THE UNIVERSITY OF MICHIGAN

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Final Report

FLEXIBLE VERSUS RESPONSIVE ENGINES

Part I. Responsive Engine Systems for Ordnance Vehicles

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ABSTRACT

This report covers a first approach to the problems of producing an engine capable of developing a constant horsepower over a wide speed range, without sacrifice in space, weight, or fuel requirements.

Several methods of producing a flat horsepower output vs. speed are considered and the performance characteristics estimated. With but one exception, all involve high manifold, compression, and firing pressures, such that a major research program would be necessary to solve the problems. In addition a considerable development in the field of turbo-chargers and fuel injectors would have to occur simultaneously.

With the range of speeds considered practicable for constant horsepower at present there is still a need for a multi-step transmission, thus there is a negligible saving in fuel burnt. The best method of fuel reduction appears to be in the development of better combustion systems for the engines.

To evaluate the responsive process in a more exact manner than has been possible in this report, an improved and expanded list of hill-climbing and obstacle-crossing requirements seems necessary.

OBJECTIVE

The main objective of this investigation was to examine present and near-future possibilities for the compression ignition engine to fulfill the need for a power plant system having the following characteristics.

1. Flexible Power System.—A flexible power system being understood to be comprised of a compression ignition engine coupled to a transmission system and of such design that the engine will operate close to its maximum economy point under all conditions of vehicle speed, load, and ground conditions.

2. Responsive Power System.—A responsive power system being one in which the power plant will deliver constant maximum horsepower output over a wide range of engine speed, reducing the need for complicated multi-speed transmissions with their accompanying losses.

In addition to the above, an examination of other power sources such as turbine, compound engine, etc., was to be make in order to explore the possibilities of various combinations in a first-approach, broad manner.

As a result of this investigation, conclusions are drawn as to the direction in which such research as may be necessary should be directed in order to achieve the most desirable power system for Ordnance vehicles.

SUMMARY

A number of methods have been investigated by which various degrees of responsive engine operation can be secured; examination of the general over-all results permit the following observations and conclusions.

1. The maximum degree of responsiveness examined and believed to be practicable at present was 3:1, using a turbo-charged engine fitted with means to maintain air flow as engine speed was reduced.

2. Such an engine would need to be developed, both engine and turbo, since it would involve mean pressures far in excess of modern practice accompanied with roughly twice the present-day firing pressures.

3. The fuel economy of such an engine shows no marked improvement over that of existing engines, and is in fact somewhat inferior under certain load conditions.

4. The need for a transmission would still exist unless the range of responsiveness could be increased from 3 to at least ≧ 12:1. This would involve prohibitive mean and maximum pressures unless a wide variation in compression ratio during operation could be achieved successfully.

5. To produce such an engine there would be the need for an aftercooler behind the compressor, adding to the weight and particularly to the over-all volume of the power package. Without such a cooler the predicted power outputs could not be achieved.

6. All other methods of producing responsiveness resulted in a degree less than 3:1 varying from a small value to as much as 2:1.

7. All methods showing any reasonable magnitude of responsiveness except one resulted in increased pressures, etc., throughout the cycle.

8. The Air-Boosted cycle indicated that a responsiveness of 2:1 could be practicable, without any serious additional strain on the engine parts, at some small increase in fuel flow during the period of responsive operation only.

9. Compounding under certain conditions gave both power increase and improved fuel flow simultaneously, though the responsive range was small.

10. In order to exactly evaluate the results of this analysis, additional rules for engine operation and requirements should be made available, such as:

(a) The exact horsepower range in which future engine design will be concentrated and vehicle weight and rolling resistance likely to be encountered.

(b) The time period during a battlefield day in which the responsive condition will be needed.

(c) The type of hill climbing and obstacle crossing considered typical.

11. If the responsive engine does meet a need, then an investigation into the fuel injector requirements for such operation must be made. Presently used equipment could not meet this wide range of conditions.

12. Research would also be needed in the fields of (1) increased manifold pressures, (2) the turbos to produce such pressures, and (3) a single-cylinder research engine to digest such pressures.

13. Responsiveness will add to the need of producing a variable compression ratio engine.

14. An immediate problem would be the analysis of a complete transmission to match the proposed responsive engine selected. This involves relating the engine and transmission phases of this contract to a concrete problem in order to produce a combined unit.

15. An investigation of an engine with a small degree of responsiveness coupled to a conventional step speed transmission should be made. It is possible that almost all the advantages of a highly responsive engine could be secured by this means without solving the difficult problem of very high mean effective pressures.

16. The combination of a simple turbine and reciprocating engine has some attractive features, provided fuel quantity per day is not an important item. The complexity of the controls may also limit its usefulness.

17. The time required for turbine warm-up could be quite important, if the turbine is used for the emergency powers only. If the unit were idled continuously to keep it hot, the amount of fuel supply consumed per 24 hr would be increased enormously.

18. The use of a regenerated turbine in combination with an oil engine does not seem too attractive.

19. A regenerated gas turbine with a high effectiveness of regeneration might prove competitive if the bulk could be kept down and a reliable regenerator which would stand up under the service conditions encountered could be produced.

20. Time did not permit much consideration of a flexible engine arrangement. It appears at first glance that this arrangement would not involve the serious problems of a responsive engine. It would transfer the problems from the engine to the transmission.

21. If fuel economy is of great concern, greater improvements in the use of the fuel appear possible by research on the cycle to improve the combustion phase (resulting in reduced fuel flow for a given power, or increased power for a given fuel flow) than can be made by the use of responsiveness.

INTRODUCTION

The compression ignition engine has shown its capabilities as a 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 characteristic 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 control methods, a wide range of engine outputs become possible which could result in simplifying the transmission between the engine and track, or in better performance of the vehicle.

Any transmission involving torque converters is notoriously inefficient when operating under conditions of torque multiplication; this condition is frequently met in military operation, with the result that when a limited fuel tank capacity is dictated by space limitations, the range of the vehicle is limited and the transportability as measured in ton/mile/gallon of fuel is greatly reduced, compared with the conventional step speed gear type of transmission. However the latter type suffers disadvantages of driver fatigue and absence of immediate response in emergencies. Secondly, the need for economy in weight and space dictates the use of high-speed engines which in turn results in high gear reductions; thus so many steps are required in a charge gear system for maximum vehicle flexibility acceleration, etc., that the transmission becomes large and heavy, offsetting much of the weight and space saving of the engine.

At the same time it must be remembered that, in the case of a track-laying vehicle, output sprocket speed is relatively low (about 400 to 550 rpm at full engine speed) on a hard surface road and as low as 20 to 50 rpm at slow vehicle speeds and heavy going on rough terrain. The result is that high torques are involved in the final drive, resulting in large heavy gears which, when coupled with the needs of braking, steering, etc., produce a certain minimum transmission casing size, weight, etc., irrespective of the power plant involved.

It follows that the ideal simplest over-all power system must (1) consist of this final drive to the sprockets, (2) be quite heavy and large, and (3) be coupled to a high-speed engine, if it could be made of the responsive type, eliminating the variable speed transmission between the sprocket drive and engine.

Alternatively, the problem could be solved by the use of the final drive mechanism, coupled to a completely variable transmission joined to an engine which operates at only one speed at its maximum economy point as far as is practicable; the mechanism must have a reasonable speed and load range of good economy along which the transmission would permit operation as required at all times.

It is proposed to examine from a theoretical and practical basis as far as is possible the conditions under which such engines will perform and evaluate the practicality and usefulness from logistical and design conditions for the applications involved in military service.

SECTION 1

ENGINE PERFORMANCE PREDICTION

1.1. METHOD

In order to carry out the terms of the contract, 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 coller 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.

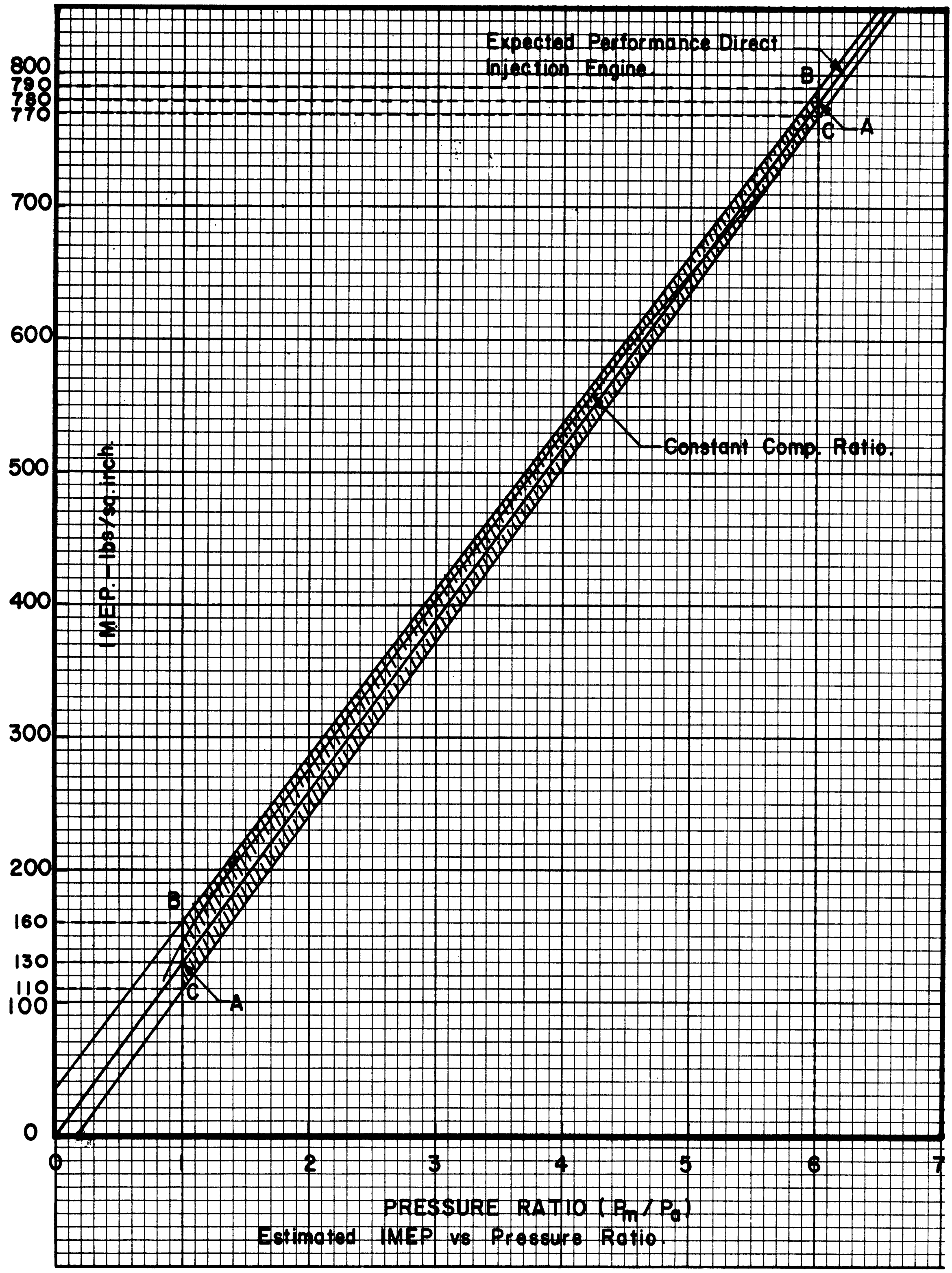

Fig. 1. I.M.E.P. vs. manifold pressure ratio.

