THE UNIVERSITY OF MICHIGAN

COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Progress Report

PRESENTATION OF ENGINE RESPONSIVENESS AND TYPICAL PERFORMANCE AND DATA

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INTRODUCTION

The work to date has been a search for a method of representing responsiveness and means of estimating typical engine performance under all conditions both now and in the foreseeable future.

This report presents the curves produced and the methods of use.

RESPONSIVENESS

The definition of responsiveness employed has been "that range of speed over which a constant horsepower output can be obtained."

Looking at the fundamental equations for horsepower in terms of engine parameters, we have

$$hp = \frac{PLAN}{33000} = \frac{TM \times 2\pi \text{ rpm}}{33000}$$

where

P = Mean effective pressure, psi
L = Length of stroke of piston, ft
rpm = Revolution per minute
A = Area of piston in square inches
N = Number of explosions per minute
$= \dfrac{n \times rpm}{2}$ for 4-cycle engines
= n x rpm for 2-cycle engine
n = Number of cylinders
T = Turning moment in lb-ft

Substituting for L and A the stroke in inches and area in terms of the piston diameter D, we get for 4-cycle engines:

$$hp = \frac{PL\,\pi D^2 \,n \text{ rpm}}{12 \times 4 \times 33000 \times 2} = \frac{TM \times rpm}{5255}$$

$$= \frac{P \times \text{Displacement} \times rpm}{792000} = \frac{TM \times rpm}{5255}$$

$$\text{Engine Displacement} = \frac{\pi D^2}{4} \times L \times n \text{ cubic inches}$$

There are an infinite number of combinations of P, displacement, and rpm; however P has a limited range and is a well-known factor for engine performance. If the hp is obtained for various values of P for any desired range of rpm for 1 cu in. of displacement, the hp/cu in. versus rpm and P can be plotted as in the left-hand side of Fig. 1a.

Now add to this a graph for various values of the displacement (the right-hand side of Fig. 1a) and the hp output is obtained for any engine in the range covered, both IHP or BHP depending upon whether P is the indicated or brake mean effective pressure. If to this diagram is added a torque curve in lb-ft/cu in. of displacement, the total responsiveness of the engine can be represented. To the torque curve can be added MEP lines as shown obtained from

$$\frac{TM_1 \times rpm}{5255} = \text{hp per cu in.}$$

where TM_1 = Turning moment in lb-ft/cu in. of displacement

Thus

$$\frac{TM_1 \times rpm}{5255} = \frac{P \times rpm}{792000}$$

$$TM_1 = \frac{P}{150.8}$$

Figure 1b shows such a torque curve.

It is now possible to represent any engine performance in terms of its responsiveness since responsiveness is a constant hp for a variable rpm. It follows that it is also a constant hp/cu in. for variable rpm and a line such as AB of Fig. 1a represents a completely responsive engine developing 1.0 hp/cu in. from 1950 to 3950 rpm. It is immediately apparent that, to operate with this degree of responsiveness, the engine must develop 400 BMEP at 1950 rpm, falling off to 200 BMEP at 3950 rpm, the reverse of the usual trend where maximum mean pressure is usually near maximum rpm.

In Fig. 1a a typical performance curve of one version of the Continental 1790 engine has been plotted (the line CDE). Examining this for responsiveness, it is seen that at a rating of 150 MEP at 2250 rpm, the responsiveness is zero since this rating of 0.425 hp/cu in. can be obtained only at one rpm. If the rating is changed to 0.25 hp/cu in., the engine is responsive from 1350 rpm to 2400 rpm, assuming that 2400 rpm is the maximum allowable engine speed. Thus the degree of responsiveness of any engine under all operating conditions can be seen. It is also plain to see what sacrifice of maximum rating is necessary to achieve any required degree of responsiveness in a given engine.

It is also possible to obtain the various engine parameters—bore stroke rpm, etc.—from this diagram to meet any proposed specification and thus visualize rapidly the various possibilities available.

TYPICAL PERFORMANCE CHARACTERISTICS

To predict what changes would result in vehicle performance from any proposed set of engine operating conditions that could conceivably be attained within the next decade, a means of prediction having a reasonable practical significance is necessary.

The usual method of air cycle calculations are so far from actual conditions that the results obtained have little practical application. An improvement is the usual compression and combustion charts (such as those by Hottel). When these are employed a closer approach to practice is obtained, but again significant differences exist in some very important items such as SFC, MEP, and maximum pressure.

To approach the actual engine performance more closely, the writer developed a means of cycle calculation allowing for the heat losses of the engine, a factor neglected in all the above-mentioned theoretical approach (Ref. 1). The method presented in Ref. 1 was changed slightly to apply it to the compression ignition cycle, and then used to examine the effect of increasing supercharge, compression ratio, fuel-air ratio, maximum pressure, etc., on the engine performance characteristics. These were then compared with known engine performance figures, where available, and a satisfactory agreement was found (in general with not more than $\pm 5\%$ variation). This is considered to be good, particularly when the wide range of variations involved was considered.

