T H E U N I V E R S I T Y O F M I C H I G A N

COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Technical Report

ENGINE PERFORMANCE WITH DIRECTLY DRIVEN SUPERCHARGERS

Part I: Performance of Compression Ignition Engines When Fitted With a Supercharger Driven Off the Engine Shaft

E. T. Vincent
N. Tokuda

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ABSTRACT

The object of this report is a comparison of engines fitted with various direct drive supercharges with the engine presently employing a turbo-charger.

The report covers the usual types of directly driven superchargers: Displacement type, Displacement with Compression, and Centrifugal machines.

Full load and part load performances are calculated for each of the types, as well as for a turbo-charged unit for a similar set of conditions.

The engine bulk and weight as well as the battlefield day fuel requirements are estimated for various cylinder arrangements for a 500 BHP engine at 3000 rpm in each case.

The degree of responsiveness, or more accurately the lack of it, was also examined for each case investigated.

OBJECT

The object of this analysis is to establish the expected engine performance when employing an engine driven supercharger and to compare this performance with that of the current Turbo-charged engine version at approximately the same pressure ratios.

MECHANICALLY DRIVEN SUPERCHARGERS

There are quite a number of different designs of compressors suitable for supplying compressed air to an engine with the object of superhcarging. Those suitable for attachment to a high speed compression ignition engine can be divided into three groups as follows:

1. Simple displacement blower without compression.

2. Displacement blower with compression.

3. Centrifugal compressor.

The characteristics of these types are considered to be as follows:

TYPE 1

In this group the most common is the Root's type compressor and its derivatives such as the helical impeller used by the General Motors Diesel Engine. It is characterized by simplicity of construction, lack of highly complicated machining operations, low internal friction, and simple reliable bearings, drive, etc. It is not too efficient but, due to low losses, is competitive when employed at low pressure ratios. The displacement feature is a distinct advantage for engine supercharging since the air delivered to the engine manifold by piston action remains almost constant per engine revolution, the result being an almost constant manifold pressure at a constant speed as the engine load varies.

In addition, the above feature also tends to maintain manifold pressure at varying speed. There will be some reduction at low speed associated with the additional time for leakage through the clearances as the revolutions fall. This reduction is relatively small so that this supercharger is capable of maintaining high torque at low speeds—a desirable feature in many applications.

While at the high speed end of the range, the efficiency tends to fall off due to poor filling with new air plus increasing losses.

The bulk of this machine tends to be somewhat great, due to the fact that mechanical and aerodynamic limitations place an upper limit on its speed of rotation, if long life and high efficiency are to be secured simultaneously. Speeds of the order of 6-7000 rpm are common; in special cases up to 10,000 rpm can be considered.

A typical plot of efficiency against pressure ratio is given in Fig. 1 from Ref. 2. The dotted efficiency line covering high pressure ratios is to be expected at such high ratios plus higher operating speeds for compactness.

TYPE 2

In this group can be included the Lysholm, Bicera, and Ricardo superchargers. The action is similar to that in Type 1 except that there is a compression phase following the completion of the inlet stroke, before delivery begins. This feature does not seem to be of much advantage at low pressure ratios, due to additional losses usually associated with the compression phase. If, however, ratios of compression of 1.5 and above are employed, then the compression machine definitely enters the picture.

As regards bulk, weight, etc., there is little to choose between the two types; similarly, the limiting revolutions remain in about the same magnitude. The advantage resides in the improved efficiency.

There are some additional complications in manufacture, resulting from the compression process, probably accompanied by increased costs, relative to the Root machine.

The torque-sustaining character of the Root supercharger is also well retained with the compression type. Figures 1(b) and 2 record typical performance characteristics, and Fig. 3 records those of a Bicera supercharger.[2,3]

TYPE 3

The centrifugal supercharger driven directly from the engine, or exhaust-turbine driven, has been in use for about 50 years, and has given excellent performance under certain limiting conditions.

This machine has the advantage of an equivalent compression process, low losses, small bulk and weight, and very high speed of rotation. Efficiencies of the order of 80-85% can be obtained today in small, well designed units.

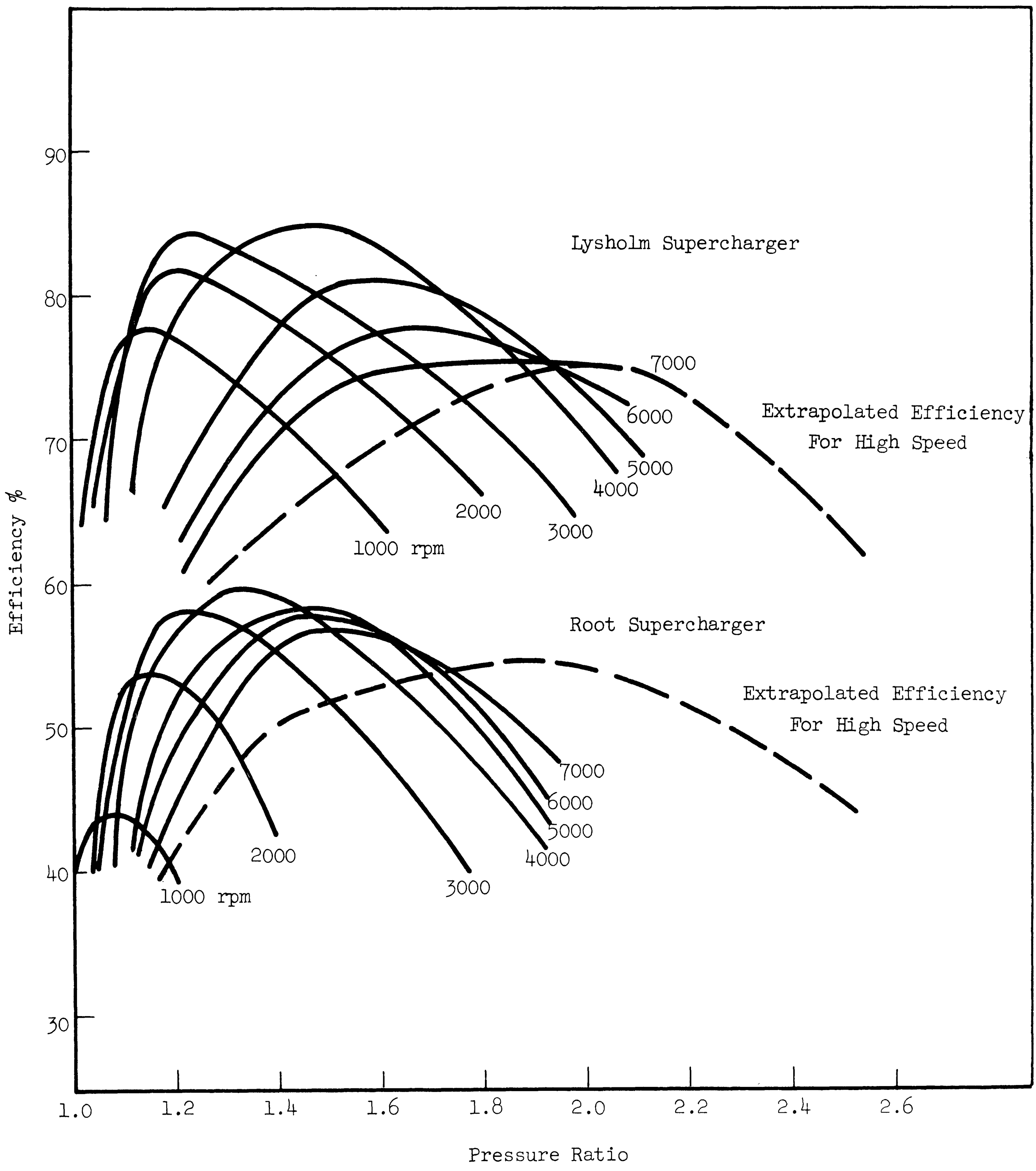

Fig. 1. Comparison of Lysholms and Roots supercharger.

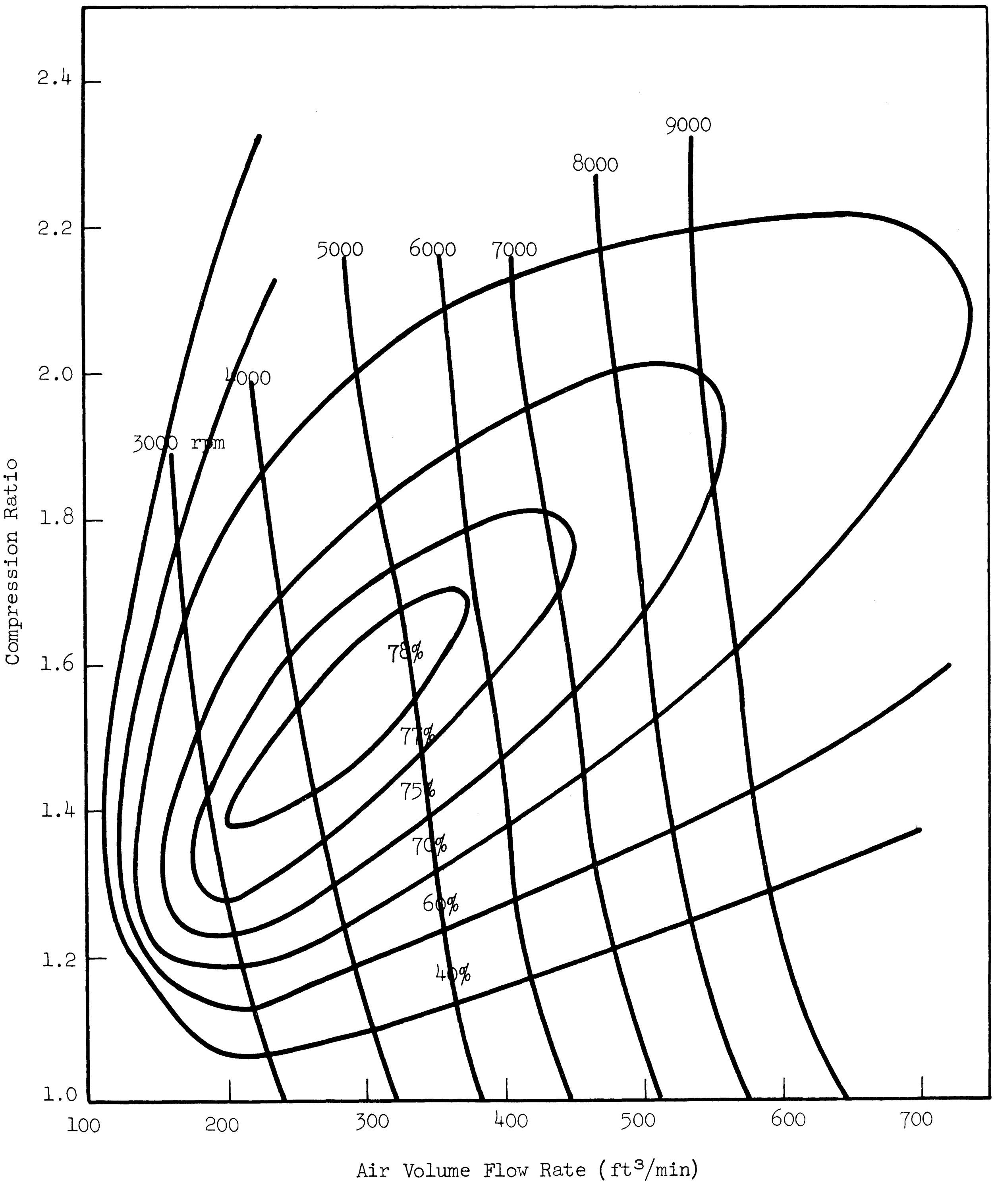

Fig. 2. Typical Lysholm performance map.

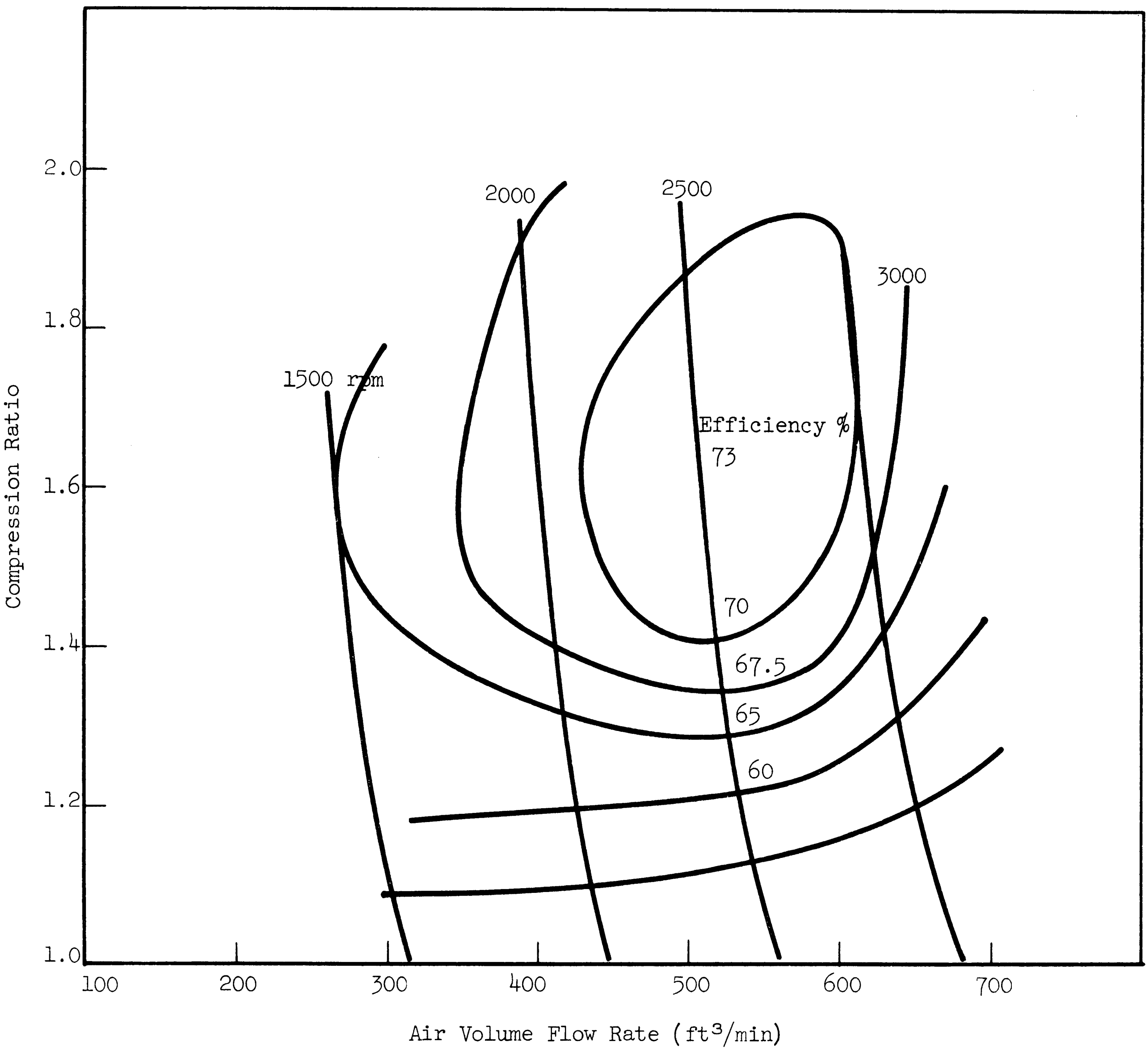

Fig. 3. Typical Bicera performance map.

The limiting conditions are mainly associated with the fact that the pressure capable of being supplied by the compressor is a function of the square of the speed of rotation. A more or less constant manifold pressure with varying engine speed is an impossibility when directly geared to the engine; thus the high torque maintainance of Types 1 and 2 is lost. The greatest use of this machine was as supercharger for aircraft engines where almost constant speed operation is common.

The centrifugal compressor can be employed for the purpose being investigated as a result of its efficiency and small bulk; and, if coupled to an engine and transmission which tend to employ a limited engine speed range, can give excellent results.