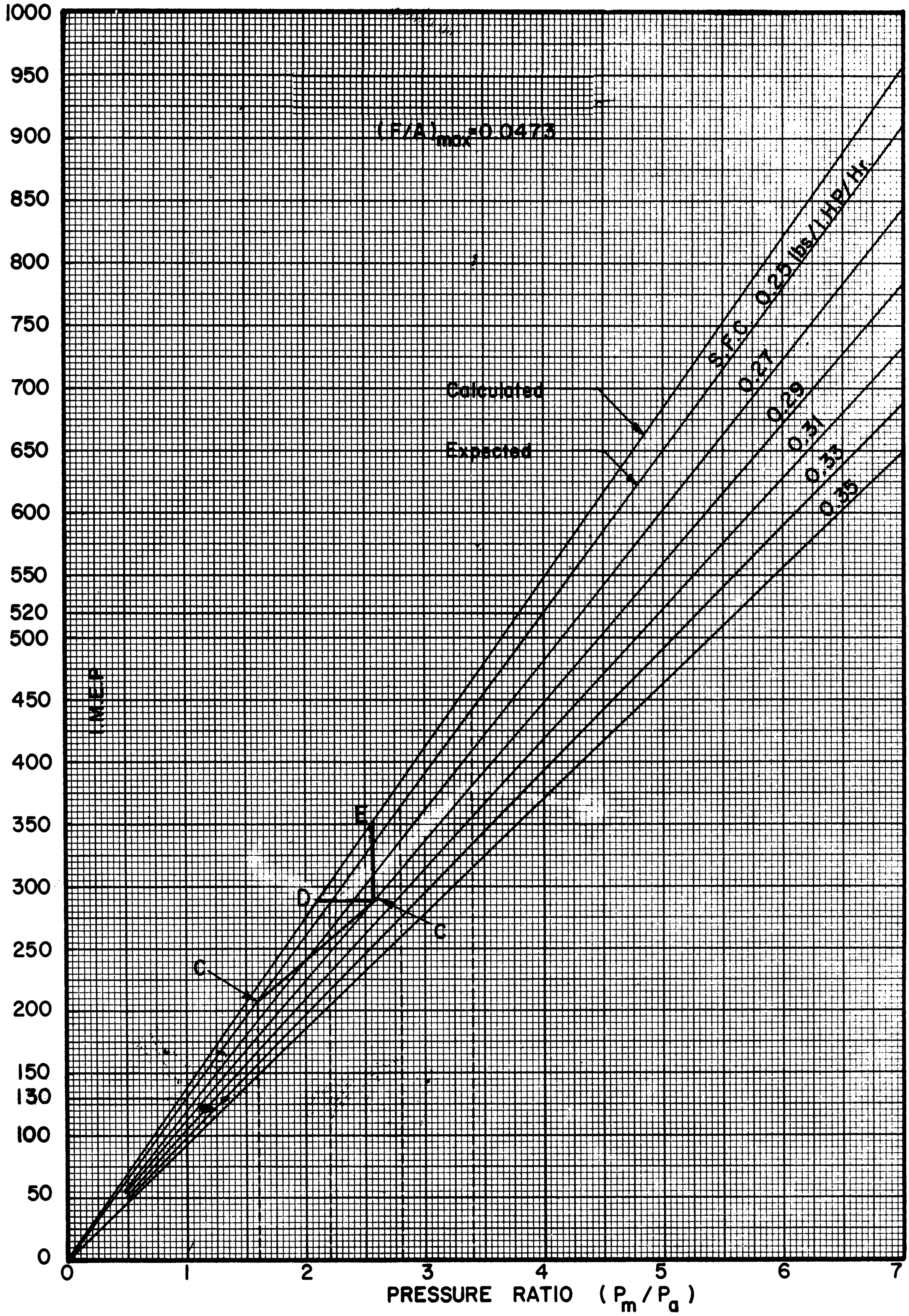

Fig. 2. I.M.E.P. vs. pressure ratio for various combustion efficiencies.

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 0.005 to the I.M.E.P. predicted for the 0.0473 ratio; e.g., if F/A of 0.005 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 0.005 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-

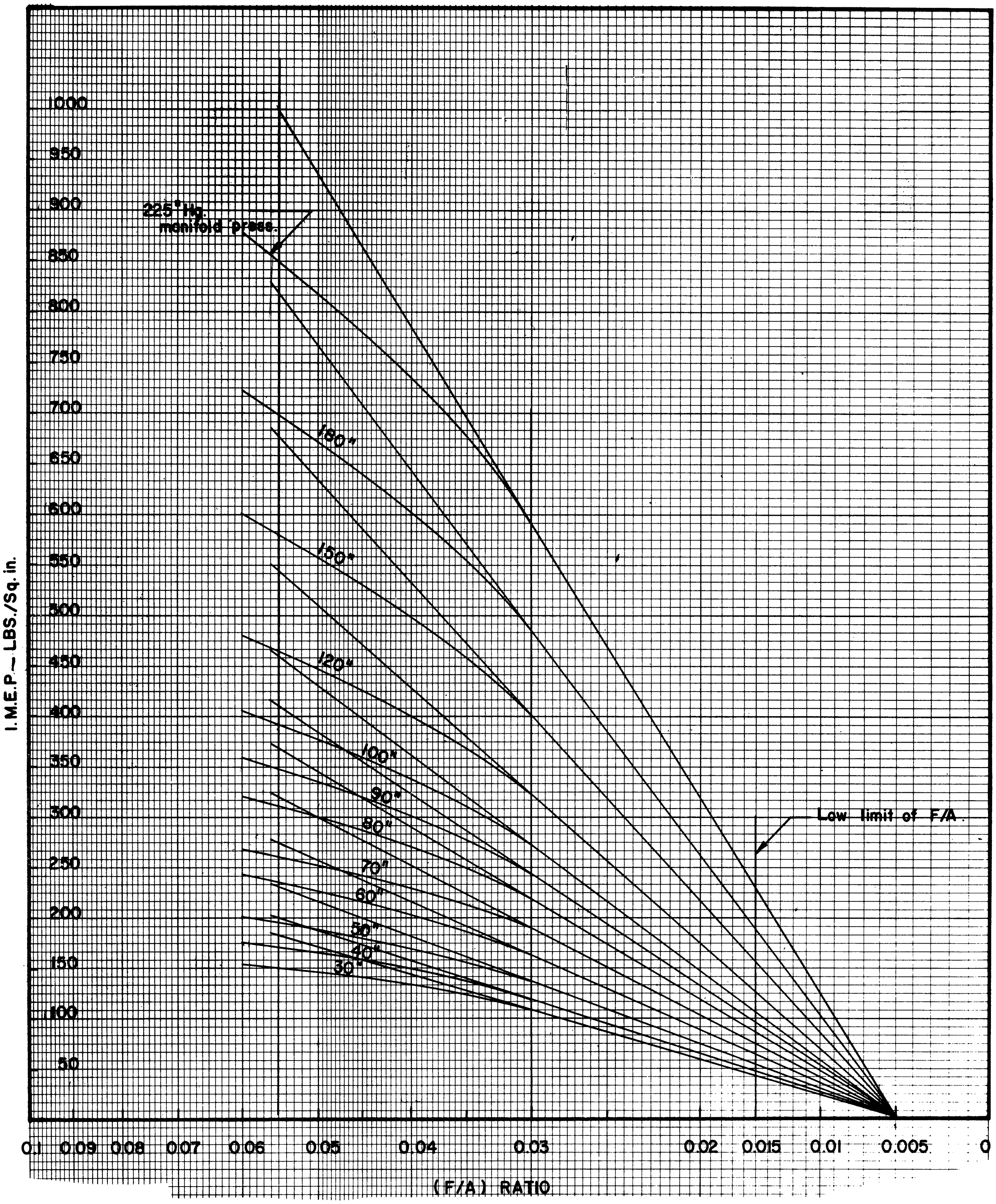

Fig. 3. I.M.E.P. vs. F/A and manifold pressures.

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 0.01. 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%

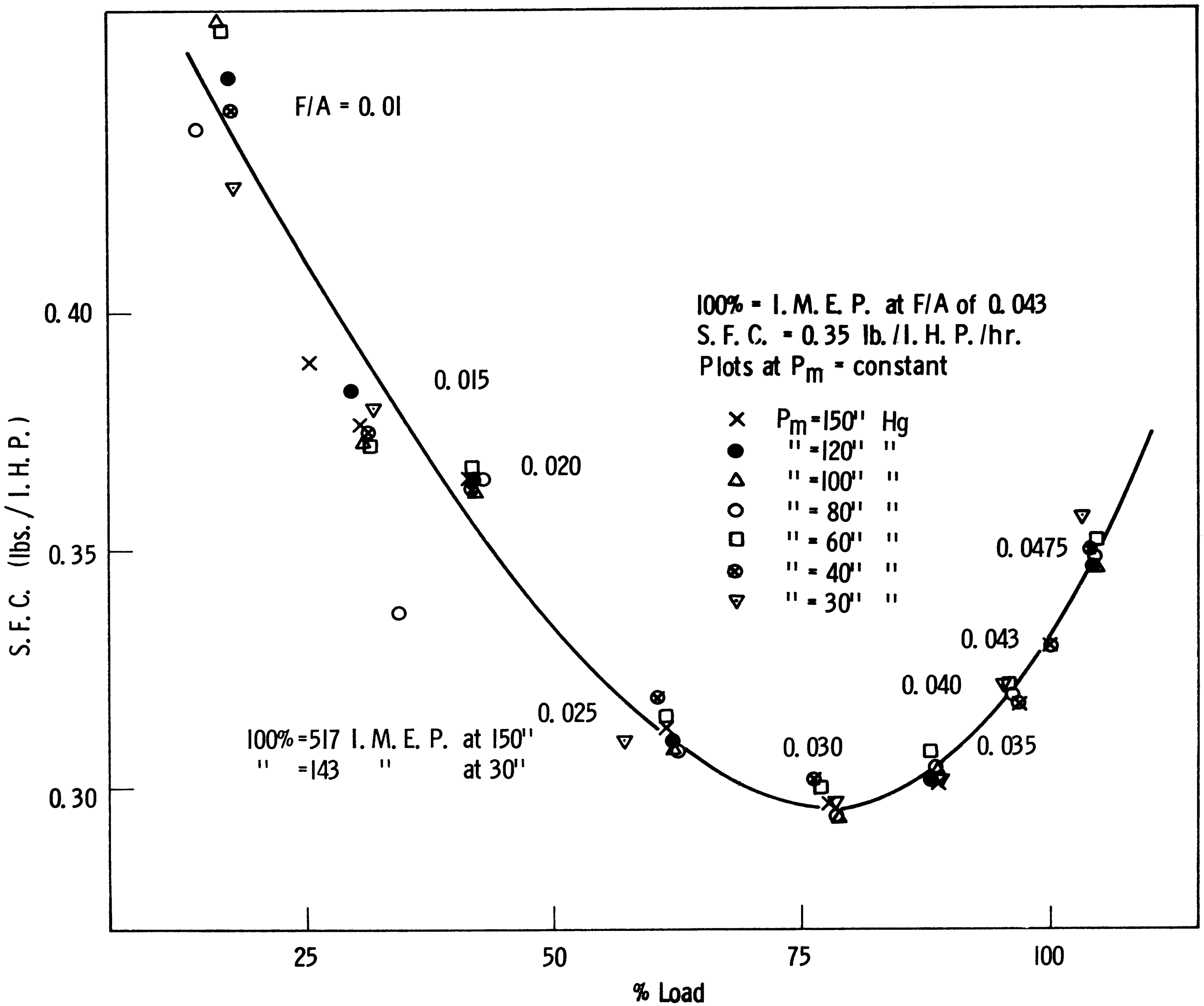

Fig. 4. S.F.C. vs. % load at constant manifold pressure.

