THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Technical Report

THE RESPONSIVE TURBO-CHARGED COMPRESSION IGNITION ENGINE: PERFORMANCE CHARACTERISTICS

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ABSTRACT

The report predicts the capabilities of a compression ignition engine designed to increase torque at low speeds. The design of a suitable variable geometry turbo-charger will allow a torque of almost four times that of a present turbo-charged engine when the engine is operating at half speed.

The turbo-charger under the proposed conditions of operation would not be in the surge state at any time during full load operation, but under part loads could enter the surge condition, though means of preventing this are proposed in the report.

A new approach to the charger design would be necessary to secure high ratio and broad range simultaneously; a two-stage design might even be necessary. The engine development program would not be difficult; only a moderate improvement in present research engine performance would be required, an improvement in the low-speed end of the operating range.

The report indicates that much can be done to obtain better performance from a compression ignition engine for vehicular propulsion. With a modest development program, torque multiplication which will make this engine almost the equivalent of a gas turbine with a free power unit seems within the bounds of possibility. Consequently, it may be possible to eliminate the torque converter from the transmission and thereby simplify the overall power package.

INTRODUCTION

Reference (1) presented the background for the development of a responsive turbo-charged engine. It was established that such an engine would be possible if a certain set of requirements regarding engine operating characteristics were met e.g. BMEP's manifold pressures, etc. Most of the assumptions made depended on relatively minor improvements in present engine and turbo characteristics improvements in fact which have already been claimed for modern research engine developments. Thus the unit proposed is not a development of the distant future, but has a reasonable prospect of success within the next five years or so, provided that the supporting program is adequately and efficiently financed, planned and excuted.

The object of this report is to present calculated performance data for the responsive turbo-charged compression ignition engine outlined in Reference (1). The results presented are for the operating engine and do not include such considerations as cooling losses, transmission, or accessories such as the generator.

The data of this report are to be used in judging the effectiveness of this engine in producing high torques at low speeds, as compared to the conventional torque converter transmission. If the proposed power plant is successful, the torque converter element will no longer be needed for the transmission.

METHOD OF CALCULATION

The performance estimates stated below were developed using methods given in Reference (2). Consequently, the indicated HP, SFC, etc., were converted to a BHP basis. To convert them to a BHP basis, an arbitrary frictional loss was employed. This was determined using a typical FMEP curve (AB given in Fig. 1) for an engine now employed in the type of vehicle under consideration, having approximately the same manifold pressure ratio at full speed as the proposed engine. An estimated increase in mean pressure at lower rpm has been added to the curve. Where the manifold pressure increases above present limits, the tow curves are respectively AB and CB of Fig. 1.

The estimated frictional loss was considered constant at any constant speed regardless of the change in manifold pressure at that speed. This assumption seems to give somewhat pessimistic values at low loads for the speed in question.

The engine conditions for the determination of the performance values were:

Maximum rpm = 2800
Maximum F/A ratio = 0.043
Manifold pressure ratio at full load and speed = 2.5.1
Exhaust pressure - 0.87 x manifold pressure
Air aftercooler temperature = 200°F
Volumetric efficiency = 93.0%

From Fig. 3 of Reference (2), the IMEP expected is 258 psi, and with the frictional loss of Fig. 1, the BMEP is 226 psi. For a 1000 BHP engine at 2800 rpm, the required engine displacement is given by:

$$\text{BHP} = \frac{\text{BMEP x Displacement x rpm}}{792000}$$

$$\text{Displacement} = \frac{1000 \times 792000}{226 \times 2800}$$

$$= \underline{\underline{1258 \text{ cu in.}}}$$

The 1000 BHP at 2800 rpm will be called the maximum BHP at 2800 rpm and considered 100%. All curves will be plotted on a percentage HP basis with the 100% being the 1000 HP at 2800 rpm. Since the change in performance, if any, will be small for outputs of say 700 to 1500 BHP, the curves presented can be employed for power outputs of various magnitudes. Of course the dis-

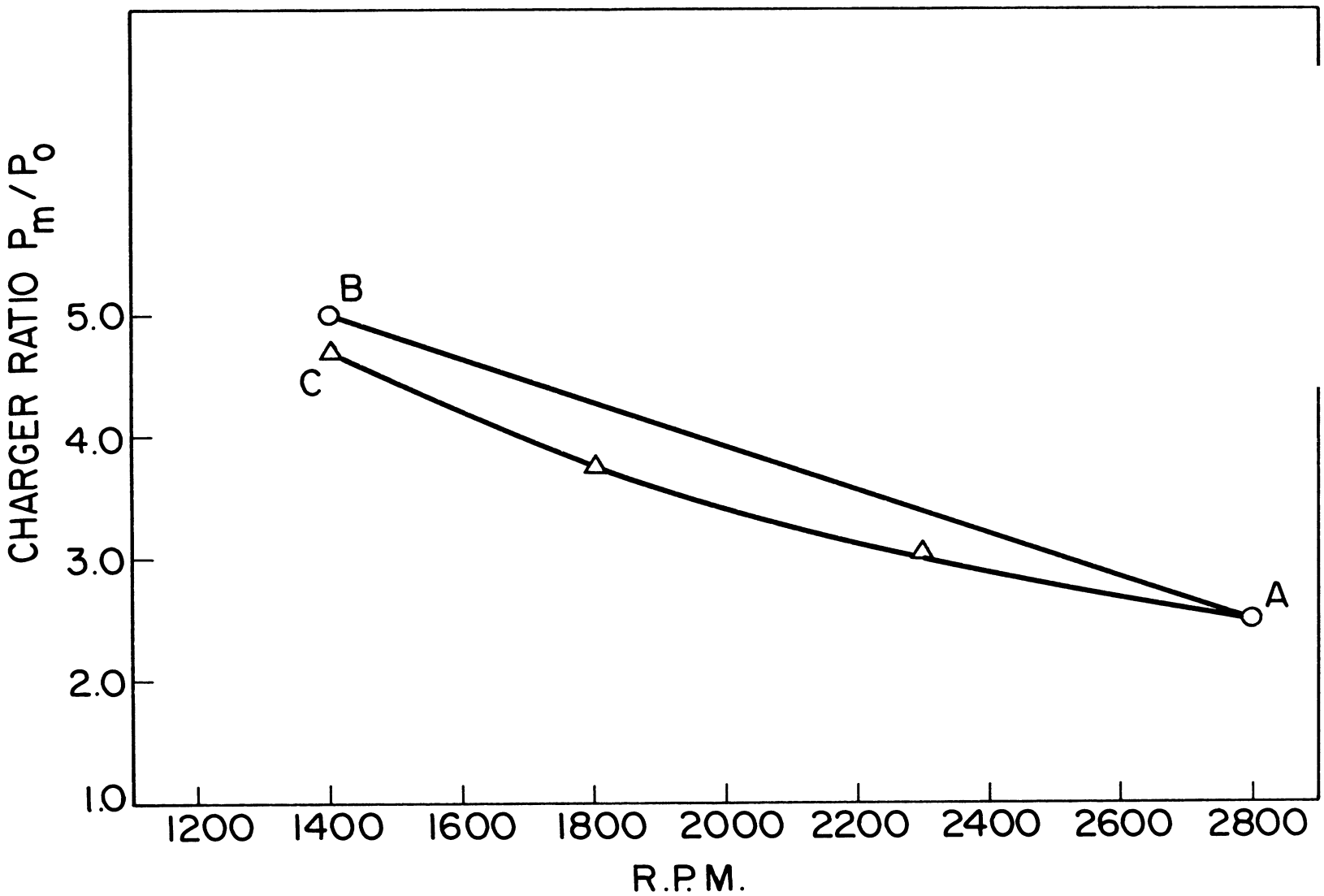
CHARGER RATIO P_m/P_o
5.0
4.0
3.0
2.0
1.0
B
C
A
1200
1400
1600
1800
2000
2200
2400
2600
2800
R.P.M.

