THE UNIVERSITY OF MICHIGAN

INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

FLEXIBLE VERSUS RESPONSIVE ENGINES

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PERFORMANCE PREDICTION FOR THE HIGHLY TURBO-CHARGED COMPRESSION IGNITION ENGINE

INTRODUCTION

The compression ignition engine has shown its capabilities, particularly when turbo-charged, as the power plant for Army vehicles due to its economy of operation as regards both quantity and quality of fuel supply. At the same time its speed range and torque characteristics leaves something to be desired, at least when fitted with present-day injection systems.

With the advent of modern turbo-chargers, materials, oils, etc., the possible engine ratings have changed in a major way and, given modern controls, a wide range of engine outputs become possible, a range which might result in simplifying the transmission system between engine and track,or alternatively in better performance characteristics.

At the request of Mr. Floyd B. Lux of the Power Plant Laboratory of the Detroit Ordnance Automotive-Tank Center the writer agreed to examine a wide range of oil engine performance factors to determine their effects upon possible future developments of interest to the Automotive-Tank Center. In order to accomplish this it was considered essential to develop a reasonable means of calculating engine performance to eliminate the general present day necessity for considerable oil engine background for such prediction.

Gasoline engine performance can be estimated with reasonable accuracy via the use of charts of gas properties plus performance coefficients. On the other hand such charts as exist for the oil engine, seen by the author, require the use of a rather large fudge factor if practical engine results are to be obtained.

The result of the investigation is given in the following pages and little in the way of oil engine experience is necessary to produce a reasonable performance characteristic, an average engineering student with a groundwork of thermodynamics can come up with an acceptable answer to such problems.

In submitting the following methods the author wishes to thank the United States Army Tank-Automotive Center, Detroit and Mr. Floyd B. Lux of that organization for the encouragement and financing which made this investigation possible and for permission to publish the results.

The methods were used in a series of investigations and appeared to give satisfactory results in line with known results and reasonable predictions in fields yet to be investigated.

SECTION 1

ENGINE PERFORMANCE PREDICTION

1.1. METHOD

In order to effect a satisfactory prediction method it is necessary to define an acceptable method by which the characteristics of a compression ignition engine can be estimated with a satisfactory degree of accuracy for all compression ratios, fuel/air ratios, manifold pressure, etc. It is well known that theoretical air cycle calculations or even prediction from actual mixture charts are not too accurate for this type of engine, due to the fact that the fuel injection characteristics alone can change the specific fuel consumption for a given set of conditions by as much as two to one. In order to approach the problem it must be assumed that a careful evaluation of the injection requirements has or will be made, so that the performance does approach the ultimate that can be expected for the type of engine involved. With this assumption and a method of calculation outlined in Ref. 1, where allowance is made for the heat losses involved throughout the cycle, and the method modified somewhat to apply to the oil engine cycle, a reasonable approximation to the best performance to be expected on an indicated basis can be obtained when using compression ignition engines of the direct-injection type.

In order to predict the complete engine performance, it is necessary to estimate the following effects:

1. Manifold pressure effect on output.

2. F/A ratio vs. I.M.E.P. for constant manifold pressures.

3. Engine speed and manifold pressure relations.

1.1.1. Manifold Pressure Effects on Output

A series of cycle calculations were made for the following conditions:

Fuel/air ratio	0.0473		
Supercharger pressure ratio P_m/P_o	1.0, 2.0, 4.0, 6.0		
Compression ratio	8:1, 16:1, and 20:1		
Charger efficiency	0.70, 0.80		
Atmospheric state	P = 14.7 psia		T = 85°F
Manifold temperature	200°F		
Total heat loss of cycle (%)	18	21	24

Heat loss during compression (%)	0.3	0.4	0.5
Heat loss during combustion (%)	1.2	1.7	2.3
Heat loss incomplete combustion (%)	2.0	2.0	2.0
Heat loss during expansion (%)	4.0	5.0	6.0
Heat loss during exhaust (%)	5.0	5.5	6.0
Heat loss to oil (%)	1.2	1.8	2.4
Heat loss miscellaneous (%)	4.3	4.6	4.8

The resulting I.M.E.P's are shown in Fig. 1, where the indicated mean effective pressure is plotted against the supercharger pressure ratio, P_m/P_a. A constant manifold temperature of 200°F was assumed which presumes the use of an aftercooler. This cooler would not be necessary for P_m/P_o = 1.0-2.0 but would be necessary for the higher ratios.

Examination of the plotted points showed little variation of mean pressure and fuel consumption with compression ratio despite the wide variation covered. This was to be expected if a careful analysis of the thermodynamics of the oil engine cycle is made. particularly so when the dual cycle (constant volume followed by a constant pressure combustion) with a limited maximum cylinder pressure is employed. All the calculated results could be included within the lines BB and CC of Fig. 1; while one constant compression ratio curve is also indicated, it is seen to curve slightly for the conditions used. It is believed that for the specification involved in the problem being considered, the line AA through the origin will represent with a sufficient degree of accuracy the best mean pressure to be expected for any set of conditions covered by the assumptions. In practice there will be some reduction in these mean values due to the efficiency of the combustion process in the actual engine, resulting from how well the turbulence can completely mix the fuel and air. An actual indicator diagram will have rounded corners in place of the square corners of the theoretical diagram. If it is assumed that the actual diagram is 0.94 times as large as that of the ideal, the plots shown in Fig. 2 can be considered as representative indicated mean pressure at a F/A = 0.0473 at various supercharge ratio with various efficiencies of combustion. The calculated line has a S.F.C. of about 0.235 lb/I.H.P./hr, the best expected performance being 0.94 of the calculated at 0.25 lb/I.H.P./hr; various other ratios of expected to calculated results are also shown. Which of these lines to use will depend upon the type and loading of the engine. For example, a direct-injection engine with compact combustion chamber and good turbulence could be expected to give a specific fuel consumption of about 0.25 lb/I.H.P./hr at about one half load, increasing to 0.30 to 0.31 at full load when the range of speed and manifold pressure is not too wide. Where the maximum output is being sought at a high speed, with some sacrifice of economy these values might change to 0.26 to 0.33 lb/I.H.P./hr respectively. In the case of a precombustion chamber, where some pumping losses exist in the cell, somewhat higher S.F.C. figures should be employed to locate the expected I.M.E.P. for any supercharge ratio.

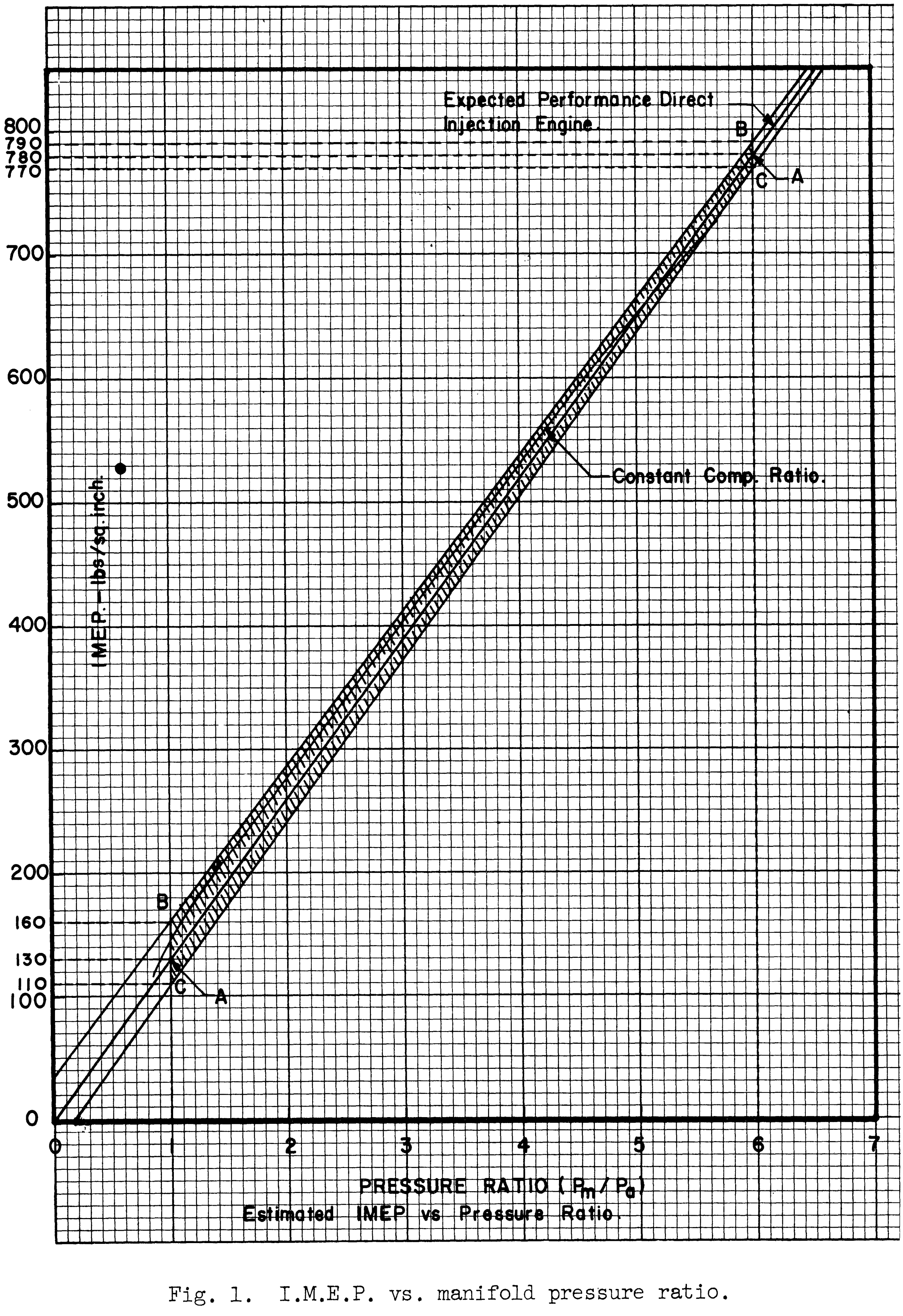

Fig. 1. I.M.E.P. vs. manifold pressure ratio.