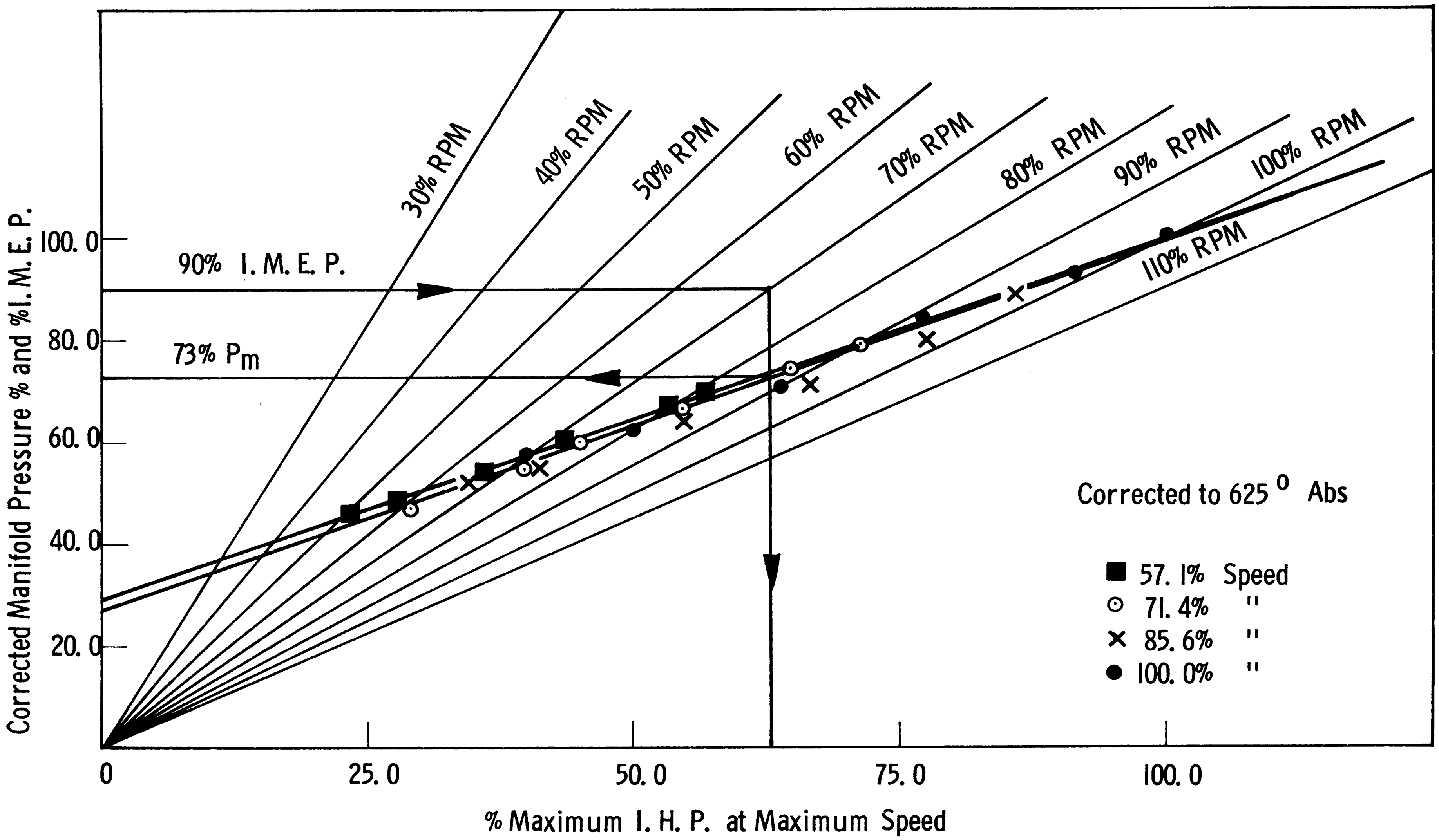

Fig. 5. % M.E.P., manifold pressure, and I.H.P. for turbo-charged engine.

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

$$\begin{aligned} F/A &= 0.0375 \times \frac{100}{57} \\ &= 0.0658 \ . \end{aligned}$$

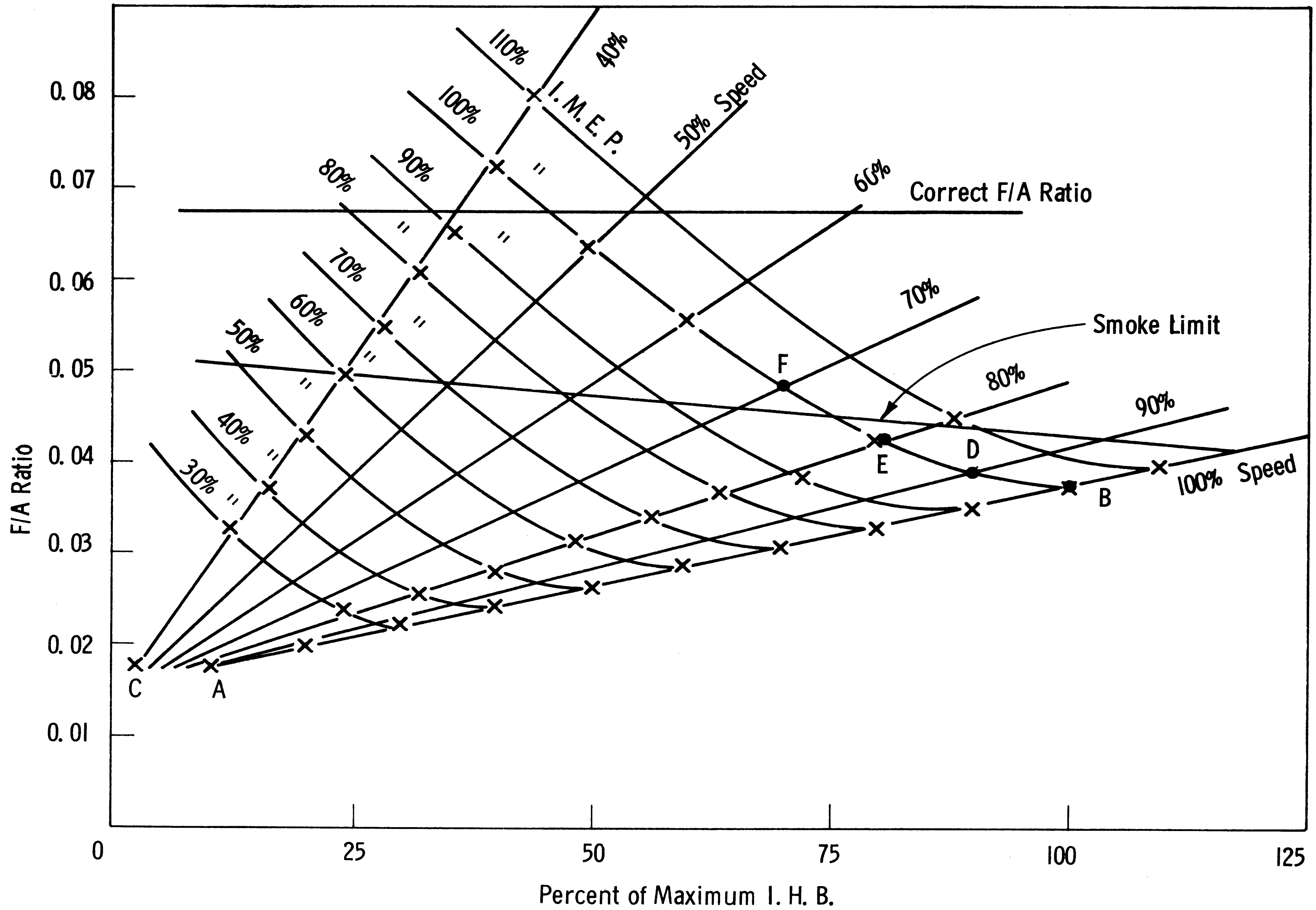

Fig. 6. Percentage performance plot of F/A, rpm, I.M.E.P., and % power.

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 Table I 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 re-

TABLE I

I.M.E.P. AND MANIFOLD PRESSURES

I.M.E.P. (%)	Speed (%)	Load (%)	Manifold Pressure			F/A Ratio		
			P_{100} (%)	P_{80} (%)	P_{40} (%)	P_{100} y=1.0	P_{80} y=.95	P_{40} y=1.1
110	100	110	107			.04		
	80	88		90			.0448	
	40	44			59			.0798
100	100	100	100	84	57	.0375		
	80	80					.0425	
	40	40						.0725
90	100	90	91			.035		
	80	72		79			.0384	
	40	36			54			.0650
80	100	80	85			.033		
	80	64		73			.0366	
	40	32			51			.0605
70	100	70	78			.0305		
	80	56		67			.0338	
	40	28			48			.0545
60	100	60	71			.0285		
	80	48		62			.0310	
	40	24			45			.0494
50	100	50	63			.0260		
	80	40		56			.0279	
	40	20			42			.0428
40	100	40	56			.024		
	80	32		51			.0252	
	40	16			40			.0369
30	100	30	50			.022		
	80	24		45			.0234	
	40	12			37			.0327

sults. 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 combustions 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.