The main disadvantage is the high speed of rotation required (25-50,000 rpm), and the higher dependency upon the impeller diameter employed. This results in very high inertia of the impeller mass, and speed fluctuations of the engine, resulting from the firing frequency, must be isolated from the supercharger drive by means of slipping torque responsive clutches, fluid couplings, etc. A very high degree of accuracy must be maintained in the manufacture of the gear train. The transmission from the crankshaft to the centrifugal impeller, though relatively small and light, can be expensive.

A typical compressor map is given in Fig. 4 taken from Ref. 5; an engine operating line has been added. Some improvement in efficiency can be expected in a modern design. Details of these various types can be checked in Refs. 2-5.

COMPRESSOR CALCULATIONS

Compressors of Type 1 are best represented by the various designs of the Root blower. The design is characterized by veing a displacement machine, with much air being taken in at the inlet and displaced into the outlet; there is no compression of the air as it flows through the machine. The delivery pressure depends on the outlet restriction, in this case the engine; each charge delivered by the blower is compressed by back flow of air compressed by previous cycles, and not by change of volume of the trapped air. As a result the theoretical indicator diagram of the process is as shown in Fig. 5. by abcd, a simple rectangle, with air inlet at P_o, the ambient pressure, and delivery at P_m, the manifold pressure.

The work done in the process is given by Eq. (1):

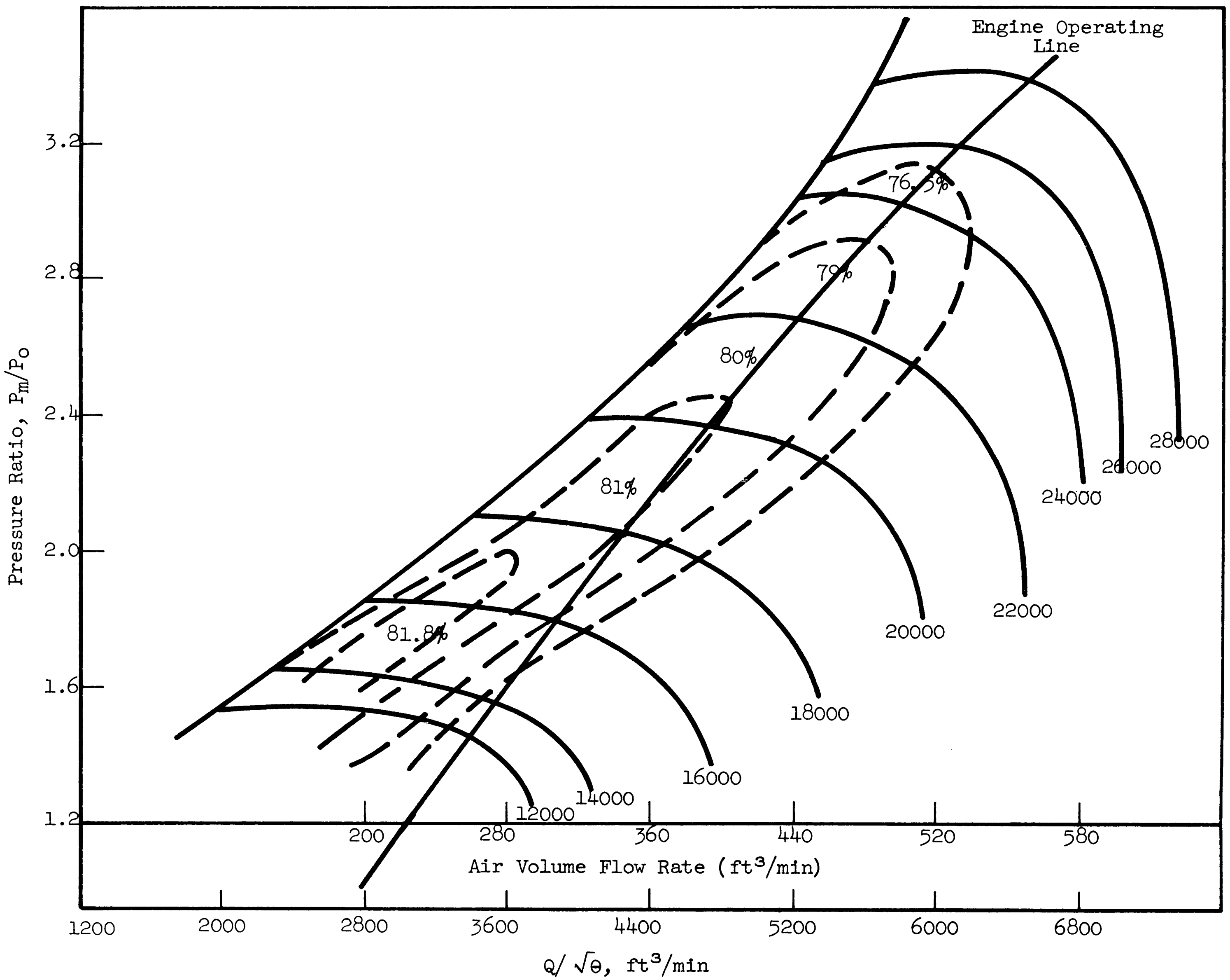

Fig. 4. Typical centrifugal performance map.

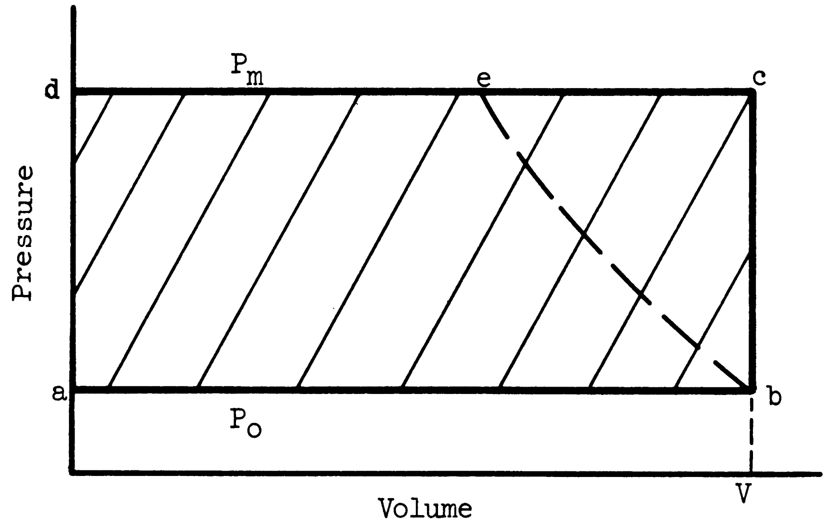

Fig. 5. Theoretical Root supercharger indicator diagram.

$$\text{Ideal Work of Compression} = 144\,(P_m - P_o)\,\frac{\mathcal{V}}{1728}$$

for suction volume $\mathcal{V}$

$$= \frac{1}{12}\,(P_m - P_o)\,\mathcal{V} \text{ ft lb}$$

where

P_m and P_o are in lb/sq in. abs

$\mathcal{V}$ is in cu in.

There will of course be some mechanical losses due to bearings, friction, etc., of the mechanical parts as well as some due to air friction of passages, turbulence, etc. The total work of the machine can be defined by the sum total of these losses as follows:

$$\text{Input Work} = \frac{(P_m - P_o)\,\mathcal{V}}{12 x \eta} \text{ ft lb} \qquad (2)$$

where

η = ratio of the ideal work to that actually required.

In order to relate the input work to some ideal process which involves no losses, and thus represents the 100% efficiency work, Eq. (2) can be related to the isentropic work of compression from P_o to P_m, a process involving no losses of any kind-friction, heat, etc. The isentropic efficiency is defined as

$$\text{Isentropic Efficiency of Compression } \eta_c = \frac{\text{Isentropic Work}}{\text{Input Work}}$$

The isentropic work is represented in Fig. 5 by the area abed in place of the area abcd of the compressor, the curve be being represented by PV^k = constant. This isentropic work is given by

$$\text{Work of Isentropic Compression} = wC_p(T_{1'}-T_o) \text{ Btu's}$$

where

$T_{1'}$ = temperature of isentropic compression from P_o to P_m

$= T_o\ (P_m/P_o)^{k-1/k}$

T_o = ambient temperature

w = air flow rate lb/engine cycle

$= \eta_v \dfrac{P_m(V_1-V_o)}{12RT_m}$

V_1-V_o = engine displacement in cu in.

T_m = manifold temperature for pressure P_m

η_v = volumetric efficiency of cylinder.

Employing an isentropic efficiency of compression η_c covering the cycle losses, plus mechanical efficiencies η_m for frictional losses,

$$\text{Work Input of Compressor} = \frac{wC_p(T_{1'}-T_o)}{\eta_c\eta_m} \text{ Btu's}$$

which becomes

$$\text{Input Work} = \frac{\eta_v 778\ P_m(V_1-V_o)C_pT_o(r^{k-1/k}-1)}{12RT_m\eta_c\eta_m} \text{ ft lb} \qquad (3)$$

where

P_m/P_o = r = pressure ratio of charger

P_m = manifold pressure, psi

T_o = ambient air temperature, °F

R, C_p, k = gas constants.

It is to be understood that the isentropic efficiency η_c is to be based upon all aerodynamic losses of the blower; suction will not be exactly at P_o, delivery will not be exactly P_m, etc. Thus η_c is the efficiency based on a pressure difference of P_m-P_o between a practical compressor and an isentropic compressor for the same pressure difference.

$$\left.\begin{array}{l}\text{Equivalent Engine Cylinder Mean Effective}\\ \text{Pressure of Blower}\end{array}\right\} = \frac{\text{Work}}{dV}$$

$$= \frac{\eta_v 778\ P_m C_p T_o (r^{k-1/k}-1)}{R T_m \eta_c \eta_m} \text{ psi.} \tag{4}$$

The mean effective pressure represented by Eq. (4) is based upon the charge of air trapped in the engine cylinder and makes no allowance for the air flow through the combustion chamber during the valve overlap period. This scavenge air usually amounts to about 5% of the air charge with normal valve overlap and can increase beyond this where large overlap is employed.

Now the average volumetric efficiency of a highly supercharged engine can be as high as 95% or better; as a first approximation it will be assumed that the 5% excess air required will balance the 5% loss due to volumetric efficiency. It can now be stated that

$$\left.\begin{array}{l}\text{Equivalent Mean Effective}\\ \text{Pressure of Charger}\end{array}\right\} = \frac{778\ P_m C_p T_o (r^{k-1/k}-1)}{R T_m \eta_c \eta_m} \tag{5}$$

Similarly, the input work of Eq. (3) can be modified by dropping the term η_v to include the excess air flow of scavenge air when necessary.

Equations (3)-(5) are in terms of ambient air, manifold pressure, and pressure ratio of charger; the equivalent effective pressure of the charger has been referred to piston displacement (V_1-V_o) and thus can be subtracted directly from the gross IMEP of the indicator diagram shown in Fig. 6.

OVERALL CYCLE

The overall engine cylinder cycle of a supercharged engine with a mechanical driven Root's type charger is represented by an indicator diagram of the type shown in Fig. 6 where the area 1278 represents the work done during the suction stroke (1-2) and exhaust stroke (7-8) of the piston; 234562 represents the work on the piston by the gas trapped in the cylinder during compression, combustion, and expansion.

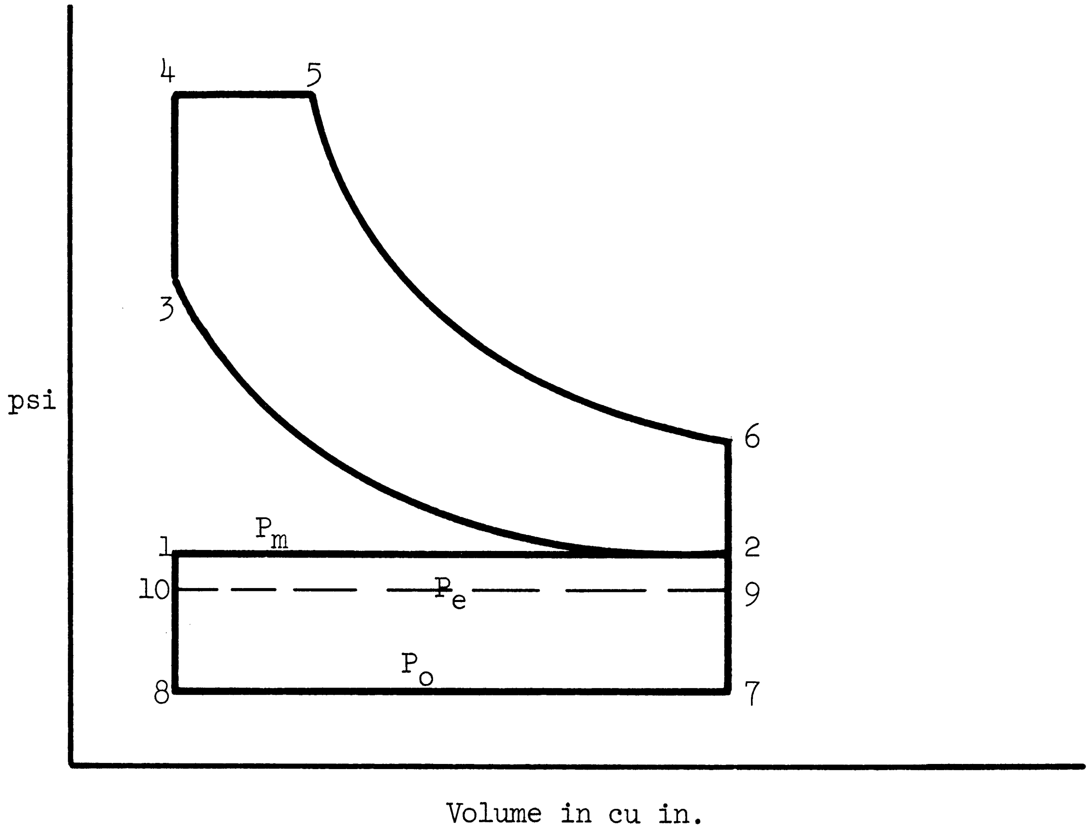

Fig. 6. Ideal engine indicator diagram.

$$\text{Gross Total Work on Piston} = (\text{Area } 23456 + \text{Area } 1278) \text{ ft lb} \qquad (6)$$

$$\underset{(\text{at } \eta_v = 100\%)}{\text{Work of Compressor}} = \frac{778\ P_m(V_1 - V_o)\,C_p T_o(r^{k-1/k} - 1)}{12RT_m \eta_c \eta_m} \text{ ft lb} \qquad (3)$$

Net effective indicated output of engine = (6) - (3).

In terms of IMEP this is given by:

$$\text{Gross IMEP of Cycle} = \frac{12(\text{gross indicated work in ft lb})}{\text{displacement } (V_1 - V_o) \text{ cu in.}}$$

$$\text{Gross IMEP of Blower} = \frac{778\ P_m C_p T_o (r^{k-1/k} - 1)}{R T_m \eta_c \eta_m}$$

$$(\text{IMEP}_{net}) \text{ of Engine Plus Blower} = \frac{12(23456+1278) \text{ in ft lb}}{V_1 - V_o} - \frac{778\ P_m C_p T_o (r^{k-1/k} - 1)}{R T_m \eta_c \eta_m} \text{ psi} \qquad (7)$$

$$\text{Net Effective IHP of Engine} = \frac{\text{IMEP}_{net} (V_1 - V_o) N}{2 \text{x} 12 \text{x} 33000} \text{ hp} \qquad (8)$$

$$\text{Net IHP/lb of Air/Sec} = \frac{\text{net IHPx2x60}}{(\text{lb of air/cycle}) N} \text{ (for 4 cycle)}$$

$$= \frac{R T_m \text{x} \text{IMEP}_{net}}{550\ P_m}$$

$$= \frac{4.36(\text{IMEP}_{net})}{r} \qquad (9)$$

where

N = rpm of engine

T_m = 200°F (intercooled temperature)

P_o = 14.7 psi (ambient pressure).

If this IHP/lb of air/sec is obtained with a fuel-to-air ratio of F/A then

$$\text{Fuel Flow/Hr} = \frac{F}{A} \text{ x } 3600 \text{ lb/hr} .$$

Thus

$$\text{Specific Fuel Consumption} = \frac{3600 \times \frac{F}{A}}{\text{IHP/lb of air/sec}} . \qquad (10)$$

In this analysis the displacement charger has been assumed; however, the same expressions apply to all superchargers if the actual isentropic efficiency is employed for each type, since the isentropic work is the same for all machines operating between pressures P_o and P_m.

