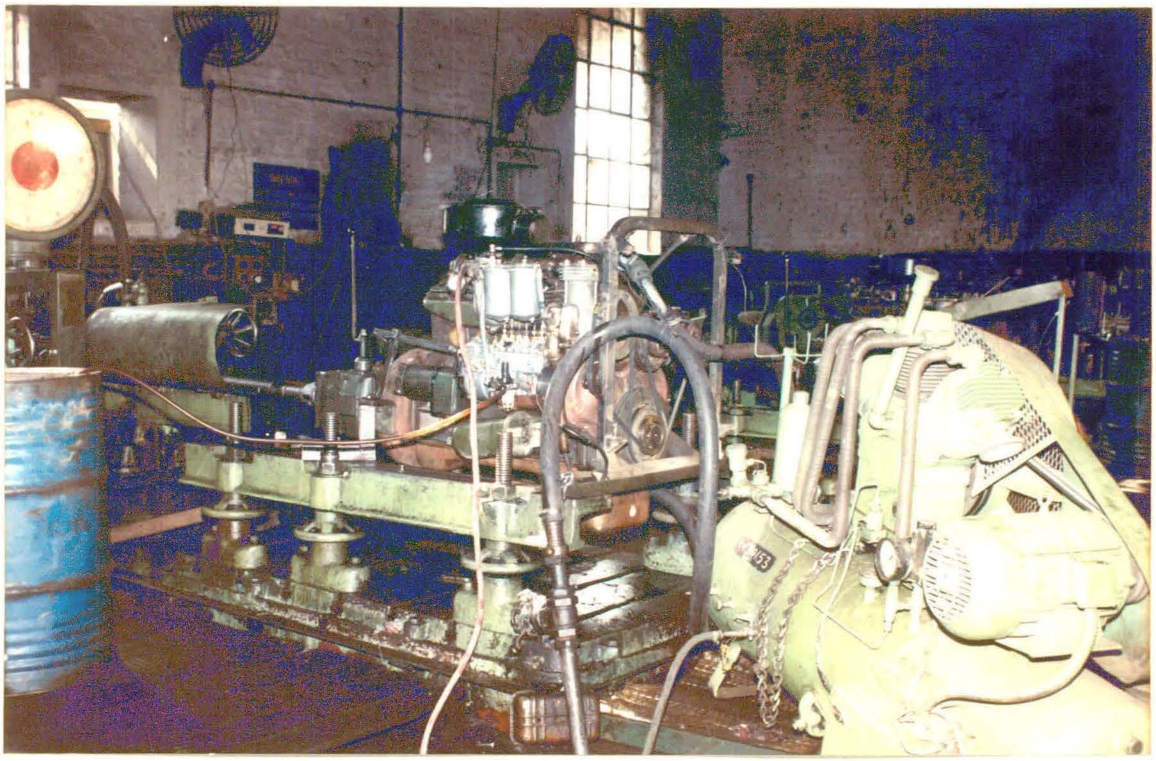


Plate No 3: An adapter fitted with a pressure gauge at the inlet manifold of a D0026 M8A diesel engine.



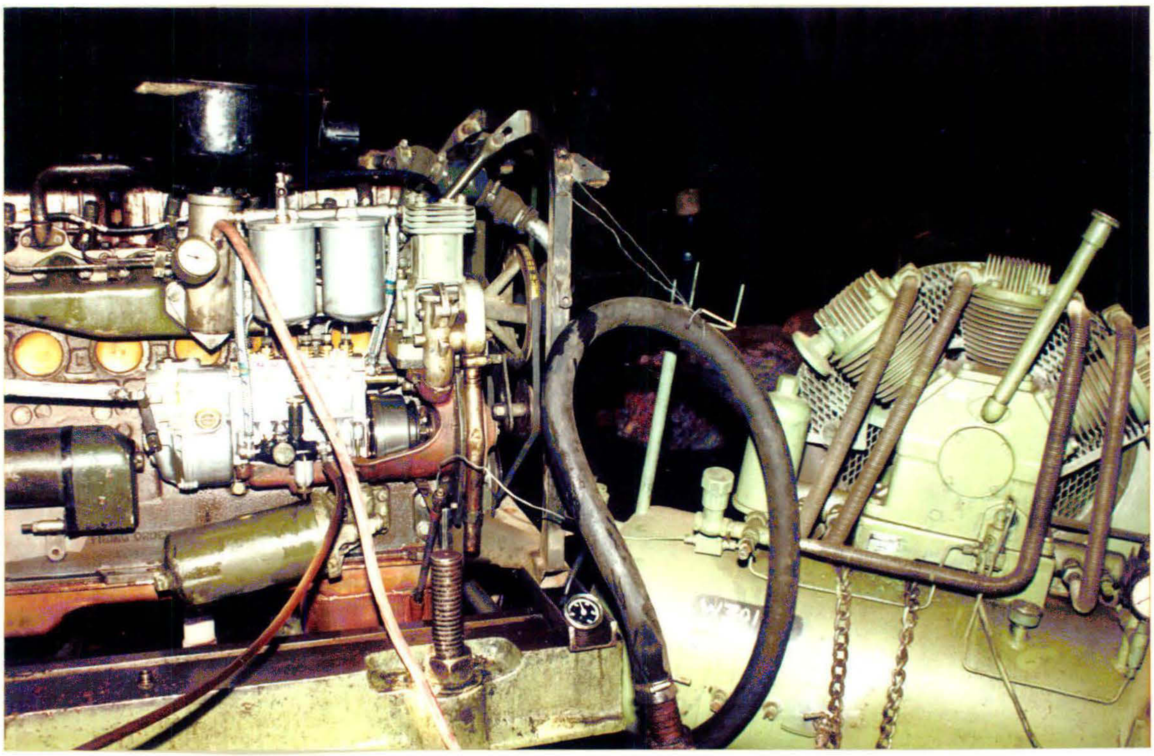


Plate Nos 4 & 5: An ELGI TC-1000 compressor connected to the inlet manifold of a DOO26 M8A diesel engine during the experiment.

147. When in action, the rotor discharges water at high speed from its periphery into the pockets formed in the casing rings, by which it is then returned at diminished speed into the rotor pocket at a point near the shaft.

148. The resistance offered by the water to motion of the rotor re-acts upon the casing, which tends to turn on its anti friction roller supports. This tendency is counter acted by means of a lever arm terminating in a weighing devices which measure the torque.

149. The test arrangement of the DOO26 M8A (Diesel) eng mounted on the bed and coupled to the AWM 300 PB, SAJ – FROUDE dynamometer in the Engine Test House of 505 Army Base Workshop are depicted in plate numbers 1 & 2 . This was the arrangement when the engine system response was investigated in naturally aspirated condition.

150. While investigating the system response to variation of air intake pressure, a half inch dia hole was drilled in the inlet manifold of the engine just below the air cleaner. An adapter with a pressure gauge mounted at one of its ends was fitted on to this half inch dia hole at the inlet manifold. This has been depicted in plate number 3. To the other end of the adapter, a hose from compressor outlet was connected (as shown in plate numbers 5 & 6) through which the air intake pressure was made to vary.

Reading Number	Engine RPM	Load in Dynamometer (kg)	Power developed (BHP)	Torque developed (Nm)
1	800	7.47	19.93	175
2	900	8.75	26.26	205
3	1000	9.93	33.09	232.5
4	1100	10.88	39.92	255
5	1200	11.74	46.97	275
6	1300	12.38	53.65	290
7	1400	13.02	60.75	305
8	1500	13.53	67.67	317
9	**1600**	13.88	74.00	**324.95**
10	1700	13.77	78.03	322.5
11	1800	13.66	81.98	320
12	1900	13.45	85.18	315
13	2000	13.23	88.24	310
14	2100	12.8	89.67	300
15	2200	12.23	89.72	286.5
16	2300	11.71	89.84	274.5
17	2400	11.25	90.00	263.47
18	**2500**	11.04	**92.00**	258.5
19	2600	9.82	85.11	230

Table 1 : Torque and Power developed by a D0026 M8A 4-stroke engine under naturally aspirated condition.