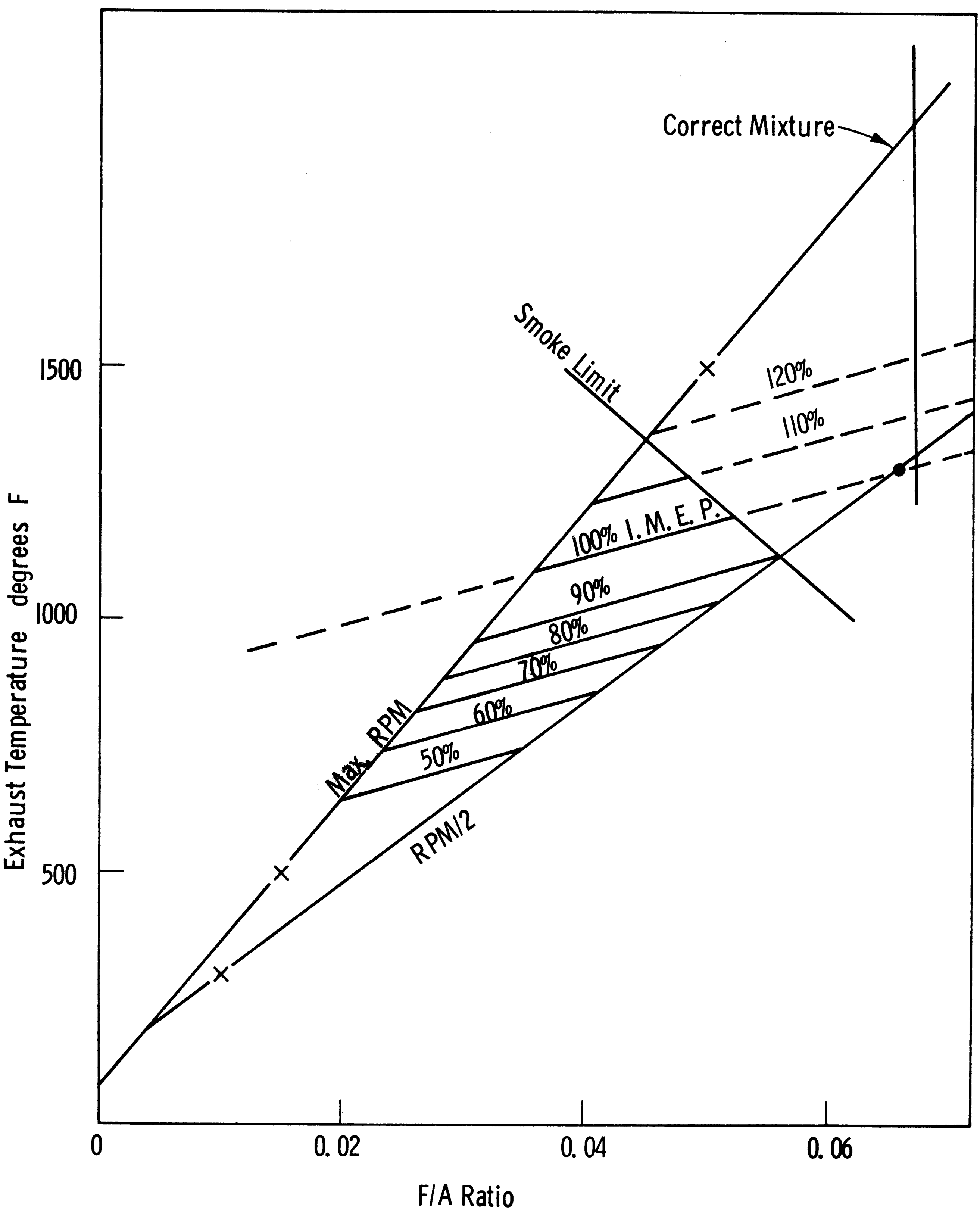

Fig. 7. Exhaust temperature vs. F/A ratio (turbo-charged engine).

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; this is expressed by Eq. (2a).

$$\text{Change of enthalpy} = wC_p[T_1 - T_2]\text{Btus} \tag{2a}$$

where

w = lb of gas flow
C_p = specific heat at constant pressure
T_1 = gas inlet temperature ° abs
T_2 = gas outlet temperature ° abs.

There are of course many methods of compressing and expanding gases; the method having the greatest efficiency occurs under conditions of no heat transfer and is called the isentropic process. This is the process that would take place in an insulated cylinder and represents what may be considered the ultimate in a practical process; hence it is employed to define the efficiency of the process.

In the case of the exhaust gas turbine, there is some pressure ratio across the turbine, at least in high turbo-charged units. Thus if expansion were isentropic, the outlet temperature (indicated by T_2' for this type of process) is given by

$$T_2' = T_1 \left(\frac{P_2}{P_1}\right)^{k-1/k}$$

where

k = ratio of the specific heats of the gas.

If T_2 indicates the actual gas temperature for the same practical expansion according to PV^n = constant, then

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{n-1/n} .$$

It follows that for an isentropic expansion process

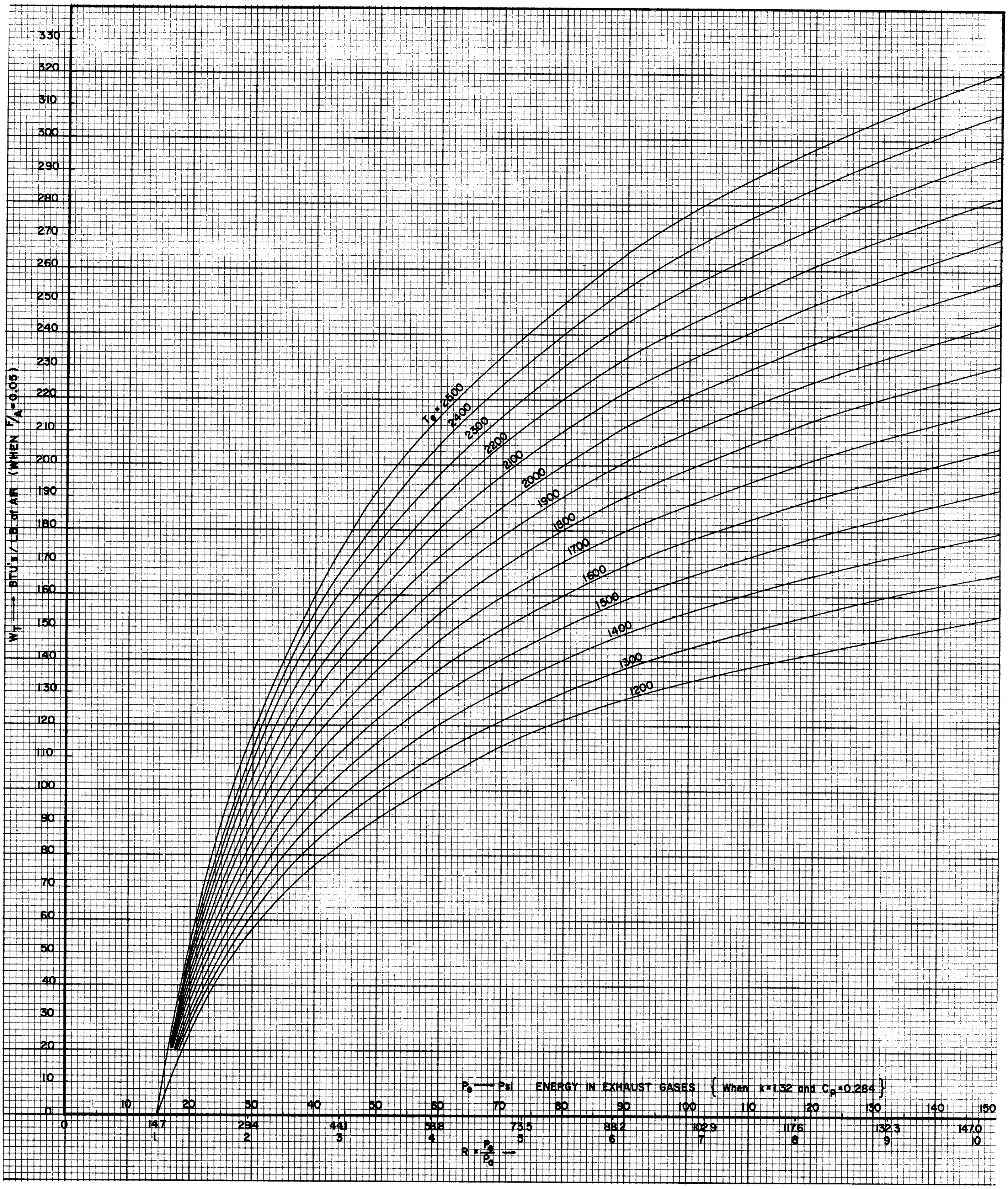

Fig. 8. Energy content of exhaust gases.

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

and for an actual process

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

The efficiency of the process is defined by

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

Now the exhaust gases of the engine when it is near full load are of a rather constant composition and temperature. It follows that the value of the gas constants and properties does not vary greatly despite the effects of variable specific heats, and that it is sufficiently accurate to employ mean values of k, C_p, and C_v to determine the energy available in the exhaust.

Since a large number of such calculations can be seen for the problems being investigated, a chart of the available isentropic energy/lb of exhaust gas was calculated for the case in which $F/A = 0.05$, $k = 1.32$, $C_p = 0.284$, and $C_v = 0.215$—a fairly average set of values. The chart was plotted on the basis that W_T Btus of work were performed by the mixture produced by 1 lb of air plus the 0.05 lb of fuel when subjected to an isentropic process. That is, it represents the maximum available energy/lb of air flow through the engine for various pressure ratios P_e/P_a where

P_e = exhaust manifold pressure
P_a = atmospheric pressure.

The range of temperatures covered is from 1200° abs to 2500° abs or from 740°F to 2040°F, which covers sufficient conditions for the average engine.

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.

In addition a corresponding evaluation for the gas turbine will be made. Here the state of the art is still in flux and it is impossible to approach the degree of accuracy existing for the reciprocating engine, since there are simple cycle engines, regenerated engines, supercharged engine cycles, etc., all employing a wide range of compression ratios and gas temperatures which produce a wide variation in size and weight. An attempt has been made to separate these various types, but each individual type is produced in a wide variation in bulk and weight, depending on the service for which it is designed. Consequently equations are developed for the average unit, even though this average can be vastly different from some of the individual designs existing. However the average does indicate what can reasonably be expected for any given unit.

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.

$$hp = PLAN/33000 \tag{3}$$

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.