It is seen from the above that the answer to the problem of the performance of a blower charged engine depends upon a knowledge of the area 23456, since the rest of the values can be easily calculated. All other superchargers are handled in the same way since the area 23456 remains constant irrespective of the type of charger employed (with intercooling to T_m = constant). Adjustment need be made for the manner in which the area 1278 may charge and for the compressor work only.

In The University of Michigan Report No. 04612-3-F, Contract No. DA-20-018-ORD-23664,[1] are given methods for obtaining the output of turbo-charged engines. If the cycle in such cases is compared with that of Fig. 6 it is seen to consist of the areas 23456 and 12910 determined by an engine back pressure of P_e which is the exhaust manifold or turbo inlet pressure. In the above report the relation

$$P_e = 0.85 \, P_m \qquad (11)$$

was maintained at all times; it follows that the gross work of the cycle presently under consideration is given by

$$\begin{aligned} \text{Ideal Gross IMEP} &= \text{IMEP of Ref. 1 + IMEP Equivalent} \\ &\quad \text{of Area (87910)} \\ &= \text{IMEP of Ref. 1} + (P_e - P_o) \\ &= \text{IMEP of Ref. 1} + (0.85 \, P_m - P_o) \\ &= \text{IMEP of Ref. 1} + 0.85 \, P_o(r - 1.18) \qquad (12) \end{aligned}$$

The full effect of the change of pressure from P_o to P_e can be employed here, since an allowance for the absence of square corners, wavey lines, etc., has already been made in Ref. 1.

It follows that the net output of a supercharged compression ignition engine in terms of Ref. 1 is given by

$$\text{Net IMEP} = \text{IMEP (Ref. 1)} + 0.85\ P_o(r-1.18) - \frac{778\ P_m C_p T_o(r^{k-1/k}-1)}{RT_m \eta_c \eta_m} \qquad (13)$$

If this value of the net IMEP is used in evaluating Eqs. (8), (9), and (10), the required performance figures can be calculated. The methods of Ref. 1 can be applied to various F/A ratios and the corrections as given above can be applied for part load operation also.

TYPICAL SAMPLE CALCULATION

We shall determine the output of a compression ignition engine when fitted with a Root's blower giving a pressure ratio of 2.4:1, the fuel-air ratio being 0.0473. An aftercooler maintaining an inlet manifold temperature of 200°F is employed.

Under the above conditions a turbo-charged engine would give an ideal IMEP of 285 psi for a SFC of about 0.27 lb/IHP/hr. See Fig. 2 of Ref. 1. $r = 2.4$, $P_m = 35.3$ psi, $P_o = 14.7$, and $T_o = 520$°F.

Expected Gross IMEP With Root's Blower Eq. (12)

$$= 285 + 0.85\ P_o(r-1.18)$$

$$= 285 + 0.85 \times 14.7(2.4-1.18)$$

$$= 300.0 \text{ psi approx.}$$

$$\text{Net IMEP of Engine (Eq. (11))} = 300.0 - \frac{778\ P_m C_p T_o(r^{k-1/k}-1)}{RT_m \eta_c \eta_m}$$

$$= 300.0 - 48.2$$

$$= 251.8 \text{ psi.}$$

This calculation assumes that $\eta_m = 0.97$ and that $\eta_c = 0.60$, an optimistic value for a charger of this type. Then if the rpm = 3000,

$$\text{Engine Suction Volume } (V_1 - V_o) \text{ Per lb of Air} = \frac{wRT_m}{\eta_v P_m}$$

$$= \frac{RT_m}{\eta_v P_m} \text{ /lb of air}$$

$$= \frac{53.34 \text{x} 660}{0.95 \text{x} 35.3 \text{x} 144}$$

$$= 7.29 \text{ cu ft/lb of air}$$

$$= 12600 \text{ cu in./lb.}$$

$$\text{Displacement/Sec/Lb of Air} = \frac{V_1 - V_o}{1728} \text{ x } \frac{N}{2\text{x}60}$$

$$= 12600 \text{ cu in./lb/sec.}$$

If there were "n" cylinders of the four-cycle type having a diameter of D" and stroke L"

$$V_1 - V_o = \frac{\pi D^2}{4} \text{ x L x n cu in./rev}$$

and suction volume is

$$\frac{\pi D^2}{4} \text{ x L x n x } \frac{N}{2\text{x}60} = 12600 \text{ cu in./lb/sec}$$

$$\text{Net IHP} = \frac{251.8 \text{x} 144}{550} \text{ x } \frac{12600}{1728}$$

$$= 481.0 \text{ IHP/lb of air/sec}$$

$$\text{Specific Fuel Consumption} = \frac{3600 \text{ x } \frac{F}{A}}{\text{IHP/lb air/sec}}$$

$$= \frac{3600 \text{x} 0.0473}{481.0}$$

$$= 0\ 354 \text{ lb/IHP/hr.}$$

By repetition of this calculation over any range of pressure ratios, efficiencies, etc., a plot of the engine performance is possible for the assumed conditions.

DISPLACEMENT SUPERCHARGER

The results of such calculations are shown in Fig. 7 and Table I for a Root's type charger; in all cases the results presented are calculated for the maximum that can be expected from the engine on the assumption that 90% of the expected performance at a F/A ratio of 0.0473 can be achieved. The same assumption is true for the other results given in this report, and thus the curves are relatively comparable.

It should be noted that an air-cooled engine is assumed and that the net BHP and SFC with cooling fan is given; this fan will be capable of cooling the engine plus normal engine and transmission oil. This should be allowed for if comparison is being made with water cooling, for example. See Figs. 9 and 10 for the fan performance.

DISPLACEMENT CHARGER WITH COMPRESSION

A displacement blower with compression is represented by the Lysholm or Bicera superchargers.[2,3] In each of these types there is compression of the charge along such a curve as be of Fig. 5. These machines have a certain built-in compression ratio that is fixed in the case of the former but is variable for the latter type. It follows that, other things being equal, the Bicera should have the higher efficiency over a variable load and speed curve; therefore it was chosen for the engine estimates. In addition, it is the easier type to calculate since the delivery pressure will correspond to any ratio desired within limits. The following calculations employ Eq. (5) for the equivalent mean effective pressure of the charger—also employing, of course, the appropriate values for the various efficiencies applicable to the type of compressor being investigated.

Examination of Ref. 3 has resulted in consideration of the diagram of Fig. 8 as the closest representative of what might be obtained for the performance of a Bicera compressor. Figure 8 has been estimated for a built-in compression ratio of 2.0:1, which should give an actual delivery ratio up to about the desired maximum of 2.6:1, which is employed in these calculations. The values used are thus hypothetical to some extent but do seem to be possible of achievement. It is believed that the results presented will also be applicable to a machine of the Lysholm type with but slight variations; thus only the one set of data are presented for the displacement supercharger with compression.

TABLE I

ENGINE PERFORMANCE WITH DIRECTLY DRIVEN ROOT'S SUPERCHARGER

(Full Load)

Manifold Pressure, P_m	atm	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6
Basic imep Ref. 1 x 0.90	psi	117	141	162	189	211	234	259	279	306
0.85 P_o (r-1.18)	psi	0	2.4	4.5	5.3	7.8	10.3	12.8	15.3	17.8
Isentropic work of compressor	Btu/lb	0	7.2	13.4	18.8	22.8	28.8	33.4	37.3	41.1
Efficiency η_c x η_m		-	0.55	0.56	0.56	0.56	0.55	0.50	0.45	0.40
T_m	°R	545	600	660	660	660	660	660	660	660
Compressor work	psi	0	6.0	11.5	20.0	25.5	36.5	51.0	70.0	93.0
Net imep	psi	117	137.4	155.0	174.3	193.3	207.8	219.8	224.3	230.8
Ihp	lb-air sec	511	500	484	476	470	453	437	407	388
Isfc	lb/ihp/hr	0.334	0.341	0.352	0.358	0.363	0.378	0.391	0.420	0.440
Fmep	psi	22.5	23.3	24.0	25.0	26.0	26.7	27.5	28.5	30.0
Bmep	psi	94.5	114.1	1.0	149.3	167.3	181.1	192.3	195.8	200.8
Bhp	lb-air sec	414.0	416.0	409.0	408.0	407.0	396.0	383.0	355.0	338.0
Bsfc	lb/bph/hr	0.412	0.410	0.417	0.418	0.419	0.431	0.445	0.480	0.504
Net bhp with cooling fan	lb-air sec	322.0	338.0	344.0	350.0	356.0	349.0	341.0	317.0	303.0
Net sfc with cooling fan	lb/bhp/hr	0.529	0.504	0.495	0.487	0.478	0.488	0.500	0.538	0.562

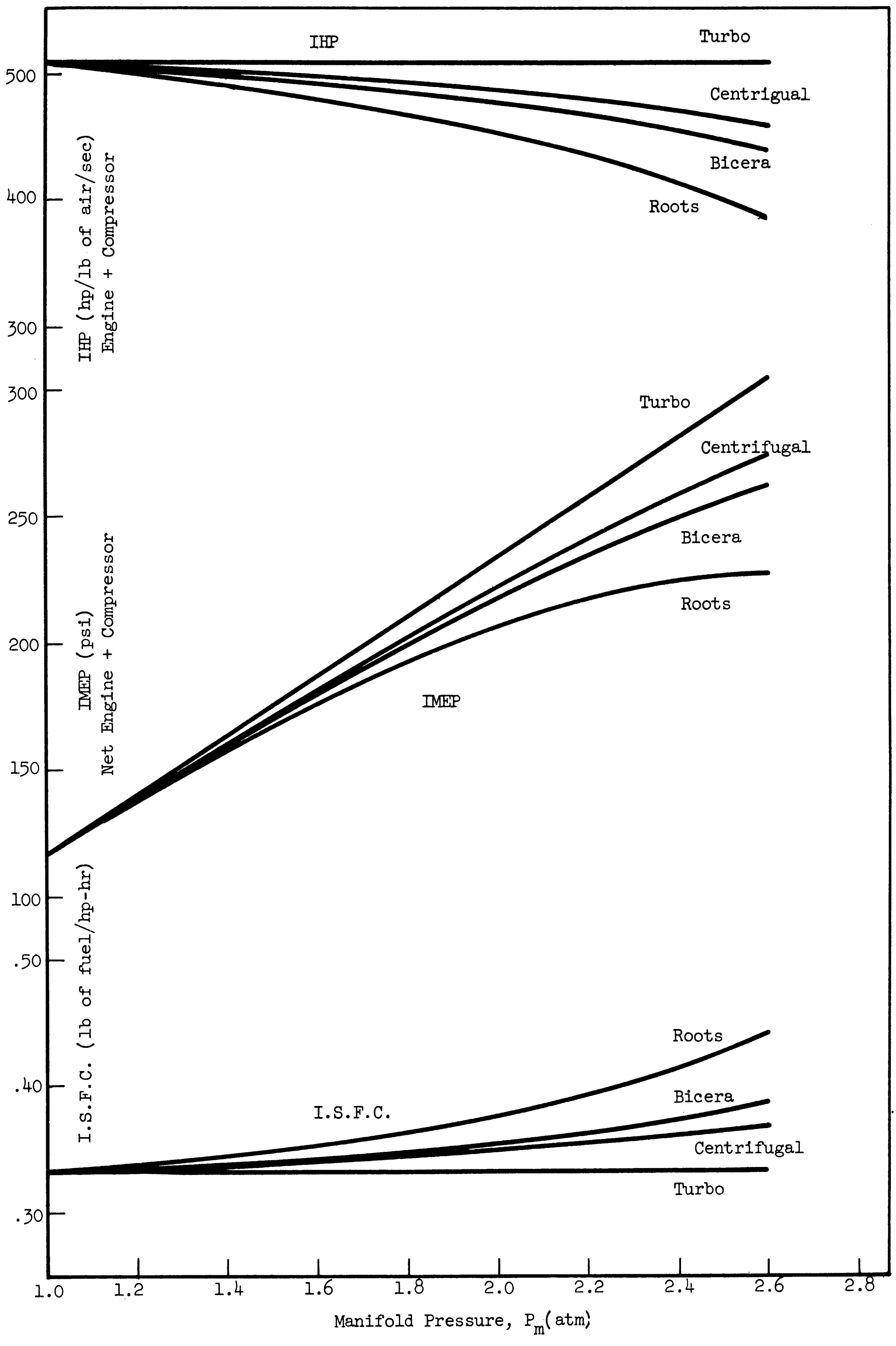

Fig. 7. Full load performance curves with various types of compressors.

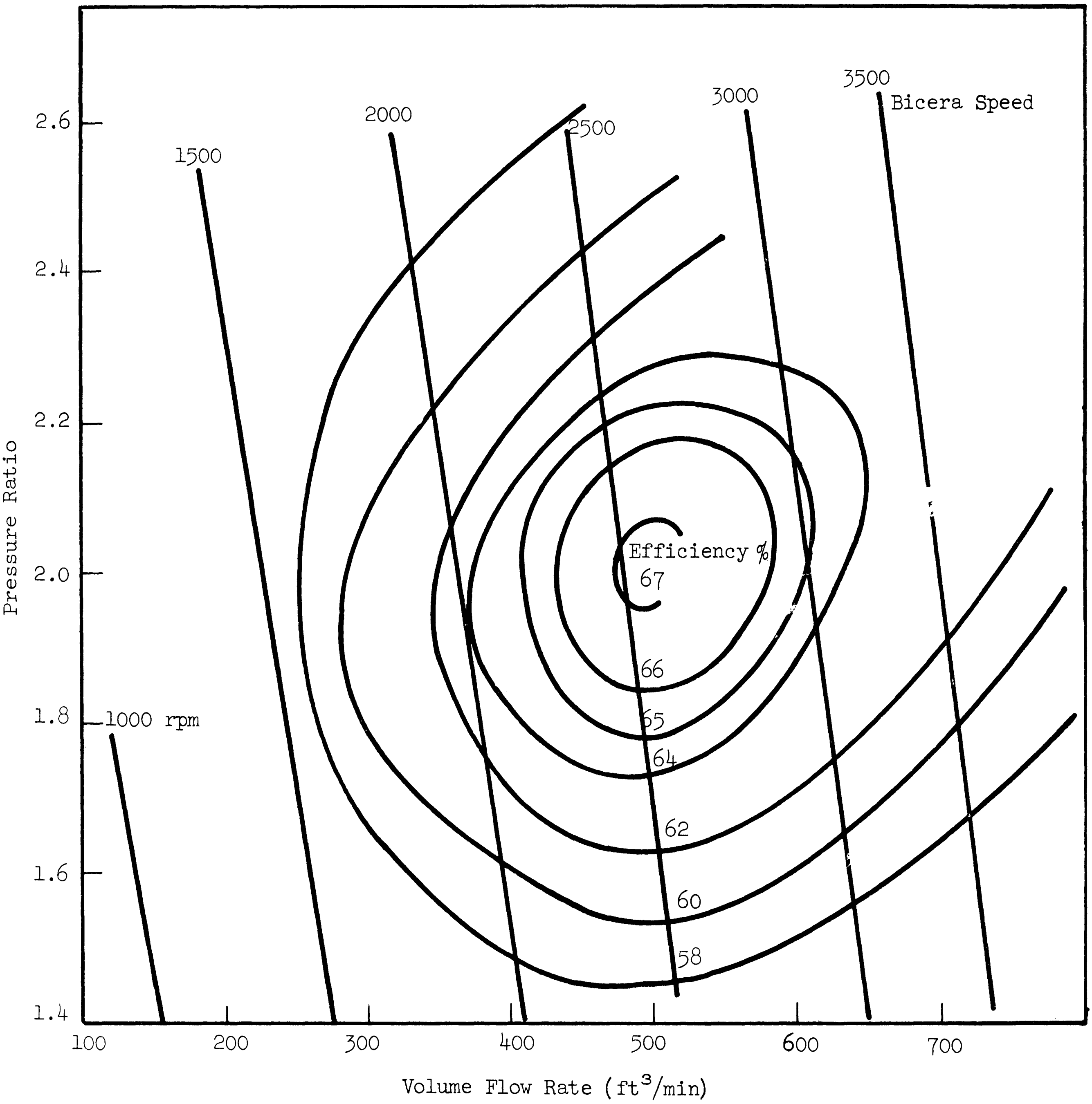

Fig. 8. High-ratio Bicera compressor.