Fig. 1

placement will change proportional to the HP as will the air and fuel flow, but the mean pressure, specific fuel flow, manifold pressure, etc., will be shown by the curves. Thus more than one engine size can be considered when necessary.

As indicated in Reference (1), the manifold pressure will be assumed to increase as the speed falls, while the F/A ratio will remain constant at 0.043 for full load at each speed.

In all full load conditions at the various speeds, the exhaust manifold pressure will be 0.87 that at inlet, obtained by adjustment of the variable geometry of the turbine. As load falls for any constant speed, the pressures in both manifolds will vary similar to any fixed geometry turbo-charger since no change in nozzle area is planned for any one speed.

In order to have a starting point for the calculations, it is assumed that a plot of manifold pressure ratio versus rpm will be the straight line AB shown in Fig. 2, starting from a ratio of 2.5:1 at 2800 rpm and reaching the estimated 5.0:1 at 1400 rpm. Because frictional losses decrease with speed, ratios less than that shown by AB probably will be possible eventually. The line AC does in fact result in a constant output of 1000 BHP at all speeds, and will be shown later.

From AB of Fig. 2, the IMEP for each speed and the manifold pressure for a F/A ratio of 0.043 can be obtained, and the expected BMEP developed. Thus the output of the 1258 cu in. engine at each speed at full load is obtained while operating along the line AB of Fig. 2.

The next point considered is the part load operation. In this report half load is considered to mean the conditions under which one half of the full load IMEP is developed for the same speed. The manifold pressure will vary, of course, with the load if no changes are made in the turbo nozzle area. It will change as indicated by Fig. 5 of Reference (2). It was established in Reference (1) that variable geometry is a fundamental feature of the proposed responsive engine; and no reason is seen why almost any nozzle area setting could not be used (within operable limits) as load and speed changes. Such an infinite variety of variables would result in calculations that would be extremely long, but an optimised engine could result. Since at this stage little is known regarding performance characteristics of this method of operation, we propose to determine what might be the least complicated control system still capable of responsive operation. This seems to be variable geometry responsive to speed alone, i.e., there will be a fixed turbine nozzle area for each speed, varying in such a way that the curve of AB of Fig. 1 will be the speed-pressure curve at full load.

It follows that at a constant speed of say 2800 rpm, the nozzle area will remain fixed at one value, and the turbo will behave as any present type of fixed geometry machine for this speed. At some other speed, the

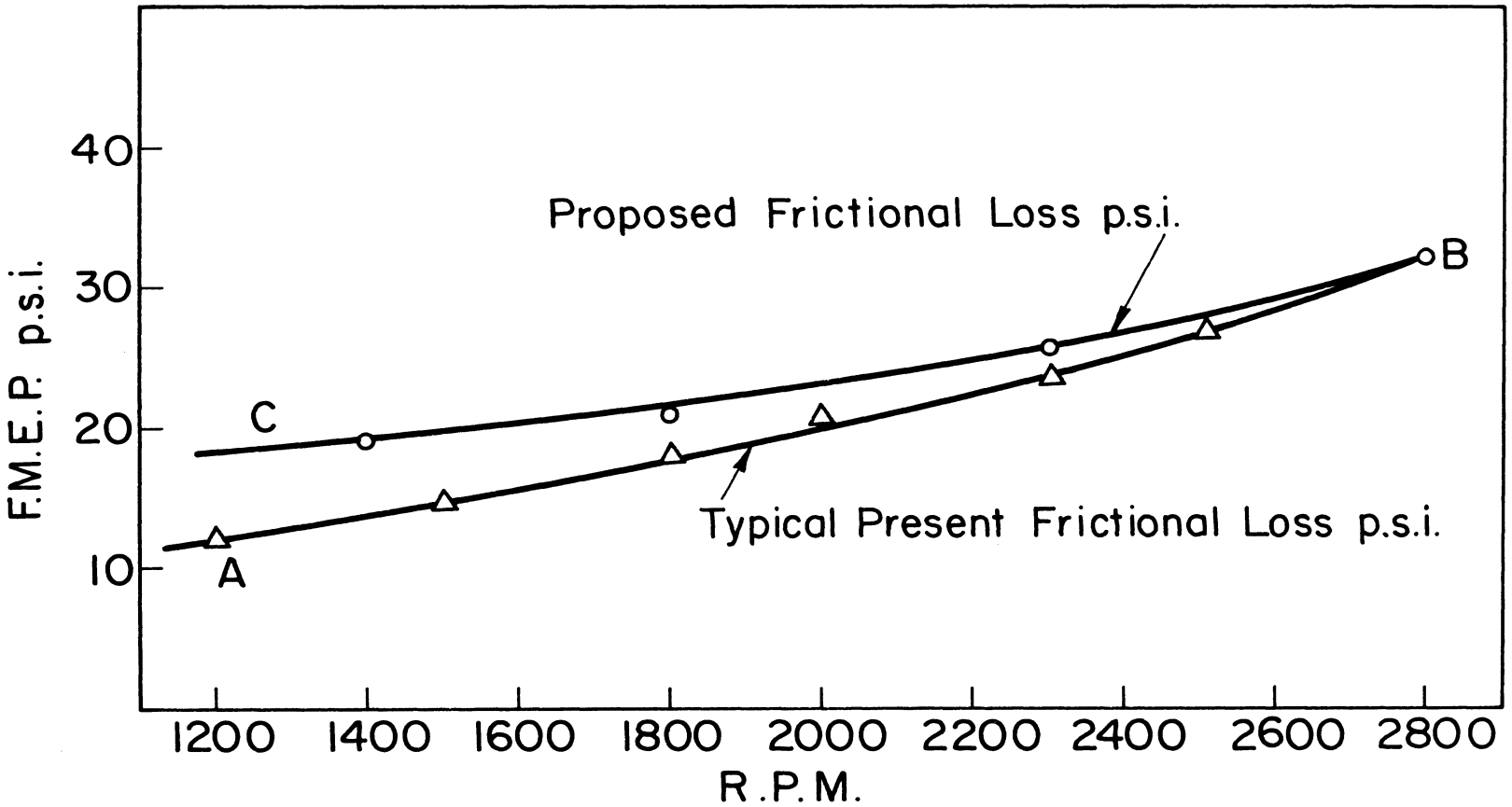
F.M.E.P. p.s.i.
40
30
20
10
Proposed Frictional Loss p.s.i.
Typical Present Frictional Loss p.s.i.
A
B
C
1200
1400
1600
1800
2000
2200
2400
2600
2800
R.P.M.

Fig. 2

area will be different, but again fixed at this value as long as the speed remains constant. In this method of operation, the control problem would be simple. As a result, the part load conditions will be determined for any one speed on the basis of fixed geometry for that speed. Figure 5 of Reference (2) can now be used for the determination of the operating manifold pressure. This method of operation need not give the optimum engine performance, but it will give a method of operation relatively easy to control.