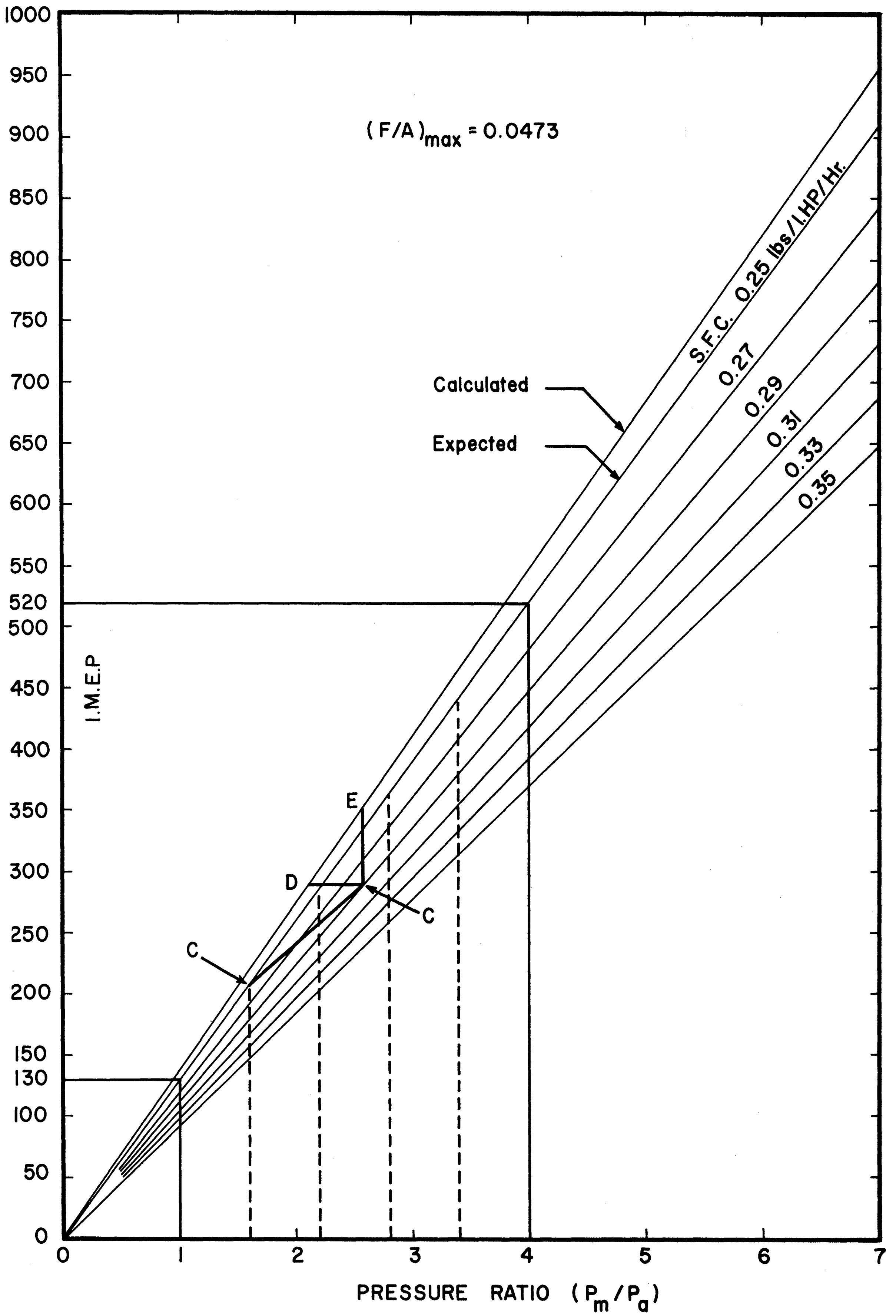

Fig. 2. I.M.E.P. vs. pressure ratio for various combustion efficiencies.

A line such as CC in Fig. 2, intersecting the various S.F.C. lines would then represent about the same load rating on an engine at the various pressure ratios as speed of the engine reduced. The line CD would represent the possibilities of further combustion research, improving the S.F.C. to about the ultimate value while the mean pressure developed remained constant. Alternatively the gains in combustion efficiency could be taken by maintaining P_m/P_o constant and going vertically along the line CE. The diagram permits the maximum output for any given engine, under a F/A ratio of 0.0473, to be predicted with reasonable accuracy. The F/A of 0.0473 was selected for maximum output as being one at which good power could be obtained with a reasonably clean exhaust, an exhaust which would be acceptable but not smoke free.

1.1.2. Fuel-Air Ratio Effect on I.M.E.P.

An investigation was made to try and correlate the mean pressure obtained with the fuel-air and pressure ratio. If the heat added to the air of the compression ignition cycle, where the air/stroke at constant manifold conditions remains approximately constant, were doubled together with the F/A, then under similar diagram conditions the work would also be doubled assuming that the thermodynamic efficiency remains constant. To achieve this relationship, the similar conditions would involve the same relations between the heat added at constant volume and constant pressure in both cases. It follows from this that the relation between F/A and I.M.E.P. would be expected to be of the straight-line variety; however, there is sufficient variation in practice to introduce errors of magnitude at high heat addition values. Investigation resulted in the plot shown in Fig. 3 as one which gave good correlation with practice, the I.M.E.P. being plotted on a base of the log of the F/A ratio for various values of the manifold pressures.

The diagram shown indicates that a straight-line relationship is obtained if a line joins the point of zero I.M.E.P. at F/A of 0.0125 to the I.M.E.P. predicted for the 0.0473 ratio; e.g., if F/A of 0.0125 at zero I.M.E.P. is joined to 734 I.M.E.P. at $P_m/P_o = 6.0$ for a F/A = 0.0473, as obtained from Fig. 2, the resulting line represents the ideal relationship for a manifold pressure of 6:1. As already indicated the performance tends to fall off somewhat at high F/A ratios and thus some correction is necessary. As an average this departure began at a F/A = 0.03 approximately and increased as the F/A increased, finally reaching an I.M.E.P. of about 85% of the theoretical expected value when the F/A was 0.055. All lines appeared to meet at the 0.0125 ratio and to have the same shape, hence the method of construction employed in producing the 225 in. Hg line of Fig. 3 by a curved line at 0.03 F/A to 85% of the I.M.E.P. of the theoretical value at F/A = 0.055. All other lines for the various manifold pressures are constructed in the manner indicated for the 225 in. Hg pressure. This diagram remains fairly accurate down to a F/A ratio of 0.015 approximately; thus part load performance can be obtained at any manifold pressure. Of course at the max-

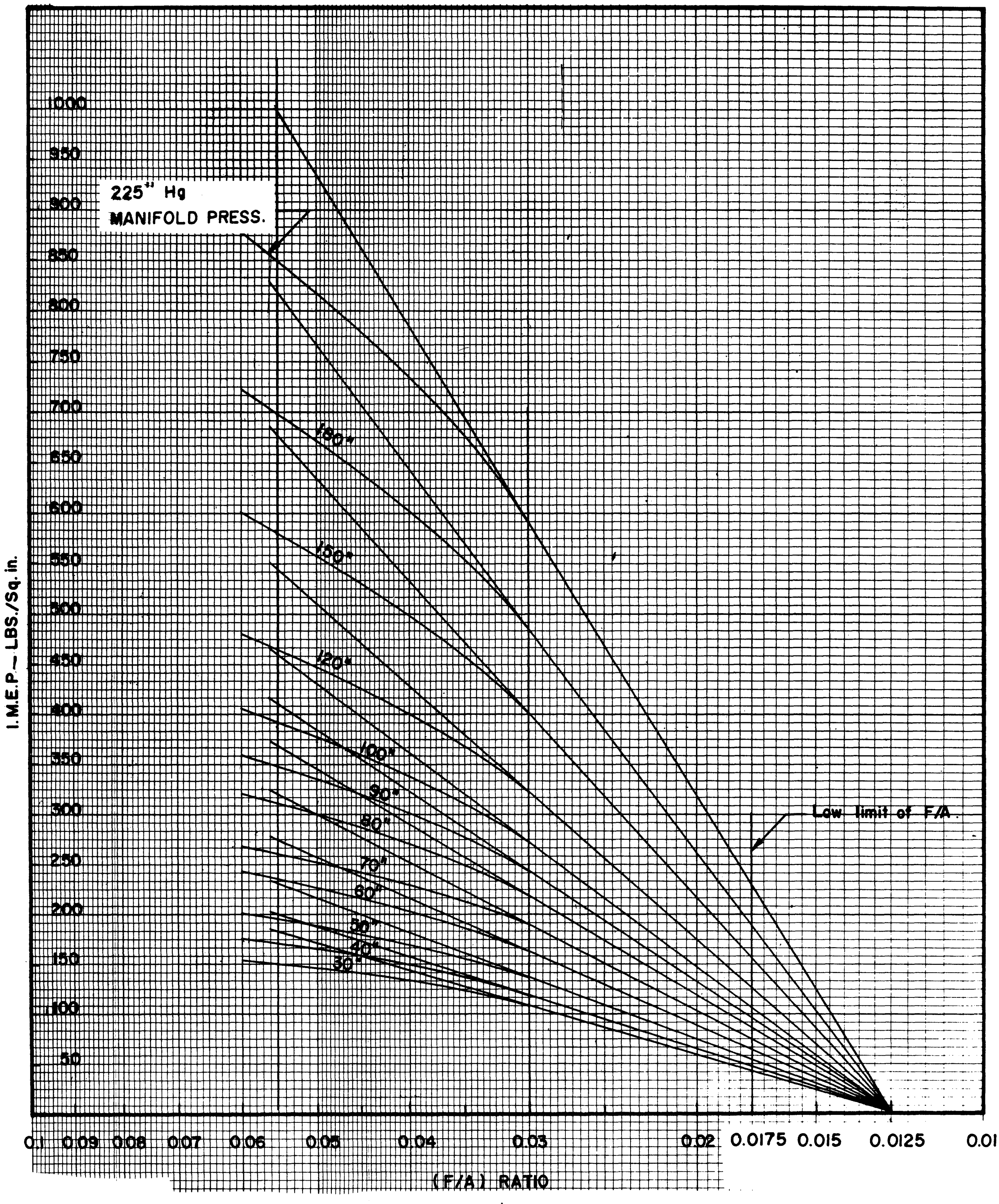

Fig. 3. I.M.E.P. vs. F/A and manifold pressures.

imum F/A of 0.055 considerable smoke would exist; an acceptable smoke limit would be at about 0.045-0.047.

If the curves of Fig. 3 are analyzed for various constant manifold pressures and plotted as percentages of the power at the F/A ratio of 0.043, then the curve of Fig. 4 is obtained for various F/A ratios from 0.0475 to 0.015. If the plotted points are considered from the broad over-all standpoint it is seen that a single curve, as shown, can represent the results with a reasonable degree of accuracy. The data covered in this diagram are for a S.F.C. of 0.33 lb/I.H.P./hr at 100% load and F/A = 0.043 for manifold pressures from 30 to 150 in. Hg lb. There is only one point at 100% since the S.F.C. was assumed constant at this point for all manifold pressures. In practice some small variation from such a value can be expected and the curve would shift slightly; but despite such detailed variations it is believed that the curve shown has many useful applications in engine analysis.