The cycle calculations were obtained over a wide range of manifold pressures employing turbo-charging, where the inlet manifold pressure was maintained slightly greater than the exhaust manifold pressure. With the high-pressure ratios examined, the air-inlet temperatures would be very high and thus the power outputs would suffer. Also, an increasing heat load of the cylinder would result. To eliminate this effect, an intercooler was employed. It was assumed that the manifold temperature could be maintained at 200°F at all times by this means.

To reduce the calculations and graphs to a reasonable amount, the maximum performance under varying rpm conditions only was aimed for, since the feature of responsiveness is that available at maximum load with various speeds;

part load responsiveness is easily achieved at all times. The next problem was that of presentation; if straight-line relationships are possible, extrapolation is possible with reasonable accuracy. Secondly, straight-line plots are easily drawn and in general represent a wider range of operation with smaller errors. Thus an attempt was made to obtain straight-line relationships as far as possible.

Examination of a number of compression-ignition engine performance curves resulted in the following type of presentation.

FUEL-AIR RATIO VERSUS IMEP

Many of the compression-ignition cycle performance factors become straight lines when plotted on a graph of heat input versus indicated values of MEP, hp, etc.

To cover the widest range of performance factors possible, the effect of fuel-air ratio, was considered most important. This is the one factor that is most difficult to plot in straight-line form. After a number of tries, the method of Fig. 2 was found to be quite accurate for the average direct injection engine.

Figure 2 is a plot of the IMEP versus the log of the F/A ratio. This plot gave a straight-line relationship up to quite high F/A values when the theoretical calculations were plotted, these have of course the minimum possible specific fuel consumption. However, as the F/A increases in a practical engine, a point is reached where effective mixing of the air and fuel begins to fail and departure from the straight-line law begins. This can be considered the point where the engine is likely to show some slight discoloration of the exhaust gases, which eventually develops into heavy smoke as F/A increases.

The point at which this occurred was somewhat variable between different engines, as would be expected, but a good average representation could be obtained if slight departure began at F/A = 0.03 and increased as F/A further increased. The result is a curve tangential to the straight line at F/A = 0.03 and reaching 85% of the IMEP predicted by the straight-line relation at a F/A of 0.055, as shown in Fig. 2. At this point considerable smoke would exist.

The problem is now to determine the IMEP and manifold pressure relationship.

INDICATED MEAN PRESSURE VERSUS MANIFOLD PRESSURE

As indicated earlier, a cycle analysis involving heat losses was employed to determine the IMEP to be expected for each compression ratio, F/A, manifold pressure, etc., when intercooled to a temperature of 200°F. As was to be expected, the results obtained are mainly proportional to the manifold pressure since the air density is proportional to pressure in this case. True, the effects of compression ratio are not linear, but these variations are of minor importance when compared to the manifold pressure at the high ratio employed in this type of engine. Figure 3 is a plot of these calculations. The average results cluster about a line through the origin, the band on each side of the line being the variations arising from compression ratio, etc. It is seen that, for a first approximation, the expected results can be read from the line OA of Fig. 3. This line represents about the ideal performance as determined by calculation for a fuel-air ratio of 0.0473 lb of fuel per lb of air and and ideal SFC can be determined for this condition. In practice the indicator diagram differs from this ideal slightly, which in turn affects the SFC also. Thus in Fig. 4 an expected performance line is drawn, allowing for these factors. This line represents about the maximum performance both in mean pressure and SFC to be expected.

Now if insufficient turbulence, poor injection characteristics, etc., exist, this line will only be approached; hence the addition of the other lines shown in Fig. 4, with increasing SFC proportional to the loss. It is thus possible to correct the performance according to the desired degree of ultimate performance aimed for. It is believed that the original line for an SFC of 0.25 lb/I hp/hr should be capable of being reached, given sufficient control of the combustion conditions,such as turbulence, period, and rate of fuel injection, absence of air pockets etc., in the combustion chamber. Engines of the slow- and medium-speed classes do give such performance when well worked out.

By combining Fig. 2, the F/A ratio effect, with the manifold pressure effect, Figs. 4a and 4b, a set of curves are produced from which the performance of a supercharged compression ignition engine can be predicted within a reasonable percentage variation of the mean for all values of the pressure ratio of the charger up to about 7:1. This upper limit of power is considered to be well above that which can be expected to be employed in the near future, but it was taken this high since experimental gasoline engines have been operated as high as 750 IMEP.