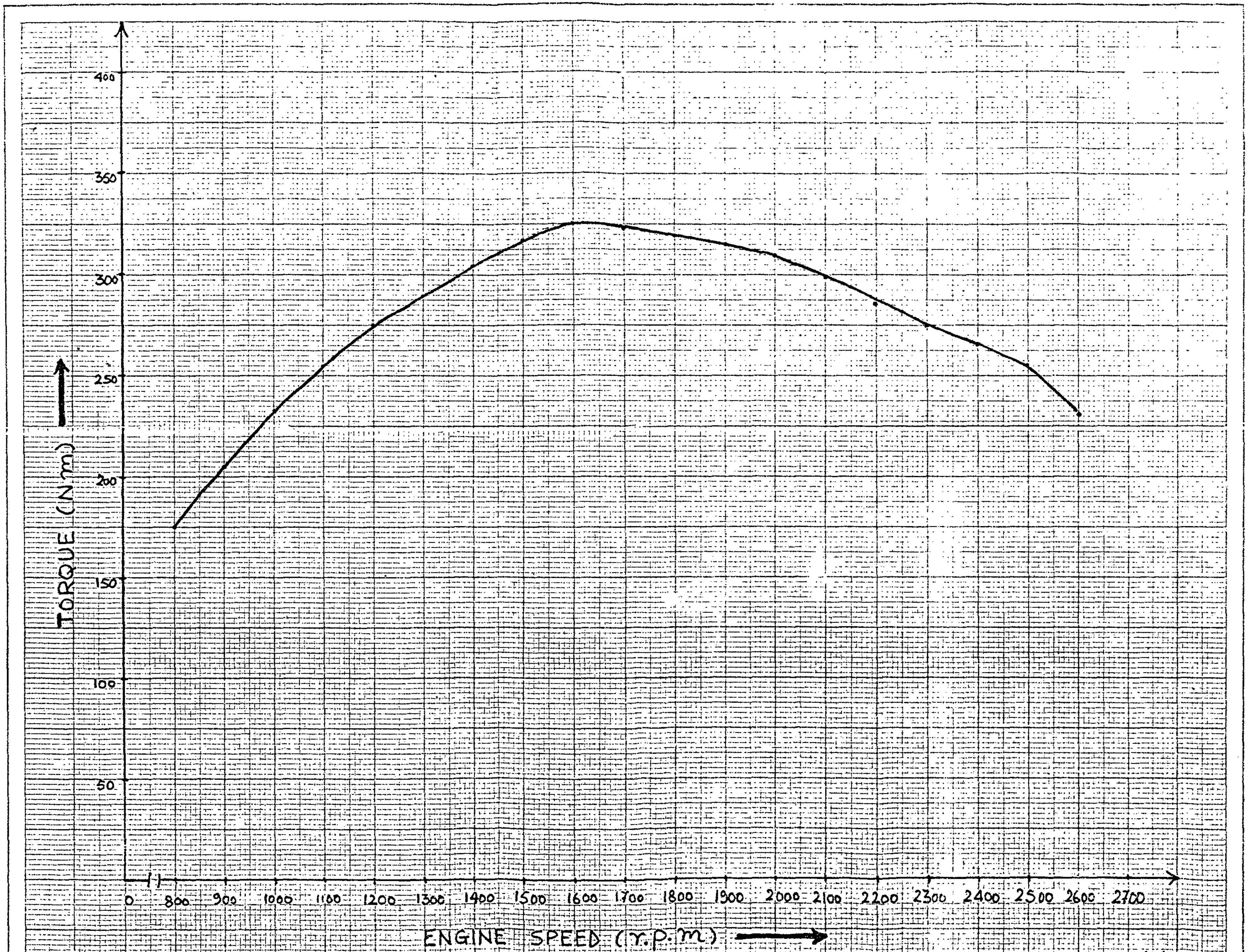


Fig 4.2 – Speed Vs Torque Characteristics of a D0026 M8A engine under naturally aspirated condition.

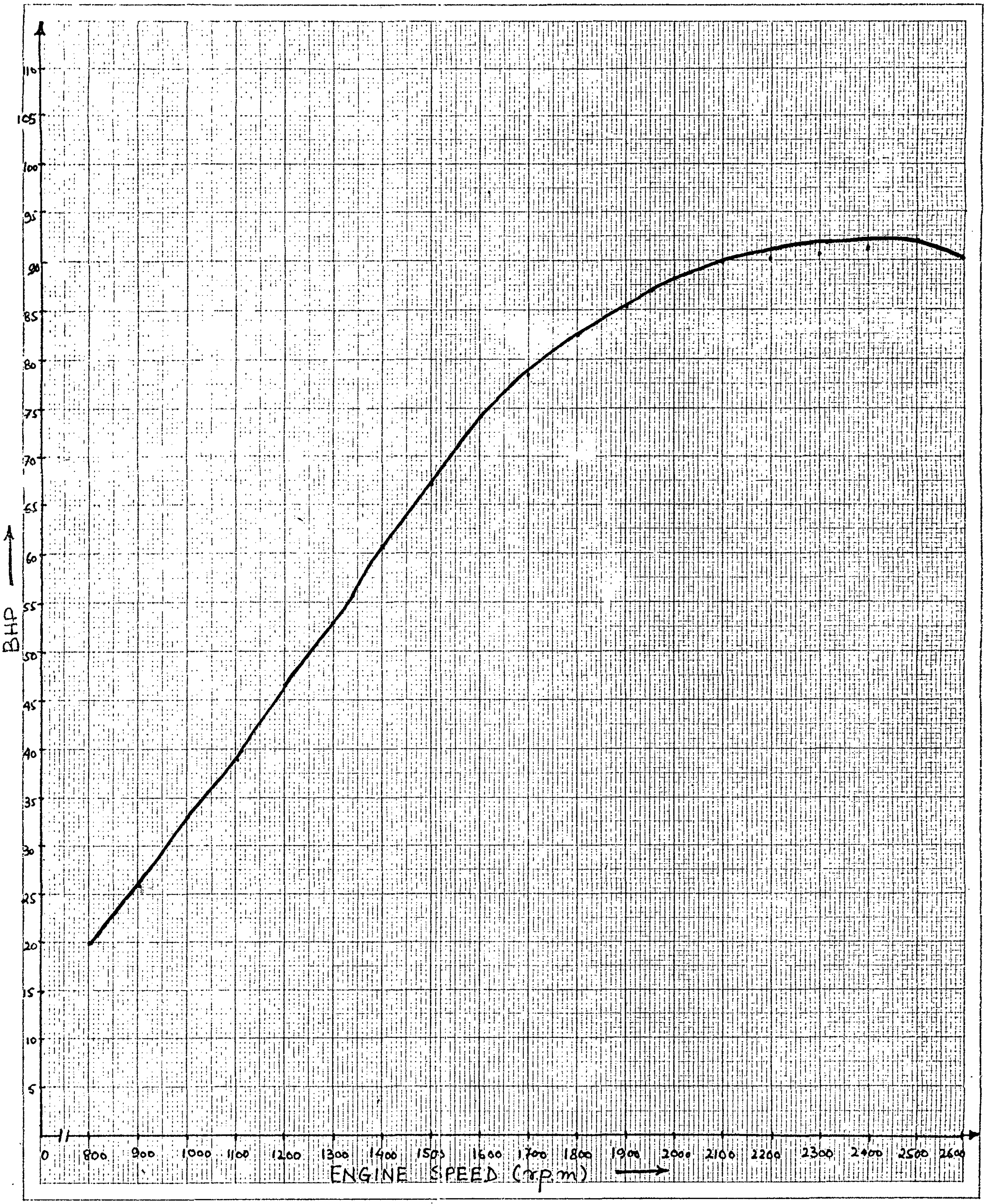


Fig 4.3 - Speed Vs BHP characteristics of a D0026 M&A engine under naturally aspirated condition

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PERFORMANCE TESTS

Natural Aspiration

151. The first requirement was to study the engine system response in the naturally aspirated conditions. The engine was mounted on the test bed and coupled with the dynamometer and made to run. The speed (rpm) of the engine was then increased gradually to its maximum and for every 100 rpm increase, the load on the dynamometer was recorded. This enabled in establishing the torque and power developed by the D0026M8A (Diesel) engine in the naturally aspirated condition as shown in table 1.

152. A set of performance curves for the engine in naturally aspirated condition were plotted for comparison with pressure charged performance curves.

153. Fig 4.2 gives the Torque Vs RPM characteristics for the naturally aspirated condition . A maximum torque of 325 Nm was obtained at 1600 RPM. This drop in maximum value of torque which should ideally be around 343 Nm can be attributed to the fact that the D0026 M8A(Diesel) Engine in the experimental set up is an overhauled one.

154. Fig 4.3 gives the BHP Vs RPM characteristics of the engine in the naturally aspirated condition. A maximum output of 92 HP was achieved at 2500 rpm which was also less than the maximum desired output of 110 HP. This drop can again be attributed to the reason mentioned in para 144.

Reading Number	Engine RPM	Load in Dynamometer (kg)	Power developed (BHP)	Torque developed (Nm)
1	800	7.47	19.93	175
2	900	8.75	26.26	205
3	1000	9.93	33.09	232.5
4	1100	10.88	39.92	255
5	1200	11.74	46.97	275
6	1300	12.38	53.65	290
7	1400	13.12	61.27	307.5
8	1500	13.64	68.21	319.5
9	**1600**	14.09	75.15	**330**
10	1700	13.98	79.24	327.5
11	1800	13.77	82.62	322.5
12	1900	13.55	85.86	317.5
13	2000	13.23	88.24	310
14	2100	12.8	89.67	300
15	2200	12.23	89.72	286.5
16	2300	11.71	89.84	274.5
17	2400	11.25	90.00	263.47
18	**2500**	11.04	**92.00**	258.5
19	2600	9.82	85.11	230

Table 2 : Torque and Power developed by a D0026 M8A 4-stroke engine under a pressure charge ratio of 1.161665.