1.1.3. Engine Speed and Manifold Pressure Relationship

In the case of turbo-charged compression ignition engines there is a definite schedule of manifold pressure speed-power relationships since the exhaust gas quantity depends upon the engine rpm and its temperature upon both rpm and F/A; thus the energy available to the turbine and in turn to the compressor depends upon the above factors. In order to predict performance over a range of speeds and loads some idea of the effects of the above combination of factors must be known.

In order to establish the approximate relationships, typical test results were analyzed and it was found possible (as shown in Fig. 5) to plot the engine performance on a percentage basis, provided that reasonably high manifold pressures were involved together with aftercooling to a constant temperature beyond the compressor when the air mass flow depended (theoretically) upon the manifold pressure. In Fig. 5 the base is plotted as the percent of full power indicated horsepower output at maximum rpm and the ordinate as the percentage of full-speed, full-load manifold pressure and percent of maximum I.M.E.P. for various percentages of the rpm. The plot in this figure consists of straight lines, as it should, at least for speed and I.M.E.P. Fortunately the manifold pressure line is also almost straight; the slight variation could easily result from observation errors. On the manifold pressure line the points for 100, 85.6, 71.4, and 57.1% speed are plotted for a typical engine to show the magnitude of the error involved; the error is seen to be small for a first broad general approach to such a complicated system.

As an example of the use of the chart, assume that a manifold pressure was desired at which 90% of the full-load I.M.E.P. would be developed at 70% full speed; then following the arrows from the 90% mean pressure to its intersection with 70% speed it is seen that the power output would be 63%

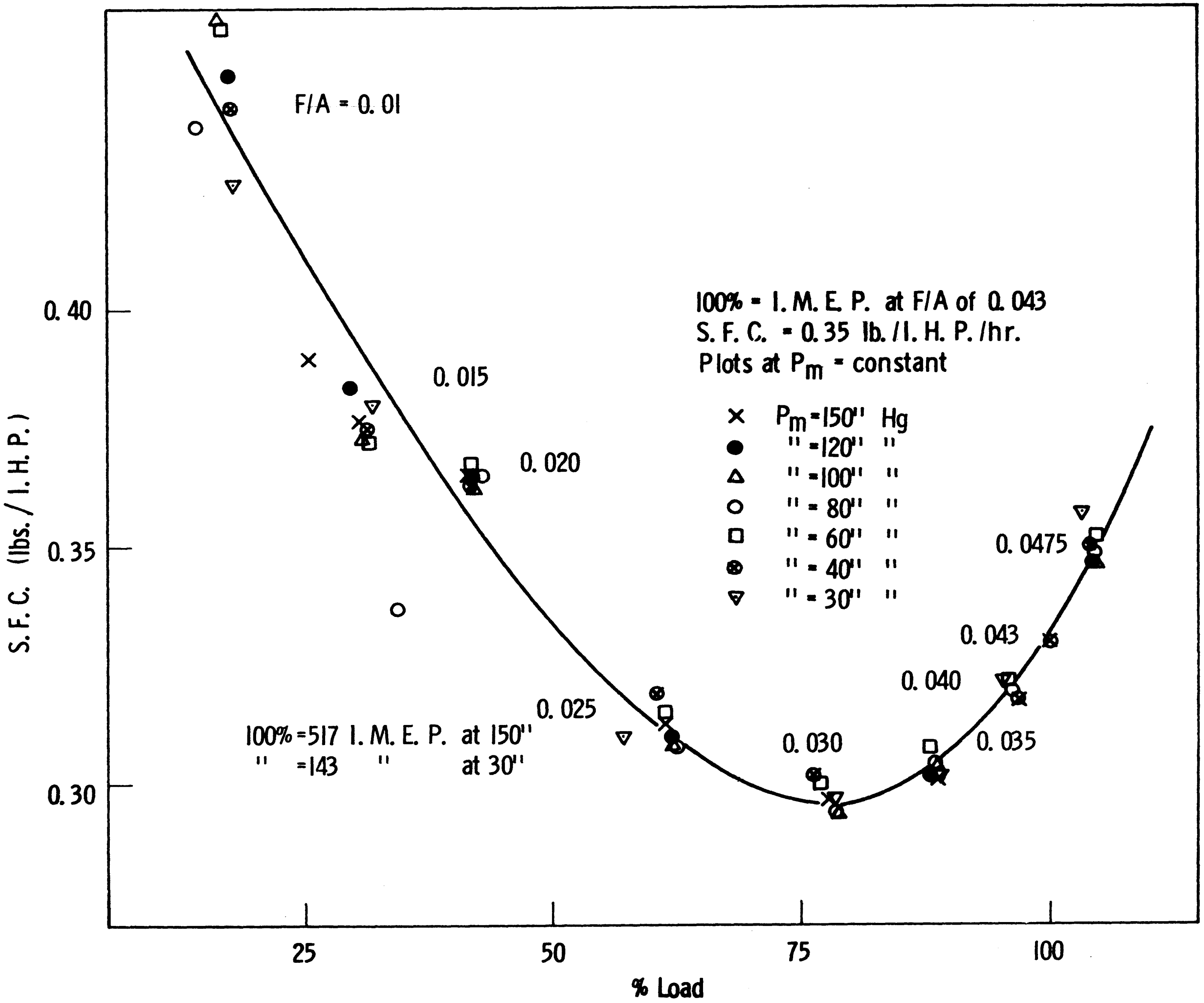

Fig. 4. S.F.C. vs. % load at constant manifold pressure.

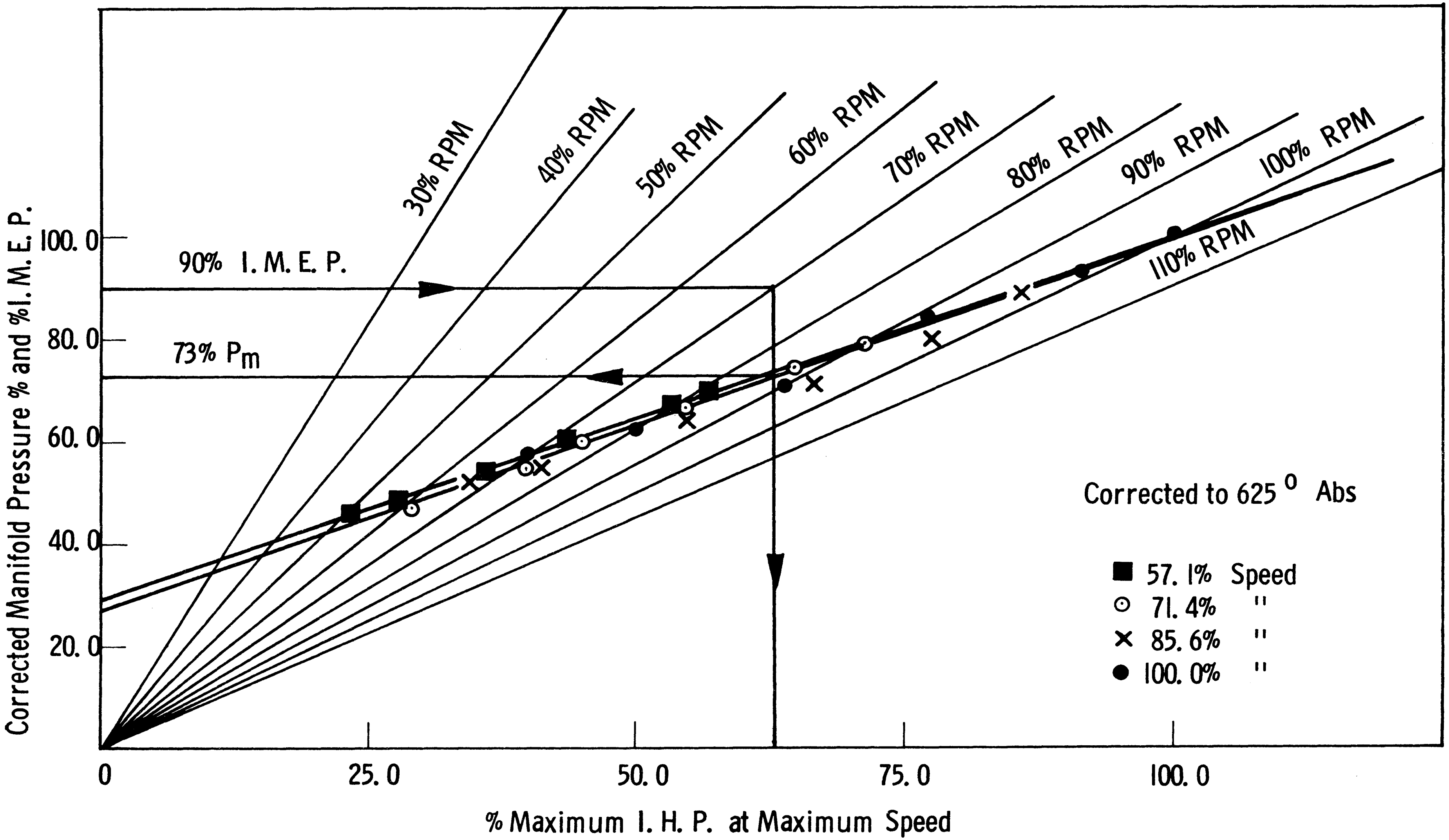

Fig. 5. % M.E.P., manifold pressure, and I.H.P. for turbo-charged engine.

of full load, i.e., 90 x 70, and would be achieved with a manifold pressure of 73% that at the full-load, full-speed condition. It follows that if the desired pressure for full load is known, all other pressures are available.

The data of Fig. 5 also indicate that 63% of the full-load, full-speed power can be carried at 73% manifold pressure at 110, 90, or 80% speed with 57, 70, and 79% I.M.E.P.

Consideration of the manifold pressure curve indicates that its actual position on the vertical scale and its slope will change slightly as the temperature to which it is corrected is changed. The line shown in Fig. 5 is corrected to a temperature of 625°R; however such variations for all practical temperatures will be of a minor character when the first approximations are being made.