It is believed that, provided the F/A ratio employed is maintained less than 0.055 and greater than 0.015, the predicted results will be of a sufficient degree of accuracy for general comparative purposes. One additional refinement can be introduced, if necessary, by the use of a slight variable expected performance line. This can be visualized as a line from a point B at the 0.03 F/A ratio values of IMEP versus F/A ratio graph transferred to the expected performance line of Fig. 4; then a straight line crossing over to slightly higher fuel values as minimum charger pressure ratio is reached and corresponding to about 0.3 to 0.31 SFC at maximum P_m/P_a and F/A = 0.473. This would make due allowance for the fact that high-duty engines are in general being pushed to the limit at their maximum power at maximum rpm conditions, and thus do not achieve the best possible SFC for that state. If a 3.4:1 pressure ratio were being employed, such a line can be represented by BB located as shown, with a reduced performance at the 3.4:1 ratio amounting to about a 10 to 15% increase in SFC at the maximum outputs. The intersection of BB with the various pressure-ratio lines will determine the manifold pressures required to produce the IMEP's indicated.

From the two graphs of Fig. 4, it is possible to predict the indicated performance of any engine and convert it to the expected BHP output basis via the use of reasonable mechanical,volumetric,efficiency assumptions over the speed range. The volumetric efficiency determines the engine displacement volume requirements for the BHP and the mechanical efficiency converts the indicated SFC to the BHP output basis.

These diagrams will be employed to predict the requirements for responsive engines under the various conditions to be assumed at a later date.

EXAMPLE

An engine of 500 hp is required at 2800 rpm having a BMEP of 250 psi. 250 BMEP at 90% mechanical efficiency for a high supercharge ratio means an IMEP of about 277 (the mechanical efficiency should be varied depending on what auxiliaries are driven off the engine). Figure 4a indicates that a charger pressure of about 77 in. Hg will be required at an F/A ratio of 0.0475 for moderate smoke; that is, $P_m/P_a = 2.57$, say 2.6 is required, and an SFC of 0.25 x 1.15 = 0.29 approx. is about the best to be expected, or 0.32 lb/B hp/hr. No allowance has been made here for cooling fan, generators, etc. The interesection of the 2.6:1 pressure-ratio line of Fig. 4b with the SFC of 0.29 gives 290 IMEP, a reasonable agreement with the previous figure of 277. The 0.03 F/A line from Fig. 4a gives an IMEP of 215 psi approx. at its intersection with 77 in. Thus 215 IMEP at a minimum SFC of 0.25 can be expected. Therefore points C and C of Fig. 4b are determined and line OCC represents the approximate output curve versus manifold pressure. In redrawing the lines these would be curved slightly. The corresponding SFC rates can be read from the interesection of the ratio lines in Fig. 4b. The performance at any constant manifold pressure, up to the 77 in., can be predicted from Fig. 4a in a similar manner.

The engine size for 500 hp is determined from Fig. 1 since with 250 BMEP at 2800 rpm about 575 cu in. of displacement is required in a 4-cycle engine. This can now be divided between any number of cylinders considered desirable.