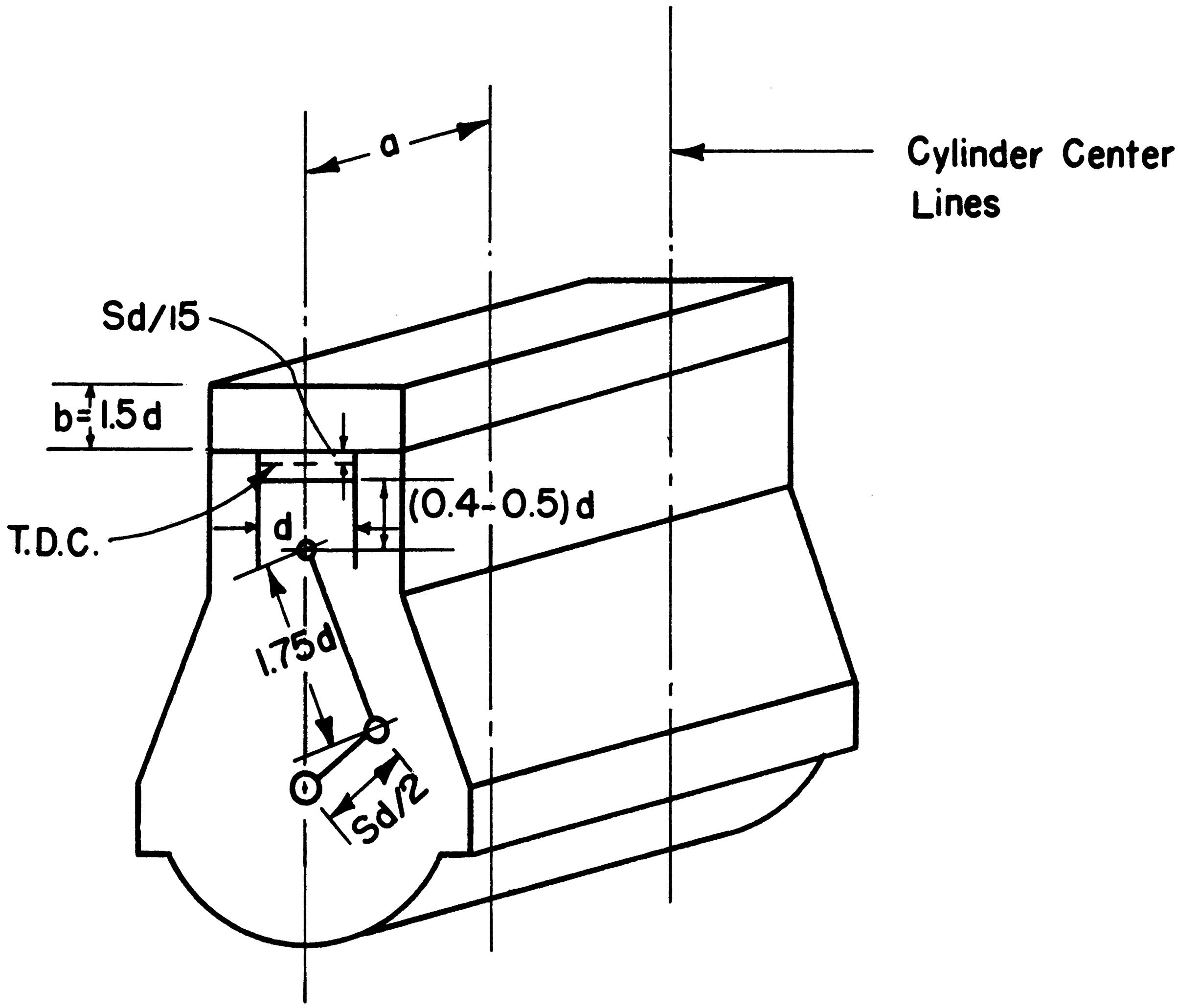

Fig. 9. Schematic diagram of in-line engine.

The value of "b" in Fig. 9, the distance from piston head at T.D.C. to the top of the walve 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:

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

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

$$\text{Over-all engine height} = (5) + (6)$$

$$= (1.067S + 4.15)d + 2 \text{ in.} \qquad (7)$$

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,

$$\text{Width of engine} = Sd + d + 2 + 1.4d$$

$$= d(S + 2.4) + 2 \quad . \qquad (8)$$

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:

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

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

(See Table II.)

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

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

TABLE II

90° VEE ENGINE VOLUME CALCULATIONS

Volumes in cu. ins $= d\left\{1.4n + 1.6\right\}\left\{(S + 2.4)d + 2\right\}\left\{(1.067S + 4.15)d + 2\right\}$

No. of CYLINDERS	n = 4			n = 6			n = 8		
CYLINDER DIA, d ins.	4	5	6	4	5	6	4	5	6
$L = (1.4n + 1.6)d$	28.80	36.00	43.20	40.00	50.00	60.00	51.20	64.00	76.80
$B = (S + 2.4)d + 2$	28.40	40.25	54.20	28.40	40.25	54.20	28.40	40.25	54.20
$H = (1.067S + 4.15)d + 2$	36.52	50.76	67.23	36.52	50.76	67.23	36.52	50.76	67.23
$V = \frac{L \times B \times H}{1728}$ cu.ft.	17.29	42.56	91.10	24.01	59.11	126.52	30.73	75.66	161.96

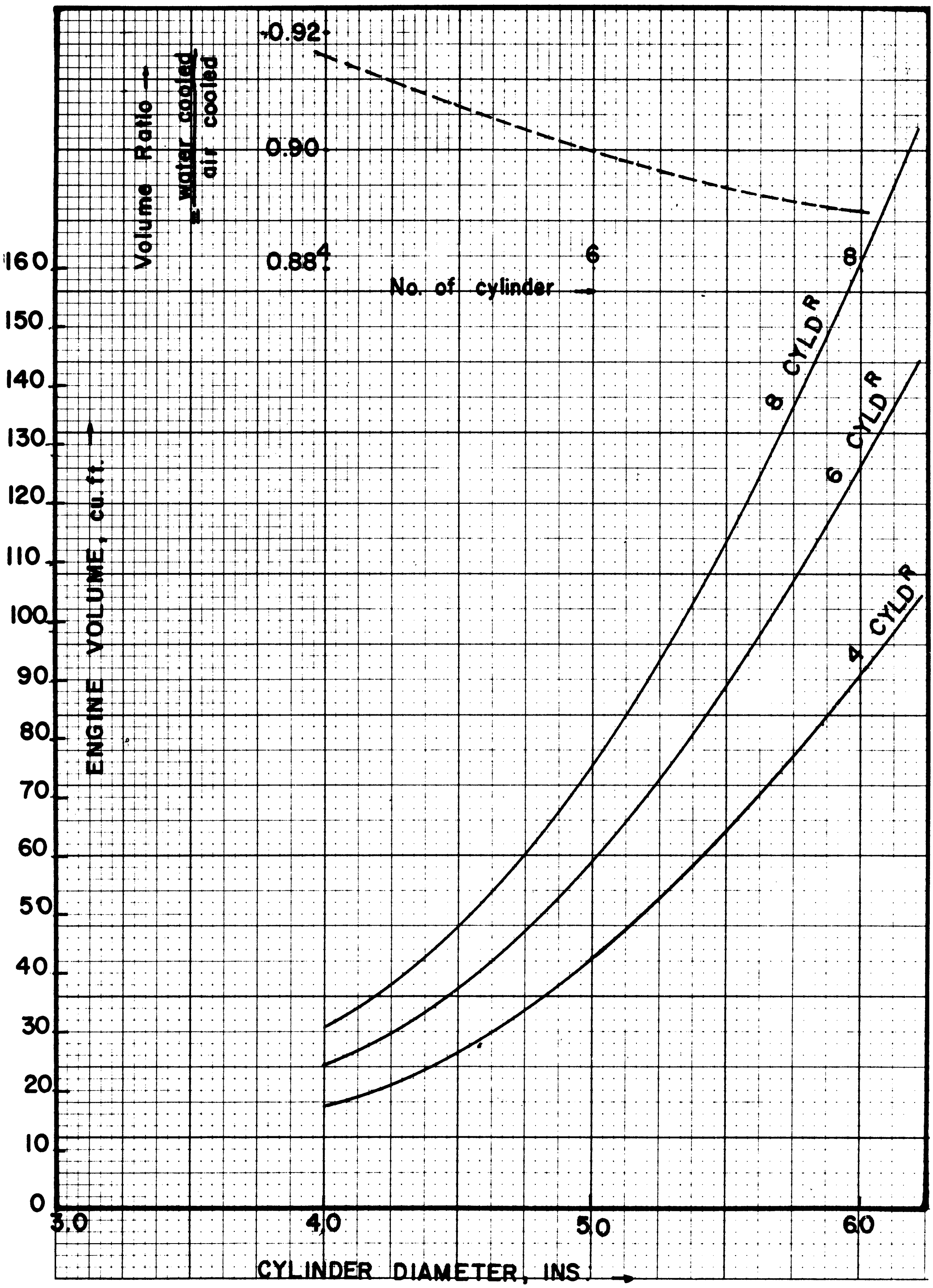

Fig. 10. Volume of in-line engines.

which for various cylinder numbers 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

$$\text{Engine height} = \left(\frac{Sd}{2} + \frac{d}{2} + 2\right) + \left(\frac{Sd}{2} + 1.75d + 0.4d + 1.5d + \frac{Sd}{15}\right) \cos\frac{\theta}{2} + 0.8d \sin\frac{\theta}{2} + 0.75d$$

$$= \left(\frac{Sd}{2} + 1.25d + 2\right) + (0.5667Sd + 3.65d)\cos\frac{\theta}{2} + 0.8d \sin\frac{\theta}{2}$$

$$= Sd\left(0.5 + 0.5667 \cos\frac{\theta}{2}\right) + d\left(1.25 + 3.65 \frac{\cos\theta}{2} + 0.8 \frac{\sin\theta}{2}\right) + 2$$

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

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

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

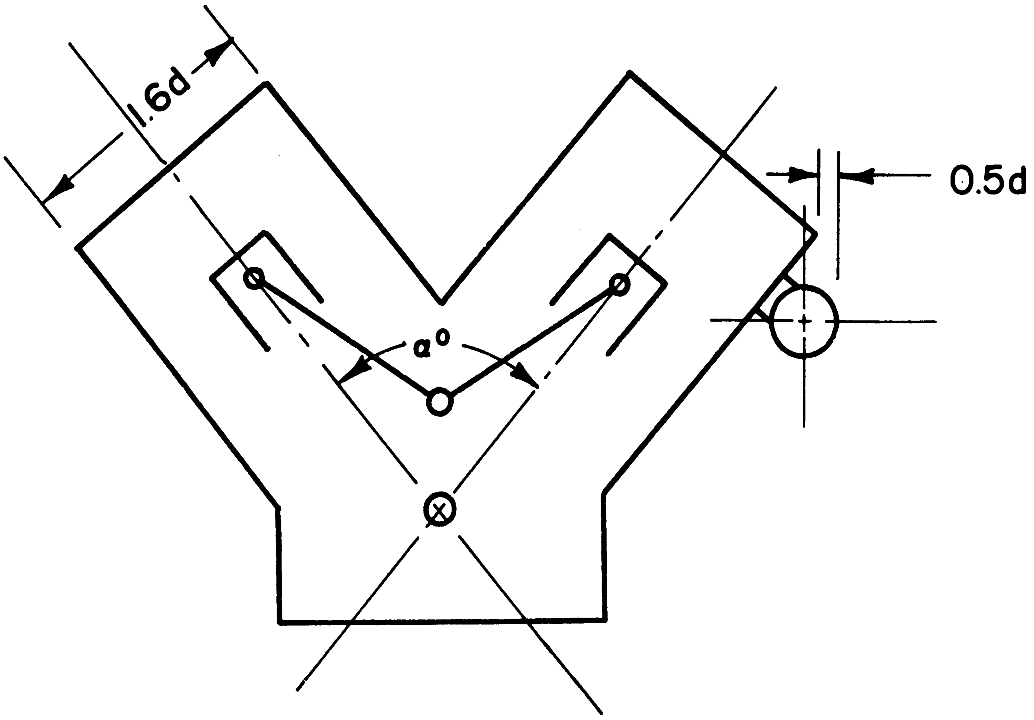

Fig. 11. Schematic diagram of Vee engine (I).