A second plot involving output, speed, and the F/A ratio is given in Fig. 6, which is constructed for a clean exhaust at a F/A ratio of 0.0375 as the 100% load at maximum rpm. Again the construction employs straight lines as far as possible. The line AB is drawn from an idle condition for maximum rpm (usually of the order of 10% of maximum I.H.P. with a F/A ratio of about 0.0175, or slightly less), through the 100% I.H.P. point at the desired 0.0375 F/A ratio. In practice this line will not be exactly straight but the error involved is of the second order. Since the line AB is a constant-speed line and since 80% I.M.E.P. corresponds to 80% load, etc., the points of reduced mean pressure can be plotted directly from the output scale. Now for a given set of theoretical conditions such as compression ratio, the I.M.E.P. does not depend upon engine speed; the cyle does depend on the F/A ratio; and to maintain a constant I.M.E.P. a constant heat release to the air in the cylinder must be achieved—i.e., the Btus added per cycle at 100% speed and 100% I.M.E.P. must be maintained at 40% speed to achieve 100% I.M.E.P. This does not mean a constant F/A since, as shown in Fig. 5, the air density changes due to change in manifold pressure with speed. Thus at 40% speed but 100% I.M.E.P., 40% power output is generated and it can be seen from Fig. 5 that this requires 57% of the manifold pressure required for 100% load and speed for the same manifold temperature. In order to release the same heat quantity it follows that

$$\text{Required F/A ratio} = \text{F/A at 100\% speed} \times \frac{\text{100\% manifold pressure}}{\text{57\% manifold pressure}} \qquad (1)$$

which for the case of 40% speed at 100% I.M.E.P. becomes

$$F/A = 0.0375 \times \frac{100}{57}$$

$$= 0.0658 .$$

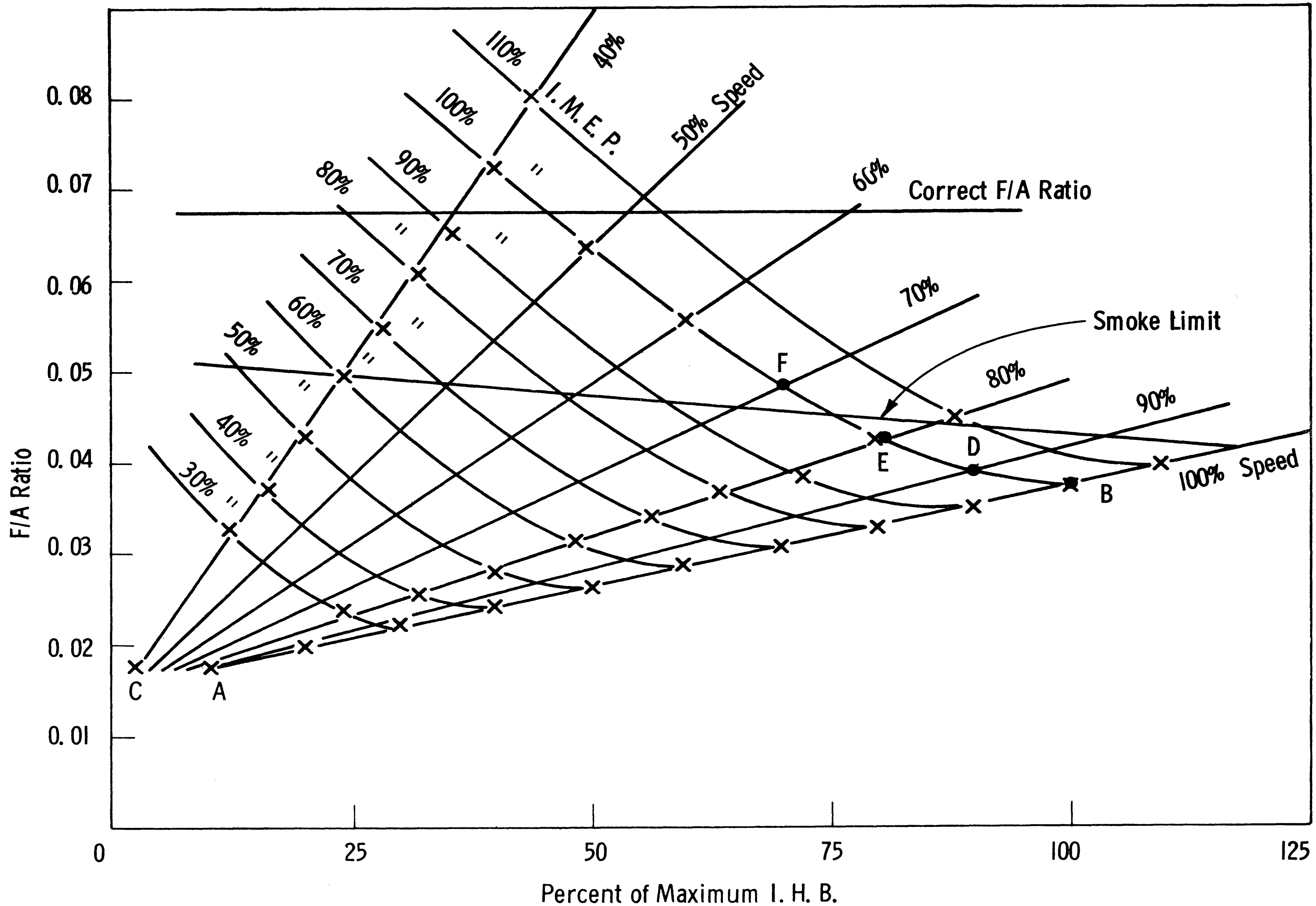

Fig. 6. Percentage performance plot of F/A, rpm, I.M.E.P., and % power.

This calculation assumes that combustion efficiency remains constant for this speed range. In practice, where the injection system is generally tailored for full load and speed, there would be some falling off in combustion efficiency with the result that to a first approximation it can be written that:

$$\text{F/A ratio at X\% speed} = y\ F/A_K \times \frac{P_K}{P_X} \qquad (2)$$

where

$X\%$ = speed at which F/A is to be determined
y = ratio of fuel at X rpm to the theoretical F/A
F/A_K = known F/A ratio at K% power
P_K = percent of manifold pressure at K%
P_X = percent of manifold pressure required for the power at X% speed.

Assuming that y = 1.1 at 40% speed, then

$$\text{Fuel air ratio at 40\% speed and 100\% I.M.E.P.} = 1.1 \times 0.0375 \times \frac{100}{57}$$

$$= 0.0725$$

in place of the theoretical value 0.0658 determined previously.

This calculation can be repeated for any number of points desired along any constant mean pressure line. However the value of y varies with speed; some improvement can be expected in combustion efficiency as speed reduces followed by an increase as the speed continues to reduce. Tests indicate that for an average engine at 80% speed, y will be about 0.95 and will gradually increase to 1.1 as used above, at the 40% speed.

It appears to be sufficiently accurate for most purposes to calculate the 100, 80, and 40% points for any I.M.E.P. and to place a smooth curve through them. This has been carried out, and Fig. 6 result.

It is seen from the plot that a straight line can be drawn through the 40% speed points with reasonable accuracy. This line passes through a F/A of 0.0175 at 5% of power (a not too unrepresentative idle condition); let this point be C. The line CA can now be divided into the required number of equal parts; each 10% speed change was used in Fig. 6 and located on the 100% I.M.E.P. line, where 80% speed means 80% power or 40% speed, 40% power, a corresponding set of points such as D, E, F, etc. If the corresponding points located between C and A are joined to D, E, F, etc., at 100% mean pressure, a performance map expressed in F/A ratio in terms of percentages results. It is believed that Fig. 6 would cover most high-speed, direct-injection engines in the 4-in. to 6-in. sizes at least. Some adjustments would be necessary for large-bore, slow-speed engines, etc. Moreover it is probable that some adjustment would be required to represent precombustion engines with accuracy, probably an adjustment to the value of the F/A ratio at 100% mean pressure and speed.

It is seen that at high mean pressures and slow speed, F/A ratios in excess of the correct mixture are involved. Such mixtures are not compatible with the C.I.E. cycle and thus represents an area in which operation is, to say the least, undesirable, since the main produce will be smoke. In addition a smoke limit line has been added to the diagram. This line is a very flexible one, which can move up or down the diagram depending on the air handling ability of the engine. The line shown can be considered fairly average.

1.2. EXHAUST GAS TEMPERATURE AND ENERGY CONTENT

One other very important factor required in determining the over-all characteristics of various engine and turbine combinations, is the exhaust gas temperature and its energy content under all exhaust manifold conditions.

This problem was investigated and it was again found that engines with different combustion systems give somewhat different results. After some investigation the plot shown in Fig. 7 was decided upon; straight lines were once again used in the construction, and although there was some slight variation from these lines it appeared to be of a minor character. The results given fit the problem of preliminary analysis of engine problems with reasonable accuracy.

The diagram is based upon an idle-exhaust temperature at maximum rpm of 500°F at a F/A of 0.015 and at half speed of 300°F at 0.010. Straight lines are drawn through these points and through 1500°F at 0.05 F/A for maximum speed and 1300°F at 0.066 for the half-speed condition. To these lines are added constant I.M.E.P. lines from 120% full-load I.M.E.P. to 50% I.M.E.P. Now the exhaust gas temperature in a correctly operating compression ignition engine is a function of load for any given speed and load is a function of F/A ratio. It follows that naturally aspirated or supercharged engines will have substantially the same exhaust gas temperature for the same fuel-air ratio; hence Fig. 7 can be employed for all engines to a first approximation, and hence the plot on percentage of full load I.M.E.P.

It will be noticed that the gas temperature increases as speed is reduced at constant I.M.E.P.; this results from the need for higher F/A at lower speed, as brought out by Fig. 6.