REFERENCE

1. E. T. Vincent, Supercharging the Internal Combustion Engine, McGraw-Hill Book Co., 1948.

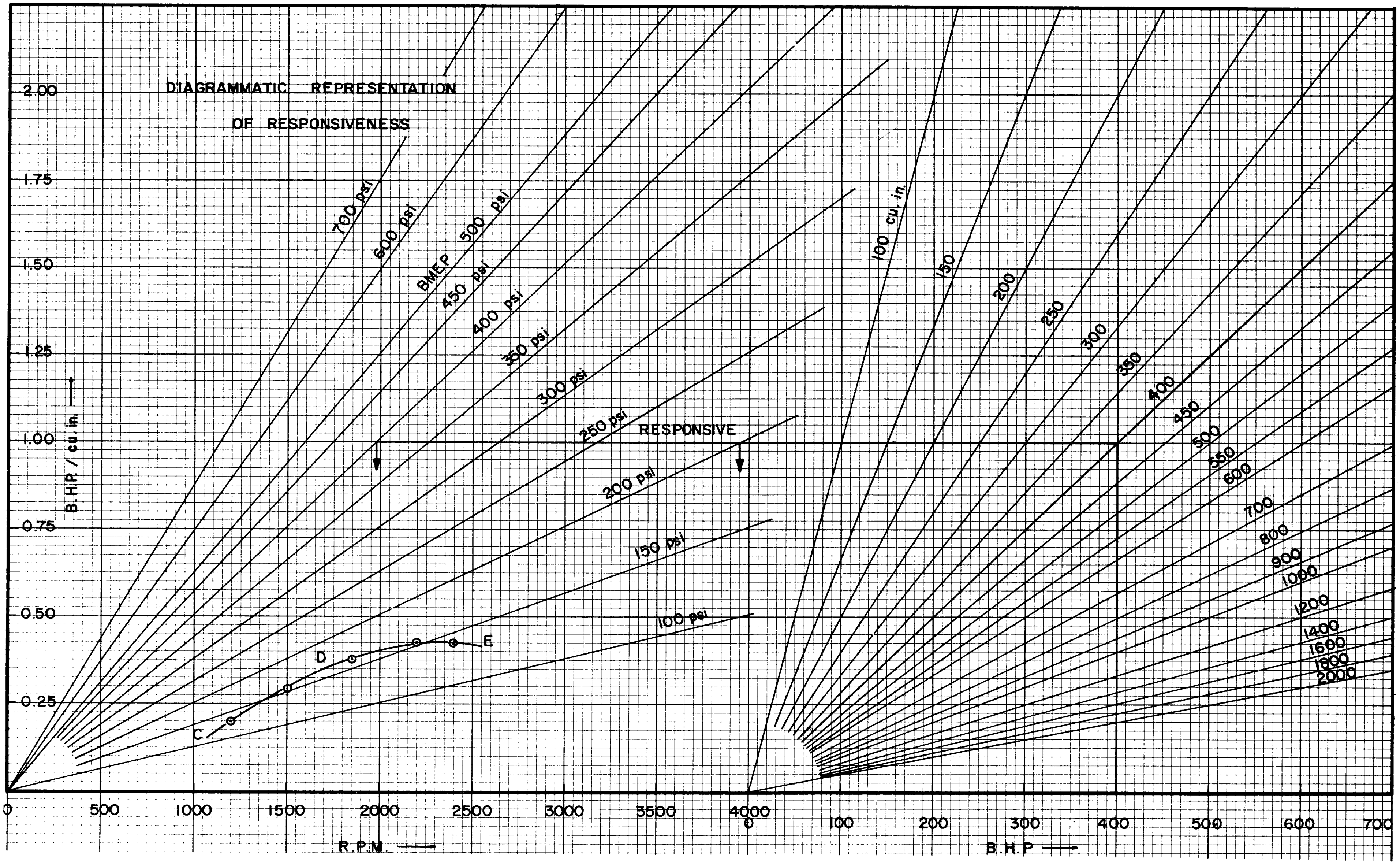
DIAGRAMMATIC REPRESENTATION
OF RESPONSIVENESS
B.H.P. / cu. in.
R.P.M.
B.H.P
BMEP
700 psi
600 psi
500 psi
450 psi
400 psi
350 psi
300 psi
250 psi
200 psi
150 psi
100 psi
RESPONSIVE
100 cu. in.
C
D
E

Fig. 1a

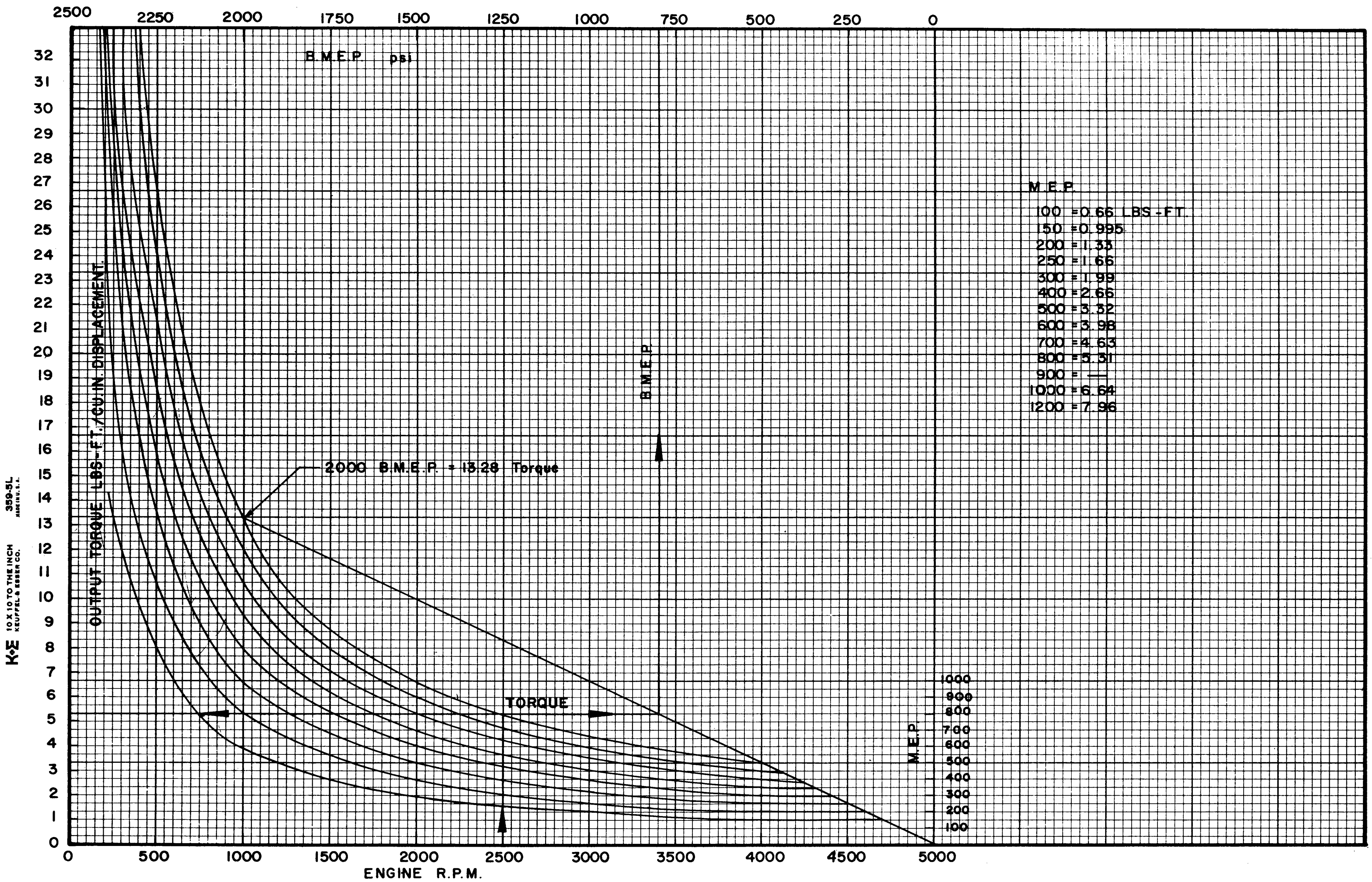
B.M.E.P. psi
2500
2250
2000
1750
1500
1250
1000
750
500
250
0
OUTPUT TORQUE LBS-FT./CU.IN. DISPLACEMENT.
2000 B.M.E.P. = 13.28 Torque
B.M.E.P.
TORQUE
M.E.P.
100 = 0.66 LBS-FT.
150 = 0.995
200 = 1.33
250 = 1.66
300 = 1.99
400 = 2.66
500 = 3.32
600 = 3.98
700 = 4.63
800 = 5.31
900 = —
1000 = 6.64
1200 = 7.96
M.E.P.
1000
900
800
700
600
500
400
300
200
100
ENGINE R.P.M.
K+E 10 X 10 TO THE INCH
KEUFFEL & ESSER CO.
359-5L