1. TYPICAL CALCULATION

As an example of the methods employed, the following calculation for 50% power at 2800 rpm is presented.

From Fig. 5, Reference (2), the manifold pressure will be 62% of the full load value for a fixed geometry charger at constant speed of 100%. Thus:

$$\text{Manifold pressure} = 14.7 \times 2.5 \times 0.62$$

$$= 22.8 \text{ psi}$$

Frictional losses at 2800 rpm (Fig. 1) = 32 psi

$$\text{IMEP} = 50\% \text{ that at full load}$$

$$= \frac{258}{2} = \underline{\underline{129 \text{ psi}}}$$

$$\text{BMEP} = 129-32$$

$$= 97 \text{ psi}$$

$$\text{BHP} = \frac{\text{P x disp. x rpm}}{792000}$$

$$= \frac{97 \times 1258 \times 2800}{792000}$$

$$= \underline{\underline{432 \text{ BHP}}}$$

$$\text{Air Flow} = \eta_V = 0.93 \frac{22.8 \times 144 \times 1258}{53.34 \times 660 \times 1728}$$

$$= 0.0633 \text{ lbs/cycle of 2 revs}$$

$$= 0.0633 \times \frac{2800}{2}$$

$$= 88.8 \text{ lbs/min}$$

η_V = Volumetric efficiency

Using Fig. 3, of Reference (2), an IMEP of 129.0 psi at Pm = 22.8 is indicated for a F/A ratio of 0.029.

$$\text{Fuel flow} = 0.029 \times 88.8$$

$$= 2.58 \text{ lbs/min}$$

$$\text{Brake specific fuel consumption} = \frac{2.58 \times 60}{432}$$

$$= 0.358 \text{ lbs/BHP/hr}$$

Thus, for the operating speed range of 2800 to 1400 rpm, the following results are obtained:

2. CALCULATED PERFORMANCE

The performance data obtained are given in Tables I through VI. The program was set up for computer operation. Therefore slight differences exist between the hand calculation of the example and the computer values. The latter are recorded in the tables.

The BHP at 0.043 F/A at 2800 rpm was taken as 100% and all other horsepowers were expressed as a percentage of this value. As a result, the 0.25, 0.5, 0.75 and 1.0 load values, which are based upon IHP, vary slightly on a BHP basis due to changes in the frictional losses as speed changes. However, these brake values are still the above fractional load performances, since IHP is the controlling influence upon heat losses, maximum pressure, and other important load fractions.

The above BHP values are the same percentage loads as those of Table I. The same is true of the readings in Tables III to VI.

The data of Tables I to VI have been plotted in Figs. 3 to 7 while in Fig. 8, a cross plot showing HP, SFC, rpm, and manifold pressure ratio has been built up from which all parameters from 1400 to 2800 rpm can be obtained.

TABLE I

BRAKE MEAN EFFECTIVE PRESSURE
psi

1400 rpm		1800 rpm		2300 rpm		2800 rpm	
Load, %	BMEP	Load, %	BMEP	Load, %	BMEP	Load, %	BEMP
110.8	501	122.0	429	117.8	324	100	226
82.0	371	90.0	316.5	86.0	236.5	71.0	161.5
53.2	241	58.1	204.0	54.1	149.0	42.9	97.0
24.5	111	26.0	91.5	22.4	61.5	14.4	32.5

TABLE II

1000 BRAKE HORSEPOWER ENGINE

rpm			
1400	1800	2300	2800
1114	1227	1184	1005
825	905	864	718
535	583	544	431
247	262	225	145

TABLE III

AIR FLOW, lb/min

rpm			
1400	1800	2300	2800
142	157	159	142
117	129	130	116
91.2	100.4	102	91
65.5	72.1	73.0	65.5

TABLE IV

FUEL FLOW
lb/min

rpm			
1400	1800	2300	2800
6.13	6.74	6.82	6.13
4.21	4.63	4.69	4.21
2.64	2.91	2.95	2.65
1.44	1.59	1.61	1.44

TABLE V

BRAKE SPECIFIC FUEL CONSUMPTION
lb/BHP/hr

rpm			
1400	1800	2300	2800
0.330	0.330	0.346	0.366
0.306	0.307	0.325	0.351
0.296	0.299	0.325	0.368
0.351	0.364	0.346	0.599

TABLE VI

MANIFOLD PRESSURE RATIO

1400 rpm		1800 rpm		2300 rpm		2800 rpm	
Ratio	psi	Ratio	psi	Ratio	psi	Ratio	psi
5.0	73.5	4.23	62.9	3.39	22.9	2.5	36.8
4.1	60.3	3.51	51.6	2.78	31.9	2.05	30.1
3.2	47.0	2.75	40.3	2.17	40.9	1.6	23.5
2.3	33.0	1.97	28.9	1.56	49.8	1.15	16.9

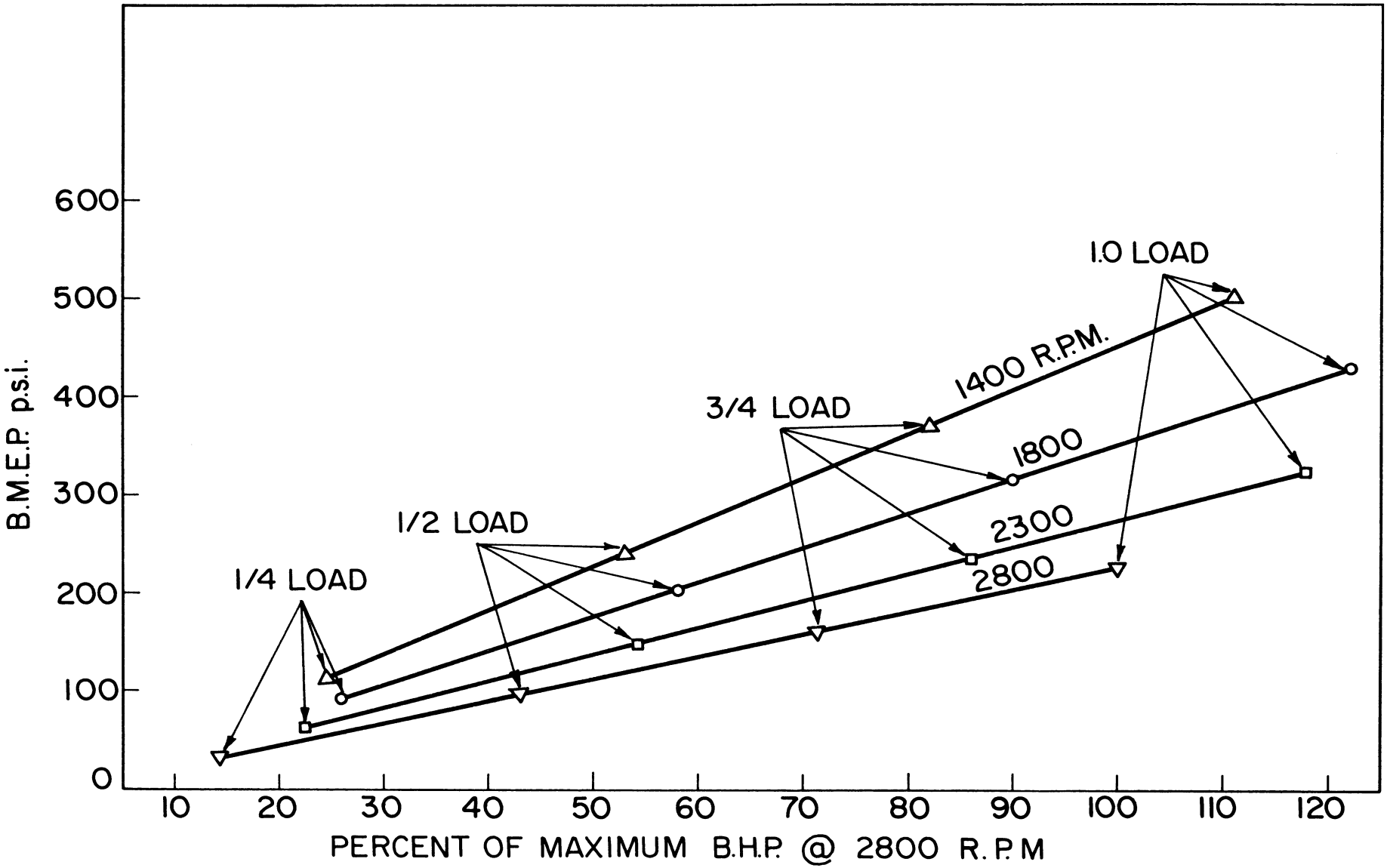
B.M.E.P. p.s.i.
600
500
400
300
200
100
0
10
20
30
40
50
60
70
80
90
100
110
120
PERCENT OF MAXIMUM B.H.P. @ 2800 R.P.M
1.0 LOAD
3/4 LOAD
1/2 LOAD
1/4 LOAD
1400 R.P.M.
1800
2300
2800