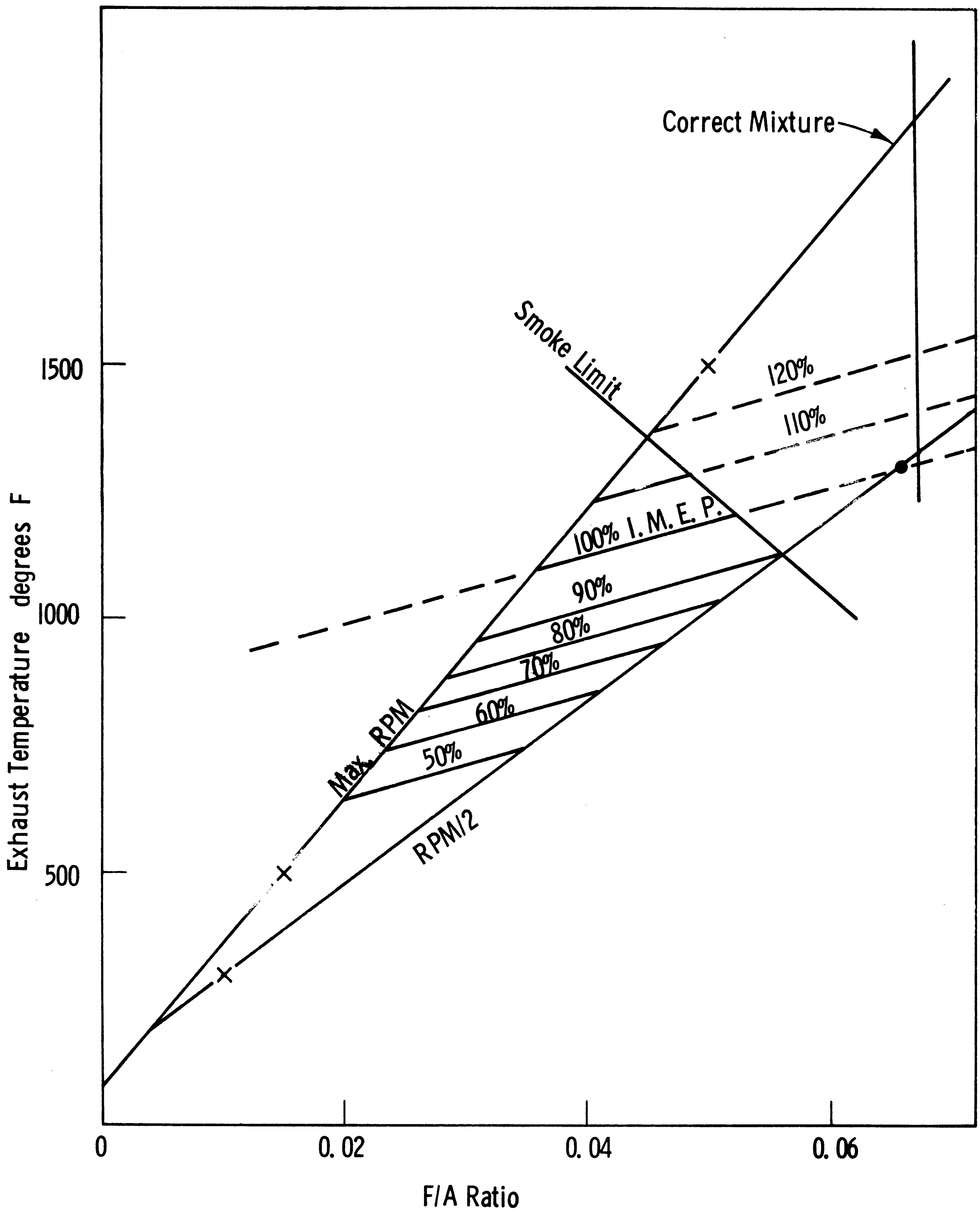

Fig. 7. Exhaust temperature vs. F/A ratio (turbo-charged engine).

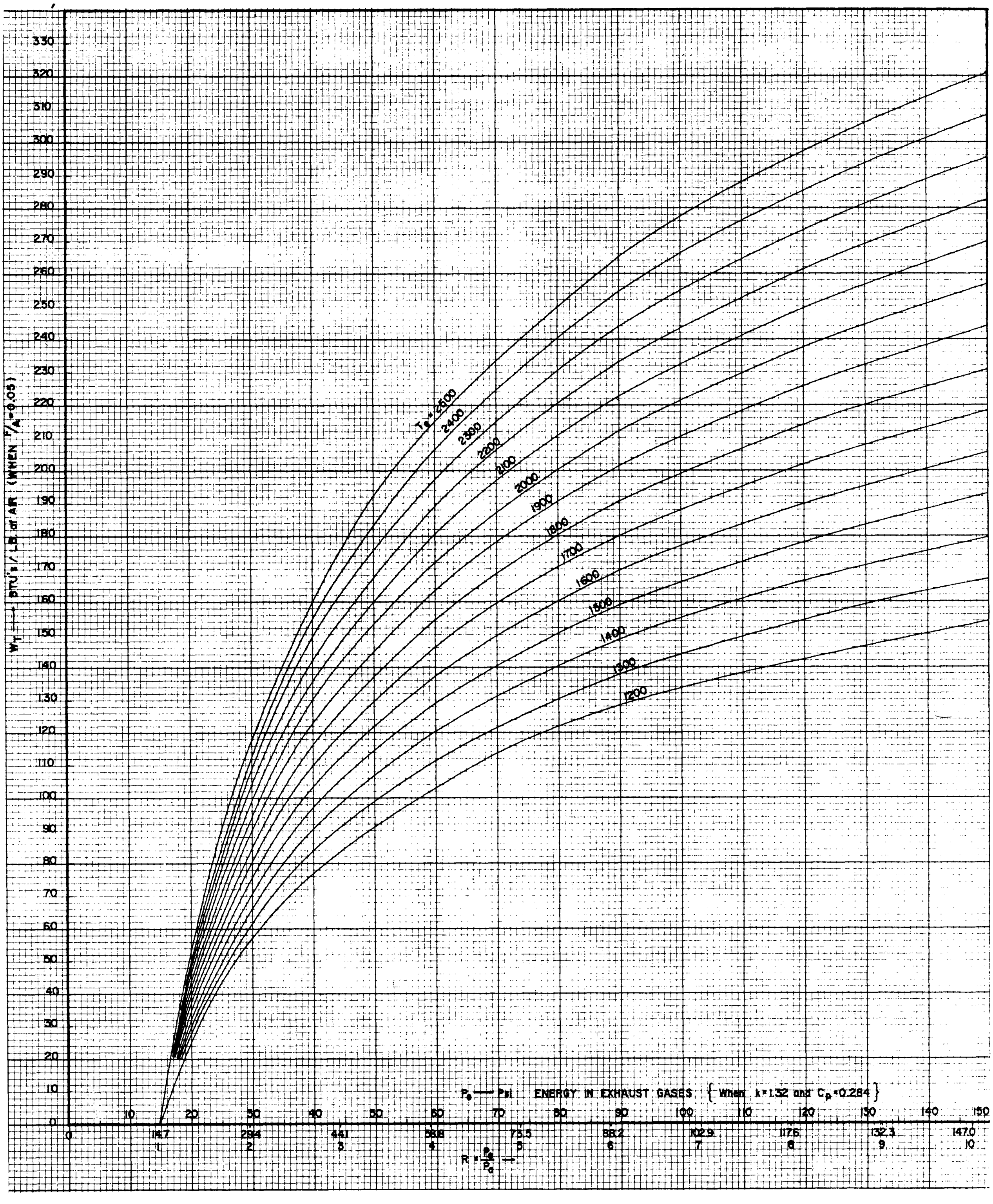

Fig. 8. Energy content of exhaust gases.

In addition to the exhaust gas temperature it is necessary to be able to calculate the energy content of the exhaust gases when turbo-charging or compounding is being investigated.

The energy available in the exhaust gases (see Fig. 8) when employed in a flow mechanism such as the turbine is given by the change of gas enthalpy during the passage through the machine.

It follows that for an isentropic expansion process

$$\text{Change of enthalpy} = \Delta h' = wC_pT_1 \left[1 - \left(\frac{T_2}{T_1}\right)^{k-1/k}\right] \qquad (2b)$$

and for an actual process

$$\text{Change of enthalpy} = \Delta h = wC_pT_1 \left[1 - \left(\frac{T_2}{T_1}\right)^{n-1/n}\right] . \qquad (2c)$$

The efficiency of the process is defined by

$$\text{Expansion efficiency} = \eta = \frac{\Delta h}{\Delta h'} . \qquad (2d)$$

1.3. ENGINE WEIGHT AND VOLUME PREDICTION

The parameters governing the over-all engine dimensions, weight, power, etc., are well established for the reciprocating type of engine. Individual engine designers will produce varying envelopes, etc., depending upon minor design variables and accessory arrangements, but in a given weight class the essential items -- stroke/bore ratio, connecting rod length, etc. -- do not vary widely. This is the result of many decades of development and the utilization of the present ultimate in materials, bearings, etc. It follows that reasonably accurate predictions can be made for a compact, lightweight engine of any selected type; the results given here are developed for what might be considered the average Army Ordnance Tank Engine of the compression ignition type. It is emphasized that the results will be "average," with sufficient accuracy to provide data that will fulfill the needs of a "Systems Analysis" type of approach to the optimization problem of any given vehicle.

1.4. RECIPROCATING ENGINES

The main engine variables which control the over-all dimensions are: (1) cylinder diameter, (2) piston stroke, (3) number of cylinders, (4) mean effective pressure, and (5) revolutions per minute. There are also secondary parameters such as the ratio of the connecting-rod length to crank radius, cylinder spacing, etc.

Starting with Eq. (3), the fundamental relationship between horsepower and size can be obtained.

$$hp = PLAN/33000 \qquad (3)$$

where

hp = indicated or brake horsepower
P = indicated or brake mean effective pressure psi
L = length of stroke in ft
A = area of piston sq. in.
N = number of explosions per min.

Then, if

S = stroke/bore ratio = ℓ/d
ℓ = stroke in in.
d = diameter of cylinder in in.
n = number of cylinders
rpm = revolutions per min.

Eq. (3) can be reduced to:

$$\begin{aligned} hp(\text{4-cycle engines}) &= 0.992 \times 10^{-6}\ Pd^3nS \times (rpm) \\ hp(\text{2-cycle engines}) &= 1.984 \times 10^{-6}\ Pd^3nS \times (rpm). \end{aligned} \tag{4}$$

Of the terms in Eqs. (4) the particular application usually determines the value of P and rpm; in the case of the turbo-charged compression ignition engine, for example, the value of P will be from 160 to as much as 250 psi, with the possibility of about 300 psi or more in the near future under certain limiting conditions. The engine speed of today's transportation units varies from a minimum of about 2000 to a maximum of about 4000 rpm in small sizes, with a maximum of about 3000 for the larger horsepowers.

The stroke-bore ratio S, which will affect height and weight materially, can be reduced to about 0.95-1.2 for today's average engine. The number of cylinders can be as low as four in small engines (though six is perhaps more general) and as high as twelve in some instances.

By selecting, within these broad limits, magnitudes for P, n, S, and rpm are established, with the result that Eqs. (4) become one in terms of d^3 and the cylinder bore is obtained from which the value of ℓ is calculated via the value S. Thus the broad engine parameters are available for the proposed powerplant, or if necessary a series of such values can be calculated from which the following parameters can be evaluated.

1.5. ENGINE VOLUME

1.5.1. In-Line Engines

The engine volume will depend upon length, breadth, and height, which can be figured from a schematic diagram such as Fig. 9.

The cylinder spacing "a" of Fig. 9 will depend a great deal upon engine arrangement, in-line, Vee, etc., and the following values are typical for the in-line engine:

Air cooled	$a = (1.4\text{-}1.5)d$
Water cooled	$a = (1.2\text{-}1.25)d.$

Of course the spacing must be sufficient to provide room for the crankshaft bearings, as well as cooling passages; the above spacing will in general make such provisions.