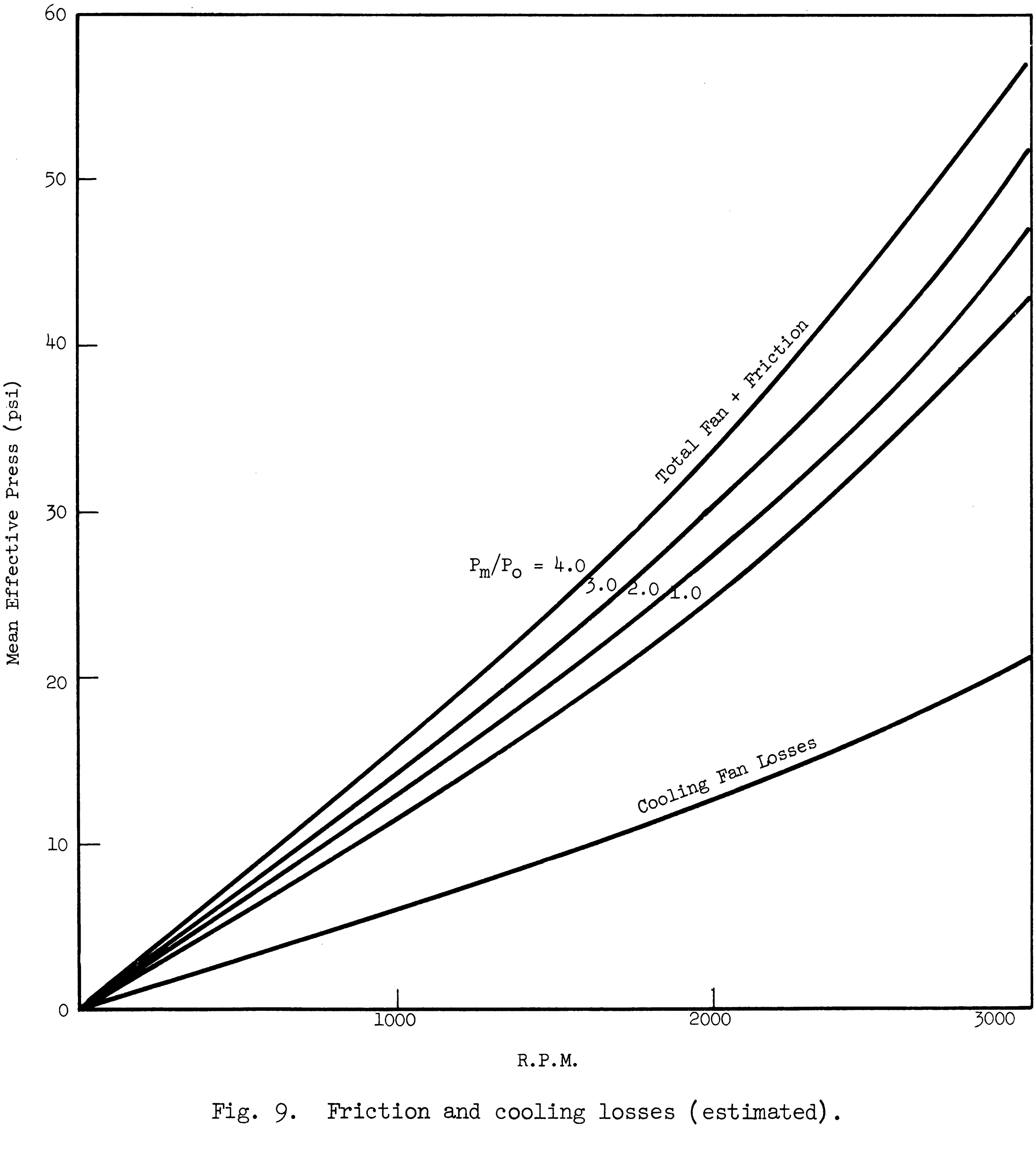

Fig. 9. Friction and cooling losses (estimated).

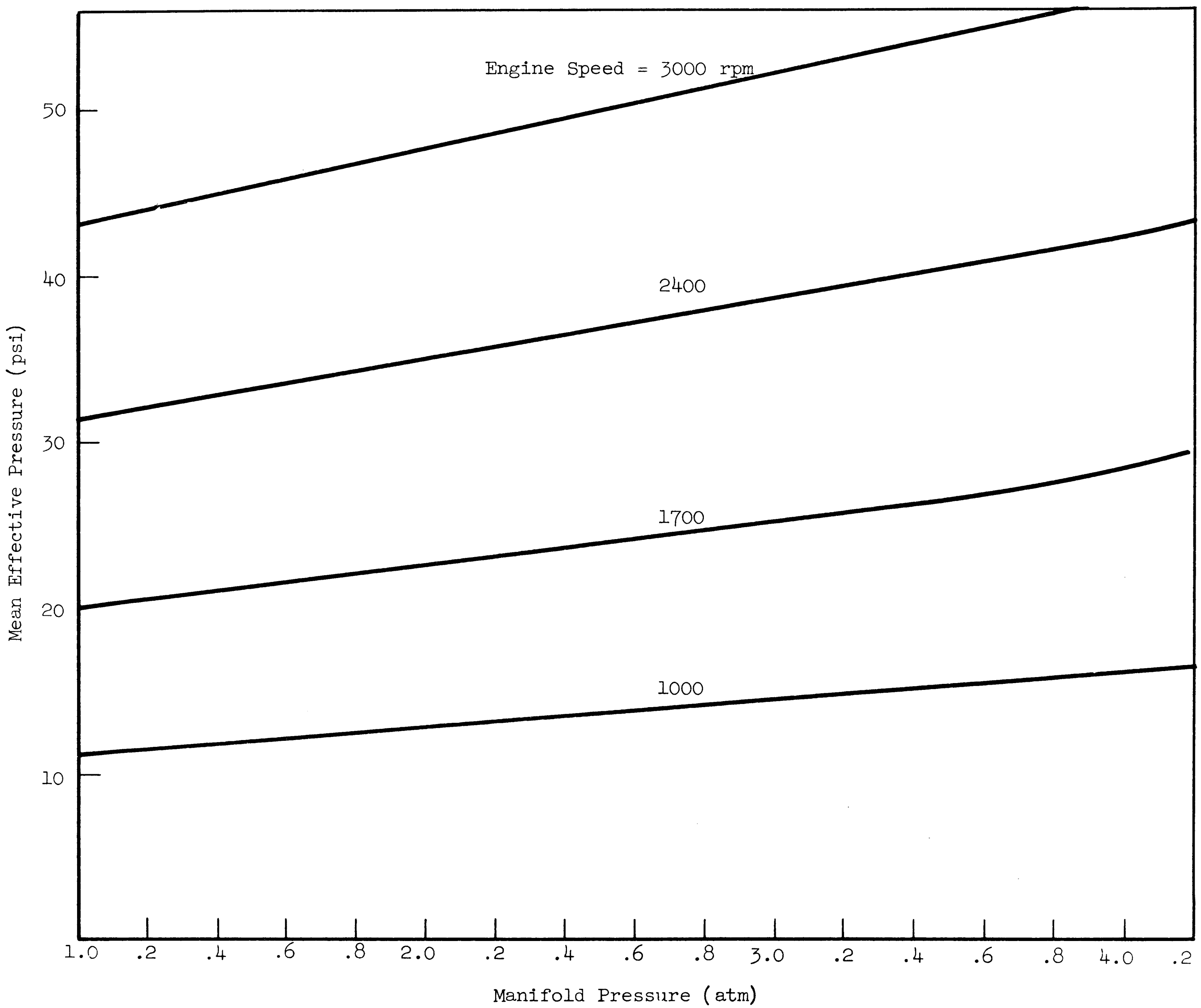

Fig. 10. Friction and cooling losses (total).

The results obtained by the methods already outlined are given in Table II and Fig. 7. These data are obtained on the BHP basis from the estimated friction and fan power curves of Figs. 9 and 10, as was done in the case of Table I.

ENGINE DRIVEN CENTRIFUGAL SUPERCHARGER

The third type of engine driven supercharger to be investigated is the conventional single stage centrifugal machine. The advantage of this is of course its great air handling ability, resulting in low size and weight. The principal of this supercharger is the employment of changes of kinetic energy to produce the pressure rise, but in the final analysis the process still involves the same equations used previously for the work required, viz. Eq. (3). The performance of an engine fitted with this type of machine can be calculated exactly as the two previous ones, provided the correct efficiencies for this type are employed.

It follows that the results calculated for the centrifugal machine can also represent the result to be achieved by any improved efficiency compression blower, irrespective of type, for the same pressure ratios.

The centrifugal supercharger has been operated at speeds up to 100,000 rpm in some instances, mainly as turbo-compressors. In order to operate at such speeds when driven from a compression ignition engine of conventional design a high ratio gear train is required plus, in general, some form of flexible or slipping drive to eliminate as far as possible the cyclic speed variations which result from the engine firing sequence and in turn from being transmitted to these highly loaded, high speed gears. These problems have been solved for such engine applications; they are only mentioned here to indicate that the centrifugal compressor application may involve a higher class of gearing and transmission apparatus than the previous type of machines.

Assuming the same conditions regarding pressure and temperature as were assumed previously, and employing a centrifugal compressor of quite modern design—aimed at exploiting the higher isentropic efficiencies proved possible in small gas turbine applications—it is estimated that a compressor map similar to that shown in Fig. 4 could be achieved in a small machine with the efficiencies given in Table III. The appropriate curve of Fig. 7 records the expected performance of an engine so equipped.

The gas volume flow at the engine cylinder inlet for a given constant engine rpm will vary directly as the manifold pressure P_m, provided after-cooling maintains a constant T_m. It follows that the engine performance line will be approximately straight as shown in Fig. 4. Employing such a compressor map for a modern design of centrifugal compressor with an efficiency schedule as shown in Table III, the performance data calculated are shown in

TABLE II

ENGINE PERFORMANCE WITH DIRECTLY DRIVEN DISPLACEMENT COMPRESSOR WITH COMPRESSION

(Bicera)

Manifold pressure, P_m	atm	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6
Basic imep Ref. 1 x 0.80	psi	117	141	162	189	211	234	259	279	306
0.85 P_o (r-1.18)	psi	0	2.4	4.5	5.3	7.8	10.3	12.8	15.3	17.8
Isentropic work of compressor	Btu/lb	0	7.2	13.4	18.8	22.8	28.8	33.4	37.3	41.1
Efficiency $\eta_c \times \eta_m$		-	0.55	0.65	0.70	0.73	0.74	0.70	0.68	0.62
T_m	°R	545	600	630	657	660	660	660	660	660
Compressor work	psi	0	5.9	9.3	14.9	19.4	26.9	36.2	45.5	60.0
Net imep	psi	117.0	137.5	157.2	179.4	199.4	217.4	235.6	248.8	263.8
Ihp	lb-air sec	511	500	490	488	482	474	466	452	442
Isfc	lb/ihp/hr	0.333	0.340	0.348	0.349	0.353	0.360	0.365	0.377	0.385
Fmep	psi	22.5	23.3	24.0	25.0	26.0	26.7	27.5	28.5	30.0
Bmep	psi	94.5	114.2	133.2	154.4	173.4	190.7	208.1	220.3	233.8
Bhp	lb-air sec	414.0	415.0	416.0	420.0	420.0	415.0	413.0	400.0	392.0
Bsfc	lb/bhp/hr	0.412	0.411	0.41	0.406	0.406	0.411	0.413	0.426	0.435
Net bhp with cooling fan	lb-air sec	322.0	338.0	350.0	363.0	368.0	370.0	371.0	362.0	356.0
Net sfc with cooling fan	lb/bhp/hr	0.529	0.504	0.487	0.470	0.463	0.461	0.460	0.470	0.478

TABLE III

PERFORMANCE WITH DIRECTLY DRIVEN CENTRIFUGAL SUPERCHARGER

Manifold pressure, P_m	atm	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6
Basic imep Ref. 1 x 0.90	psi	117	141	162	189	211	234	259	279	306
0.85 P_o (r-1.18)	psi	0	2.4	4.5	5.3	7.8	10.3	12.8	15.3	17.8
Isentropic work of compressor	Btu/lb	0	7.2	13.4	18.8	22.8	28.8	33.4	37.3	41.1
Efficiency $\eta_c \times \eta_m$		o	0.78	0.80	0.82	0.82	0.81	0.81	0.80	0.79
T_m	°R	545	584	607	641	660	660	660	660	660
Compressor work	psi	-	3.8	8.2	12.8	17.5	23.7	31.7	39.2	47.0
Imep	psi	117	139.6	158.3	181.5	201.3	220.6	240.1	255.1	276.8
Ihp	lb-air sec	511	507	494	495	488	480	479	464	463
Isfc	lb/ihp/hr	0.334	0.336	0.346	0.344	0.349	0.355	0.356	0.367	0.366
Fmep	psi	22.5	23.3	24.0	25.0	26.0	26.7	27.5	28.5	30.0
Bmep	psi	94.5	116.3	134.3	156.5	175.3	193.9	212.6	226.6	246.8
Bhp gross	lb-air sec	414.0	423.0	418.0	427.0	425.0	423.0	422.0	412.0	413.0
Bsfc	lb/bhp/hr	0.412	0.402	0.417	0.399	0.40	0.403	0.404	0.414	0.411
Net bhp with cooling fan	lb-air sec	322	346	354	370	374	376	382	374	378
Net sfc with cooling fan	lb/bhp/hr	0.529	0.492	0.481	0.46	0.455	0.453	0.446	0.455	0.45

that table and Fig. 7. As mentioned previously, the data at the low MEP is somewhat adversely affected by the use of the same direct drive cooling fan for both the high and low ratings; a slip drive would improve the results.

It should be understood that the compressor map proposed is associated with an existing conpressor, one that has not been specially designed for this particular problem. It follows that the results obtained are typical for such a mechanism and do not represent a specific combination to yield the best overall results.

TURBO-CHARGED ENGINE

In order to complete these comparisons, typical data for a straight turbo-charged engine are given in Table IV. These data are as calculated in Ref. 1, and the main difference between this and the previous machines is seen to be in the fact that turbo and compressor efficiencies, and their work factors, do not enter into the problem since it is assumed (and well established in practice) that the energy of the exhaust gas is sufficient to carry the work of the compressor without the aid of any engine power. The only change necessary is the employment of sufficient back pressure on the engine to secure this requirement. The back pressure employed in these calculations was constant at 0.85 P_m, i.e., at a 15% drop below the inlet pressure.

It must be recognized that the back pressure will probably exceed the manifold pressure at low engine speed and power. As is normally the case, this does not affect the data of Table IV since in all cases the compressor has been assumed to be designed for the particular case being investigated, with the necessary margin of pressure across the manifolds for satisfactory scavenging of the cylinder.

PART LOAD PERFORMANCES

The material presented above records the full load and speed data expected for the different types of superhcargers covered. Of equal importance is the part load fuel requirements, etc.

This side of the problem was examined in the manner outlined in Ref. 1 with the following results.

DISPLACEMENT SUPERCHARGER

Employing a displacement machine without compression, directly driven at some desirable ratio from the engine, and assuming a constant manifold pres-

TABLE IV

PERFORMANCE WITH A TURBO-CHARGER

Manifold pressure, P_m	atm	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6
Basic imep Ref. 1	psi	130	156	180	210	255	260	288	310	340
0.9 basic imep	psi	117	140.3	162.0	189	211.2	234.0	259.0	279.0	306
Ihp	lb-air sec	511	510	505	515	512	510	514	507	513
Isfc	lb/ihp/hr	0.334	0.334	0.337	0.330	0.332	0.334	0.331	0.336	0.332
Fmep	psi	22.5	23.3	24.0	25.0	26.0	26.7	27.5	28.5	30.0
Bmep gross	psi	94.5	117.0	138	164	185.2	207.3	231.5	250.5	276
Bhp gross	lb-air sec	414	425	430	446	449	452	459	455	463
Fan mean pres.	psi	21.0	21 0	21.0	21.0	21.0	21.0	21.0	21.0	21.0
Net bmep	psi	73.5	96.0	117	143	164.2	186.3	210.5	229.5	255.0
Net bhp with cooling fan	lb-air sec	322	349	365	390	398	406	417	418	427
Net sfc with cooling fan	lb/bhp/hr	0.529	0.488	0.466	0.437	0.428	0.419	0.408	0.407	0.399

sure as the F/A ratio varies, a condition closely followed for a constant engine speed. The calculated performance was then obtained as follows.