Fig. 1b

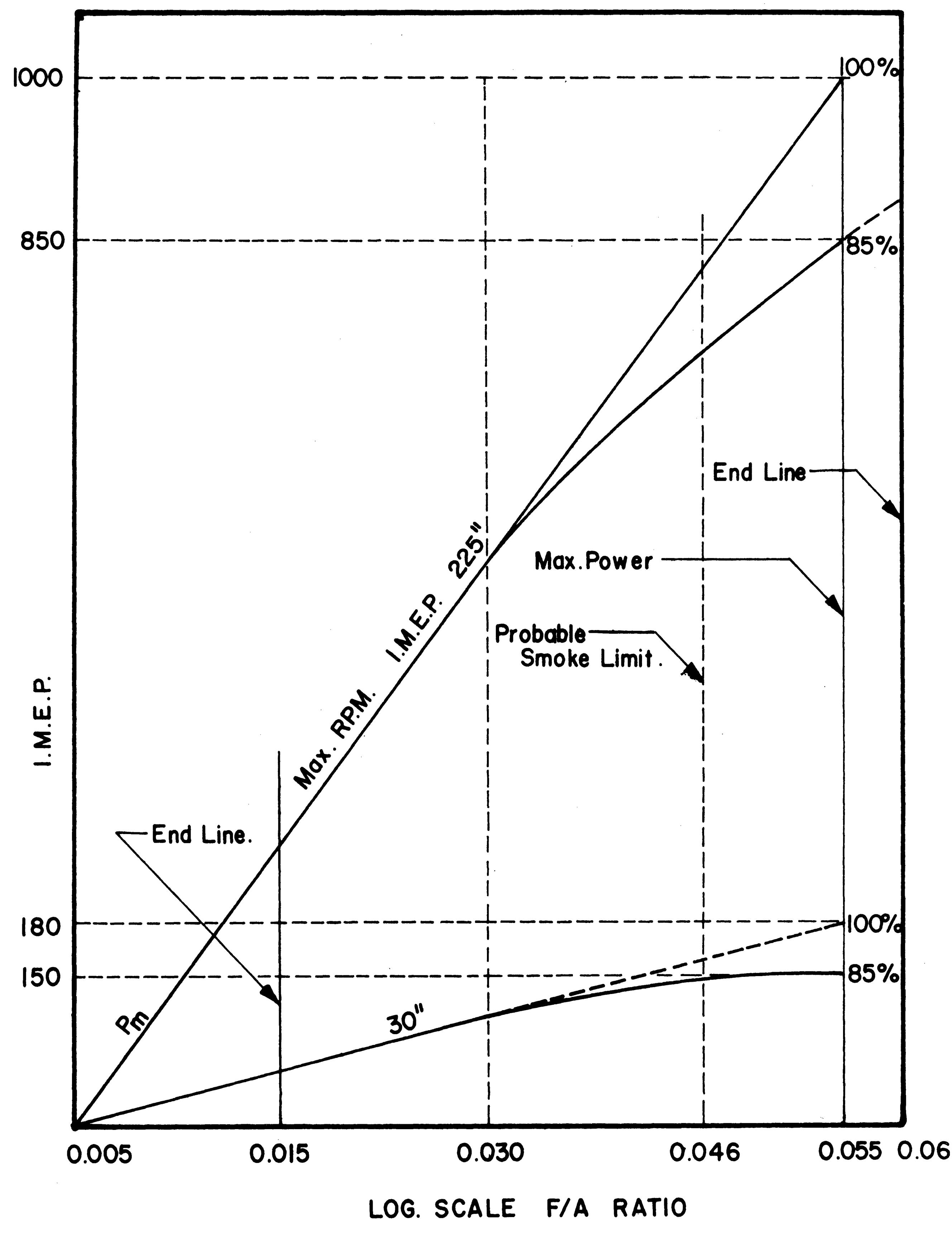

Fig. 2. Plot of IMEP vs. log of F/A ratio.

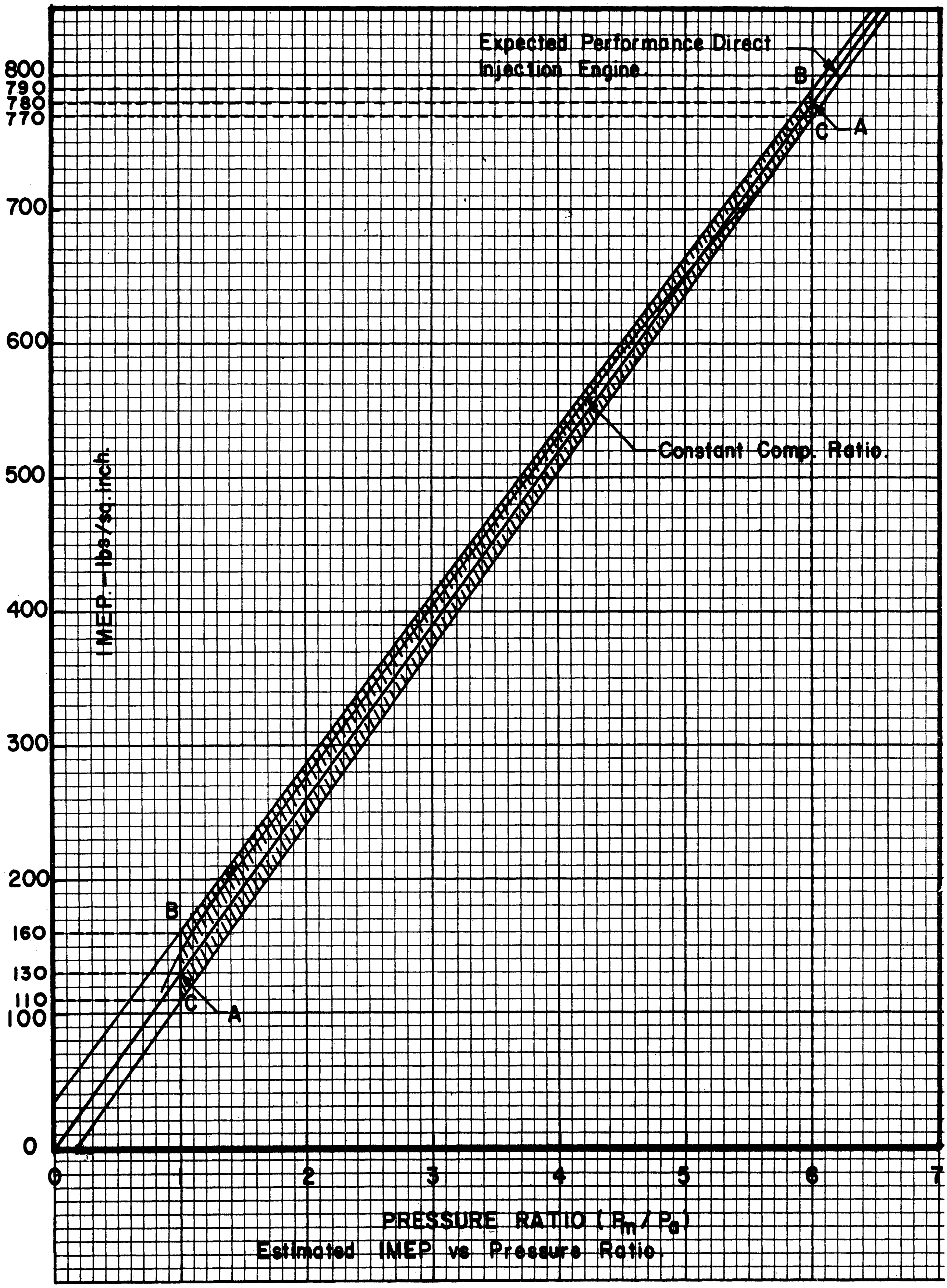
Expected Performance Direct Injection Engine.
Constant Comp. Ratio.
IMEP.—lbs/sq.inch.
800
790
780
770
700
600
500
400
300
200
160
130
110
100
0
B
C
A
0
1
2
3
4
5
6
7
PRESSURE RATIO (P_m / P_a)
Estimated IMEP vs Pressure Ratio.