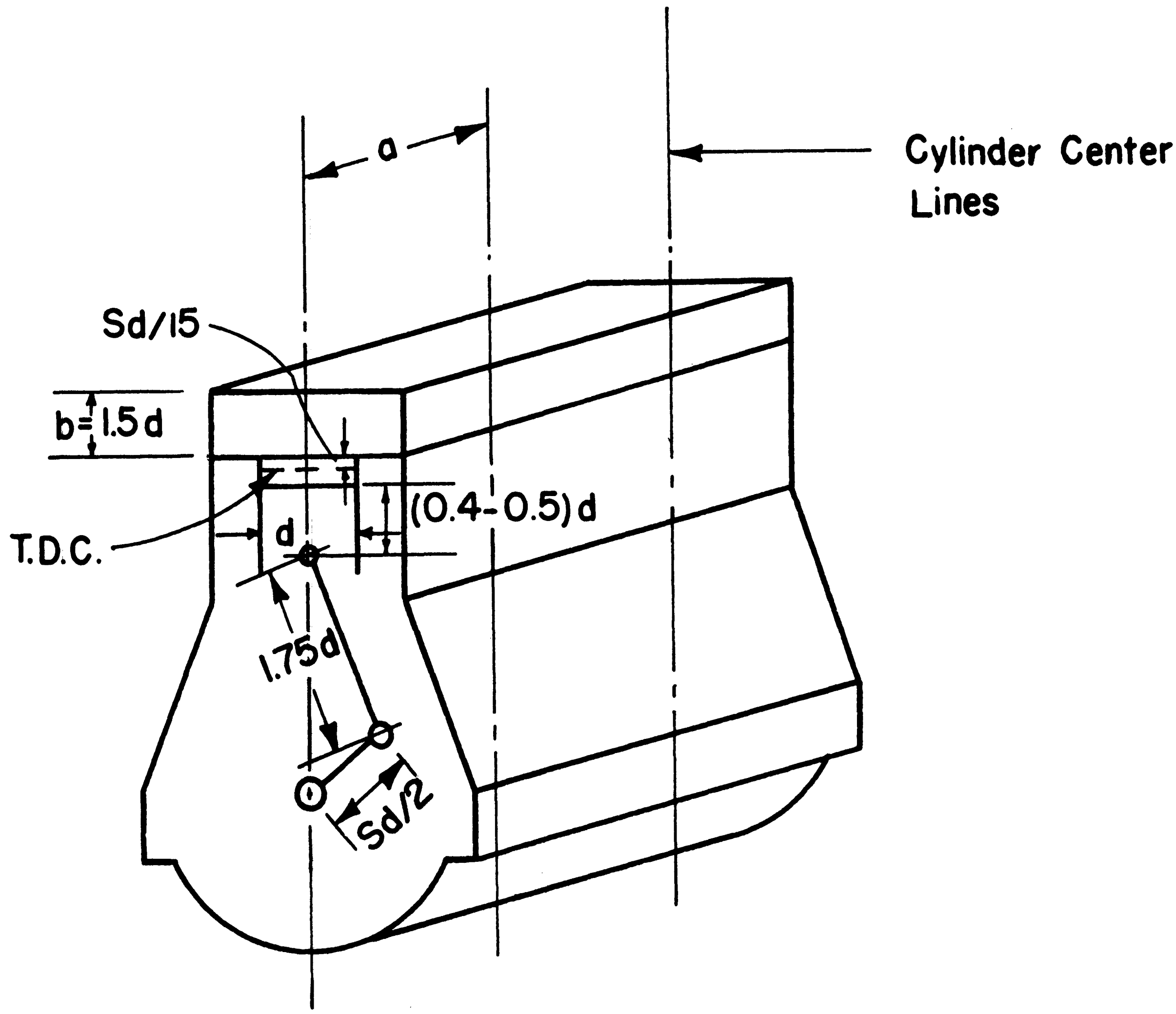

Fig. 9. Schematic diagram of in-line engine.

The value of "b" in Fig. 9, the distance from piston head at T.D.C. to the top of the valve cover, is given approximately by $b = 1.5d$ and will vary but little from one design to another, be the engine air-cooled or water-cooled.

The length of the connecting rod for a modern compact light design is generally about 1.75d.

One other factor involved in the over-all height of an engine is the over-all equivalent diameter of the big end of the connecting rod; this can be taken as equal to d, the cylinder diameter. This dimension is involved in estimating the depth below the crankshaft center line to determine the requirements for the bottom of the oil sump. It will be assumed that all the engines under consideration are of the dry sump type, or at least that oil capacity is provided by means other than sump depth itself. In such a case an allowance of about 2 in. below the bottom of the rod at B.D.C. should be provided. It thus follows that:

$$\text{Centerline of crankshaft to bottom of sump} = \frac{Sd}{2} + \frac{d}{2} + 2 \text{ in.} \tag{5}$$

$$\text{Centerline of crankshaft to top of cylinder} = \frac{Sd}{2} + 1.75d + 0.4d + 1.5d + \frac{Sd}{15} \tag{6}$$

$$\begin{aligned} \text{Over-all engine height} &= (5) + (6) \\ &= (1.067S + 4.15)d + 2 \text{ in.} \end{aligned} \tag{7}$$

The engine width by a similar process becomes

$$\text{Width of crankcase} = 2\left(\frac{Sd}{2} + 0.5d + 1\right).$$

Assume that asscessories, manifolds, etc., overhang the crankcase by 0.7d per side; then,

$$\begin{aligned} \text{Width of engine} &= Sd + d + 2 + 1.4d \\ &= d(S + 2.4) + 2 \quad . \end{aligned} \tag{8}$$

This dimension is somewhat difficult to decide before a design is finished, since so much depends upon the accessory arrangement, position of turbo-chargers, manifolds, etc.

The over-all length for "n" cylinders in-line becomes:

$$\text{Engine Length (Air-cooled, 4-cycle)} = (n-1)1.4d + \text{flywheel} + \text{accessory case, etc.}$$

$$= 1.4(n-1)d + 3.0d \text{ in. approx.} \tag{9}$$

$$\text{Engine Length (Water-cooled, 4-cycle)} = 1.2(n-1)d + 3.0d \tag{10}$$

$$\text{Engine Length (Water-cooled, 2-cycle)} = 1.45(n-1)d + 3.0d \tag{11}$$

When these relationships are used, the enclosing rectangular volume of such an engine amounts to:

$$\text{Air cooled engine volume} = \text{length x width x height}$$

$$\begin{aligned} &= [1.4(n-1)d + 3.0d][d(S+2.4) + 2][(1.067S + 4.15)d + 2] \\ &= d(1.4n + 1.6)[(S + 2.4)d + 2][(1.067S + 4.15)d + 2] \text{ cu in.} \end{aligned} \tag{12}$$

When the volume given by Eq. (12) is plotted for cylinder diameters from 4 to 6 in. with 4 to 8 cylinders, the curves of Fig. 10 are obtained (assuming that S = 1.05).

By way of comparison, the water-cooled engine for the same range of cylinder sizes is also shown, the volume ratio being given by Eq. (13). Since engine width is determined by manifolds and height by connecting rods, crank radius, etc., the only change is in the engine length; thus

$$\begin{aligned} \frac{\text{Volume of water-cooled}}{\text{Volume of air-cooled}} &= \frac{[1.2(n-1) + 3.0]d}{[1.4(n-1) + 3.0]d} \\ &= \frac{1.2n + 1.8}{1.4n + 1.6} \end{aligned} \tag{13}$$

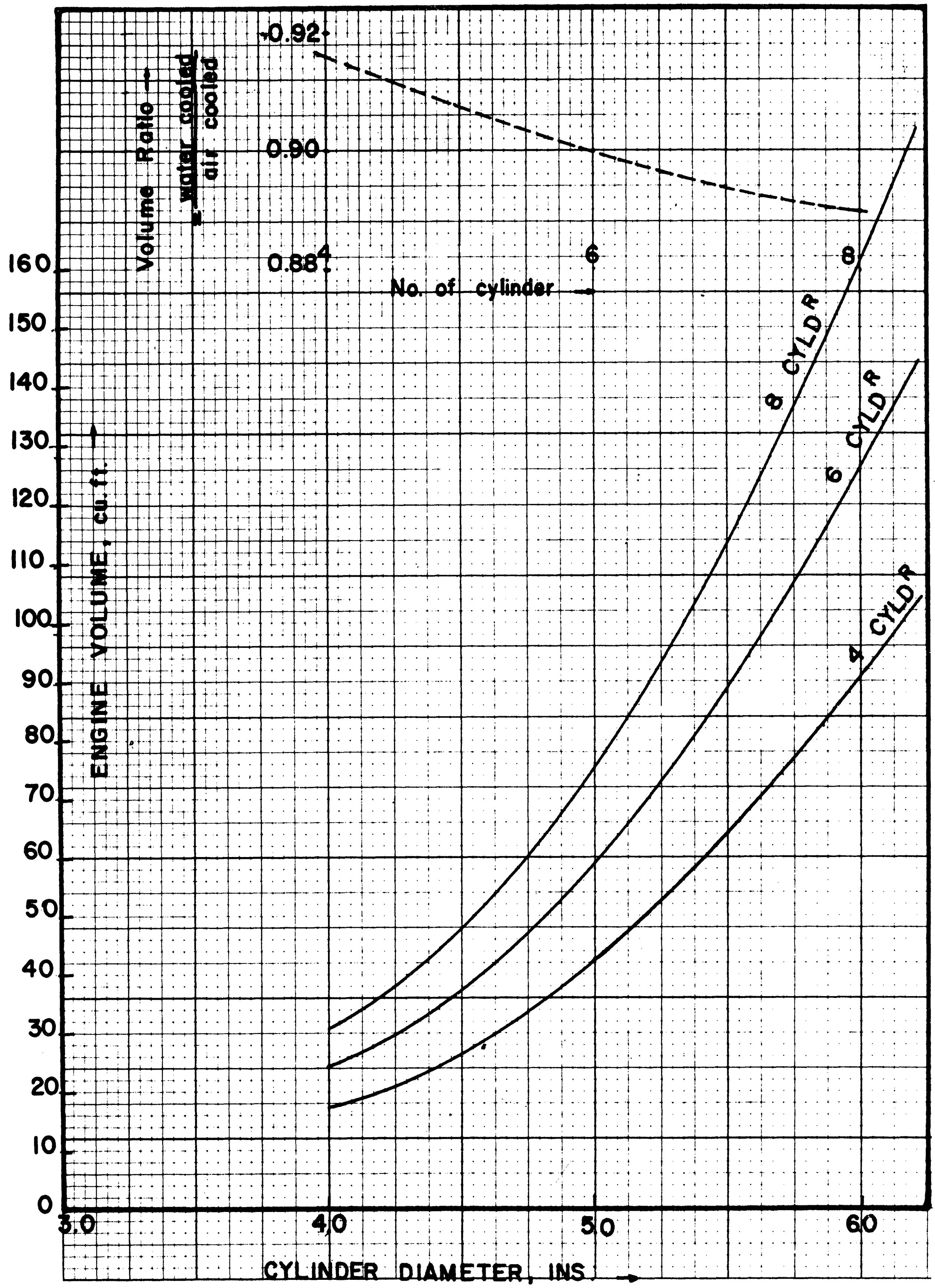

Fig. 10. Volume of in-line engines.

which for various cylinder number become

$n =$	4	6	8
V_w/V_a	.916	0.90	0.89

It must be remembered in using this comparison that values for engine radiator, fan, etc., have not been included; these items will add space to both types of engines but the air-cooled engine will be affected to a smaller extent since the air to be handled will be of much less volume due to the higher temperature difference involved between the air and cooling surface.