Fig. 3

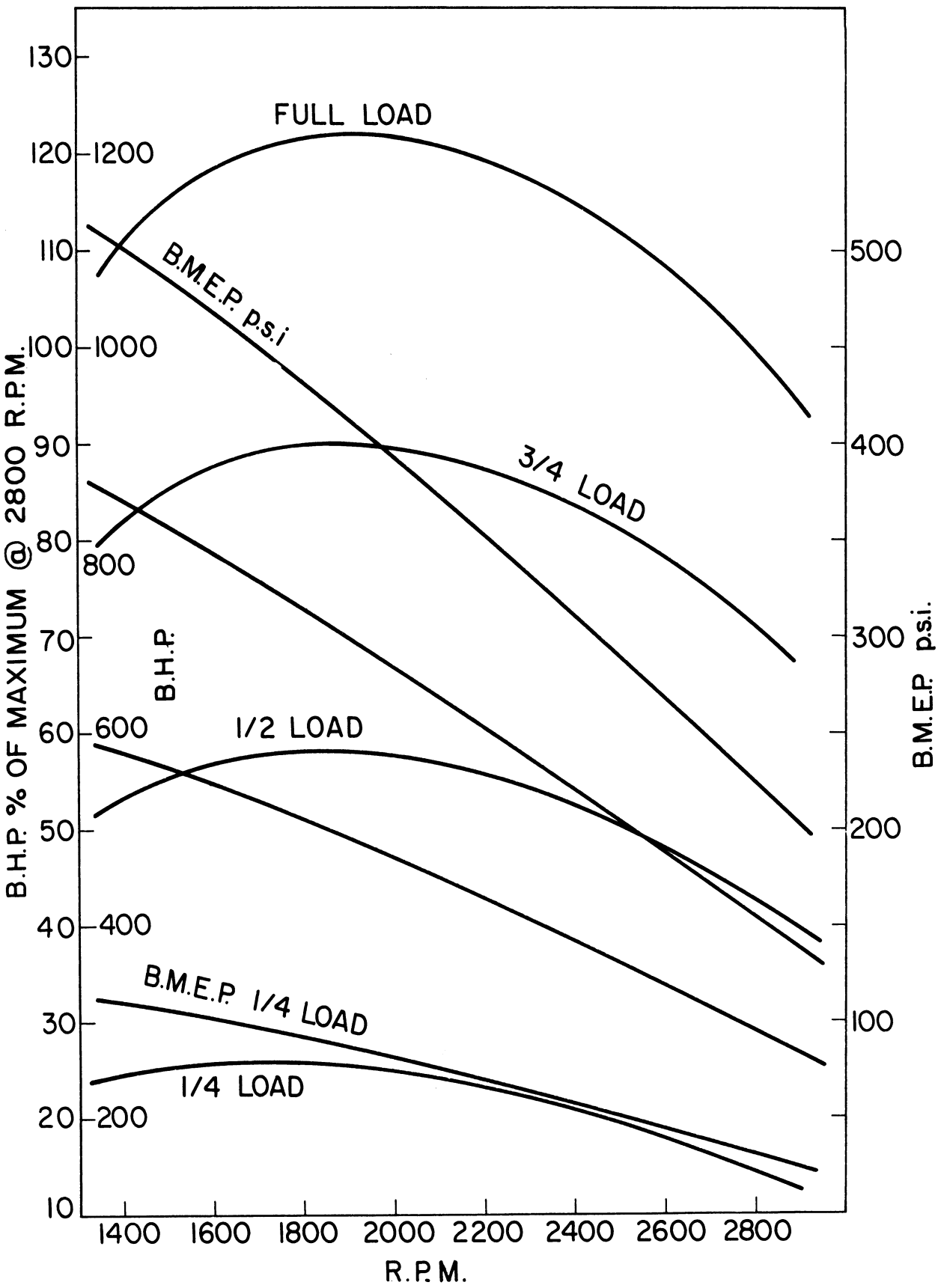
FULL LOAD
B.M.E.P. p.s.i
3/4 LOAD
B.H.P.
1/2 LOAD
B.M.E.P. 1/4 LOAD
1/4 LOAD
B.H.P. % OF MAXIMUM @ 2800 R.P.M.
B.M.E.P. p.s.i.
R.P.M.
130
120
110
100
90
80
70
60
50
40
30
20
10
1200
1000
800
600
400
200
500
400
300
200
100
1400
1600
1800
2000
2200
2400
2600
2800