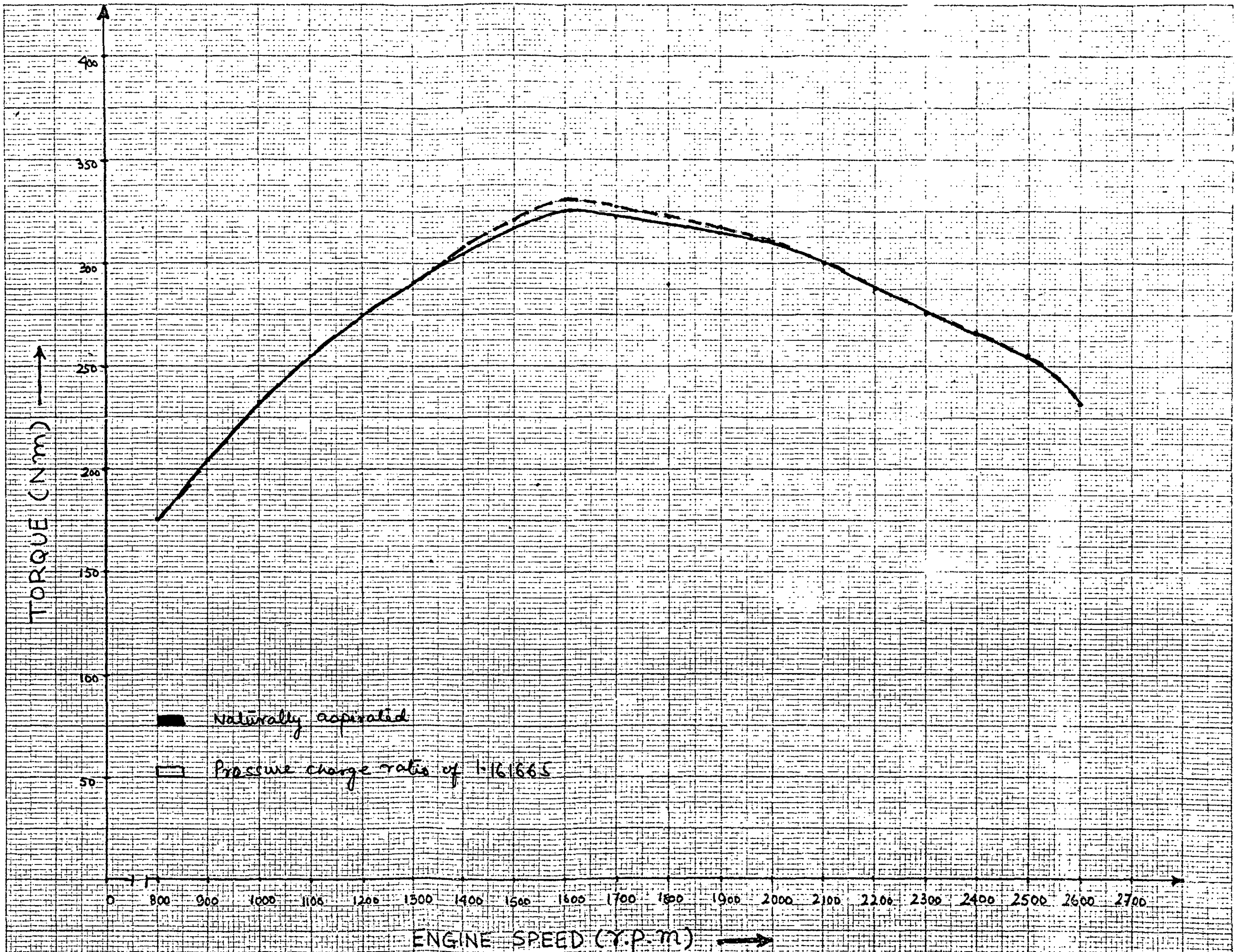

Fig 4.1 Speed Vs Torque characteristics of a D0026M8A engine under pressure charge ratio of 1.161665 and naturally aspirated conditions

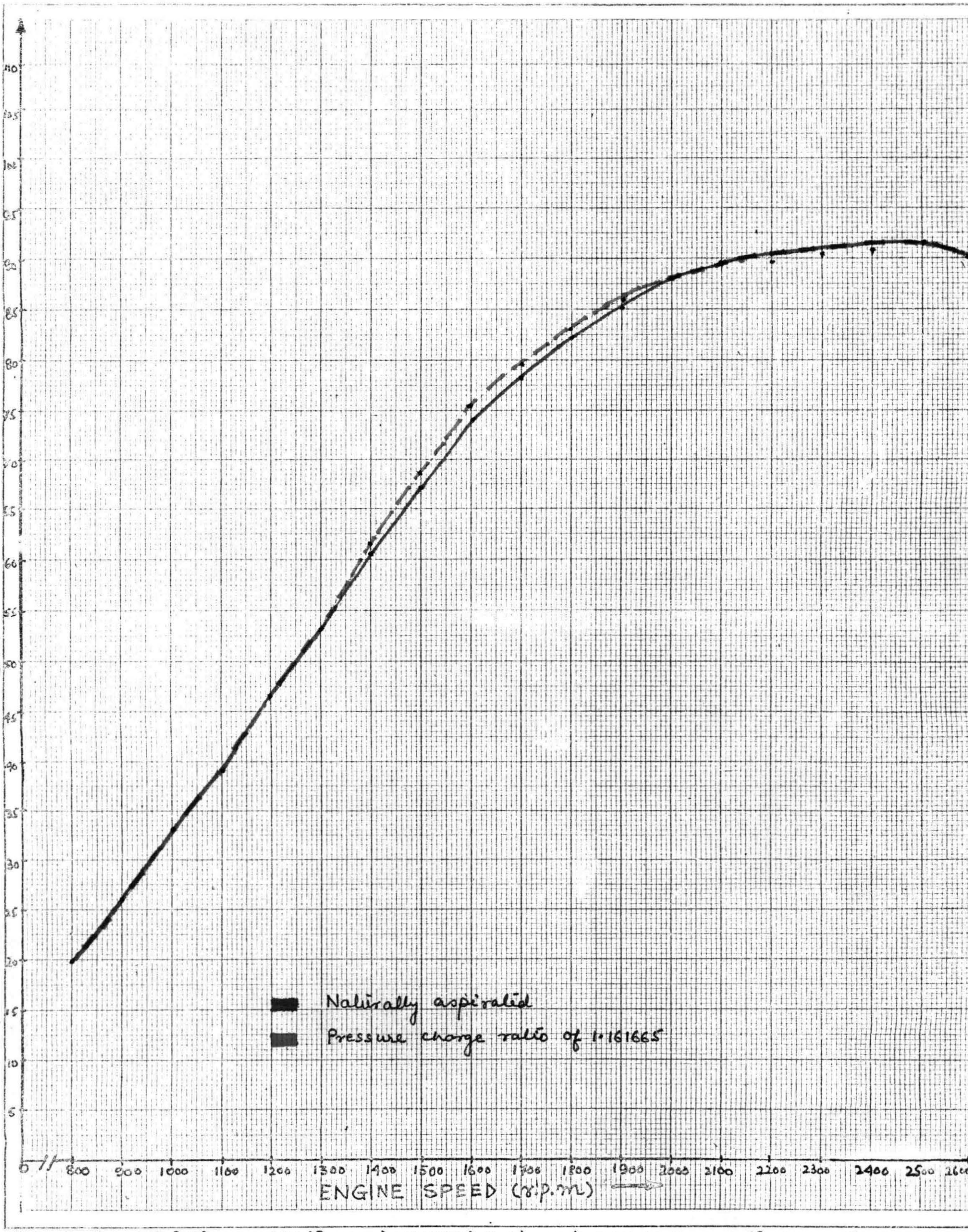

Fig 4.5 – Speed Vs BHP characteristics of a D0026 MBA engine under pressure charge ratio of 1.161665 and naturally aspirated conditions.

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155. Thus, in this experimental set-up we proceed with the ideal characteristics of the engine in naturally aspirated condition as

(a) Maximum Torque - 324.95 Nm at 1600 rpm

(b) Maximum output - 92 BHP at 2500 rpm

156. The compression pressure of the engine in the naturally aspirated condition was measured and was found to be 25 Kg/Cm^2.

Pressure Charging

157. For studying the engine system response to the variation of air intake pressure, that is, pressure charging of the engine, the compressor was connected to the intake manifold of the engine as shown in fig 4.1 and plate numbers 5 & 6 . The rest of the test arrangement and parameters were same as they were in the time of naturally aspirated condition.

158. For the first pressure charging test, the charge pressure at the inlet manifold was kept at 1.2 Kg/cm^2, that is, a charge pressure ratio of 1.161665. A set of readings for load on the dynamometer, the torque and the power developed by the engine were obtained for every 100 rpm increase in the engine speed. This has been shown in table 2. The characteristics curves were then plotted for comparison with naturally aspirated performance curves.