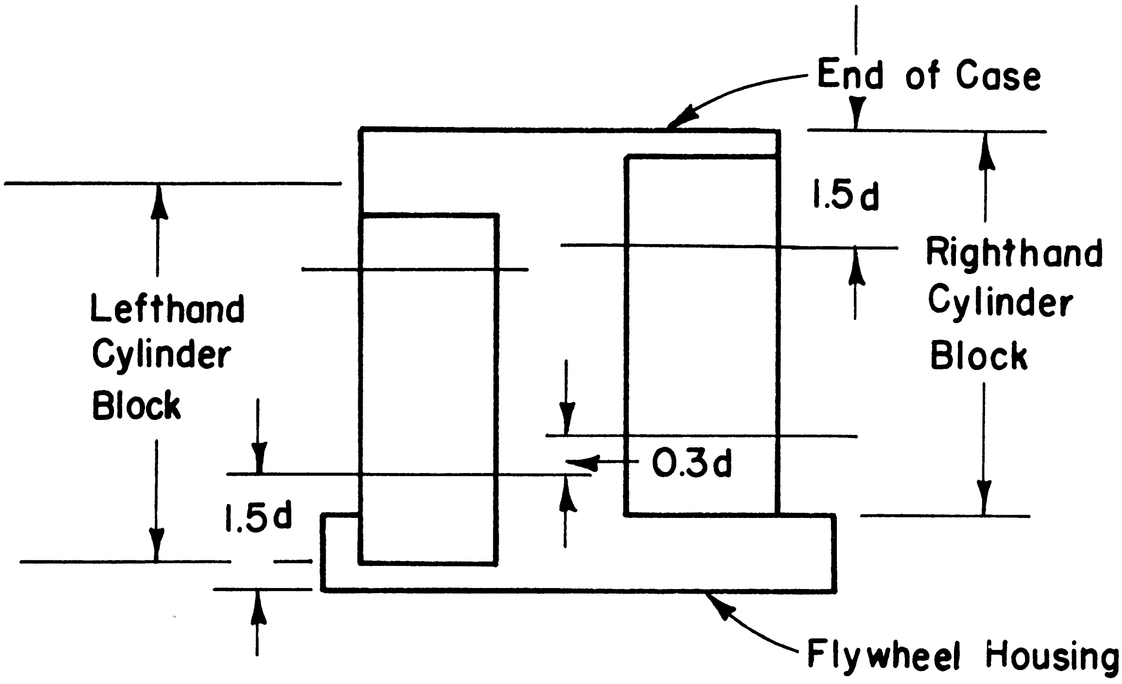

Fig. 11a. Schematic diagram of Vee engine (II).

Engine width

$$= 2\left(\frac{Sd}{2} + 1.75d + 1.9d + \frac{Sd}{15}\right)\sin\frac{\theta}{2} + 2 \times 0.8d \cos\frac{\theta}{2} + 0.5d$$

$$= 2(0.5667Sd + 3.65d)\ \sin\frac{\theta}{2} + d(1.6 \cos\frac{\theta}{2} + 0.5)$$

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

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

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).

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

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

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

Thus for the 90° Vee engine the volume becomes

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

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 Table III and Fig. 12.

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

$$\text{Height} = Sd(0.5 + .5667 \times .866) + d(1.25 + 3.65 \times .866 + 0.8 \times 0.5) + 2$$

$$= 0.9905Sd + 4.81d + 2$$

TABLE III

60° VEE ENGINE VOLUME CALCULATIONS

No. of CYLINDERS	n = 4			n = 6			n = 8			n = 12		
CYLINDER DIAMETER, d"	4	5	6	4	5	6	4	5	6	4	5	6
L = LENGTH of ENGINE. $L = 0.75nd + 1.8d$	19.20	24.00	28.80	25.20	31.50	37.80	31.20	39.00	46.80	43.20	54.00	64.80
B = BREADTH of ENGINE $B = 0.801Sd + 6.79d$ where, $S = 1.05d$	40.60	54.97	71.02	40.61	54.97	71.02	40.60	54.97	71.02	40.60	54.97	71.02
H = HEIGHT of ENGINE $H = 0.9Sd + 4.296d - 2$	34.30	47.10	61.79	34.30	47.10	61.79	34.30	47.10	61.79	34.30	47.10	61.79
VOLUME cu.ft. $= \frac{L \times B \times H}{1728}$	15.475	35.967	73.143	20.310	47.206	96.00	25.146	58.446	118.85	34.82	80.92	164.57

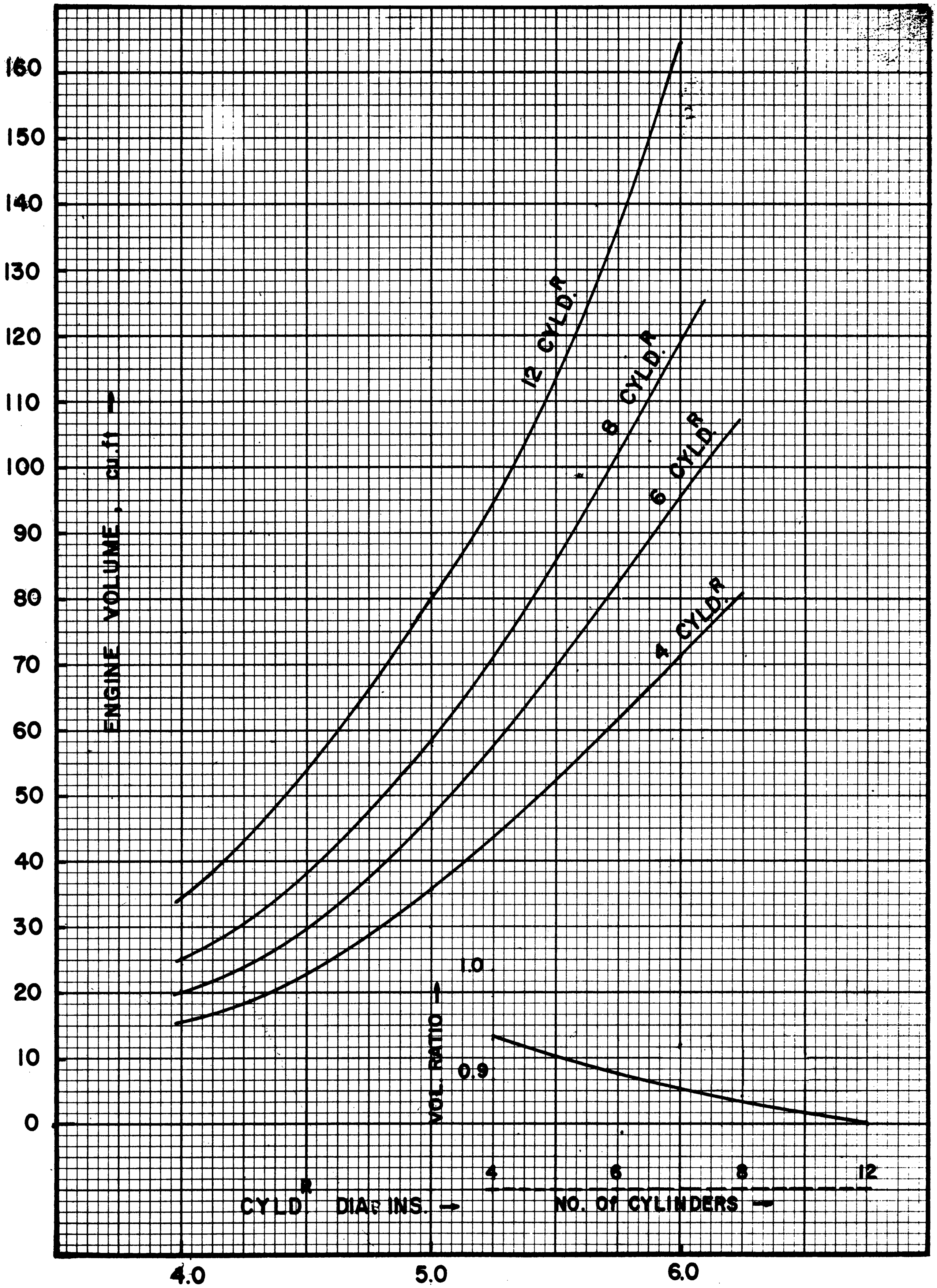

Fig. 12. Volume of 90° Vee-type engines.

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

$$= 1.1334Sd \times 0.5 + d(7.30 \times 0.5 + 1.6 \times .866 + 0.5)$$

$$= 0.5667Sd + 5.536d = 6.131d$$

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

calculating from these equations the values of Table IV result and the curves have been plotted in Fig. 13.

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

$$\text{Length of a water-cooled engine} = (\frac{n}{2} - 1)1.2d + 1.5d + 0.3d + 1.5d$$

$$= 0.6nd + 2.1d$$

$$= (0.6n + 2.1)d.$$

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

TABLE IV

500 HP RESPONSIVE ENGINE

No. of CYLINDERS	n = 4			n = 6			n = 8			n = 12		
CYLINDER DIAMETER, d"	4	5	6	4	5	6	4	5	6	4	5	6
L = LENGTH of ENGINE L = 0.75 nd + 1.8 d	19.20	24.00	28.80	25.20	31.50	37.80	31.20	39.00	46.80	43.20	54.00	64.80
B = BREADTH of ENGINE B = 0.567 Sd + 5.53 d	31.645	42.534	54.612	31.645	42.534	54.612	31.645	42.534	54.612	31.645	42.534	54.612
H = HEIGHT of ENGINE H = 0.99 Sd + 4.81 d + 2	37.87	52.03	68.28	37.87	52.03	68.28	37.87	52.03	68.28	37.87	52.03	68.28
VOLUME cu.ft. $= \frac{L \times B \times H}{1728}$	13.31	30.74	62.14	17.46	40.34	81.55	21.62	49.94	100.98	29.98	69.15	139.81