Fig. 3

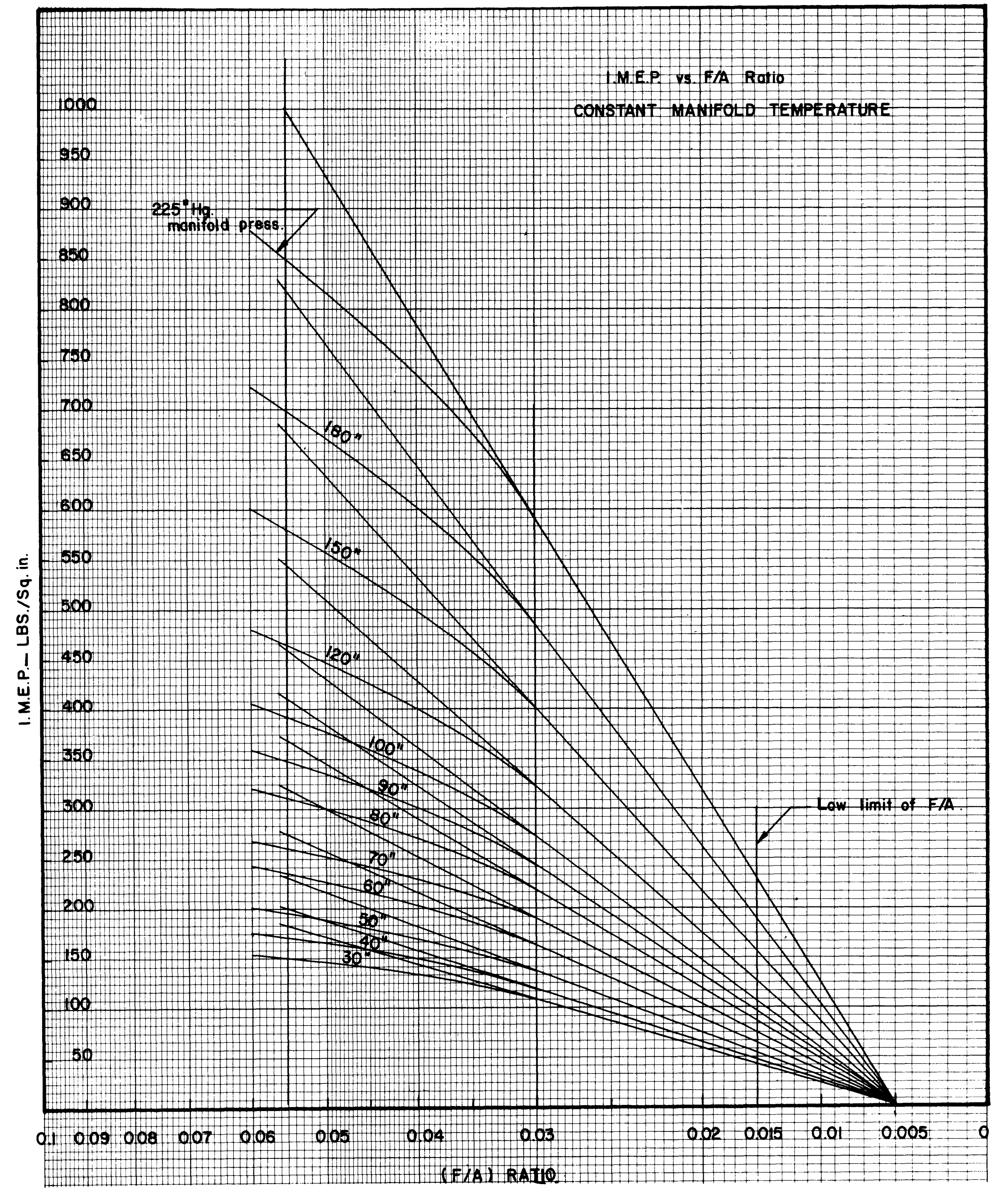
I.M.E.P. vs. F/A Ratio
CONSTANT MANIFOLD TEMPERATURE
225" Hg. manifold press.
180"
150"
120"
100"
90"
80"
70"
60"
50"
40"
30"
Low limit of F/A
I.M.E.P.— LBS./Sq. in.
1000
950
900
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.015
0.01
0.005
0
(F/A) RATIO