159. Figures 4.4 and 4.5 give a comparison of the characteristics curves obtained for the naturally aspirated engine and for the pressure changed engine with a pressure charge ratio of 1.161665. In this case we find that

Reading Number	Engine RPM	Load in Dynamometer (kg)	Power developed (BHP)	Torque developed (Nm)
1	800	7.8	20.80	182.5
2	900	9.18	27.54	215
3	1000	10.76	35.87	252
4	1100	12.16	44.62	285
5	1200	13.45	53.8	315
6	1300	14.73	63.84	345
7	1400	15.79	73.73	370
8	**1500**	16.65	82.26	**390**
9	1600	16.43	87.67	385
10	1700	16.11	91.30	377.5
11	1800	15.8	94.79	370
12	1900	15.47	98.00	362.5
13	2000	15.15	101.05	355
14	2100	14.73	103.11	345
15	2200	14.09	103.33	330
16	2300	13.66	104.75	320
17	**2400**	13.02	**106.00**	310
18	2500	12.27	102.30	287.5
19	2600	11.63	100.84	272.5

Table 3 : Torque and Power developed by a D0026 M8A 4-stroke engine under a pressure charge ratio of 1.9361.

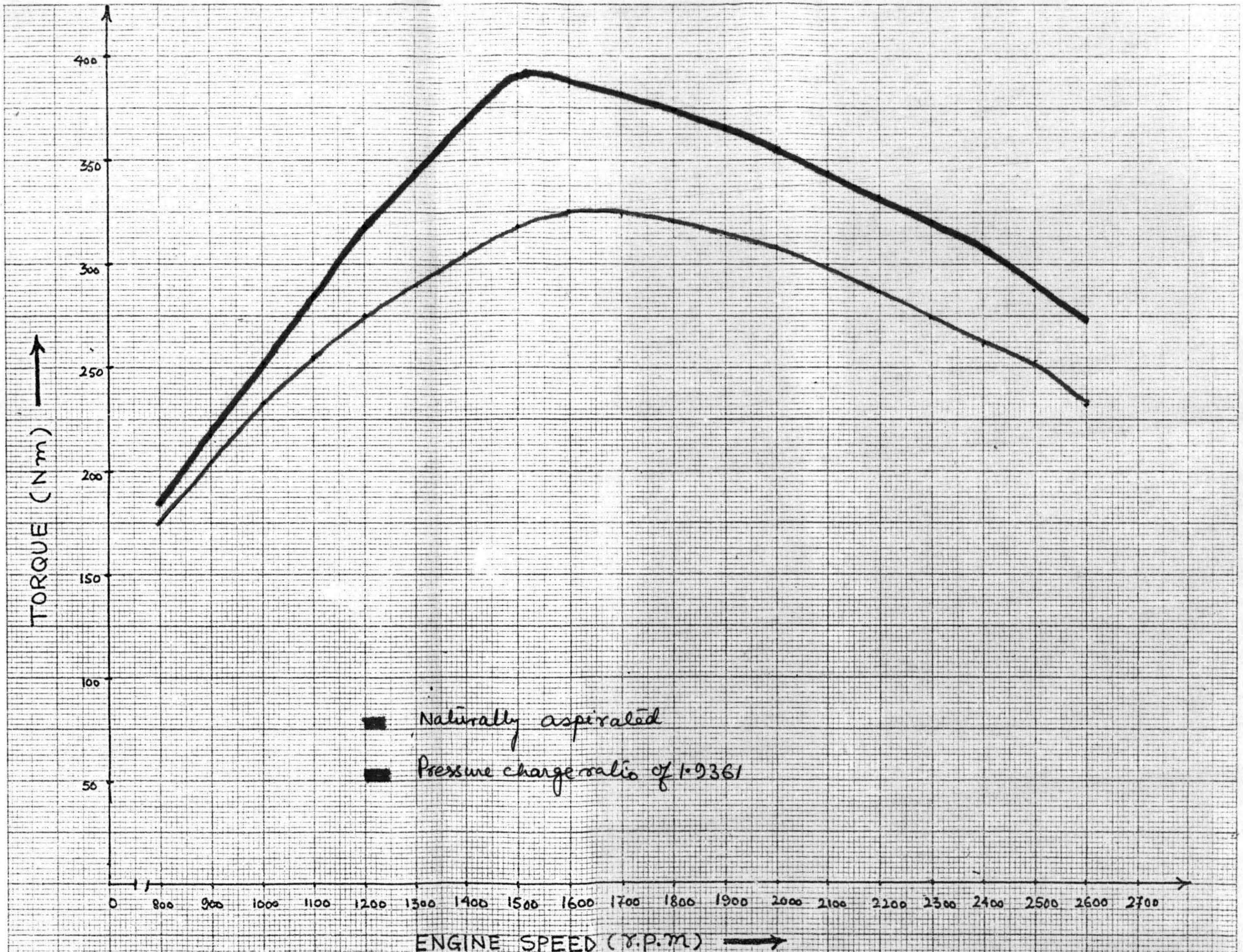

Fig 4·6 Speed Vs Torque characteristics of a DD026M8A engine under pressure charge ratio of 1·9361 and naturally aspirated conditions.

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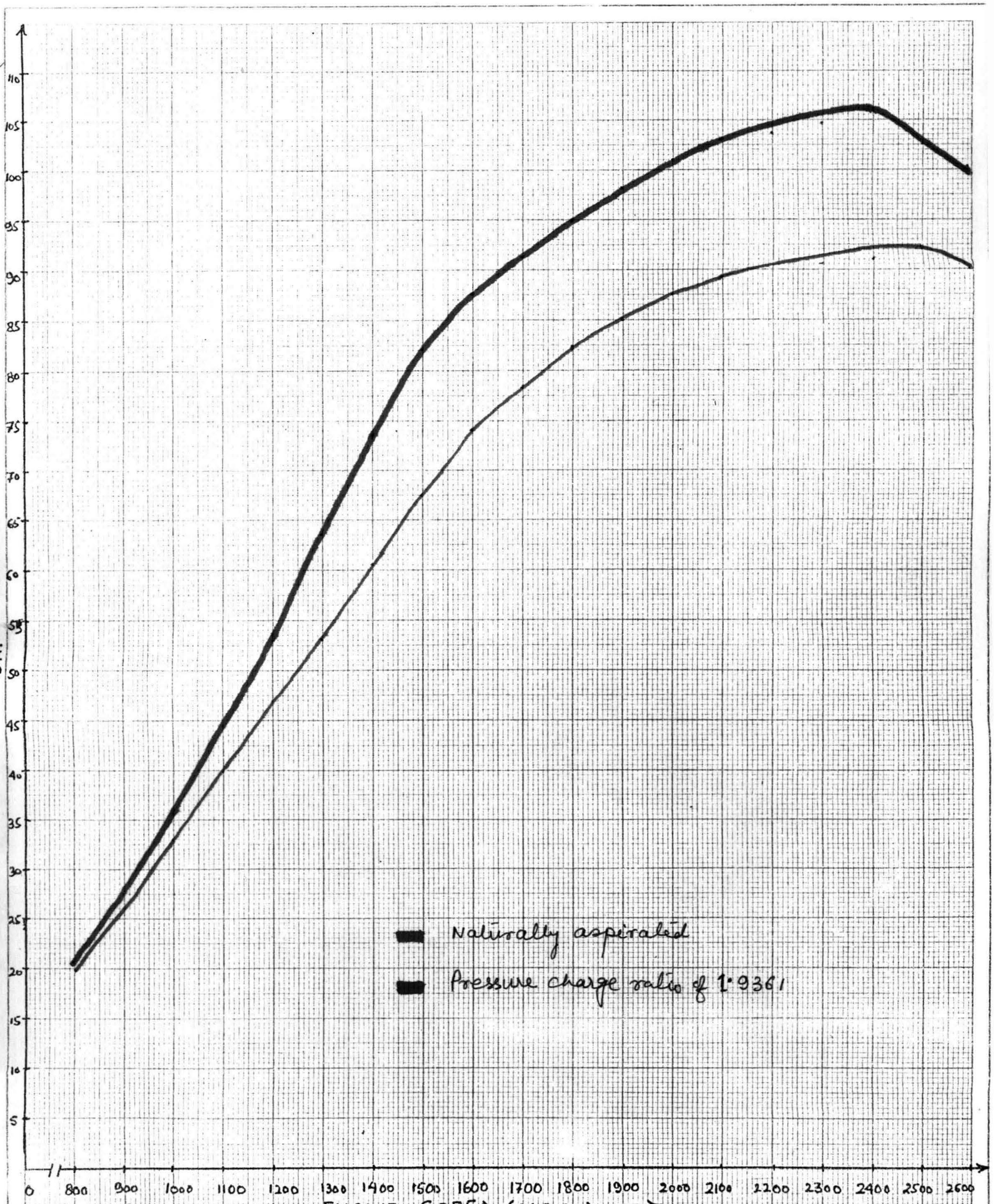

Fig 4.7 - Speed Vs BHP characteristics of a D0026M8A engine under pressure charge ratio of 1.9361 and naturally aspirated conditions.