The engine is assumed to have a maximum speed of 3000 rpm and F/A_{max} = 0.05 approx.; the data are calculated for 3000, 2400, 1700, and 1000 rpm. With the above engine revolutions, typical compressor maps indicate that, if the design is set for a pressure ratio of 2.6:1 at 3000 rpm, ratios of 2.2:1, 1.8:1 and 1.4:1 can be expected at 2400, 1700, and 1000 rpm at the air flow requirements for these speeds. Under these conditions the data of Table V are calculated. The frictional and cooling fan losses are considered constant at constant speed and pressure ratio, while the compressor input is also considered constant, since little if any change in air flow will occur under such conditions.

If the engine is designed for 500 BHP net at the 2.2:1 ratio and 3000 rpm, then the air flow at that condition becomes

$$\text{Required Air Flow for 500 hp} = 1 \times \frac{500.0}{344.0}$$

$$= 1.452 \text{ lb/sec.}$$

Since the manifold temperature (at least at the high ratios) has been maintained at 660° abs., the air flow varies directly as the engine speed and pressure, as below:

Manifold Pressure Atms.	Air Flow Lb/Sec			
	3000 rpm	2400 rpm	1700 rpm	1000 rpm
2.6	1.72	---	---	---
2.2	1.452	1.17	---	---
1.8	1.19	0.951	0.675	---
1.4	0.925	0.74	0.524	0.308

It follows that the engine performance data for a 500 BHP engine fitted with a 2.2:1 displacement blower becomes as given in Table VI and Figs. 11-14.

The 2.2:1 ratio was chosen for full load in this case since there was a distinct drop in performance of the system when the ratio was increased to 2.6, as can be seen from Table VI. This indicates the limitations arising from low efficiencies at the high pressure ratios that are associated with this type of charger. Ratios in the 1.4 to 1.6 range give the best performance from a SFC point of view.

TABLE V

PART LOAD PERFORMANCE OF AN ENGINE WITH A DISPLACEMENT COMPRESSOR

Rpm and Pres. Ratio	F/A Ratio	Air For = 1 lb/sec Imep	Bmep	Ihp	Bhp	Specific Fuel Consumption, lb/hp/hr ihp	Net bhp
	0.015	102.8	---	172.0	---	0.313	---
	0.020	139.8	---	234.0	---	0.308	---
	0.025	192.8	48.8	323.0	81.9	0.278	1.01
3000	0.030	237.8	93.8	398.0	157.0	0.272	0.689
r = 2.6:1	0.035	267.8	123.8	449.0	207.5	0.281	0.606
	0.040	292.8	148.8	491.0	249.0	0.293	0.579
	0.045	307.8	163.8	516.0	274.0	0.314	0.592
	0.050	317.8	173.8	533.0	291.0	0.338	0.619
	0.015	92.8	---	184.0	---	0.293	---
	0.020	117.8	18.3	233.0	36.2	0.309	1.99
	0.025	162.8	63.3	322.0	125.3	0.280	0.717
3000	0.030	206.8	107.3	409.0	213.0	0.264	0.507
r = 2.2:1	0.035	227.8	128.3	451.0	254.0	0.279	0.496
	0.040	247.8	148.3	491.0	294.0	0.293	0.490
	0.045	257.8	158.3	511.0	314.0	0.317	0.516
	0.050	272.8	173.3	540.0	344.0	0.334	0.524
	0.015	67.8	---	163.0	---	0.330	---
	0.020	91.8	18.6	222.0	45.0	0.324	1.60
	0.025	132.8	59.6	321.0	144.0	0.280	0.625
3000	0.030	157.8	84.6	382.0	205.0	0.283	0.528
r = 1.8:1	0.035	177.8	104.6	430.0	253.0	0.293	0.498
	0.040	195.8	122.6	474.0	297.0	0.304	0.485
	0.045	207.8	134.6	503.0	326.0	0.322	0.497
	0.050	215.8	142.6	522.0	346.0	0.345	0.520
	0.015	54.5	---	169.5	---	0.318	---
	0.020	69.5	13.0	216.0	40.5	0.333	1.78
	0.025	108.5	52.0	334.0	162.0	0.269	0.555
3000	0.030	134.5	78.0	418.5	243.0	0.258	0.445
r = 1.4:1	0.035	149.5	93.0	465.5	289.5	0.271	0.436
	0.040	158.5	102.0	494.0	318.0	0.292	0.453
	0.045	174.5	118.0	544.0	367.5	0.298	0.442
	0.050	179.5	123.0	559.0	383.0	0.322	0.470
	0.015	92.8	6.3	184.0	12.5	0.293	4.31
	0.020	117.8	31.3	233.0	62.0	0.309	1.16
	0.025	162.8	76.3	321.0	151.0	0.281	0.596
2400	0.030	206.8	120.3	410.0	238.0	0.261	0.454
r = 2.2:1	0.035	227.8	141.3	450.0	280.0	0.280	0.450
	0.040	247.8	161.3	490.0	319.0	0.294	0.451
	0.045	257.8	171.3	510.0	339.0	0.318	0.478
	0.050	272.8	180.3	540.0	358.0	0.333	0.503

Rpm and Pres. Ratio	F/A Ratio	Air Flow = 1 lb/sec Imep	Bmep	Ihp	Bhp	Specific Fuel Consumption, lb/hp/hr ihp	Net bhp
	0.015	67.8	8.2	164.0	19.8	0.328	2.72
	0.020	91.8	32.2	222.0	78.0	0.324	0.924
	0.025	132.8	73.2	321.0	177.0	0.281	0.509
2400	0.030	157.8	98.2	382.0	238.0	0.283	0.454
r = 1.8:1	0.035	177.8	118.2	430.0	286.0	0.293	0.441
	0.040	195.8	136.2	474.0	330.0	0.304	0.436
	0.045	207.8	148.2	503.0	359.0	0.322	0.451
	0.050	215.8	156.2	522.0	379.0	0.345	0.476
	0.015	54.5	11.0	169.5	34.2	0.318	1.58
	0.020	69.5	26.0	216.0	81 0	0.333	0.889
	0.025	108.5	65.0	334.0	202.5	0.270	0.444
2400	0.030	134.5	91.0	418.5	283.0	0.258	0.382
r = 1.4:1	0.035	149.5	106.0	465.5	330.0	0.271	0.382
	0.040	158.5	115.0	494.0	358.0	0.292	0.402
	0.045	174.5	131.0	544.0	408.0	0.298	0.399
	0.050	179.5	136.0	559.0	424.0	0.322	0.429
	0.015	67.8	20.3	164.0	49.2	0.328	1.096
	0.020	91.8	34.3	222.0	83.0	0.324	0.867
	0.025	132.8	85.3	321.0	206.0	0.280	0.436
1700	0.030	157.8	110.5	382.0	267.0	0.283	0.404
r = 1.8:1	0.035	177.8	130.5	430.0	316.0	0.293	0.399
	0.040	195.8	148.5	474.0	360.0	0.304	0.400
	0.045	207.8	160.5	503.0	389.0	0.322	0.416
	0.050	215.8	168.5	522.0	408.0	0.345	0.441
	0.015	54.5	21.5	169.5	67.0	0.318	0.805
	0.020	69.5	36.5	216.0	114.0	0.333	0.632
	0.025	108.5	75.5	334.0	235.0	0.270	0.383
1700	0.030	134.5	101.5	418.5	316.0	0.258	0.342
r = 1.4:1	0.035	149.5	116.5	465.5	363.0	0.271	0.347
	0.040	158 5	125.5	494.0	391.0	0.293	0.368
	0.045	174.5	141.5	544.0	441.0	0.298	0.368
	0.050	179.5	146.5	559.0	456.0	0.322	0.395
	0.015	54.5	32.0	169.5	99.6	0.318	0.541
	0.020	69.5	47.0	216.0	146.5	0.333	0.491
	0.025	108.5	86.0	334.0	268.0	0.270	0.336
1000	0.030	134.5	112.0	418.5	349.0	0.258	0.310
r = 1.4:1	0.035	149.5	127.0	465.5	396.0	0.271	0.318
	0.040	158.5	136.0	4 4.0	422.0	0.293	0.341
	0.045	174.5	152.0	544.0	472.0	0.298	0.343
	0.050	179.5	157.0	559.0	489.0	0.322	0.368

TABLE VI

PERFORMANCE OF A 500 BHP (NET) ENGINE WITH DISPLACEMENT SUPERCHARGER
DELIVERING 1.452 LB OF AIR PER SEC AT 2.2:1 AND 3000 RPM

Manifold Pressure, atm	F/A	RPM 3000 Bhp	3000 Bsfc lb/hp/hr	2400 Bhp	2400 Bsfc lb/hp/hr	1700 Bhp	1700 Bsfc lb/hp/hr	1000 Bhp	1000 Bsfc lb/hp/hr
2.6	0.015	---	---						
	0.020	---	---						
	0.025	141	1.01						
	0.030	270	0.689						
	0.035	356	0.606						
	0.040	429	0.579						
	0.045	469	0.592						
	0.050	500	0.619						
2.2	0.015	---	---	14.6	4.31				
	0.020	52.6	2.00	72.5	1.16				
	0.025	182	0.72	177.0	0.60				
	0.030	310	0.51	279.0	0.45				
	0.035	369	0.50	328.0	0.45				
	0.040	427	0.49	374.0	0.45				
	0.045	456	0.52	397.0	0.48				
	0.050	500	0.52	419.0	0.50				
1.8	0.015	---	---	18.8	2.72	33.2	1.1		
	0.020	53.6	1.60	74.2	0.92	55.8	0.87		
	0.025	172	0.63	168.0	0.51	139.0	0.44		
	0.030	244	0.53	226.0	0.45	180.0	0.40		
	0.035	301	0.50	272.0	0.44	213.0	0.39		
	0.040	354	0.49	314.0	0.44	243.0	0.40		
	0.045	388	0.50	342.0	0.45	262.0	0.42		
	0.050	412	0.52	360.0	0.48	275.0	0.44		
1.4	0.015	---	---	25.5	1.58	35.0	0.81	30.7	0.54
	0.020	37.4	1.78	60.0	0.89	59.7	0.63	45.1	0.49
	0.025	150	0.56	150.0	0.44	123.0	0.38	82.6	0.34
	0.030	225	0.45	209.0	0.38	165.0	0.34	108.0	0.31
	0.035	267	0.44	244.0	0.38	190.0	0.35	122.0	0.32
	0.040	294	0.45	265.0	0.40	205.0	0.37	130.0	0.34
	0.045	339	0.44	302.0	0.40	231.0	0.37	145.0	0.34
	0.050	354	0.47	314.0	0.43	239.0	0.40	151.0	0.37

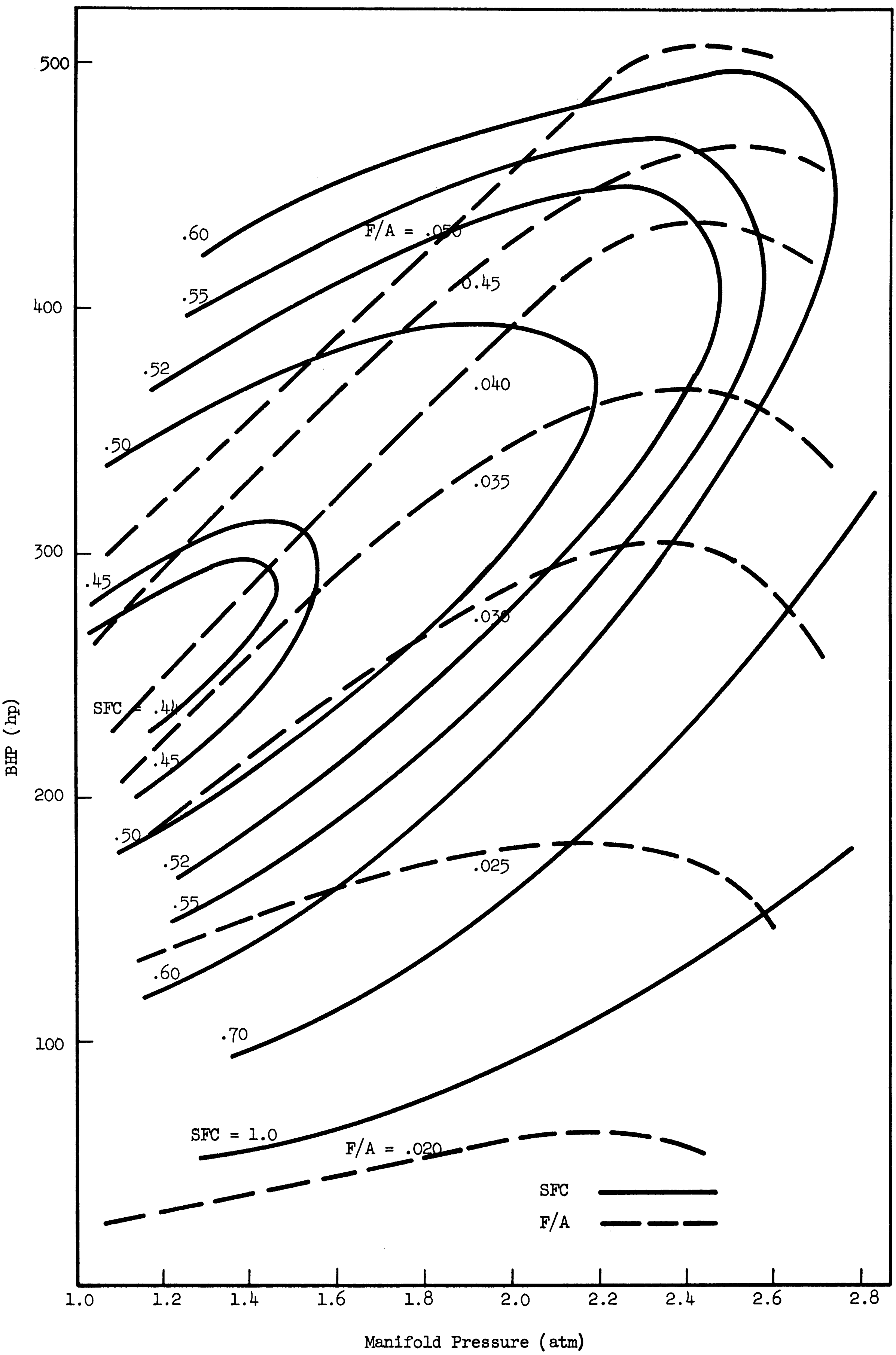

Fig. 11. Part load performance with displacement compressor at 3000 rpm.