1.5.2 Vee Engines

The cubic capacity of Vee engines can be determined in the same manner as for the in-line engine by use of the outline shown in Figs. 11 and 11a. The outline can be considered as two in-line engines placed at an angle of $\alpha°$ to one another; then

$$\text{Engine height} = Sd(0.5 + 0.5667\cos\frac{\theta}{2}) + d(1.25 + 3.65\cos\frac{\theta}{2} + 0.8\sin\frac{\theta}{2}) + 2 .$$

In the case of a 90° Vee the above equation reduces to

$$\text{Engine height} = (0.9S + 4.396)d + 2 \tag{14}$$

$$\text{Engine width} = 1.1334Sd\sin\frac{\theta}{2} + d(7.30\sin\frac{\theta}{2} + 1.6\cos\frac{\theta}{2} + 0.5)$$

which for a 90° Vee becomes

$$\text{Width} = 0.801Sd + d(5.16 + 1.13 + 0.5)$$

$$= (0.801S + 6.79)d. \tag{15}$$

Engine length is determined as for the in-line engine, with one block of cylinders displaced by about 0.3d with respect to the other block (see Fig. 11a).

$$\text{Length} = (\frac{n}{2} - 1)1.5d + 1.5d + 0.3d + 1.5d$$

$$= \frac{n}{2}1.5d - 1.5d + 3.3d$$

$$= (0.75n + 1.8)d .$$

Thus for the 90° Vee engine the volume becomes

$$\text{Engine volume} = (0.75nd + 1.8d)(0.801Sd + 6.79d)(0.9Sd + 4.396d + 2) \tag{16}$$

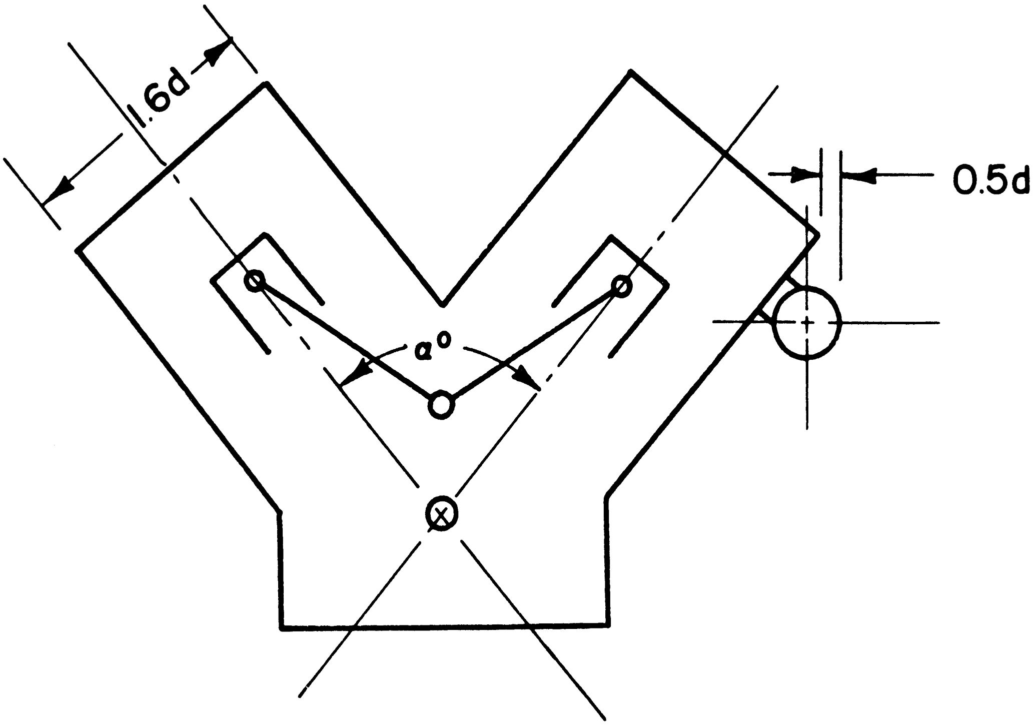

Fig. 11. Schematic diagram of Vee engine (I).

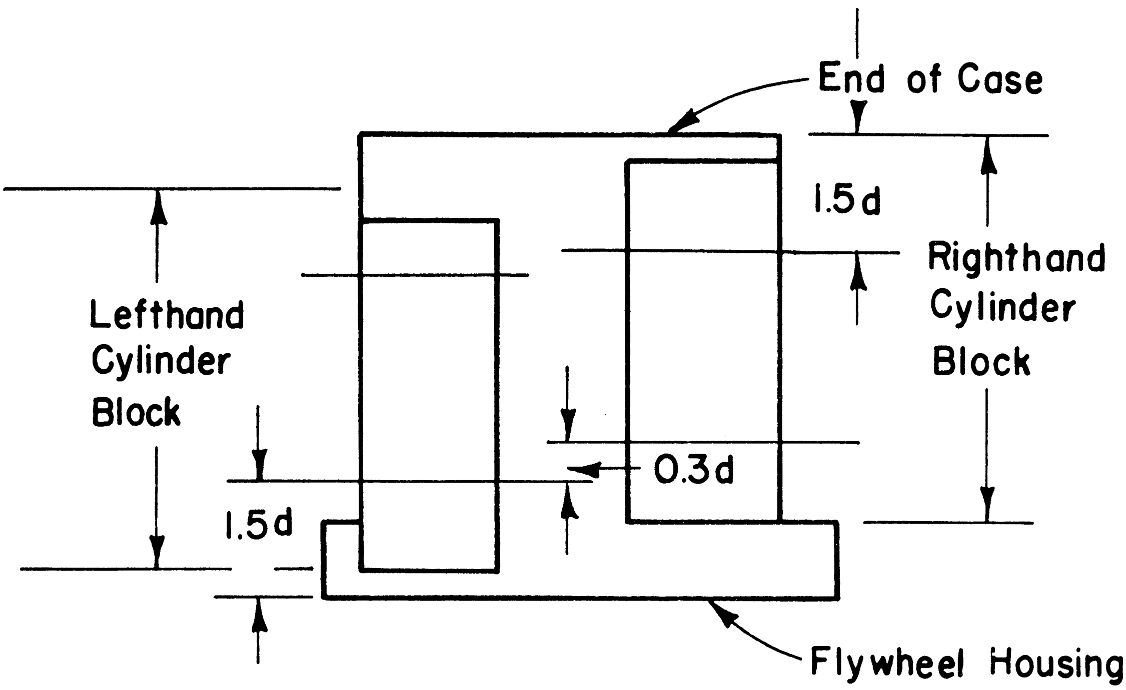

Fig. 11a. Schematic diagram of Vee engine (II).

This expression has been evaluated for engines of cylinder diameter from 4 to 6 in. and number of cylinders from 4 to 12; the results are shown in Fig. 12.

In the case of 60° V $\quad \cos\frac{\theta}{2} = .866$ and $\sin\frac{\theta}{2} = 0.5$; thus

$$\text{Height} = 0.9905Sd + 4.18d + 2$$

$$\text{Width} = 0.5667Sd + 5.536d = 6.131d$$

$$\text{Length} = (0.75n + 1.8)d$$

calculating from these equations the values of Fig. 13 result.

Relation Between Air- and Liquid-Cooled Vee Engines.—In this case the only change will, to a first approximation, be in the engine length as a result of smaller cooling passages.

$$\begin{aligned}\text{Length of a water-cooled engine} &= (\tfrac{n}{2} - 1)1.2d + 1.5d + 0.3d + 1.5d \\ &= 0.6nd + 2.1d \\ &= (0.6n + 2.1)d.\end{aligned}$$

Ratio of water-cooled to air-cooled becomes

$$\text{Volume ratio for Vee engines} = \frac{0.6n + 2.1}{0.75n + 1.8}$$

n = 4	6	8	12
Ratio = 0.937	0.905	0.884	0.861

Again radiators and fan for water cooling and fan for air cooling are not provided. In the case of the air-cooled Vee engine most of the fan can be included inside the Vee and little extra volume is required.