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initially with the increase in engine speed, there is no increase in the torque developed till about 1300 rpm. At around 1400 rpm there is a marginal increase in the torque developed. The torque keep on increasing, though again marginally, till 1800 rpm after which there is again no change in torque developed by the engine. The maximum power developed by the engine remains the same. However, there is a notable increase in the compression pressure which was 26.5 Kg/Cm^2.

160. The next pressure charging test was conducted with a charge pressure of 2 Kg/cm^2 at the inlet manifold, that is, a charge pressure ratio of 1.9361. Table 3 gives the set of readings obtained under this pressure charged conditions. Here we observe that in a pressure charged condition with a pressure ratio of 1.9361, the valve of maximum torque is 390 Nm at 1500 rpm and the maximum power developed by the engine is 106 HP at 2400 rpm.

161. Figures 4.6 & 4.7 give the comparative pictures of the response of the engine under the naturally aspirated and the pressure charged conditions with the pressure charge ratio being 1.9361.

162. In figure 4.6 we observe that there has been an approximately 20% increase in the maximum torque developed by the engine when pressure charged with a change pressure ratio of 1.9361. Another important observation to make note of here is that the maximum torque has been

Reading Number	Engine RPM	Load in Dynamometer (kg)	Power developed (BHP)	Torque developed (Nm)
1	800	8.85	23.62	207.5
2	900	9.16	27.54	215
3	1000	11.74	39.14	275
4	1100	13.34	48.92	312.5
5	1200	14.51	58.00	340
6	1300	15.79	68.46	370
7	1400	16.86	78.7	395
8	**1500**	17.72	88.6	**415**
9	1600	17.5	93.36	410
10	1700	17.07	96.78	400
11	1800	16.54	99.27	387.5
12	1900	16.22	102.76	380
13	2000	15.69	104.61	367.5
14	2100	15.15	106.1	355
15	2200	14.51	106.46	340
16	2300	13.98	107.21	327.5
17	**2400**	13.45	**107.60**	315
18	2500	12.8	106.74	300
19	2600	12.06	104.54	282.5

Table 4 : Torque and Power developed by a D0026 M8A 4-stroke engine under a pressure charge ratio of 2.90416.

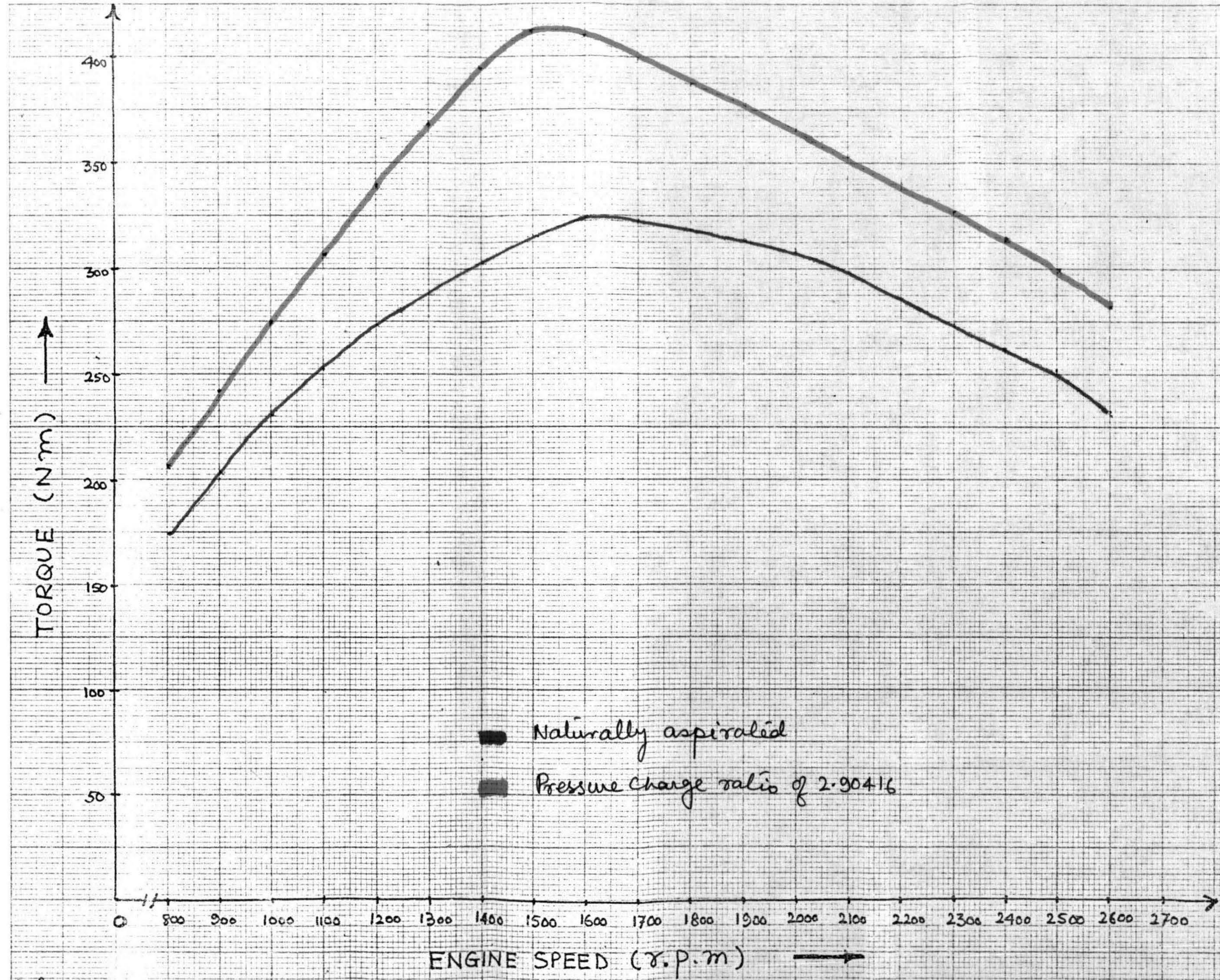

Fig 4.8. Speed Vs Torque characteristics of a 30026 M8A engine under pressure charge ratio 2·90416 and naturally aspirated condition

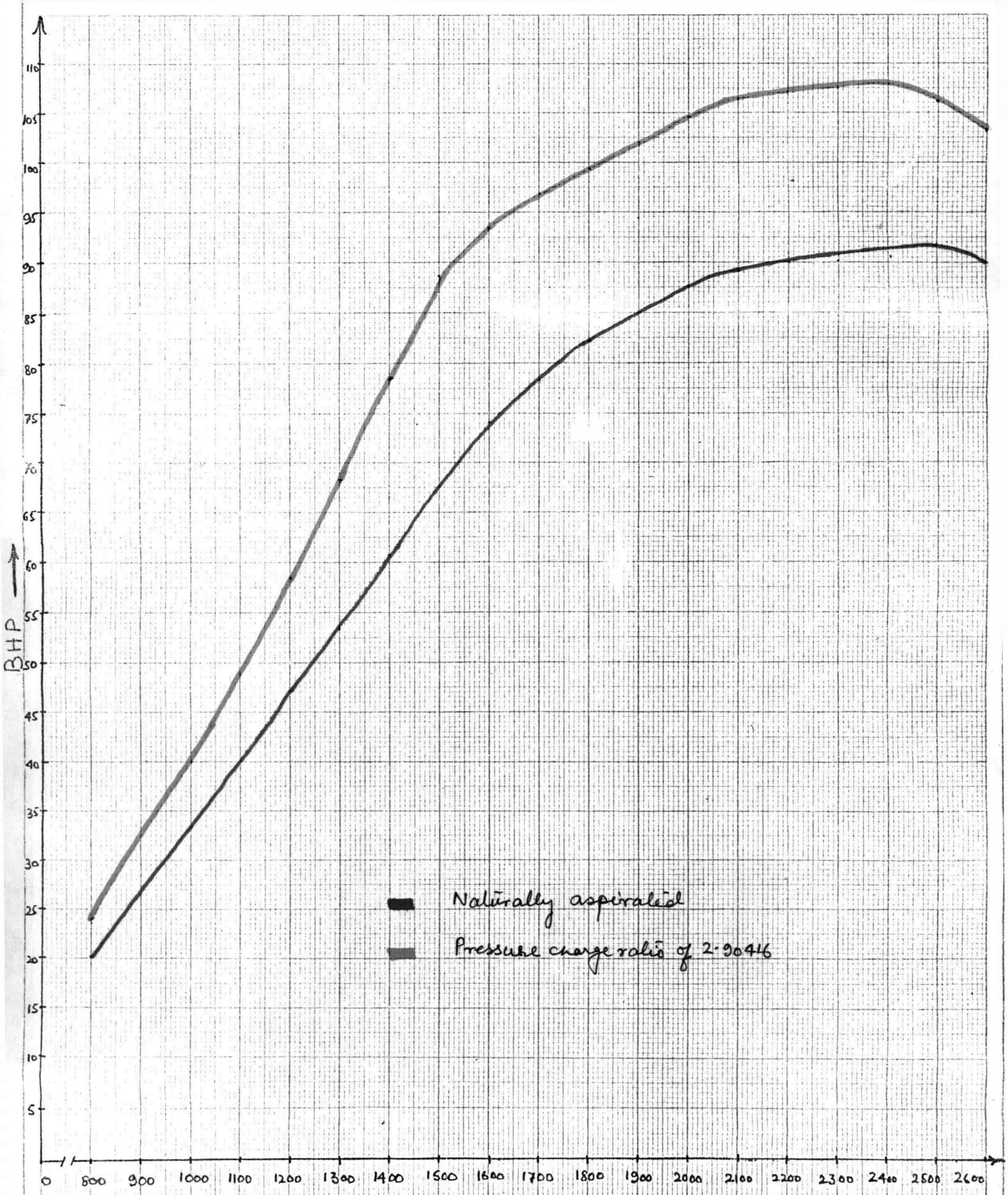

Fig 4.9 Speed Vs BHP characteristics of a D0026M8A engine under pressure charge ratio of 2.90416 and naturally aspirated conditions.