Fig. 4

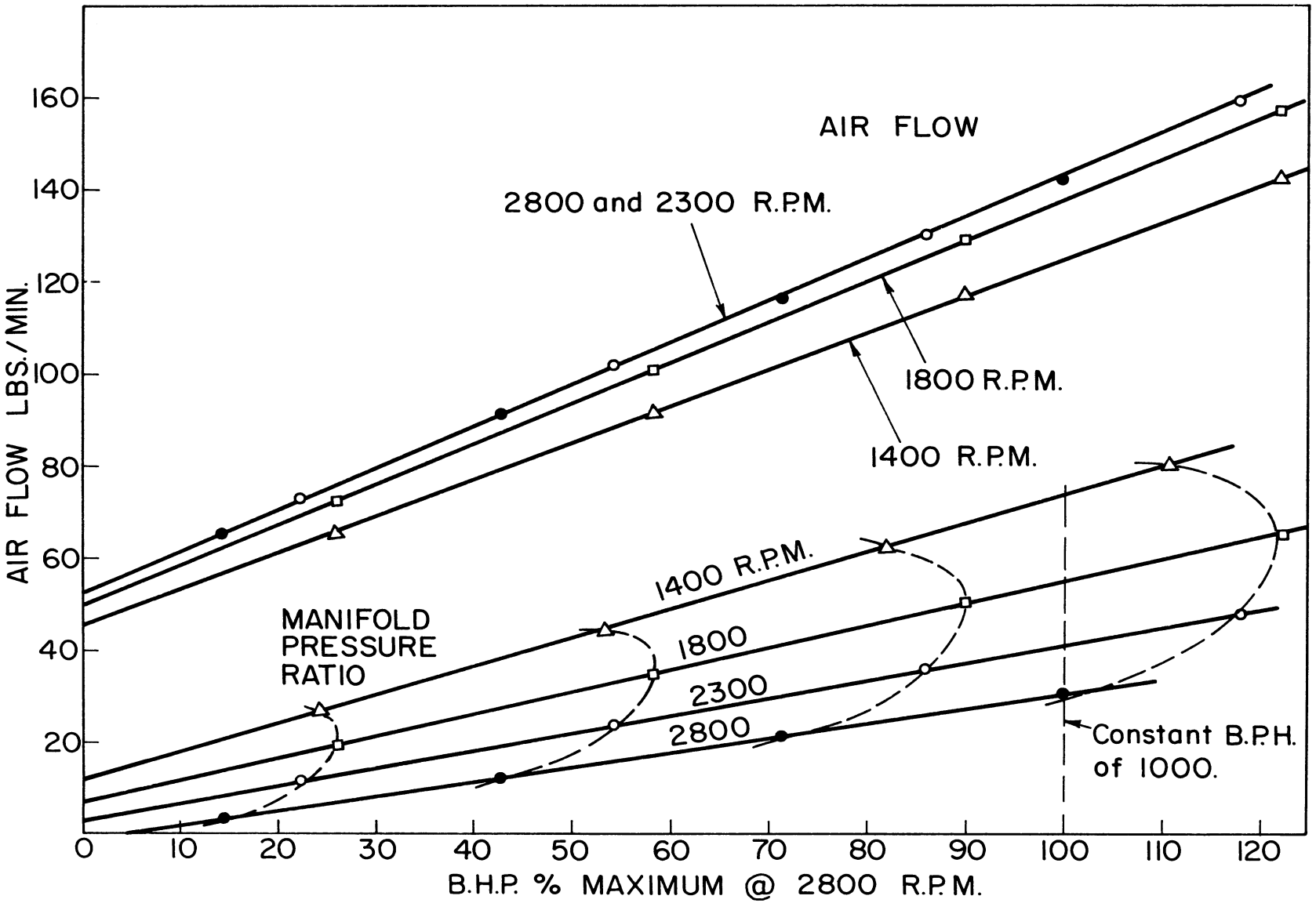
AIR FLOW
2800 and 2300 R.P.M.
1800 R.P.M.
1400 R.P.M.
1400 R.P.M.
1800
2300
2800
MANIFOLD
PRESSURE
RATIO
Constant B.P.H.
of 1000.
AIR FLOW LBS./MIN.
160
140
120
100
80
60
40
20
0
10
20
30
40
50
60
70
80
90
100
110
120
B.H.P. % MAXIMUM @ 2800 R.P.M.

Fig. 5

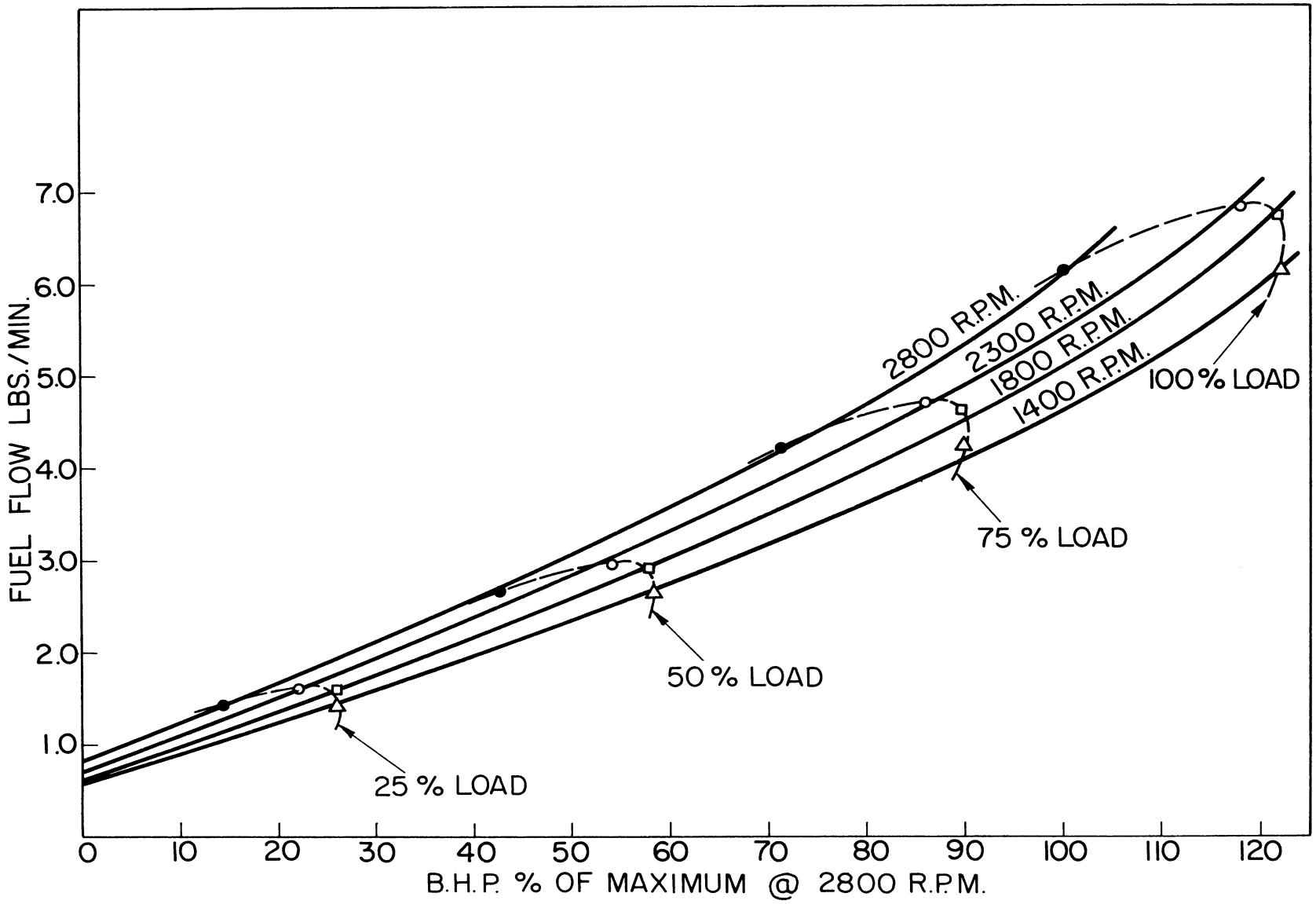
FUEL FLOW LBS./MIN.
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
10
20
30
40
50
60
70
80
90
100
110
120
B.H.P. % OF MAXIMUM @ 2800 R.P.M.
2800 R.P.M.
2300 R.P.M.
1800 R.P.M.
1400 R.P.M.
100 % LOAD
75 % LOAD
50 % LOAD
25 % LOAD