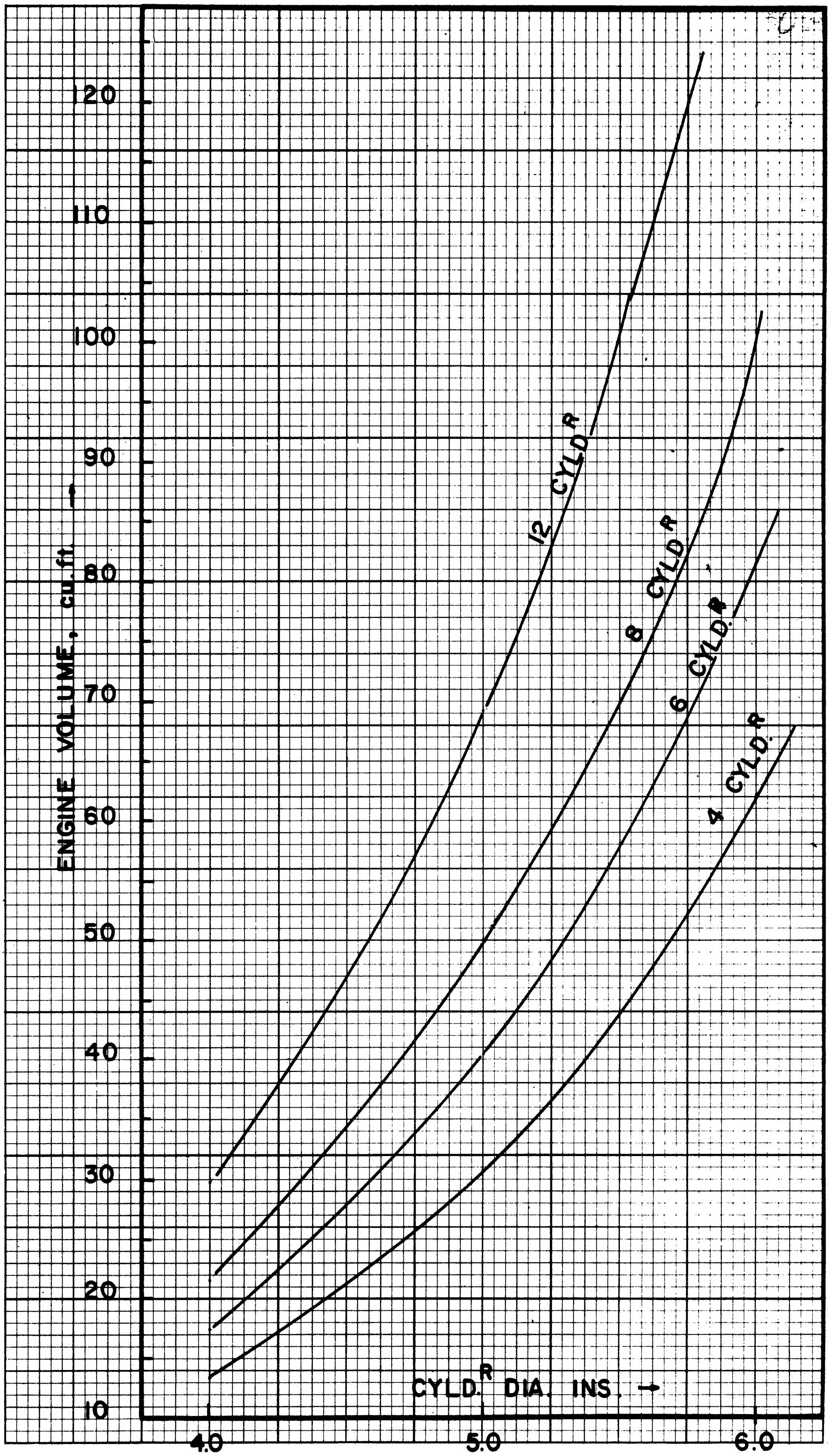

Fig. 13. Volume of 60° Vee-type engines.

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. GAS TURBINE POWER PLANTS

The mass of data available on the turbine engine is somewhat restricted at the present time due to lack of a large number of applications. Most turbine units are employed as jet engines, which is an altogether different application to that being considered.

In order to approach the weight and size relationship for this type of power plant, use was made of the Statistical Issue of Automotive Industries for March 1962, where data are given regarding published units for all purposes.

Calculations from the data available were made regarding volume, volume per hp, lb/cu ft, lb/hp, etc., for all the units for which sufficient data were given. This material was then plotted in various ways but in all cases there was a rather wide scatter in the points for similar machines. This scatter reflects the lack of past history; it is believed that time will eliminate unsatisfactory units and that the band of points will contract until only a moderate variation (such as that provided by standard design ideas) will exist. The gas turbine size is also affected by the air flow per hp, which changes rapidly with the gas temperature employed; this temperature seems to vary from a low of 1380° to a high of 1740° at the present time. The life expected at 1740° is of course lower than that at 1380° for corresponding heat resistant materials; hence engine specification again enters the problem when considering the gas temperature to be employed.

One other variable was the number of compression stages which ranged from 1 to 4; some units were aircraft conversions, others had regenerators of various types, and still others were of the simple cycle. In view of all these variations some scatter is understandable.

An attempt was made to take major variations, with or without regenerators, into account. The results obtained are as follows:

Engine Volumes

Units up to 300 hp

$$\text{Total volume} = 0.058 \text{ hp} + 3.0 \text{ cu ft} \tag{18}$$

Units from 150 to 750 hp

$$\text{Total volume} = 0.0153 \text{ hp} + 12.5 \text{ cu ft} \tag{19}$$

Units from 0 to 3600 hp

$$\text{Total volume} = 0.0903 \text{ hp}^{0.844} \text{ cu ft} \tag{20}$$

The different formulas given above for the various ranges indicated result in the best prediction being made at the large end of the range. It follows that the smallest units should be projected by the use of Eq. (18) and the largest by 20.

It is believed that the engine volume can be predicted with somewhat greater accuracy if the data are plotted in the form of cu ft/hp vs. hp and the equations given below result.

Units of 100-750 hp unregenerated

$$\text{Volume/hp} = \frac{8.27}{\text{hp}^{0.92}} \text{ cu ft/hp} . \tag{21}$$

Here again the greatest variation is in the small sizes:

Regenerated units 50-300 hp

$$\text{Volume/hp} = \frac{79.8}{\text{hp}^{1.27}} \text{ cu ft/hp.} \tag{22}$$

Engine Weights

Plotting the lb/cu ft of engine volume, the following relations result:

Units of 50-750 hp

$$w = 6.58 \text{ hp}^{0.2} \text{ lb/cu ft} \tag{23}$$

where

w = lb/cu ft of over-all volume
hp = engine output.

Employing the lb/hp relationship the equation becomes for unregenerated units

Units of 50-750 hp unregenerated

$$W = \frac{35.05}{\text{hp}^{0.644}} \text{ lb/hp} \tag{24}$$

where

W = lb of weight/hp

50-350 hp regenerated units

$$W = \frac{111.8}{\text{hp}^{0.71}} \text{ lb/hp} . \tag{25}$$

In all cases the most scatter appeared in the lower power ranges, as would be expected.

With the above relations it is believed that a first approximation to the weight and volume of gas turbine power plants can be made. The results will be sufficiently close for a broad systems analysis approach to the problem of the relative merits of turbine and reciprocating engines.

Individual designs to meet special cases can vary considerably.

1.8. 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.

So far as is known these methods are the first comprehensive set for such calculations. Of course, one skilled in the art does not need such a complete calculated analysis.

In the sections to follow, examples are given of how these methods as well as other available methods can be used. The practical use of all the methods are illustrated so that the reader can make some selection to suit his problem.

SECTION 2

THE RESPONSIVE ENGINE

The work schedule specified by the contract for this project covers "Flexible Power Systems," and "Responsive Engines." Of these two it is believed that the latter needs more attention since it involves the use of an entirely new method of engine operation.

2.1. DEFINITION OF RESPONSIVE ENGINE

In the conventional engine and transmission system this combination is employed since the power output of an internal combustion engine depends upon its speed of rotation; it is high when the speed is high, and low when the speed is low. It follows that for hill climbing or heavy going where the vehicle speed is low the engine speed would also be low if the engine is coupled directly to the wheels; the power output would be low only when high power is demanded. The function of the transmission is to adjust the engine speed to vehicle requirements so that full engine power at full speed can be made available to the wheels under all vehicle-speed conditions. In other words, the transmission is designed to permit full engine power to the road wheels under any conditions of vehicle speed when necessary, by keeping the engine speed at its peak value at all times and changing its gear ratio to the wheels.

As is well known, all transmissions have losses of one sort or another; thus for a given vehicle performance the engine power required will vary as the losses vary. A good example of this is the modern automobile with torque-converter transmissions; complete speed variation is possible but the losses are considerable at anything but high speed. The fuel consumption in town driving is poor but not too bad on the highway. The result is that an engine horsepower of 250-300 must be employed when only about 40-50 hp is used for a large percentage of the time; the high power is required to provide the required acceleration during those few minutes in starting from rest when the efficiency of the transmission is poor—approaching zero, in fact. By eliminating the losses, the engine power, size, and weight can be reduced. Hence the idea of a responsive engine—one which will deliver constant horsepower equal to its maximum under all conditions of engine and vehicle speed, with the result that no transmission is required and no losses are involved. The responsive engine can be defined as follow.

A "responsive engine" is an internal combustion engine capable of delivering a constant power equal to its maximum horsepower, irrespective of engine speed and over all conceivable operating speeds of the vehicle.

The ultimate or ideal responsive engine would be directly coupled to the road wheels without transmission of any sort. Such a power plant would of course be large and heavy due to the low speed of operation. It follows that a practicable responsive engine would be high-speed, of light-weight design, of the required size and power, and geared to the road wheels in such a way that at full speed of the vehicle on a level road the engine speed is that at which the designed full power can be developed. The required gear ratio would be fixed at such a value to effect this combination and would be incapable of being varied; thus the minimum number of gears are required involving the least losses, and the maximum in engine power output is available to the wheels.

The problem then arises of reduced vehicle speed which directly reduces engine speed proportionally, and in turn reduces the power available to the wheels. This combination would soon stall out on rough terrain or hills and the vehicle would be immobile.

In order to achieve the desired objective, maximum horsepower must be maintained; and since

$$B.H.P. = PLAN/33000$$

where L and A are fixed the product PN must be a constant for all engine speeds, the plot of mean pressure vs. rpm being a hyperbola; if the speed is halved the mean pressure doubles, etc.

Assume a clutch is provided in addition to the gear to permit engine operation while the vehicle is stationary; then the speed ratio of the engine would be equal to the normal vehicle operation speed ratio. In the case of a tank this might be from 3 to 45 mph or a range of 1 to 15; it follows that the mean effective pressure must also vary by 1 to 15 to fulfill the desired requirements.

If a modern or advanced turbo-supercharged engine is considered, its mean effective pressure at full power and speed would be of the order of 200 to 300 psi. Thus at low vehicle speed the engine would have to be capable of developing 3000 to 4500 psi mean effective pressure in order to be responsive. This is obviously an impossibility at the present time and probably in the far distant future also. The ratio of maximum to mean pressure in present-day engines is about 7-10:1. If such a ratio were maintained in the responsive engine, impossible peak cylinder pressures of 21,000 to 45,000 psi would be involved. (A scheme is outlined later according to which little if any increase in cylinder pressure would be involved for a moderate amount of responsiveness.)

It follows that some partial degree of responsiveness is the best that can be achieved within the forseeable future, with the result that some portion of the transmission must be retained to offset the lack of complete responsiveness. If a 2:1 responsiveness were possible (i.e., constant horsepower over a 2:1 speed range) then theoretically the over-all gear ratio could be reduced by 2, etc.

So far nothing has been said regarding the requirement for the steering of tracked vehicle; the need for this function to be incorporated into the transmission also dictates the need for some gearing to remain in the system. It thus becomes necessary to evaluate the advantageous effects, mainly ton-miles per gallon of fuel, against the disadvantages of extremely high mean effective pressure at low engine speed.

The reduction in over-all transmission ratio due to responsiveness does not seem to provide any major reduction in size or weight of the transmission since the part still needed—the larger, heavy, slow-speed components—generally will be unchanged while the high-speed, normally small, light-weight gears will have to be increased in size and weight in order to standup to the high torque resulting from the increased mean pressure at low speeds.

2.2. RESPONSIVENESS

The degree of responsiveness of an engine can be represented on a diagram such as Fig. 14; for details of its construction see the first Progress Report for this contract.[2] The diagram in Fig. 14 plots M.E.P., B.H.P./cu in., and rpm in one quadrant, with engine displacement against B.H.P. in the other.