Neelgagan

obtained at 1500 rpm, whereas the same was obtained in the case of naturally aspirated condition at 1600 rpm.

163. Referring to figure 4.7 we find that under the pressure charge the condition with a pressure charge ratio of 1.9361, the maximum power developed by the engine is 106 Nm at 2500 rpm which is about 15% more than 92 HP at 2600 rpm that was achieved in naturally aspirated conditions. In this case also the maximum power developed is about 100 rpm earlier than in naturally aspirated conditions. Also the maximum power of 92 HP for the naturally aspirated engine was achieved as early as at 1700 to 1800 rpm in this pressure changed condition when pressure ratio was 1.9361.

164. Under this pressure charge ratio of 1.9361, the compression of the engine was found to be 29 kg/cm^2

165. The engine was next pressure charged with a pressure charge ratio of 2.90416, that is, a charge pressure of 3 Kg/cm^2 at the inlet manifold. The results obtained under this condition is given in table 4.

166. Figures 4.8 & 4.9 give a comparative picture of the torque and power developed by the engine under naturally aspirated and pressure charged conditions with the pressure charge ratio being 2.90416. Here we find that in this particular case of pressure charge ratio of 2.90416, the maximum torque developed by the engine was 415 Nm at around 1500 rpm which is about 28% more and that the maximum power developed was 107.60 HP at

around 2200 rpm which is about 17% more than that obtained in naturally aspirated condition.

167. Further experiments with increasing the air intake pressure were not conducted because of safety reasons as the compression pressure under the pressure charged condition of 3Kg/cm² at the inlet manifold was found to be 33.5 Kg/cm² and which is more than 10% of the designed limit of the D0026 M8A engine and hence the investigation on the response of the engine to variation of air intake pressure was concluded at this juncture.

CHAPTER V

ANALYSIS OF RESULT

General

168. The two most important factors pertaining to an engine system response, that is, the Torque and the power developed under varying the air intake pressure will be discussed during the course of this chapter.

169. Before an attempt is made to analyse the test results, it is important to see as to what are the main factors limiting the engine performance. These factors and their effect on the engine performance are discussed in the following paragraphs:-

Engine Performance Limitations

170. The key factors limiting the engine performance are the thermal loading and stress, the mechanical loading and resultant stress and the minimum air fuel ratio.

171. The thermal loading of the engine is related with the exhaust valve temperature and hence the engine performance is limited by this factor. Since the increase in the exhaust valve temperature is also directly related to the variation in air intake pressure, it is pertinent that increase in air intake pressure beyond a certain limit is not always feasible.

172. Mechanical loading is usually characterized by maximum cylinder pressure and is related to boost pressure, engine compression ratio and injection timing etc. Since the factors like the compression ratio and injection timing are fixed during the design stage for a given engine, these cannot, or rather, should not be varied. Hence in the experimental set up, we have considered the only other variable, that is the maximum boost pressure as the limiting factor.

173. Based on the above facts, it was decided at the beginning of the experiment not to mechanically load the engine by more than 10% of the upper limit of the designed value.

Analysis of the Results

174. The effect of variation of air intake pressure on the torque and power developed by the engine in the experimental setup are discussed in the subsequent paragraphs.

Torque

175. The torque developed by an engine is directly related to its BMEP. The BMEP is affected by the air/fuel ratio. The maximum value of BMEP is achieved in the vicinity of chemically correct air/fuel ratio.

176. The poor distribution of fuel and its limited inter-mixing with the air results in incomplete combustion. Hence the CI engines are always operated with excess air varying from 35 – 50%. Thus, when the speed of the engine increases, the quantity of air available in the combustion chamber increases,

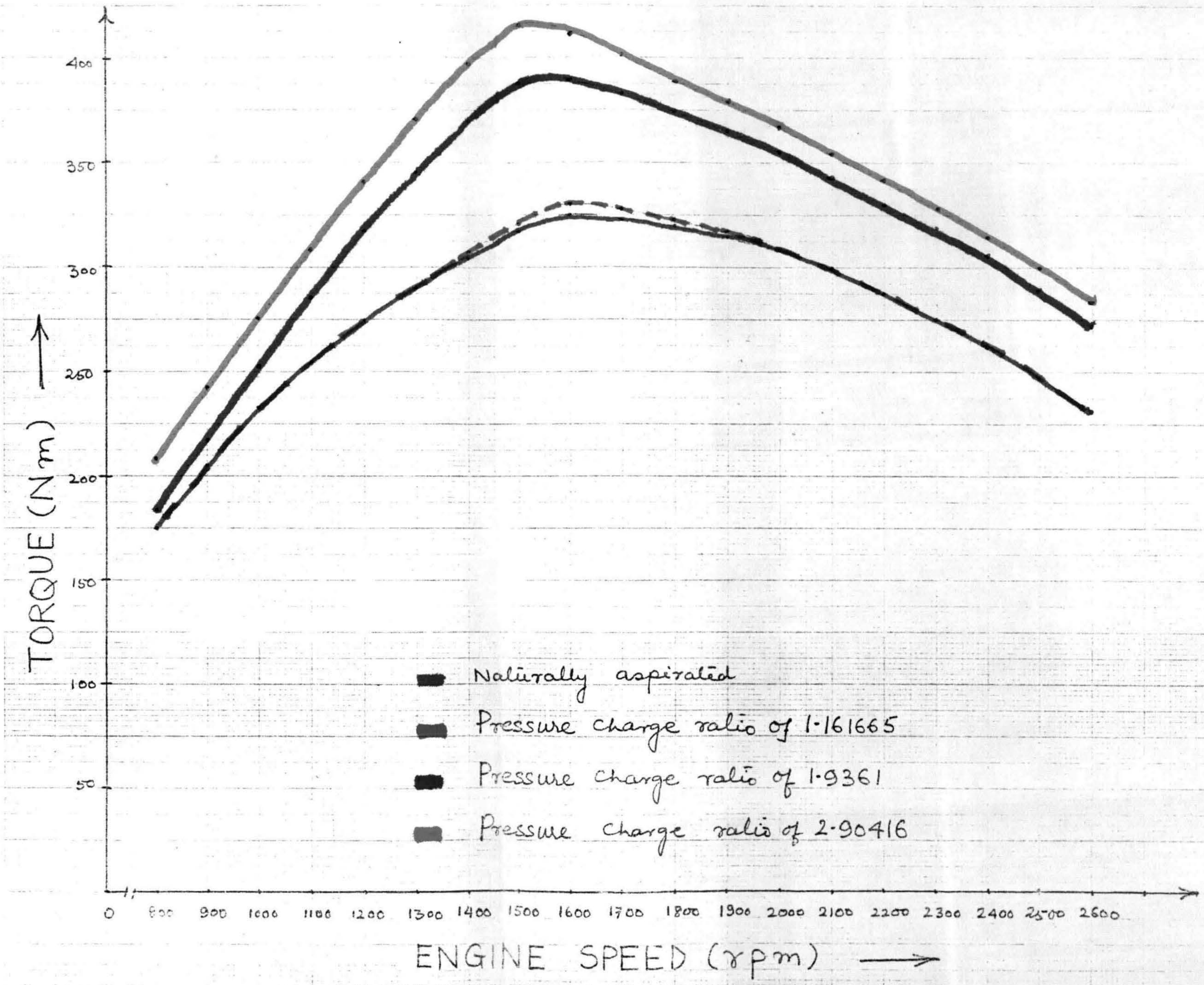

Fig 5.1 - Speed Vs Torque characteristics of a D0026 M8A engine under varying air intake pressures.