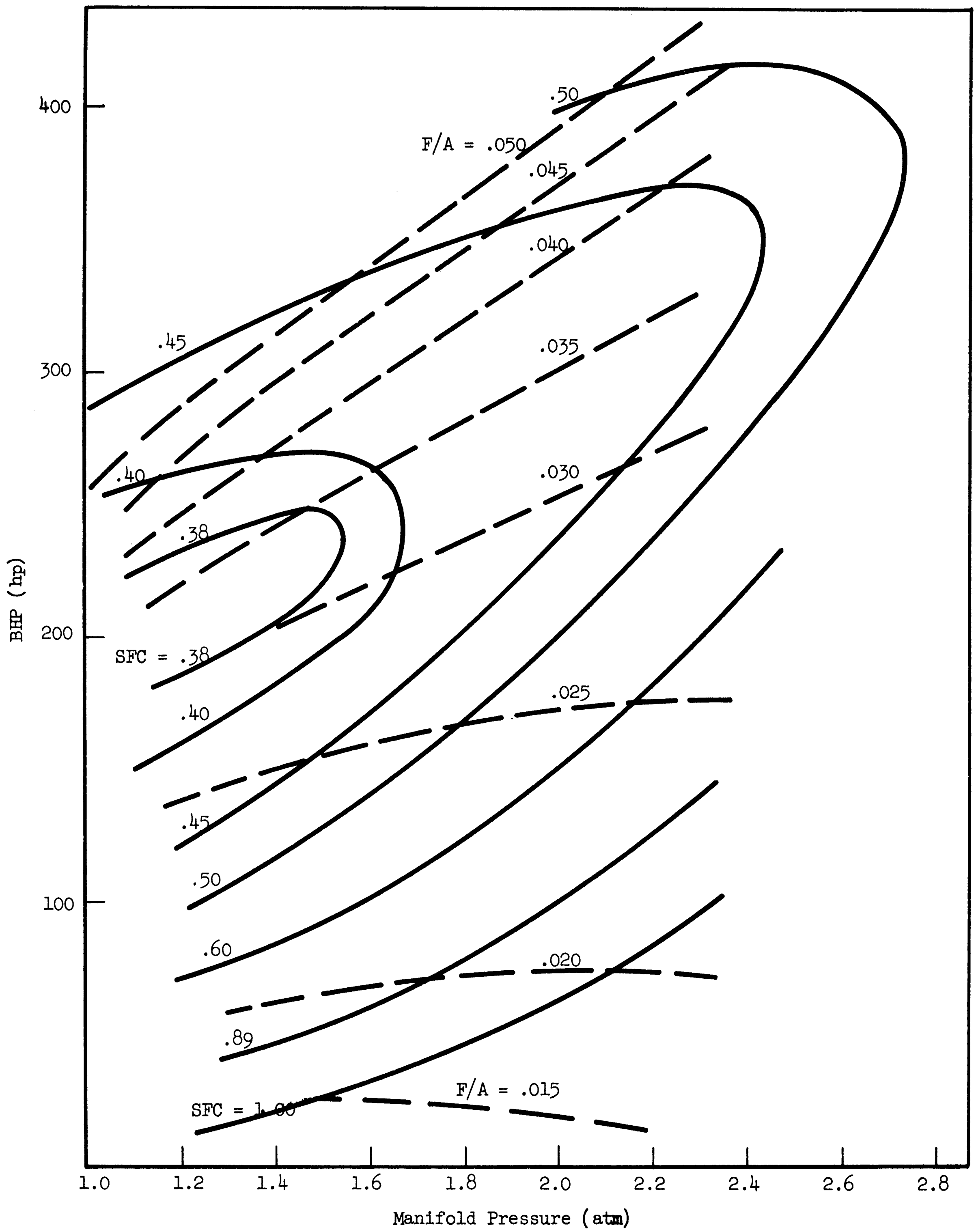

Fig. 12. Part load performance with displacement compressor at 2400 rpm.

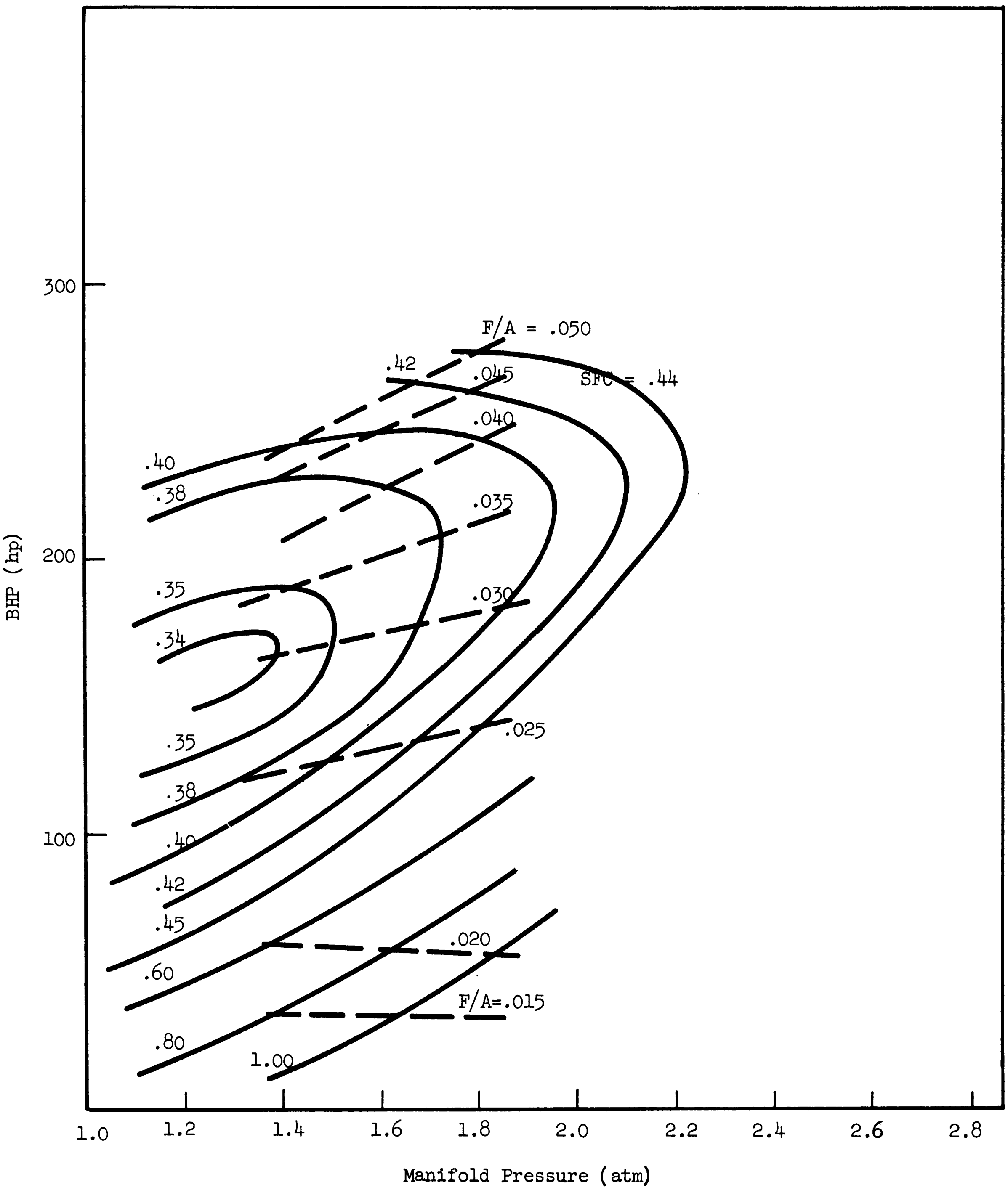

Fig. 13. Part load performance with displacement compressor at 1700 rpm.

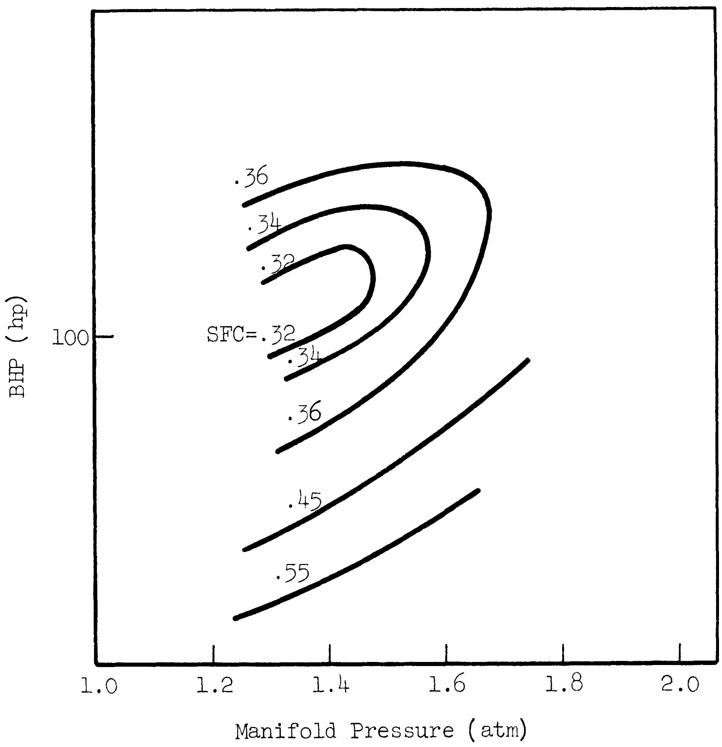

Fig. 14. Part load performance with displacement compressor at 1000 rpm.

CENTRIFUGAL SUPERCHARGER

If the centrifugal machine, of modern design, is employed as the example to be investigated for part load performance characteristics it will, due to the high efficiency of this type, give about the best performance of the directly driven superhcarger. The Lysholm and Bicera machines would lie between the displacement and centrifugal machines. It is not proposed to calculate part load for each of these machines at this time; the centrifugal is examined as presenting the most efficient.

The centrifugal machine has a definite compressor map depending upon the resistance encountered at the discharge; a typical map has already been represented in Fig. 4. The problem then arises as to how the engine requirements are supplied by the compressor. If a complete performance chart was available for the engine, the compressor and engine could be matched accurately. What is to be examined in this investigation is that, at full load and engine speed of 3000 rpm, the compressor will deliver air sufficient to maintain a 2.6:1 pressure ratio at a constant manifold temperature of 660° abs. The compressor being driven off the engine will also operate at constant speed as the F/A ratio and thus the power is varied at a constant engine speed of 3000; similar conditions will exist at other engine speeds with the manifold pressure ratio varying approximately as the square of the speed as given by Eq. (14). Now at constant engine speed and manifold conditions the air flow of the engine will be considered constant as the F/A ratio varies. This is a reasonable first assumption; without it the solution becomes quite involved and lengthy. With this assumption, the data of Table VII and Fig. 15 are calculated, for a flow of 1 lb/sec. The data are then corrected to an engine of 500 BHP at 3000 rpm for an air flow of 1.36 lb/sec at 2.6:1 pressure ratio, as for the other chargers. The higher ratio of 2.6 was chosen for full power in this case since the results showed improved performance at this ratio, as a result of the improved compressor performance.

$$(P_m/P_o)^{0.286} - 1 \propto rpm^2 \qquad (14)$$

The effect of the higher pressure ratio, relative to the displacement machine, shows up in a reduced air flow, higher net BMEP, and thus a reduced engine volume and weight. These details will be summarized later.

TURBO-CHARGER

The part load performance of the turbo-charged engine is calculated from Ref. 1 and presented in Table VIII. In this case it is assumed that the turbine will always drive the compressor and that no power generated by the pistons is lost in compressing the air. This is true except at very low outputs, when the exhaust back pressure may exceed the inlet manifold pressure

TABLE VII

PART LOAD PERFORMANCE WITH CENTRIFUGAL COMPRESSOR

Rpm, P_m	F/A	Hp for 1 lb of air per sec				Hp at 1.36 lb/air/sec	Sfc, lb/hp/hr	
		Imep	Bmep	Ihp	Bhp	at 3000 rmp	Ihp	Bhp
	0.015	102.8	4.8	172.2	8.5	11.5	0.313	6.35
	0.020	139.8	41.8	234.0	70.1	95.3	0.308	1.03
	0.025	192.8	94.8	323.0	159.0	216.0	0.279	0.566
3000	0.030	237.8	139.8	398.0	234.0	318.0	0.272	0.462
2.6:1	0.035	267.8	169.8	449.0	285.0	387.0	0.281	0.442
	0.040	292.8	194.8	491.0	326.0	443.0	0.293	0.442
	0.045	307.8	209.8	516.0	352.0	478.0	0.314	0.460
	0.050	317.8	219.0	533.0	368.5	500.0	0.338	0.490
	0.015	44.0	---	101.0	---	---	0.534	---
	0.020	66.0	3.0	151.5	6.9	5.48	0.475	1.04
	0.025	129.0	66.0	296.0	151.0	120.0	0.304	0.596
2400	0.030	169.0	106.0	388.0	244.0	194.0	0.278	0.443
1.9:1	0.035	184.0	121.0	422.0	277.0	220.0	0.299	0.455
	0.040	200.0	137.0	459.0	314.0	251.0	0.314	0.459
	0.045	219.0	156.0	503.0	358.0	285.0	0.322	0.452
	0.050	230.0	167.0	528.0	383.0	304.0	0.341	0.470
	0.015	34.5	4.8	107.8	14.9	6.2	0.502	3.62
	0.020	74.5	44.8	232.0	139.0	57.7	0.310	0.518
	0.025	103.5	73.8	322.0	230.0	95.5	0.280	0.391
1700	0.030	129.5	99.8	403.0	311.0	129.0	0.268	0.347
1.4:1	0.035	149.5	119.8	466.0	373.0	155.0	0.270	0.338
	0.040	159.5	129.8	497.0	404.0	167.5	0.290	0.356
	0.045	166.5	136.8	519.0	426.0	177.0	0.312	0.380
	0.050	174.5	144.8	544.0	450.0	187.0	0.331	0.400
	0.015	28.1	14.4	112.2	57.7	10.9	0.480	0.93
	0.020	61.1	47.4	244.0	190.0	36.0	0.295	0.378
	0.025	86.1	72.4	344.0	289.0	55.0	0.262	0.312
1000	0.030	111.1	97.4	444.0	389.0	74.0	0.243	0.278
1.09:1	0.035	126.1	112.4	505.0	450.0	85.5	0.250	0.280
	0.040	135.1	121.4	541.0	486.0	92.4	0.266	0.296
	0.045	143.1	129.4	573.0	518.0	98.4	0.283	0.313
	0.050	151.1	137.4	605.0	550.0	104.3	0.298	0.328

TABLE VIII

PART LOAD PERFORMANCE WITH TURBO-CHARGER

Rpm	Pres. Ratio	F/A	Hp per lb of air per sec				Air Flow, lb/sec	Bhp	Sfc, lb/hp/hr	
			Imep	Bmep	Ihp	Bhp			Ihp	Bhp
	1.07	0.02	60.0	17.0	244	69.3	0.492	34.0	0.295	1.04
	1.30	0.025	90.0	46.0	302	154.0	0.598	92.0	0.298	0.584
	1.48	0.029	120.0	75.0	353	221.0	0.681	150.5	0.296	0.474
	1.77	0.036	165.0	119.0	406	293.0	0.814	238.0	0.319	0.442
3000	2.06	0.041	210.0	162.5	444	344.0	0.947	326.0	0.333	0.430
	2.23	0.046	240.0	191.5	469	374.0	1.027	384.0	0.353	0.443
	2.4	0.051	270.0	219.5	490	398.0	1.103	439.0	0.375	0.462
	2.6	0.052	300.0	249.0	503	418.0	1.197	500.0	0.372	0.448
	2.66	0.055	306.0	254.5	502	417.0	1.221	510.0	0.395	0.476
	1.04	0.0185	52.3	20.4	218	85.5	0.383	32.7	0.305	0.778
	1.33	0.027	105.0	72.5	345	238	0.489	116.0	0.282	0.409
	1.48	0.032	131.0	97.9	386	289	0.545	155.0	0.298	0.399
	1.61	0.0375	157.0	123.6	425	334	0.592	198.0	0.318	0.404
2400	1.77	0.041	183.0	149.2	450	368	0.651	239.0	0.328	0.401
	1.90	0.0445	209.0	175.2	480	402	0.699	281.0	0.334	0.399
	2.08	0.049	235.0	200.6	492	420	0.766	322.0	0.359	0.420
	2.21	0.052	248.0	212.3	489	418	0.814	340.0	0.383	0.448
	2.29	0.065	275.0	239.8	524	456	0.842	384.0	0.447	0.514
	1.0	0.019	52.0	31.3	227	137	0.261	35.8	0.301	0.50
	1.0	0.0235	78.0	57.3	340	249	0.261	65.0	0.249	0.34
	1.0	0.029	104.0	83.3	453	363	0.261	94.6	0.231	0.288
1700	1.03	0.038	130.0	109.2	502	421	0.268	113.0	0.273	0.325
	1.23	0.050	156.0	135.1	554	479	0.322	154.0	0.325	0.376
	1.32	0.052	168.0	146.9	556	485	0.344	167.0	0.336	0.386
	1.44	0.060	182.0	160.7	551	486	0.376	183.0	0.392	0.444
	1.0	0.015	40	25.8	174.5	112.5	0.184	20.7	0.309	0.48
	1.0	0.0185	50	35.8	218.0	156.0	0.184	28.7	0.306	0.426
	1.0	0.02	60	45.8	261.0	199.8	0.184	36.8	0.276	0.360
1200	1.0	0.024	80	65.8	348.0	287.0	0.184	52.9	0.249	0.302
	1.0	0.028	100	85.8	436.0	374.0	0.184	68.9	0.232	0.270
	1.0	0.0365	122	107.8	532.0	470.0	0.184	86.5	0.247	0.280
	1.0	0.039	132	117.8	575.0	514.0	0.184	94.5	0.244	0.273