Fig. 6

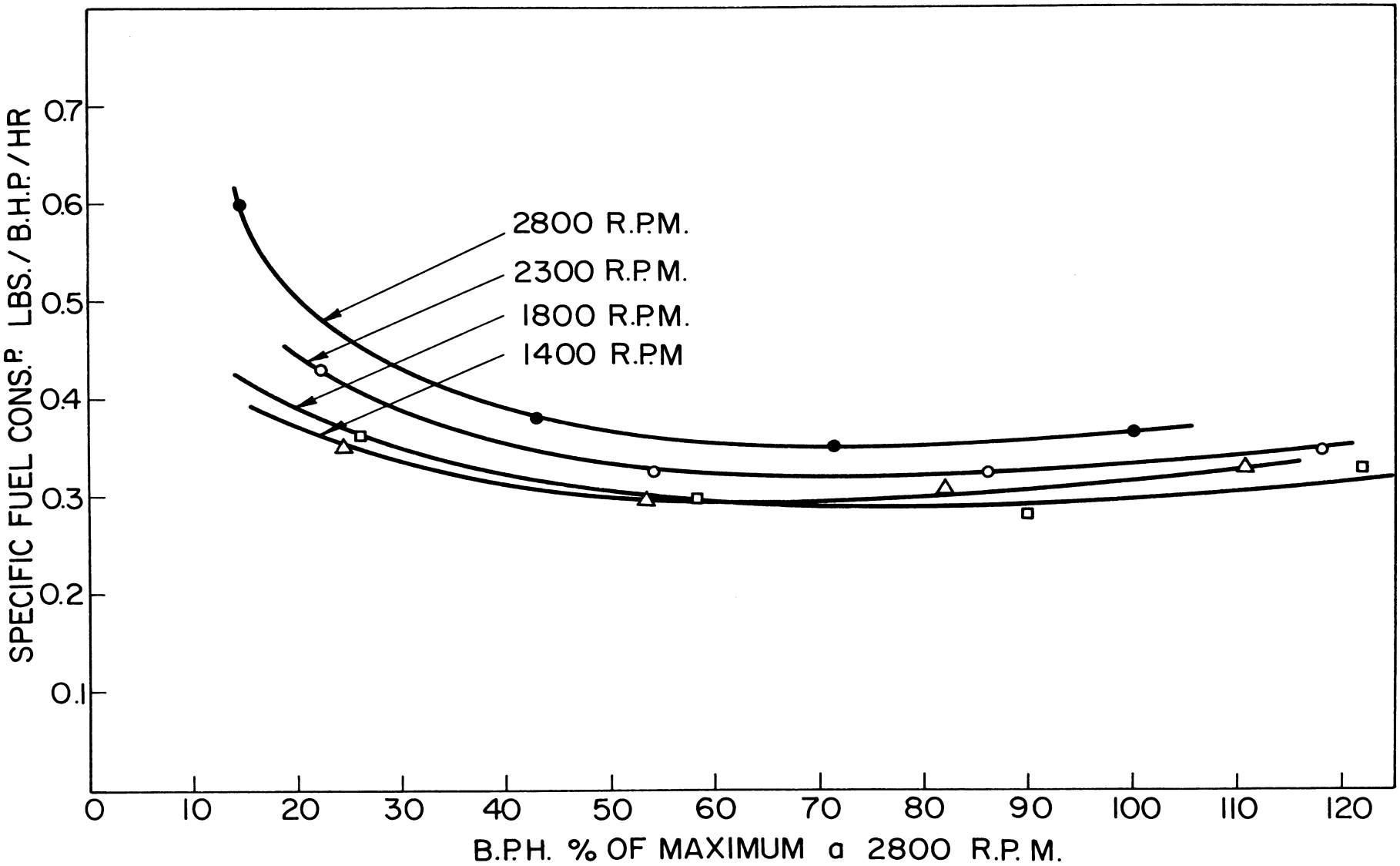
SPECIFIC FUEL CONS.P. LBS./B.H.P./HR
0.7
0.6
0.5
0.4
0.3
0.2
0.1
2800 R.P.M.
2300 R.P.M.
1800 R.P.M.
1400 R.P.M
0
10
20
30
40
50
60
70
80
90
100
110
120
B.P.H. % OF MAXIMUM a 2800 R.P.M.

Fig. 7

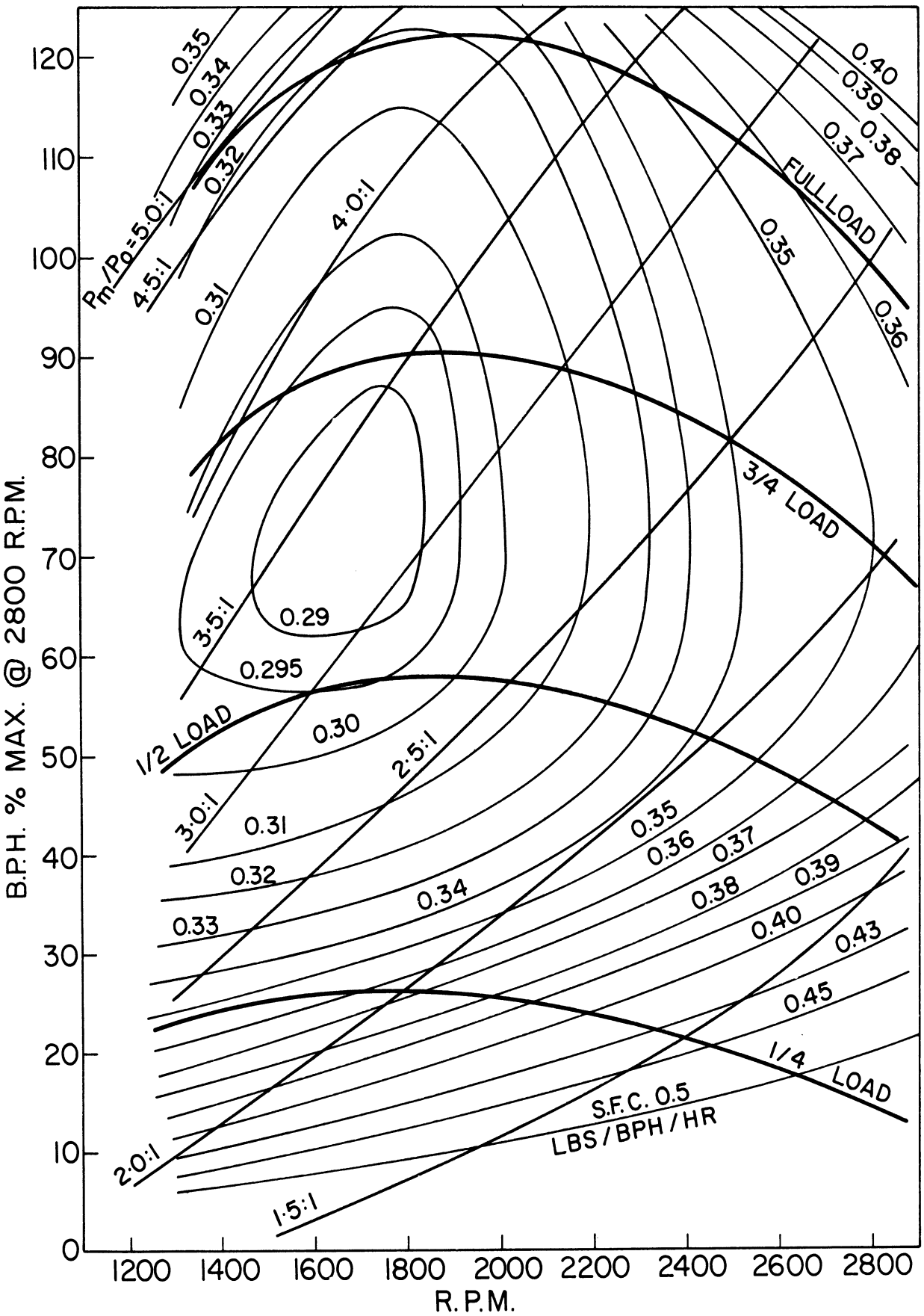
B.P.H. % MAX. @ 2800 R.P.M.
R.P.M.
0
10
20
30
40
50
60
70
80
90
100
110
120
1200
1400
1600
1800
2000
2200
2400
2600
2800
FULL LOAD
3/4 LOAD
1/2 LOAD
1/4 LOAD
P_m/P_a=5.0:1
4.5:1
4.0:1
3.5:1
3.0:1
2.5:1
2.0:1
1.5:1
0.35
0.34
0.33
0.32
0.31
0.29
0.295
0.30
0.31
0.32
0.33
0.34
0.35
0.36
0.37
0.38
0.39
0.40
0.43
0.45
S.F.C. 0.5
LBS/BPH/HR