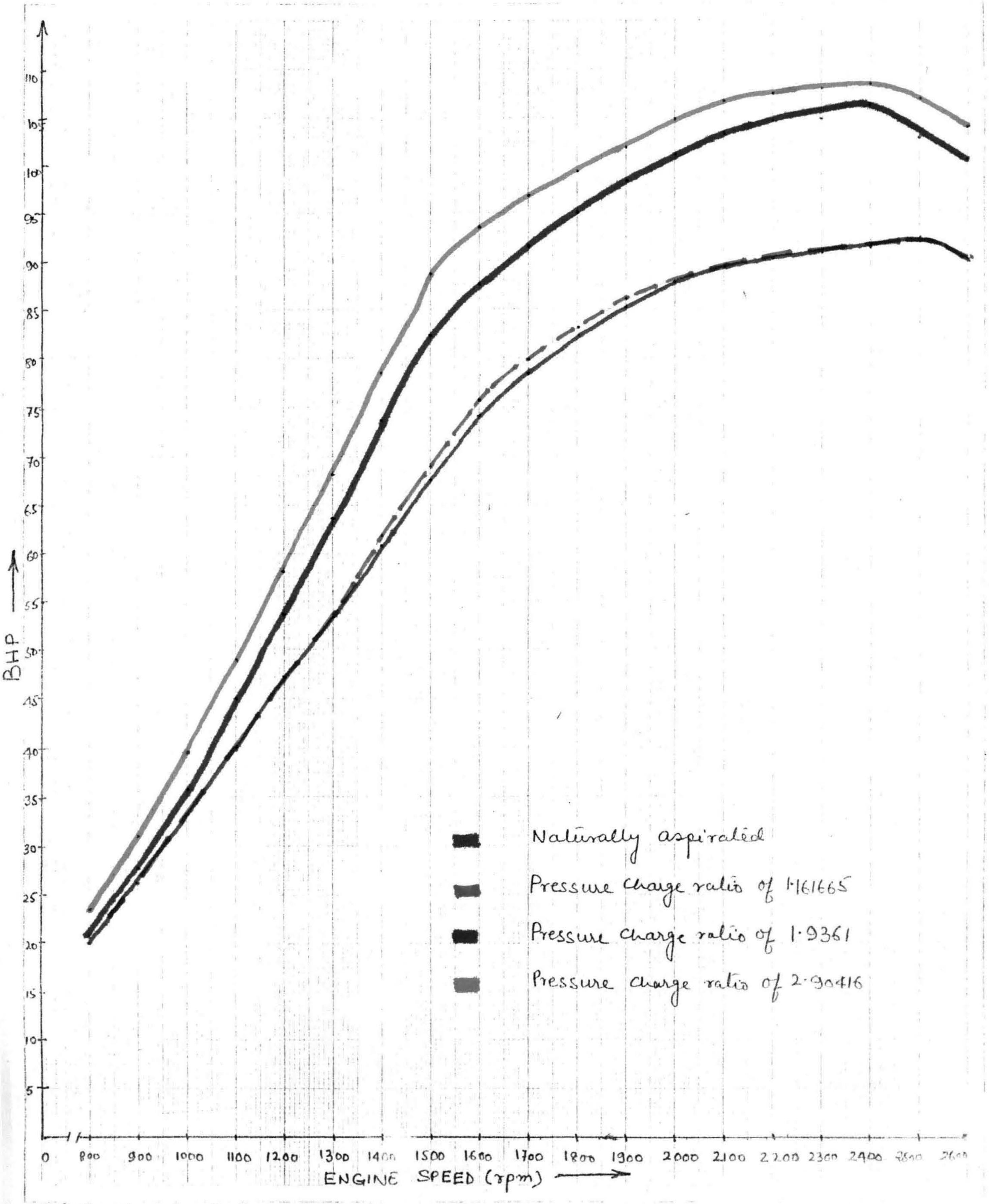

Fig 5.2 - Speed Vs BHP characteristics of a D0026M8A engine under varying air intake pressures.

as a result of which better combustion is achieved and as a result an increase in BMEP. Since the torque is related to BMEP, it also increases as the speed of the engine increases till the peak point of BMEP. After this peak point the BMEP starts falling down and so does the torque. This is attributed to the combustion process.

177. Due to the above mentioned facts, when the air intake pressure was increased above the atmospheric pressure, the torque also increased accordingly.

Power

178. The power developed by the engine depends on the rpm and torque developed by the engine. Initially, as the speed of the engine increases the torque also increase and thus the power also increases. After the maximum torque point is reached the drop in torque is compensated by the increase in engine rpm . Hence the power curve keeps on increasing till the maximum designed speed after which the rate of drop in torque overcomes the increase in engine rpm and there is a sharp drop in power developed.

179. Thus, when the air intake pressure is increased, which resulted in increased torque, the power developed by the engine also increased.

180. Figures 5.1 and 5.2 give a comparative picture of the torque and power developed by the D0026 M8A Diesel engine under investigation at the Engine Test House of 505 Army Base Workshop when tested under varying

air intake pressure. It can be seen that with the increase in the air intake pressure both torque and power developed by the engine showed a distinct improvement.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

181. The results of the test conducted on the DOO26 M8A Diesel engine confirmed that by increasing the density of charge air between the air intake and cylinder, the engine system response can be improved to a very large extent.

182. This improvement in the engine system response is due to the fact that as the density of charge air between the air intake and cylinder is increased, the weight of air induced per working stroke also increases which results in a greater weight of fuel to be burnt and consequently, an increase in the power output.

183. A distinct improvement in the engine characteristics were noticed with every increment of air intake pressure.

184. Though the D0026 M8A Diesel engine showed a remarkable 28% increase in torque and 17% increase in power output at a pressure charge ratio of 2.90416, the compression pressure shot up by more than 10% of the designed value . Thus it is not considered to safe to pressure charge the engine upto a pressure charge ratio of 2.90416.

185. The engine showed about 20% increase in Torque and about 15% increase in power developed under a pressure charge ratio of 1.9361. The

compression pressure developed in this case was 29 Kg /cm^2 which was also within the designed limit.

186. It is therefore recommended that the increase in air intake pressure of D0026 M8A Diesel engine should be upto, and not beyond, a pressure charge ratio of 1.9361.