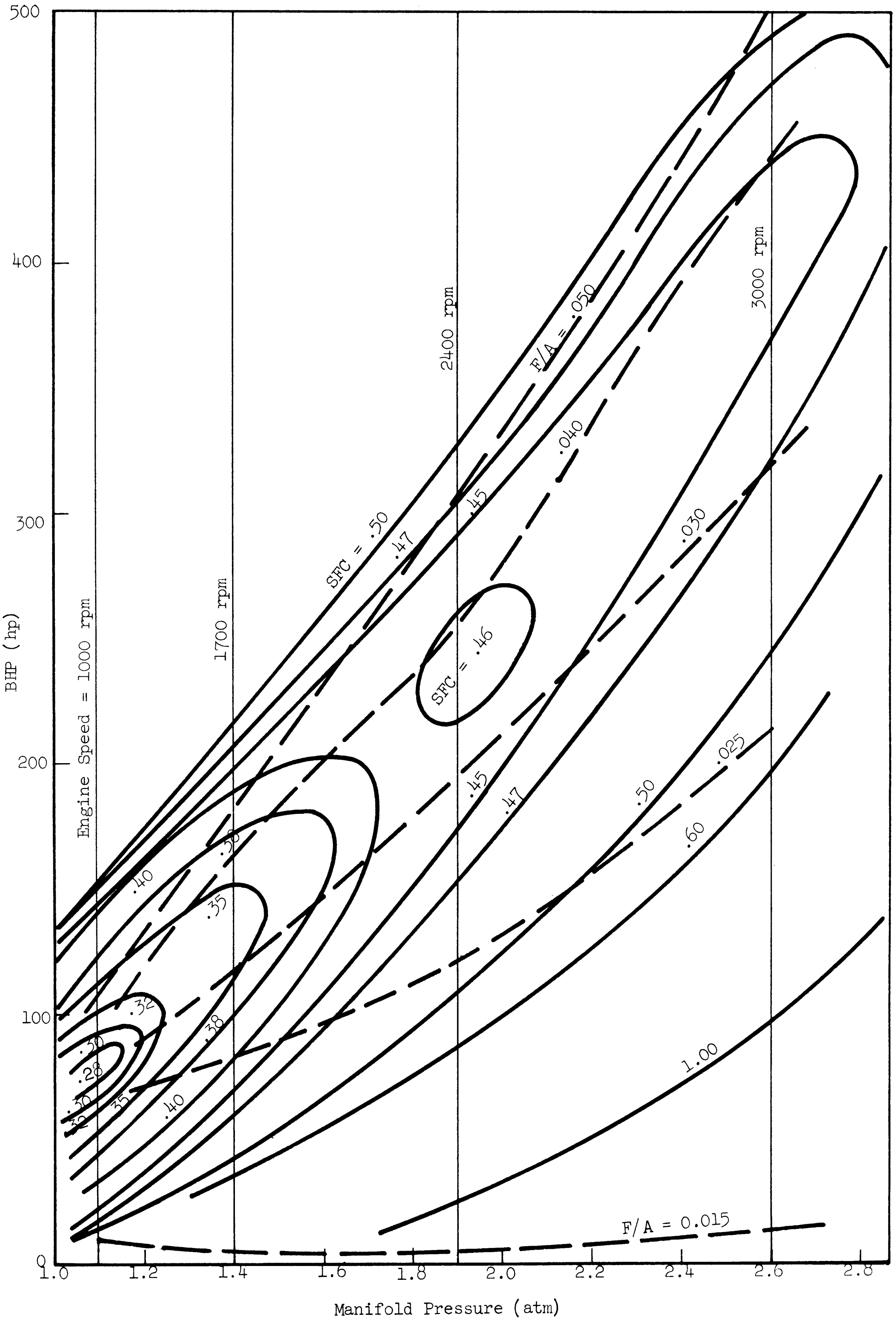

Fig. 15. Part load performance with centrifugal compressor.

by a small amount, which in turn may reduce engine power slightly.

The engine performance line is calculated with the use of Fig. 5 of Ref. 1 for the variation of manifold pressure with load and speed. It is seen from this diagram that P_m will vary, at a constant engine speed, as the F/A ratio varies due to the changes in the exhaust gas temperature, etc. The starting point was taken as 2.6:1 at F/A = 0.0473 and 3000 rpm, as before, with ratios of 2.2, 1.77, and 1.0 at 2400, 1700, and 1200 rpm with full load IMEP. This means that the F/A ratio for the lower pressure ratios and speeds would have to change in a manner similar to that shown in Fig. 19 of Ref. 1, to keep the mean pressure constant. These combinations take the F/A up into the smoke regions and thus there will be some limitation to these maximum values.

The diagram of Fig. 19 of Ref. 1 constructed for a F/A = 0.043 can be used for the present case by increasing it proportionally to 0.0473, the required ratio at full speed. Then, for example, at 80% speed, viz. 2400 rpm with a smoke limit of 0.052, the maximum IMEP that can be expected would be given at a F/A ratio of

$$\text{F/A at 80\% for Max. IMEP} = 0.052 \times \frac{0.043}{0.0473}$$

$$= 0.0472.$$

In other words, 0.0472 of Fig. 19 of Ref. 1 corresponds to 0.052 on a diagram built up for a F/A of 0.0473. At this value the IHP at 80% speed is given by Fig. 5 of Ref. 1 as 76%; thus the IMEP will be 76/80 x 260 = 248 psi, the 260 being the IMEP at 3000 rpm for 0.0473 F/A. The starting point for the 2400 rpm thus becomes F/A = 0.052, P_m = 67 approx. (see Fig. 3 of Ref. 1), which is 86% of the 2.6:1 ratio at full speed. Checking this with Fig. 5 of Ref. 1, the ratio is there given as 84% approx. Taking the mean of these two values, say 85% manifold pressure resulting in a ratio of 2.21:1 being required from the charger, the balance of the 2400 rpm data can be filled in on Table VIII. At 1700 rpm, maximum IMEP will also be at F/A = 0.0472 on Fig. 19 of Ref. 1; the IHP = 36.5%, giving an IMEP of 36.5/56.6 x 260 = 168 psi. For this value Fig. 3 of Ref. 1 gives that a manifold pressure of 40.5" Hg or 51.5% is required, or ratio = 1.35. Figure 5 of Ref. 1 predicts 49.0% of full speed pressure, or a ratio of 1.28. Averaging these two estimates the pressure ratio of 1.32:1 will be employed. Similar calculations were employed for the 1200 rpm, resulting in an IMEP of 122 psi for 0.052 F/A. This can be obtained with no supercharging at a manifold pressure of 22" Hg approx.; Fig. 5 of Ref. 1 predicts 31" Hg. It will be assumed that an unsupercharged condition will meet this case.

It is seen that the mean pressure and supercharger ratio varies slightly, depending upon which diagram of Ref. 1 is used. The change is quite small, however, in the major operating range of the engine. By taking the average of the results, sufficient accuracy is believed to be secured. This method does result in a slight variance of the final F/A ratio from that originally assumed, but the results are affected only by a small change in the mean pressure.

Table VIII and Fig. 16 present the results of these calculations.

ENGINE OVERALL VOLUME

A casual examination of Tables VI, VII, and VIII shows little difference for engines of 500 BHP output at 3000 rpm. However it must be remembered that the air flow in lb/sec has been varied to obtain this constant output. It follows that the engine bulk is not identical even if the output and fuel flow are constant.

Taking these differences in air flow into account, it is possible to determine the engine bulk for each of the cases examined. Let it be assumed that a constant stroke/bore ratio of 1.08 is employed for all engines.

DISPLACEMENT SUPERCHARGER

Air flow = 1.452 lb/sec at 660° abs and at 2.2:1 pressure ratio and 3000 rpm. If volumetric efficiency is assumed at 96% the engine size is given by

$$\text{Displacement Volume of Engine} = \frac{\pi D^2}{4} \times L \times \frac{n \times N}{2} \text{ cu in./min}$$

where

D = diameter of cycle in inches

L = stroke of cycle in inches = 1.08D

n = number of cylinders

N = rpm

$$\text{Volume of Air} = \frac{wRT_m}{P_m} \text{ cu ft}$$

$$= \frac{60 \times 1.452 \times 53.34 \times 660 \times 1728}{144 \times 14.7 \times 2.2} \text{ cu in./min}$$

$$= 12.54 \times 10^5 \text{ cu in./min.}$$

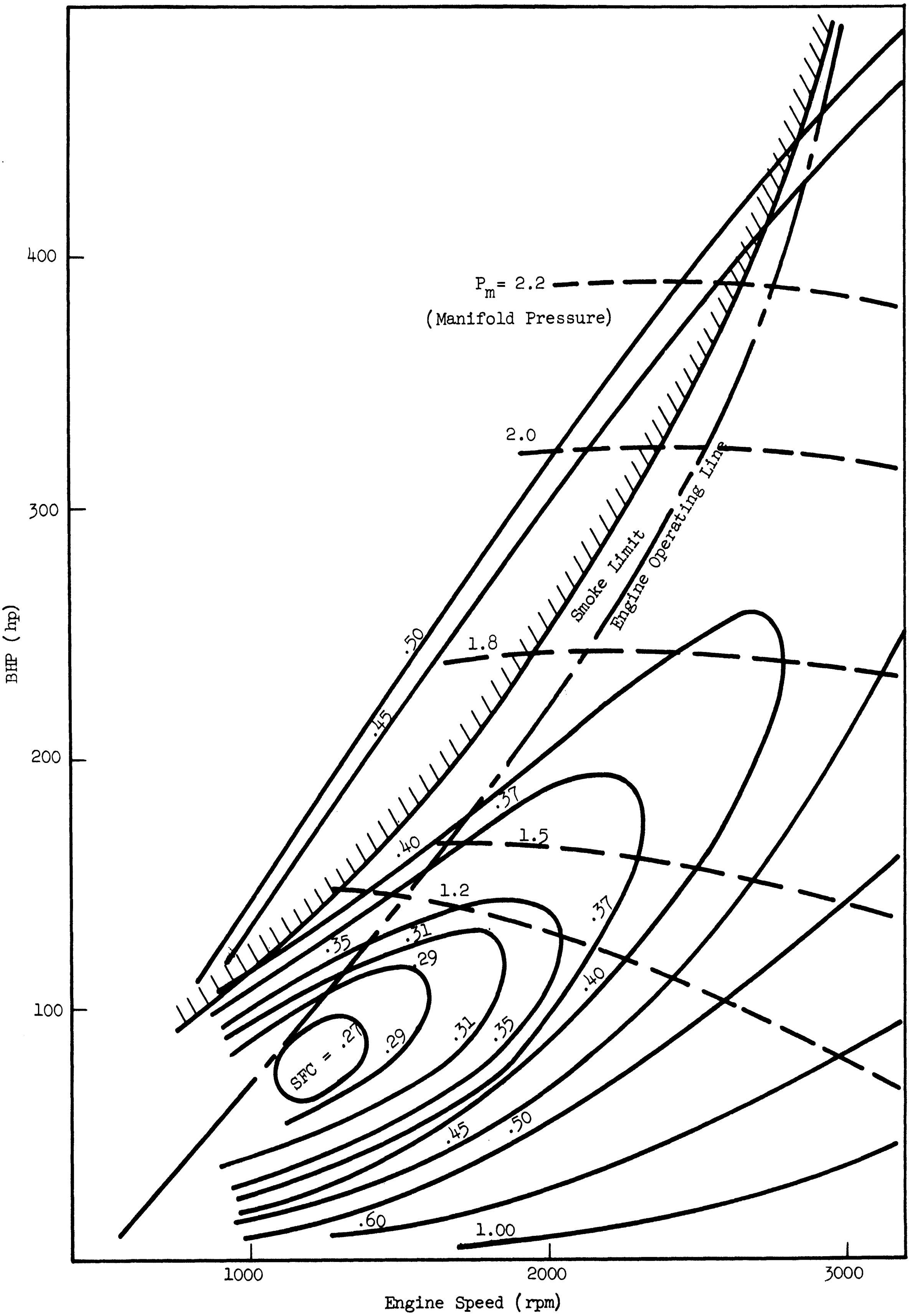

Fig. 16. Part load performance with turbo-charger.

at 96% volume efficiency.

$$\text{Required Cylinder Volume} = \frac{12.54\times10^5}{0.96}$$

$$= 13.08 \times 10^5 \text{ cu in./min}$$

$$= \text{engine displacement}$$

therefore

$$\frac{\pi D^2}{4} \times 1.08D \times \frac{n\times3000}{2} = 13.08 \times 10^5$$

$$nD^3 = \frac{13.08\times10^5\times4\times2}{\pi\times1.08\times3000}$$

$$= 1.028 \times 10^3$$

n	6	8	12
D^3	1.711×10^2	1.282×10^2	0.855×10^2
D	5.55	5.04	4.4
L	6.0	5.45	4.75

The possible engine combinations would then be as below:

n	6	8	12
D	5-5/8	5	4-1/2
L	6.0	5-1/2	4-3/4
Approx. Vol. cu ft In-line Engine	97	76	---
Approx. Vol. cu ft V- Engine 60°	63	50	47
Approx. Vol. cu ft V- Engine 90°	76	58	54

There would be, of course, some difference of weights despite the output being constant at 500 BHP. Assuming 4.5 lb/hp for the V-engines, it is believed that the range would be from about 2300 lb to 3000 lb. This change of weight is quite small and rather negligible when the total vehicle weight is considered.

The above calculations are based upon Figs. 10, 12, and 13 of Ref. 1, which are for turbo-charged engines where the superchargers are quite small. The displacement compressor will be of considerable bulk, relatively, and its volume will be estimated on the basis of a Root machine where the average area of the impeller is about 63% of the circle swept out by the tip.

In one revolution of the machine, the theoretical displacement of the blower will be, for the two lobes,

$$\text{Displacement} = 0.37 \times 2 \times \frac{\pi D^2}{4} \times L \text{ su in.}$$

where

D = diameter of impeller in inches

L = length of impeller in inches.

A good value of L for low leakage, etc., is given by $L = 1.5D$

$$\begin{aligned} \text{Displacement} &= 0.37 \times 2 \times \frac{\pi D^2}{4} \times 1.5D \\ &= 0.873D^3 \text{ cu in./rev} \\ &= 0.873D^3N \text{ cu in./min} \\ N &= \text{rpm of blower.} \end{aligned}$$

If it is assumed that there is a leakage of 7%, then

$$\text{Required Displacement} = 0.94D^3N \text{ cu in./min.}$$

This type of blower can be operated at some 7000 to 10,000 rpm. Assume it is geared to the proposed engine at 3:1, giving 9000 rpm of the blower, which a relatively high speed for this type: then the dimensions become

$$\text{Required Displacement} = \frac{wRT_o}{P_o} \text{ cu in./min}$$

$$= \frac{1.452x60x53.34x545x1728}{144x14.7}$$

$$= 2.068 \times 10^6 \text{ cu in./min.}$$

Assuming a 6% air flow during valve overlap, the displacement must be increased to allow for this; thus

$$\text{Displacement} = 2.19x10^6 \text{ cu in./min}$$

$$= 0.94D^3N$$

$$D^3 = \frac{2.19x10^6}{0.94x9000}$$

$$= 259.0 \text{ in.}$$

$$D = 6.37 \text{ or, say, } 6\text{-}3/8.$$

With the above impeller dimensions, the overall dimensions would be approximately as shown in Fig. 17 with a length of at least 16 in., giving a total volume of about 16x14x8.5/1728 = 1.1 cu ft.