Fig. 8

DISCUSSION

All data in this discussion applies to an engine cooled by an external source. Results are those of a typical test of a water cooled engine, one in which both oil and water cooling are provided by make-up cold water rather than via radiators and cooling fans as in the installed condition, or they represent tests of an air cooled engine in which cooling air is provided by some external source. Thus, the bare engine dynamometer results are being presented. This places the air and water cooled engine on the same basis, and the one set of values satisfactorily applies to both types of engine.

1. RESPONSIVENESS

The main object of this investigation was to examine the degree of responsiveness obtained, i.e., constant HP with varying speed.

In Fig. 8, the plotted full load line starting at 100% HP and 2800 rpm shows a rise in HP to 122% at about 1900 rpm falling to 110% at approximately 1400 rpm. Thus, the engine is more than responsive. This results from the first choice of manifold pressure change with speed, viz., AB of Fig. 2. It is apparent that the required manifold pressure was overestimated somewhat in the first approach since it was based on IHP. The increased HP at reduced speed is mainly a function of the effect of reduced frictional losses.

From Fig. 5 or 8, it is now possible to read the exact manifold pressure required for each engine speed in order to develop a constant 100% HP at all speeds. The pressure required is shown by AC of Fig. 2, a curved line replacing the straight one originally assumed, with a reduction of manifold pressure ratio at 1400 rpm from 5.0:1 to 4.7:1. This change could be quite an important factor in the design of the turbo-charger.

The torque curves that will result from operation according to AB and AC of Fig. 2 are shown in Fig. 9 together with the BHP's developed. To this diagram has been added AD, a torque curve for a present-day turbo-charged engine of 1000 HP.

There is the possibility of a torque increase of about 3.9:1 at 1600 rpm provided that the assumptions made in this report can be fulfilled. These assumptions will be collected and considered in detail later.

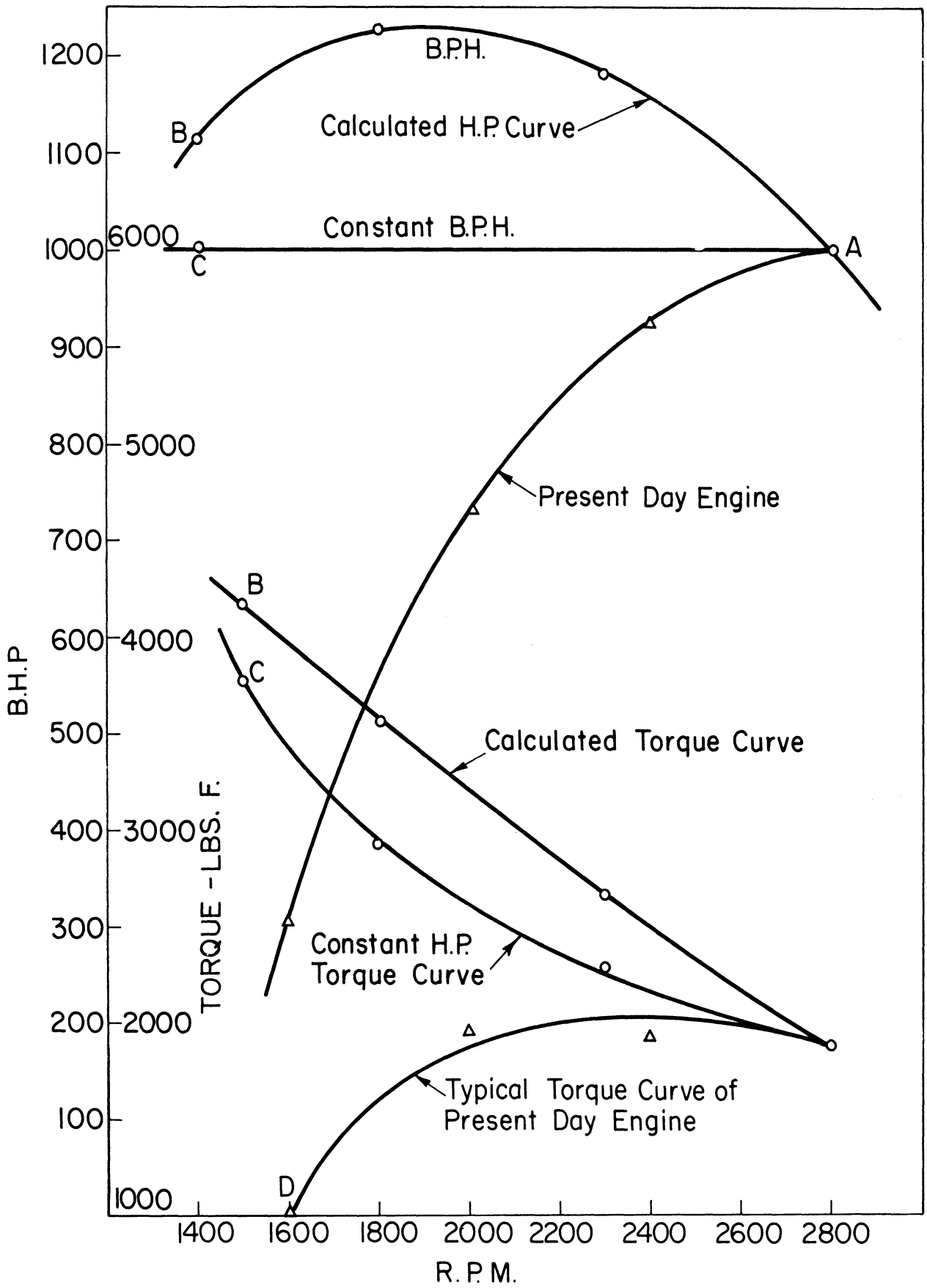
B.P.H.
Calculated H.P. Curve
Constant B.P.H.
Present Day Engine
Calculated Torque Curve
Constant H.P. Torque Curve
Typical Torque Curve of Present Day Engine
B.H.P
TORQUE - LBS. F.
R. P. M.
A
B
C
D
1200
1100
1000
900
800
700
600
500
400
300
200
100
6000
5000
4000
3000
2000
1000
1400
1600
1800
2000
2200
2400
2600
2800

Fig. 9

2. SPECIFIC FUEL CONSUMPTION

Examination of Fig. 8 with reference to SFC indicates that a broad range of powers and speeds exist for moderate changes in the specific fuel consumption. Engine cooling losses are not included, but, allowing 85 HP for this purpose if an air cooled engine is involved, SFC = 0.397 lbs/BHP/hr at full load, and 2800 rpm is obtained changing to about 0.7 lbs/BHP/hr at 100 HP and 2800 rpm. At 1400 rpm the full load fuel consumption will be approximately 0.370 with 0.70 at an output of 60 HP at the same speed. These values assume constant cooling for HP at full load of 85 HP under responsive operation, reducing to 60 HP at the idle condition for the same two speeds.