Fig. 4a

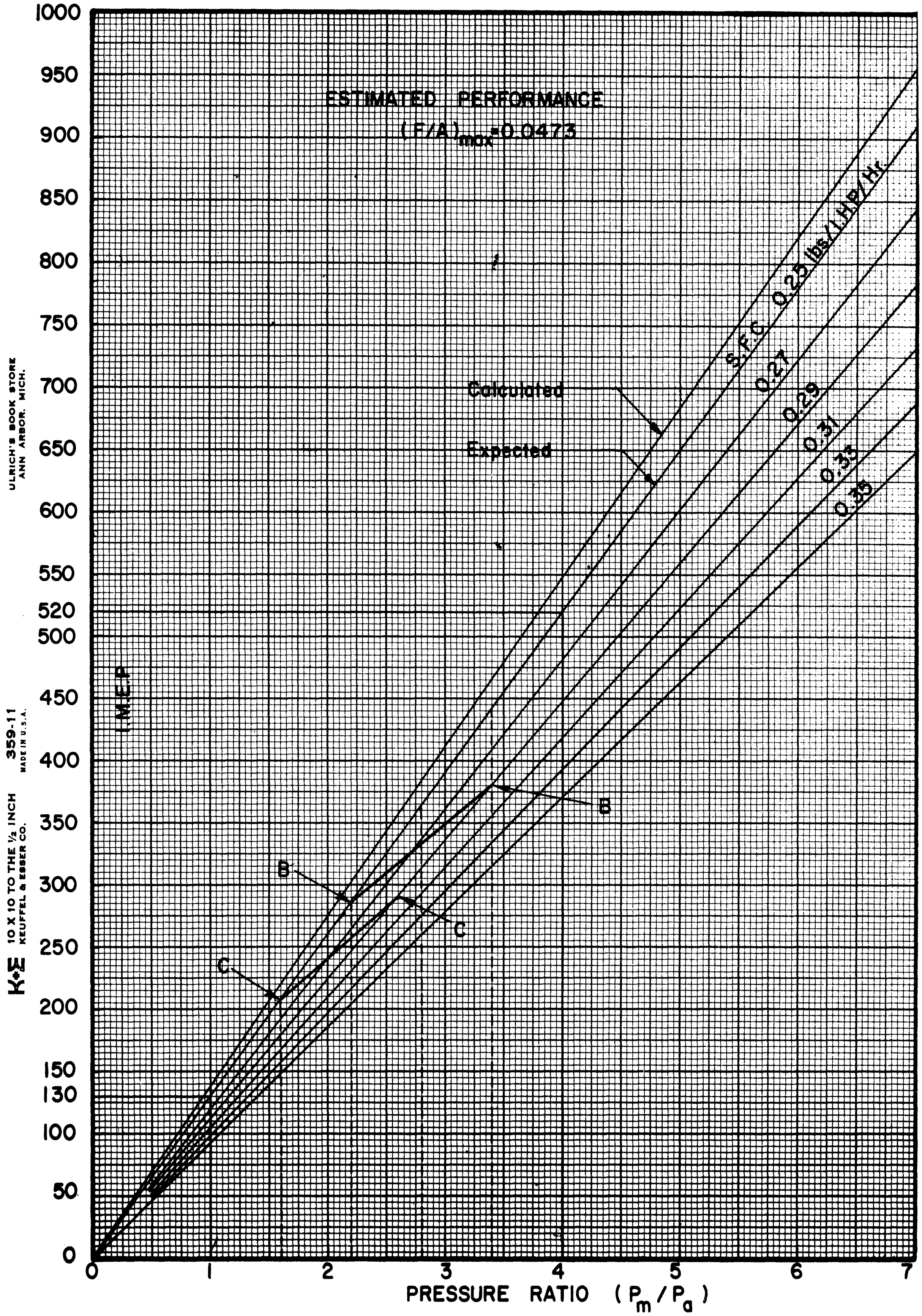
ESTIMATED PERFORMANCE
(F/A)max=0.0473
S.F.C. 0.25 lbs./I.HP/Hr.
0.27
0.29
0.31
0.33
0.35
Calculated
Expected
B
B
C
C
I.M.E.P.
PRESSURE RATIO (Pm / Pa)
1000
950
900
850
800
750
700
650
600
550
520
500
450
400
350
300
250
200
150
130
100
50
0
0
1
2
3
4
5
6
7
K+E 10 X 10 TO THE 1/2 INCH 359-11
KEUFFEL & ESSER CO. MADE IN U.S.A.
ULRICH'S BOOK STORE ANN ARBOR, MICH.

Fig. 4b