This volume would be added to the engine volume and would be almost constant irrespective of the type of displacement machine employed.

CENTRIFUGAL SUPERCHARGER

In the case of the centrifugal supercharger driven directly from the engine by gearing, the physical dimensions will be small, comparable with the sizes already allowed for in the turbo-charger in Figs. 10, 12, and 13 of Ref. 1. It follows that the engine bulk can be read directly from these figures when the engine size necessary to handle the air volume has been determined. Employing the same methods and assumptions as for the displacement machine, the following data are determined:

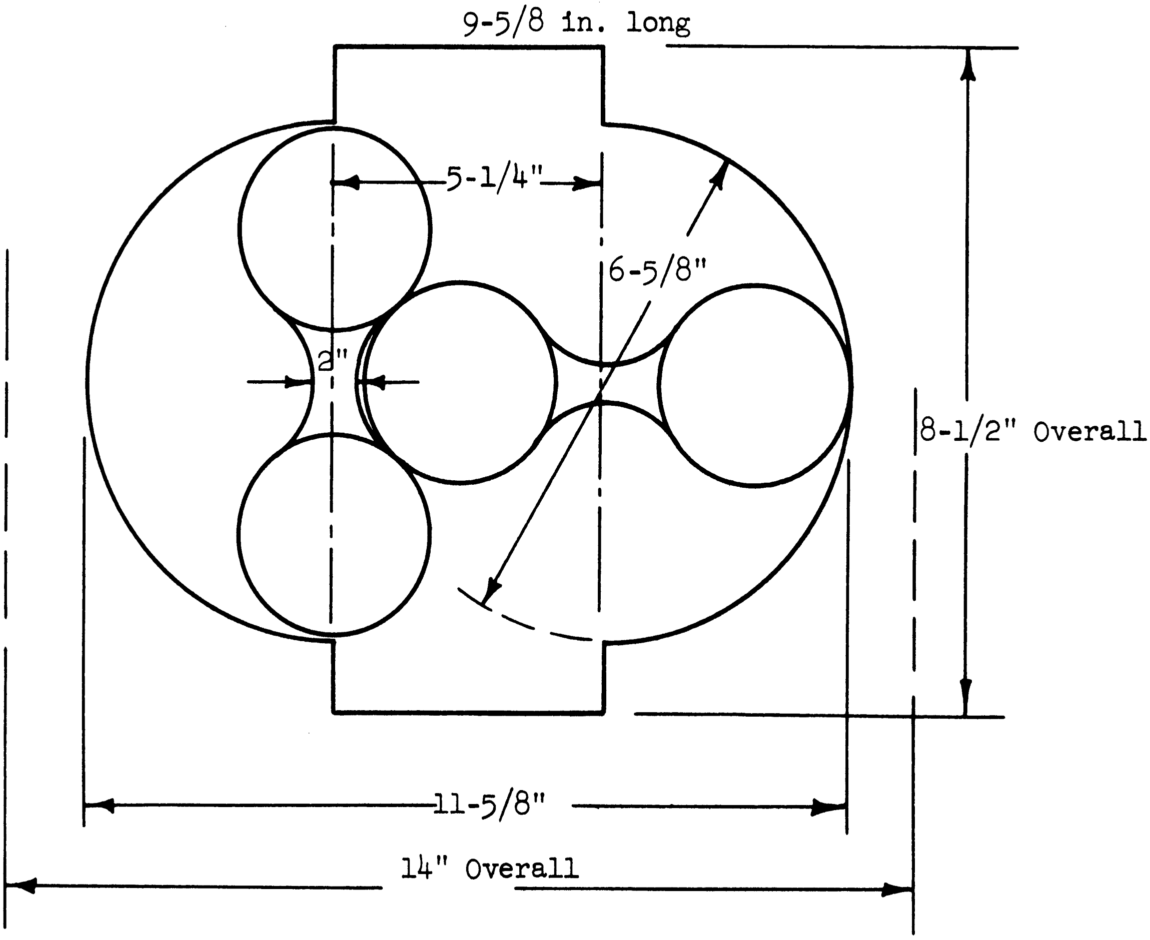

Fig. 17. Overall dimensions of displacement supercharger.

Air Flow at 3000 rpm = 1.36 lb/sec, ratio 2.6:1

No. of Cylinders	6	8	12
Bore	5-1/8	4-3/4	4-1/8
Stroke	5-5/8	5-1/8	4-1/2
Engine Volume cu ft			
In-Line	66	60	--
V 60°	44	42	33
V 90°	53	48	38

TURBO-CHARGER

In the case of the turbo-charged engine, the data calculated are as given below:

Air Flow at 3000 rpm = 1.197 lb/sec, ratio 2.6:1

No. of Cylinders		6	8	12
Bore		5	4-1/2	4
Stroke		5-3/8	4-7/8	4-1/4
Engine Volume cu ft				
In-Line		58	48	--
V	60°	48	34	30
V	90°	47	38	35

BATTLEFIELD DAY REQUIREMENTS

Taking the various engines investigated above and determining their fuel consumption over a 24-hour battlefield day based on the following vehicle data, the relative positions of the various engines are as shown.

Vehicle Weight—43 tons

Pitch Diameter of Sprocket—22.19 in.

Resistance on First Class Roads

mph	2	3	4	5	6	7	8	9	10	15	16	17	18	19	20	30
Resistance lb/ton	67	67.5	68	68	68	68.5	69	70	73.5	72	69	68	67	67	67	70

Time Schedule

20% of day	15, 16, 17, 18, 19, 20 mph	Resistance 1.57 times that of first class roads
40% of day	2, 3, 4, 5, 6, 7, 8, 9, 10 mph	2.00 times resistance of first class roads
40%	Engine Idling	

The average ground hp for the 20% period becomes 260.0 hp; for the 40% period, the hp averages out at 96 hp. Assuming a 10% ground slip plus an 82% transmission efficiency, the engine power becomes 348 for the 20% and 129 hp for the 40% period. With these powers the fuel requirements for the 24 hours becomes as shown in Table IX.

TABLE IX

FUEL REQUIREMENTS FOR A BATTLEFIELD DAY

Condition	Displacement Compressor (500 hp)	Centrifugal Charger (500 hp)	Turbo-Charger (500 hp)
20% at 348 hp 4.8 hr	845.0	754.0	730.0
40% at 129 hp 9.6 hr	470.0	451.0	435.0
40% Idle 9.6 hr	69.0	57.6	48.0
Total Fuel lb	1384.0	1262.6	1213.0

Of the figures in Table IX those for the idling condition are perhaps subject to the greatest error since both the engine speed and fuel requirements for idling are a function of the injection system in addition to the engine characteristics.

The calculations do not permit evaluation of the fine gradations such as the possible difference in the battlefield day consumption of a six and eight cylinder in-line engine. No attempt has been made to evaluate this; it would be small, resulting mainly from change of friction with engine size.

COMMENTS

No attempt has been made in this report to include any method for achieving responsiveness with these systems of supercharging. It is proposed to examine this phase of engine operation as applied to direct connected chargers in Part II of the report.

This analysis has been made in accordance with the methods of Report No. 04612-3-F, Contract No. DA-20-018-ORD-23664,[1] with suitable adjustments to meet the required conditions. The present report is aimed at the highly supercharged cycle with some wider spread of results to be expected at low pressure ratios. The present work includes some predictions at low ratios; these predictions will possibly possess a greater error than most of the work. These low ratios were necessary due to the fact that with some of the types of superchargers examined, high ratios are not possible. In fact, in the case of the displacement superhcarger, the ratios of 2.2 to 2.6:1 are above those generally recommended for this machine. It is believed that, at the engine powers of interest for the purpose in view, the results are sufficiently accurate for a systems analysis. Also, if the results obtained with the various machines are compared with one another, the correct trends will result; any deviation of actual engines, when constructed, will affect each one in about the same proportion. For ease of comparison the various combinations have been summarized in Table X.

ENGINE TORQUE CHARACTERISTICS

Torque curves have been plotted for the results of each of the assumed combinations in Fig. 18. These curves are plotted on a net output BHP including fan for engine, engine oil, and transmission oil cooling. In Fig. 18 is also included the degree of responsiveness that each system has. It is seen that the displacement supercharger does have a slight degree of responsiveness down to about 75% speed. The centrifugal supercharger falls off in responsiveness most rapidly with the turbo-charger in an intermediate position.

In any case the responsiveness is far below that needed for the approximate maintainance of constant horsepower with variable speed.

The responsiveness of the displacement compressor is secured at the expense of a considerable increase in fuel consumption.

The centrifugal and turbo-charged engines both fall off in torque characteristics, but the gear driven centrifugal due to the positive drive to the compressor wheel appears to hold up fairly well when the responsiveness has fallen to 0.6 approx. The turbo-charger continues on downward due to the fact that with engine speed reduction, the gas flow reduces and the power to drive the compressor falls rapidly. In fact, little or no supercharging occurs at 1200 to 1400 rpm. This feature could be corrected to a considerable extent by the use of a variable area turbine nozzle, increasing the availability of the heat in the exhaust gases at low speed.

TABLE X

SUMMARY OF DATA FOR 500 BHP ENGINES AT 3000 RPM

	Compressor								
	Displacement			Centrifugal			Turbo-Charged		
No. of cylinders	6	8	12	6	8	12	6	8	12
Bore, in.	5-5/8	5	4-1/2	5-1/8	4-3/4	4-1/8	5	4-1/2	4
Stroke, in.	6	5-1/2	4-3/4	5-5/8	5-1/8	4-1/2	5-3/4	4-7/8	4-1/4
Arrangement		In-Line			In-Line			In-Line	
Volume, cu ft	98	77	--	66	60	--	58	48	--
Weight, lb	3000	2800	--	2800	2600	--	2750	2550	--
Arrangement		60° Vee			60° Vee			60° Vee	
Volume, cu ft	63	50	47	44	42	33	48	34	30
Weight, lb	2600	2450	2300	2500	2350	2200	2450	2300	2150
Arrangement		90° Vee			90° Vee			90° Vee	
Volume, cu ft	76	58	54	53	48	38	47	38	35
Weight, lb	2800	2600	2450	2600	2450	2300	2550	2400	2250
Fuel used **Battlefield Day,** lb		1384			1263			1213	

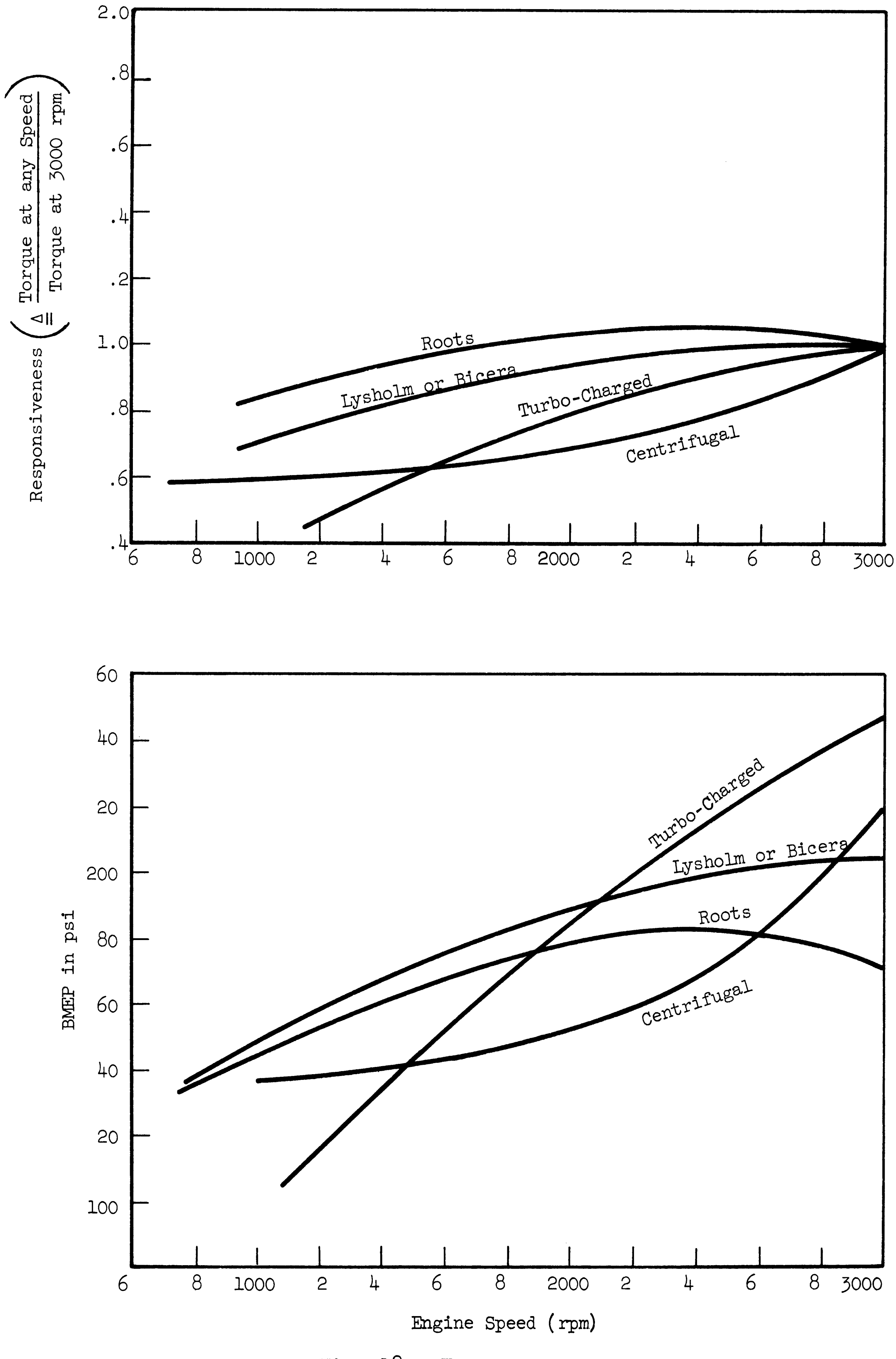

Fig. 18. Torque curves.

CONCLUSIONS

Examination of the data calculated in this report leads to the following conclusions, when a series of engines of similar design fitted with the different types of superchargers are considered.

1. The engine cycle, as represented by the gross IHP at the pistons with no deduction for the compressor, remains constant per lb of air supplied for all types of direct driven superchargers.

2. The engine cycle for a turbo-charged engine with the same manifold pressure as a directly driven engine yields less gross IHP per lb of air due to the higher engine back pressure in the exhaust manifold.

3. Of the compressors examined, the engine performance, on the basis of power output and SFC for the net IHP (engine plus compressor), increases as the isentropic efficiency of the compressor increases.

4. The turbo-charger, involving little if any power loss from the piston unit for compressor drive, provides the best maximum output with the lowest SFC.

5. The various systems arrange themselves in the same order of merit approximately for the part load performance characteristics.

6. Improved efficiency of the units in the case of a turbo-charged engine would only result in very small changes in performance if the inlet manifold pressure remained unchanged (represented by the reduction in back pressure on the engine).

7. Improved efficiency of the turbo units would permit an increase in manifold pressure accompanied by a corresponding increase in the horsepower if operated at the same back pressure as low efficiency units.

8. The direct drive superchargers must use up some of the oxygen supplied to the engine cylinder to provide the power for the compressor.

9. The turbo-charger obtains its power from the heat of the exhaust gases by making it available by increase of back pressure.

10. The turbo-charger employs all of the oxygen supplied to the cylinder in the production of work available to the output. The air flow, and thus engine size and weight, is reduced for a given BHP at any speed.

11. The turbo-charged unit maintains a fairly constant SFC as the pressure ratio is varied.

12. The positive displacement compressor has the advantage of maintaining torque at its maximum value approximately as the engine speed varies.

13. The turbo-charged engine has the least bulk for a given arrangement in most cases, with the centrifugal supercharger a close second.

14. The bulk of the engine plus fuel for a battlefield day is least for a turbo-charged unit.

FUTURE WORK

The above report deals only with the direct driven supercharger geared to the engine through a fixed gear ratio.

Part II of this work will cover the case of similar superchargers driven off the engine in such a manner as to exploit the possibilities of producing engines with the maximum degree of responsiveness.

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