1.6. ENGINE WEIGHT

The weight in lb/B.H.P. of the reciprocating engine is fairly standardized and depends to a great extent upon the duty for which the engine is designed. Engines for tractors, trucks, marine purposes, etc., are all in different weight classes, mainly determined by the life requirements of the application. Engines designed for the same purpose by different manufactures will vary somewhat, but in secondary factors only. It follows that a tank engine designed by different manufacturers to meet the same specification

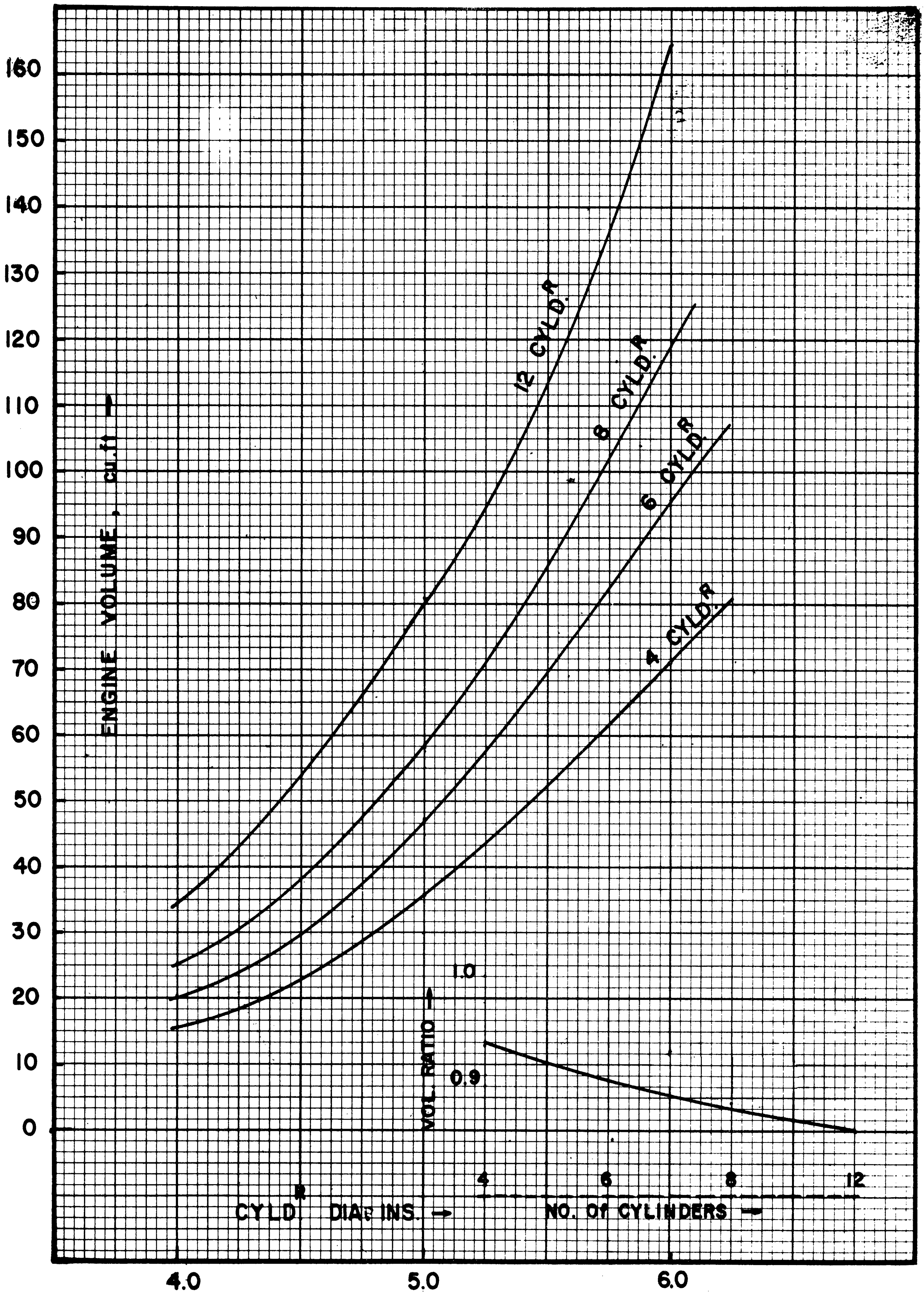

Fig. 12. Volume of 90° Vee-type engines.

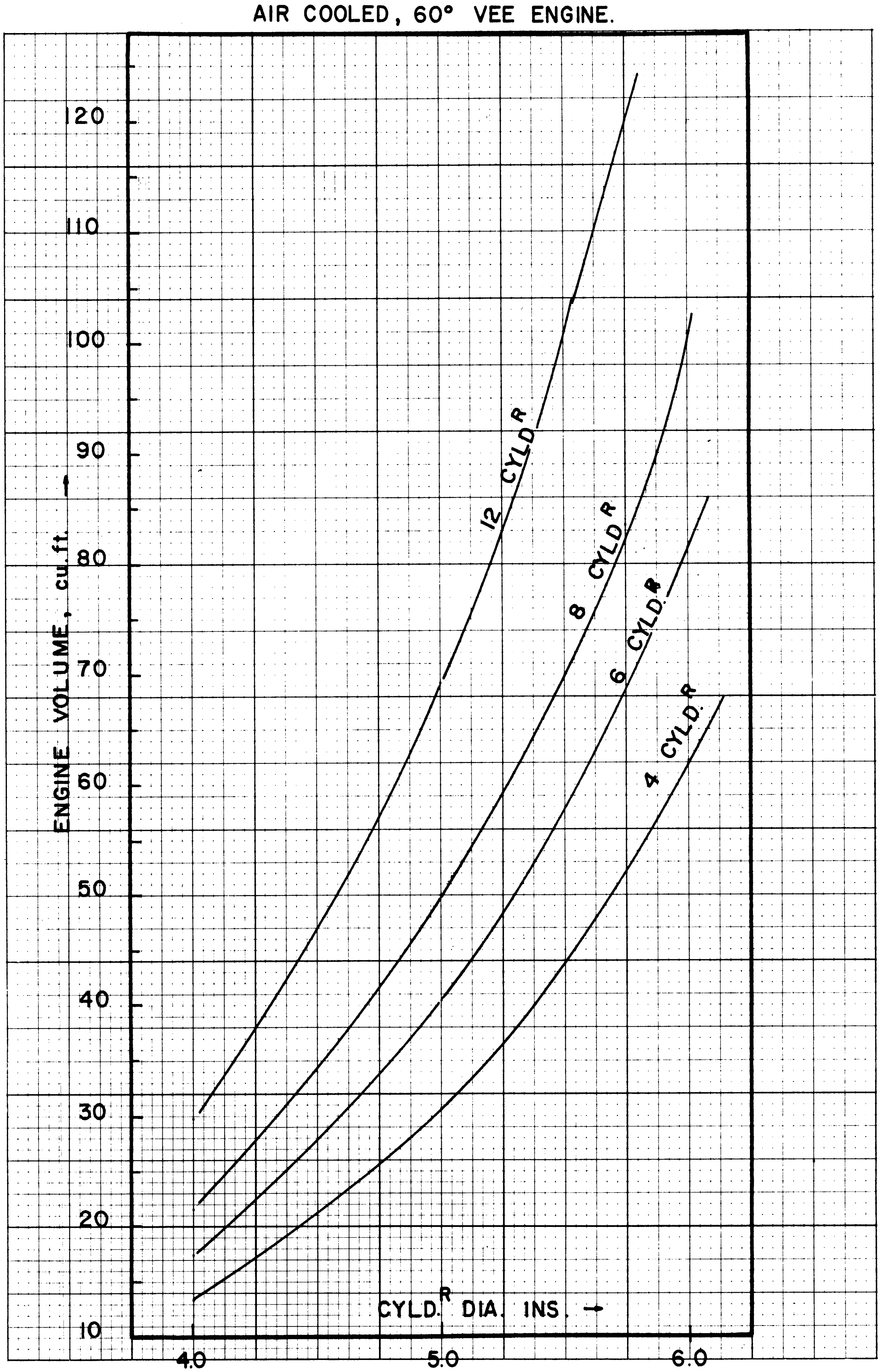

Fig. 13. Volume of 60° Vee-type engines.

will be roughly the same weight/hp. Any departure from this (particularly a weight reduction) involves a major development program; and a major reduction in weight entails a corresponding increase in frequency of maintenance, etc. Aircraft engines have been designed for as little as 0.6 to 0.7 lb/hp but this type was only employed for racing, fighters, etc., where short life was accepted. The commercial plane engine ran about 1 lb/hp. It follows that military gasoline engines for tanks, etc., could be produced at, say, 1 to 2 lb/hp if the increased maintenance were to be accepted.

It is believed that, for the present engine-type test, an engine weight or about 4 to 5 lb/hp cannot be improved to any major extent without shortening the length of life.

A second alternative would be the initiation of an extensive, fundamental combustion research program. This program should not be tied to any one concept of combustion but should be of wide latitude to permit a number of leads to be followed. Improvement of the combustion efficiency would increase engine output without involving increased engine load or heat stresses, as can be seen from the fundamental efficiency expression

$$\text{Engine efficiency} = \frac{\text{Work done}}{\text{Heat added}} . \tag{17}$$

Assume a given fuel-air ratio is employed; it follows that the heat added in Eq. (17) remains constant, and given the same weight of air to the engine, the same fuel flow in lb/hr occurs. Improvement in combustion efficiency must thus result in increased work,i.e., hp, for this same fuel flow. The peak cylinder pressures need not necessarily change under such a development; thus engine weight remains unchanged for the increased output. One other important factor which can outweigh all others in such a development program is the fact that, if more heat is converted into work, there is less heat to be disposed of; i.e., cooling is improved, temperatures of rings, pistons, etc., are all reduced and the life expectancy of the engine is increased simultaneously.

One factor which may limit the possible reduction in weight is the problem of fuel. If multi-fuel requirements are necessary, the problem of rough combustion must be faced along with those of very high compression ratios and accompanying peak pressure. All of these tend to increase the specific weight.

The use of variable compression ratio will overcome these problems to some extent, for it would allow the increased weight to be held to a minimum. Variable compression ratio can be very valuable when normal diesel fuels are used; in the present case, under normal operating conditions sufficient ratio can be employed to give efficient cold starting coupled with a lower ratio and improved power and fuel economy. It has been well established that most compression ignition engines give maximum power with minimum fuel consumption at a lower compression ratio than that at which cold starting occurs. This does not invalidate thermodynamics, for reasons which are to long and complicated to be given here. It follows that, other things being equal, a variable ratio could help the specific weight problem also.

It can be concluded that the weight per horsepower of compression ignition engines for tank operation can be improved and adequate service life maintained if:

(1) Specific power output, hp/cu in. of displacement is improved.

(2) Improved output can be achieved by:

(a) improved combustion efficiency;
(b) variable compression ratio;
(c) increased supercharging;
(d) improved design and stress control; and
(e) control of the rate of combustion and peak pressures by injection system and combustion factors.

Of the items listed in (2), all except (c) present a possiblity of general over-all improvement without the need for increased engine structure. Of course (c) would involve increased pressure and thus weight, unless a highly variable compression ratio engine could be designed.

In the meantime the specialized engine for military purposes can be expected to weigh some 4 to 10 lb/hp, depending on output and materials (aluminum or iron).

1.7. SUMMARY OF METHODS OF CALCULATION

In the previous pages are presented methods by which other than skilled Compression Ignition Engine Engineers can approach almost any problem of power, fuel consumption manifold pressure, prediction, etc., with reasonable accuracy and confidence.

It must always be kept in mind that the methods proposed are for the broad aspect of the problem. Individual details will result in some variations from the mean.

The above methods were employed in a series of investigations concerning power plants for military vehicles with, it is believed, satisfactory results.

Methods were also prepared for similar predictions for gas turbines and combinations of C.I. Engines and gas turbines which might serve the purpose of a second paper.