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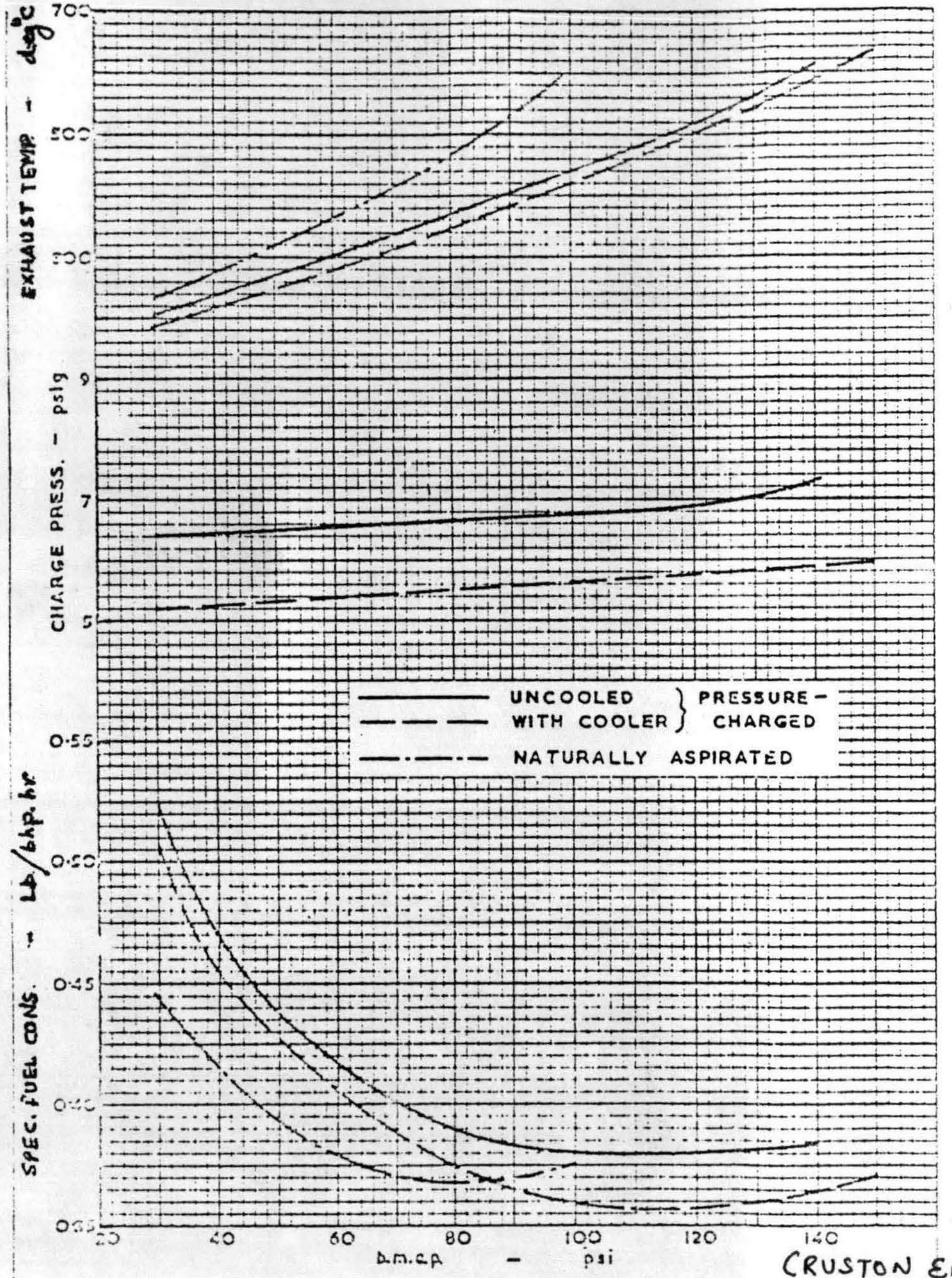
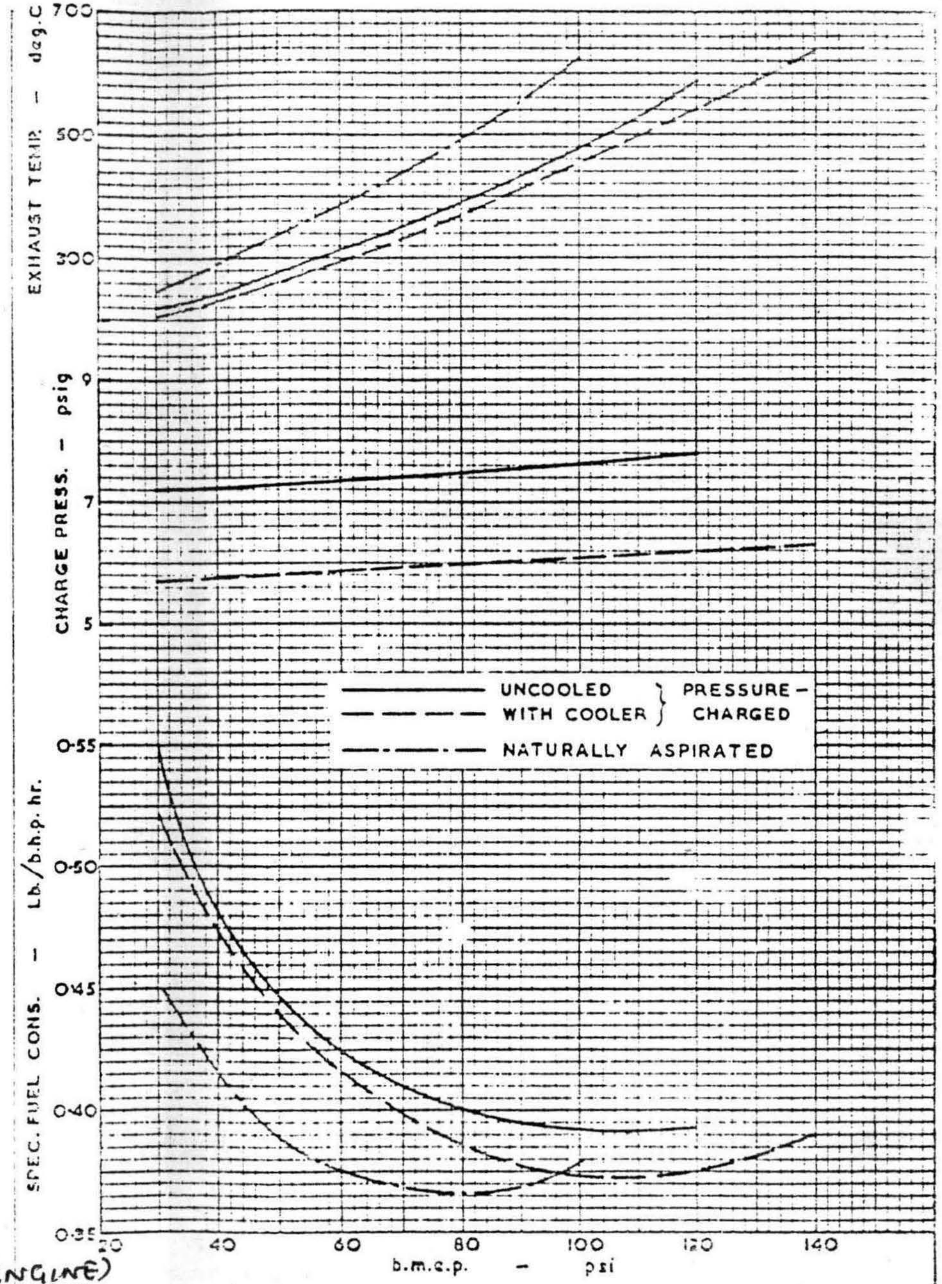
Appx 'A'
(Ref to para 78)



Comparative performance curves at 1100 r.p.m.

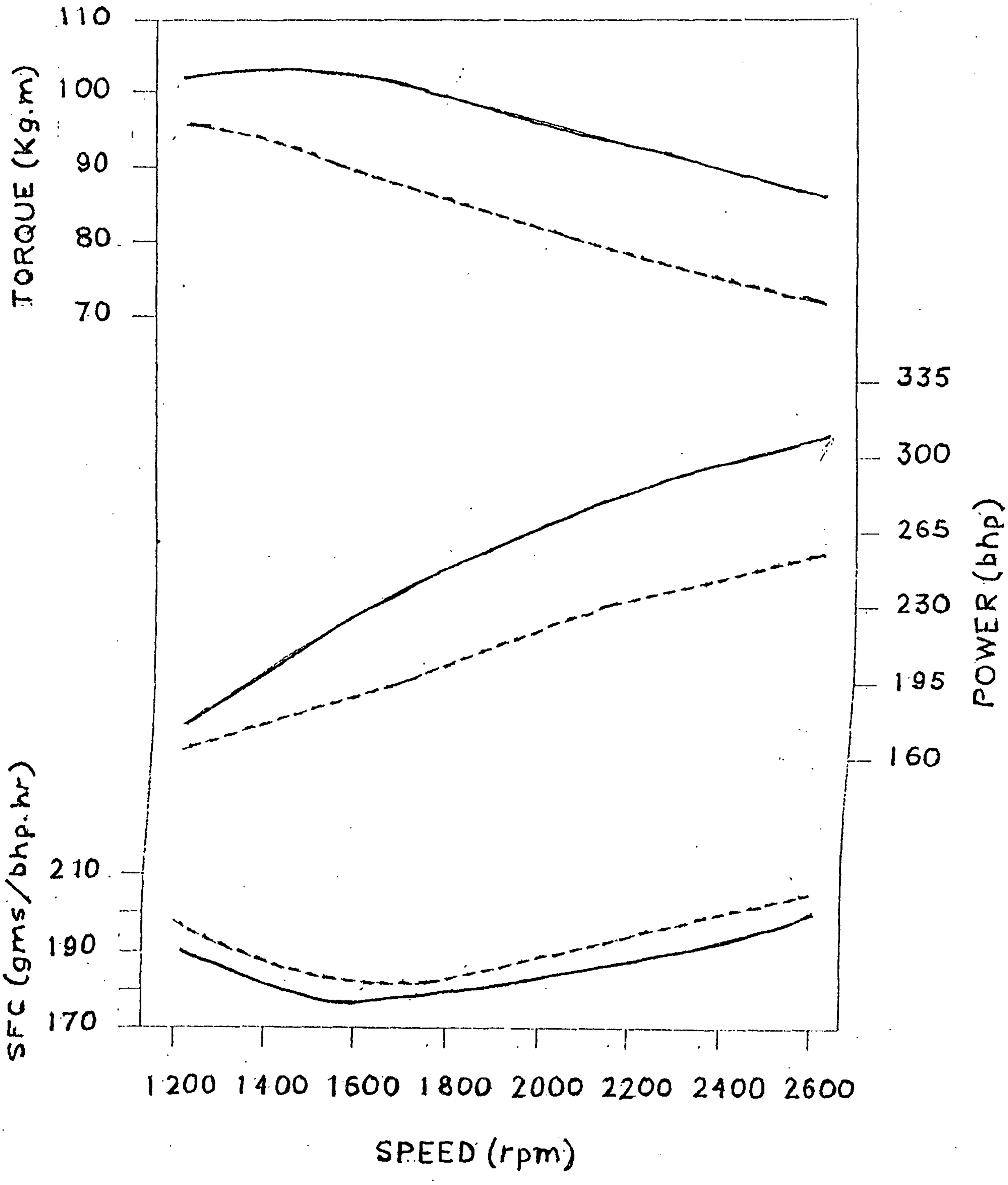


Comparative performance curves at 1300 r.p.m.

(RUSTON ENGINE)

APPX B
(Refer to para 78.)

PERFORMANCE CURVES



COMPARATIVE ENGINE PERFORMANCE (BMP - II ENGINE)

——— TURBOCHARGED ENGINE WITH EJECTOR & AIRCLEANER

- - - - N.A. ENGINE WITH EJECTOR & AIRCLEANER