3. ASSUMPTIONS

In order to produce an engine giving the performance indicated in Fig. 8, the following assumptions were made:

a. A turbo-charger capable of giving a pressure ratio of up to 5:1 was assumed. As pointed out in Reference (1), this was stretching the limits as far as possible. However it has now been shown that the ratio need not exceed 4.75:1 in order to achieve 100% responsiveness over a 2:1 speed range.

The 5:1 ratio can be readily obtained in a two-stage machine. This type would also benefit from a wider flow range without surge, perhaps the deciding factor in selecting the type of charger.

The diagram of Fig. 1, Reference (1), illustrates how closely the above conditions can be approached in the present design.

The employment of high ratio at low speed is not new. It is the combination of low ratio at high and high ratio at low engine speed that is new.

It should be mentioned that the air flow variation between 2.5:1 at 2800 rpm and 5.0:1 at 1400 rpm is negligible; approximately the same mass flow exists under the two conditions. Thus, the impeller passage areas and flow quantities are not as incompatible as may appear at first sight.

b. Efficiencies—A compressor efficiency of 75% was assumed at high and low speeds. Figure 1, Reference (1), shows that this efficiency at a ratio of 4.5:1 was considered possible while at 2.5:1 efficiencies of 78% to 79% were possible for the same mass flow. Reference (1) shows that for turbine efficiency, values of 64% at 2.5:1 and 62.3% at 5:1 would have to be achieved for a self-sustaining charger.

These values can be readily increased. If necessary, and if the turbine efficiency exceeds the above values, compressore values lower than those given can be used.

c. Engine Back Pressure—To maintain the desired conditions, an exhaust manifold pressure of 0.87 that at inlet was employed giving good engine scavenging at TDC of the suction stroke. This value is commonly employed, no difficulty in air flow requirements should be encountered with it. In all present installations both the manifold and the back pressure fall off rapidly as speed is reduced. Eventually the exhaust pressure exceeds the inlet manifold value, and scavernging is lost.

However, is it possible to operate with a back pressure of 4.3:1 at 1400 rpm when the inlet manifold is 5.0:1? There is no difficulty anticipated since the engine (of the four-cycle type) is, in fact, capable of operating as a pump if necessary. This is demonstrated by present engine low-speed manifold pressures when a changeover occurs at low loads.

In this case, the high exhaust pressure will be produced by a low area turbine nozzle, which, due to the use of the variable geometry feature, will open up as speed increases. Scavenging will still be possible at low speed, and there is no reason why good performance cannot be obtained.

As pointed out in item (b), the variation in air flow is actually slight. Thus, the area variation required will not be excessive.

One other factor involved in high back pressure at low speed is leakage of exhaust gases up the stem of the valve. This occurs with the present engines, and will be increased somewhat in the case being investigated. Possibly some form of sealing ring might prove necessary on the value stems at the high pressures being considered.

d. Surge—It has been assumed that surge will not be encountered at any speed with full load. This is a safe assumption according to Reference (1), Fig. 1. The question of surge at other loads cannot be answered until a compressor map has been developed for the conditions of operation used in this analysis. As pointed out in Reference (1), this surge factor could curtail some of the operating conditions given.

4. DEVELOPMENT REQUIRED

Engines producing 350 psi brake mean pressure with manifold pressure ratios of 3.5:1 or better have already been tested at a speed of 2800 rpm. The proposed engine will have a somewhat lower rating at 2800 rpm than the above, but a correspondingly higher one at 1400 rpm. This is not too different from present practice since the higher rating proposed is for a

relatively larger manifold pressure ratio, i.e., the engine will run with a somewhat lower F/A ratio and therefore be cooler than the high output development engines with which comparison can be made. Consequently, a high ratio compressor with an efficiency of 75% or better is the main improvement required. If possible, it should be accompanied by a variable geometry turbine to match. The compressor should have as wide a flow range as possible at all points particularly in the 2.5 to 3.0:1 ratio.

A second consideration will be the operation of the engine at the involved high mean pressures. This may be closer to realization than the compressor.

Thirdly, the cooling problem will need solution in order to meet the needs of the responsive operation at low speed as well as normal operation at the same speed. A variable fan drive ratio or its equivalent may be necessary.

A problem of satisfactory lubrication for conditions to be encountered may also need solution.

The turbo-charger stands out as the critical item requiring development. Without it, performance of the proposed engine is impossible.

The use of some form of variable compression ratio mechanism in the engine would be desirable.

To avoid surge of the compressor under sudden accelerating conditions, some form of by-pass from the compressor to the exhaust pipe may be needed to permit the mass flow (required by the compressor) to occur. The by-pass will be responsive to one of the surge indicators employed in gas turbine operation. Since this by-passing of air would be temporary, its effect upon the performance will be negligible (1) because the compression is already performed by the energy of the exhaust gases, and (2) becuase the compressed air would pass through the turbine and most of its energy would be recovered.

CONCLUSIONS

A responsive turbo-charged compression ignition engine can be developed for a speed range of about 2:1 provided that:

(1) A charger is available to provide a pressure ratio range of about 2.0:1, the higher ratio being at low engine speed.

(2) That the compressor has an efficiency of about 70 to 75% for the extreme pressure ratios, 2.5 and 4.7:1 in the case tested.

(3) That the turbine has an efficiency of about 70% or better in the extreme pressure ratios of 2.5 and 4.7:1.

(4) That the turbine driving the charger has a system of variable blade geometry for the turbine nozzles. In order to maintain good performance at reduced loads and give the desired speed increase of the compressor at low speed and during acceleration, the turbine nozzle variation should be capable of as wide an area range as possible.

(5) That the aftercooler has the capacity to deliver air to the engine at a temperature of about 200°F at all times.

As far as specific fuel consumption is concerned, there is possibly a slight gain in economy at the low speed responsive condition, due to the smaller proportion of power lost in friction.

The torque multiplication achieved by the engine without the aid of any torque converter is of the order of 1.9 to 2.2:1. This is based upon the torque achieved at half speed relative to that at full speed. If it were based upon that of a present turbo-charged engine at half speed, there would be a torque increase of almost 4.0:1 at the same speed.

Response to the engine controls should be improved since higher manifold pressures would then exist at low speeds, and, due to the variable geometry of the turbine, a faster acceleration of the turbine would occur, assisting in the elimination of smoke during acceleration.

GENERAL CONCLUSION

Although the proposed scheme may prove unacceptable for one reason or another, there is one important general conclusion which may be applied to all types of turbo-charged engines:

There is still considerable improvement possible in the performance of a compression ignition engine used for vehicular purposes if the energy capabilities of the exhust gases are employed appropriately.

REFERENCES

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