For a given engine operating responsively a constant horsepower is developed and thus a constant hp/cu in. of displacement. Hence as shown in Fig. 14 a 400 hp engine with a 400 cu in. displacement is responsive from 3950 to 1975 rpm if it can develop 200 B.M.E.P. at the high speed and 400 at the low, with no mean pressure falling below the horizontal line between these two points at any speed. A typical present-day mean pressure curve is shown by CDE. If this engine is rated at 2400 rpm it is seen that it can respond at its maximum hp/cu in. only over the speed range from 2400 to 2250 rpm. Increased responsiveness could be obtained, for example, by reducing the rated mean pressure from 150 to 100 psi; the responsiveness would then be from 2400 to 1500 rpm, not a wide range by any means. What would be the result? The engine maximum power has been reduced in the ratio of 100 to 150, and the output available is 66.6% of the engine's capabilities. It is seen that responsiveness is easily secured. To be ridiculous let 20 M.E.P. be the full load; then a 10:1 responsive speed range would only involve 200 M.E.P. at the low speed and these conditions could easily be met today. However for the engine to develop the required power it would have to be of tremendous size and weight, incapable of being installed on any sort of transportation vehicle.

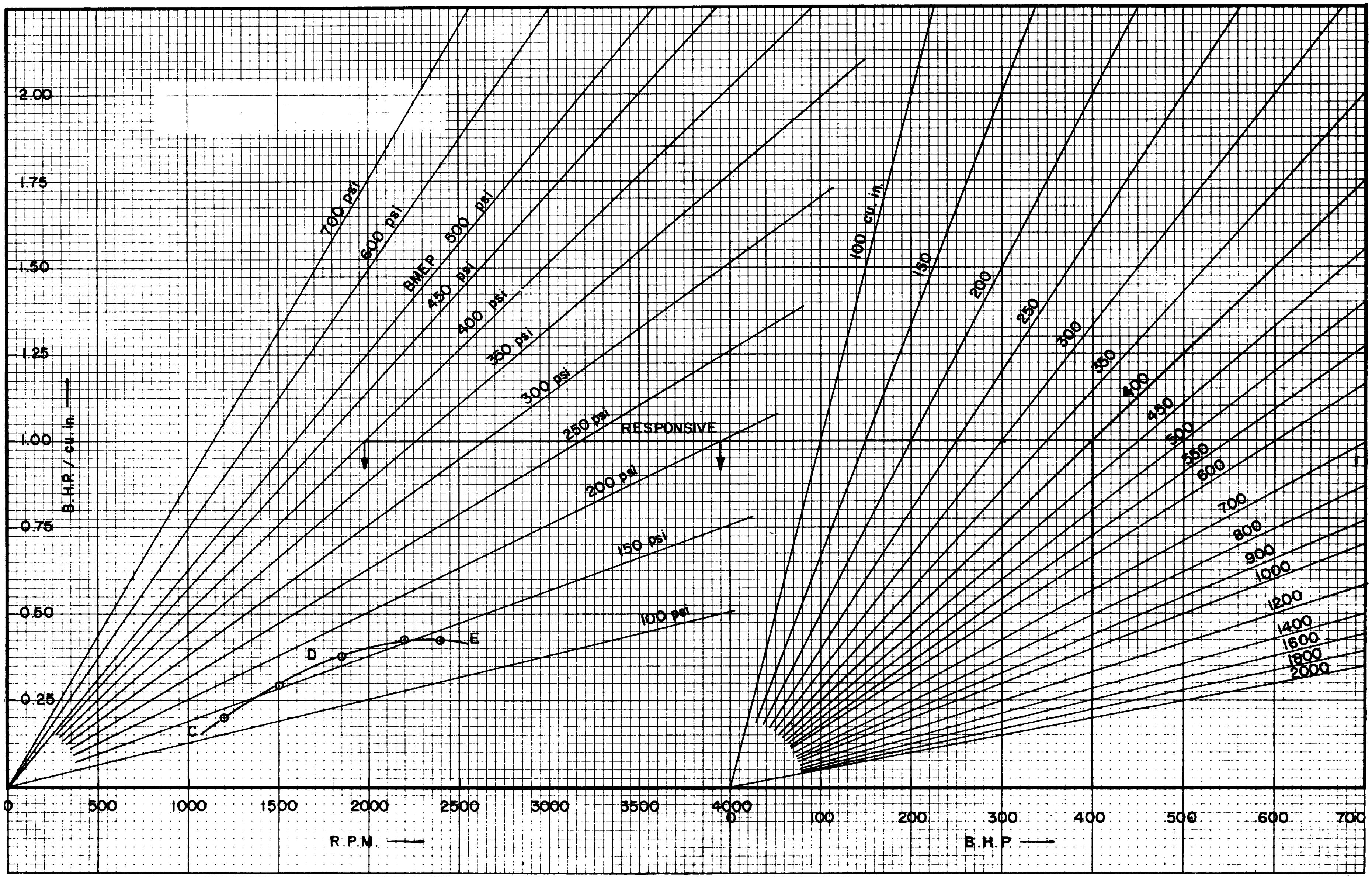

Fig. 14. Diagrammatic representation of responsiveness.

It is thus established that to be competitive little if any sacrifice can be made in what is normally understood as the full-load, full-speed operation. If this is the case the only solution to the problem is an increasing mean pressure as the speed is reduced. A small degree of responsiveness, say a 2:1 constant hp speed range, involves a 2:1 change in mean pressure. A mean pressure change of this magnitude is of major concern to the engine builder who has taken many years to increase engine output from about 100 B.M.E.P. to 200 psi; here by one change in method of operation 400 psi is needed immediately in order to be successful.

The manner in which the engine torque must vary for responsiveness is shown in Fig. 15, where torque/cu in. of displacement is plotted for speeds from 500 to 4000 rpm in a series of lines of constant horsepower. A B.M.E.P. scale is added at the top of the diagram and the arrows indicate the method of relating rpm torque and B.M.E.P. at any point. The rapidity with which the mean pressure increases is easily seen, together with the high torque for which the first gear on the engine output shaft must be designed if any large degree of responsiveness is aimed at.

From this brief discussion of responsiveness the following observations can be made:

1. A responsive engine will not eliminate the large, heavy, slow-speed gearing of a typical tank drive since it will still be necessary for braking, steering, etc.

2. The over-all gear ratio from engine to tracks can be reduced by responsiveness but only a small change in size and weight of the transmission results, at least for the degree of responsiveness that can be visualized for the next five years.

3. The engine weight and size would tend to increase slightly due to increased stress; under some conditions, which will be discussed later, increased size of turbo-chargers and controls would occur.

4. The over-all package of engine and transmission would show little if any change in the process.

5. Some small reduction in fuel supply per day can be seen by the use of a responsive engine attached to a change gearset. If this type of engine resulted in the elimination of torque converters from the transmission a substantial saving in fuel would result.

6. A high degree of responsiveness could relieve the driver of the vehicle of considerable labor and strain.

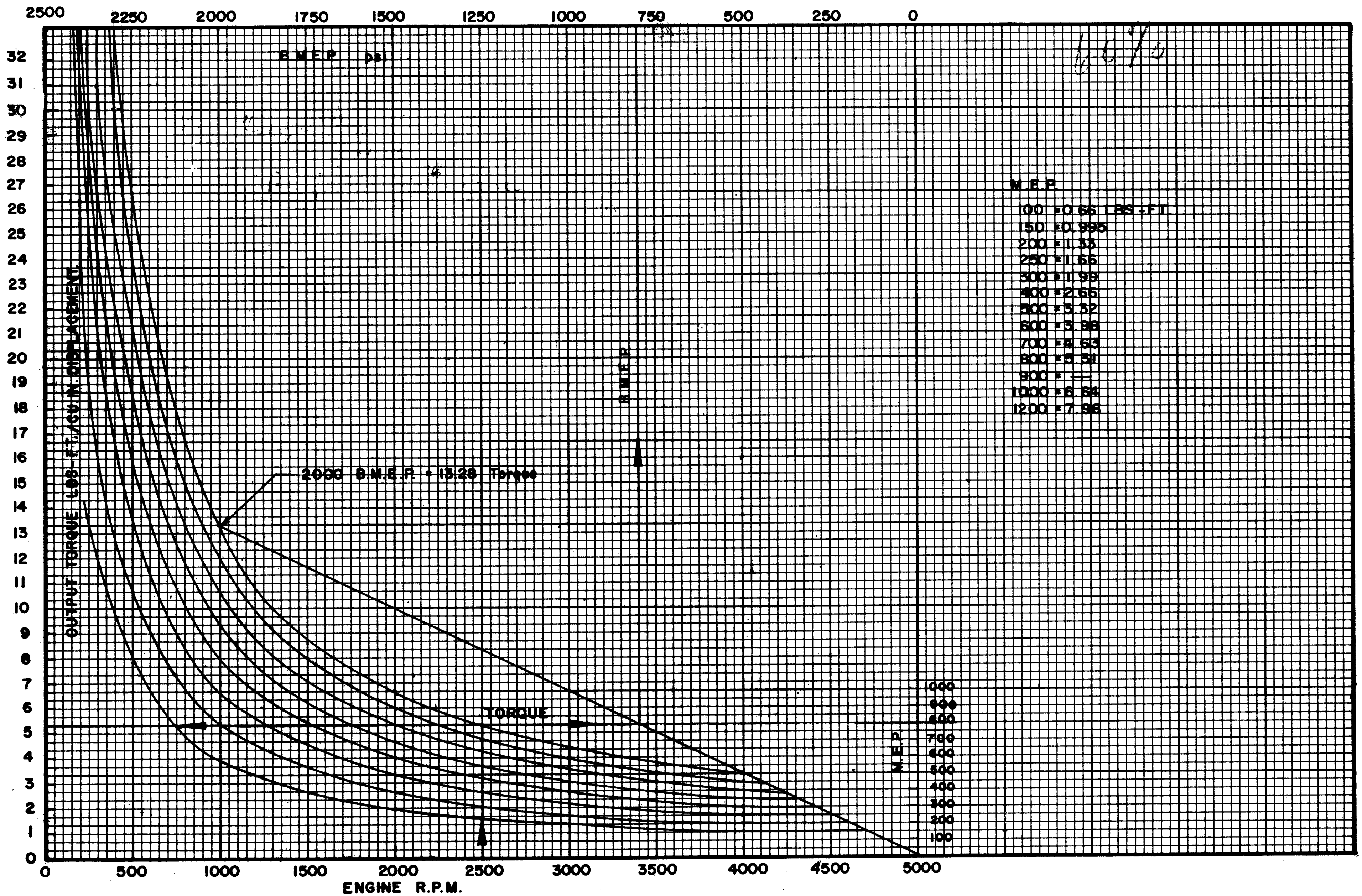

Fig. 15. Torque requirements for a responsive engine.

2.3. METHODS OF OBTAINING RESPONSIVENESS

It is now proposed to examine various means by which some degree of responsiveness can be obtained from the compression ignition engine. These methods may not be the only way but are those which occur to the reporter. All of these methods are based upon one fact, viz., no major sacrifice is to be made in the present performance characteristics at full load and speed.

The following methods are examined in some detail and the results as far as responsiveness are given for each method.

1. Variable degree of turbo-charging.

2. Variable air mass flow by auxiliary means.

3. Engine used as a pump to increase the turbine output of a compound unit.

4. Turbo-charged engine with afterburner and by-pass air flow.

5. Compression ignition engine in tandem with simple gas turbine.

6. Simple and regenerated turbine performance.

2.4. THE RESPONSIVE TURBO-CHARGED COMPRESSION IGNITION ENGINE

This arrangement is the present type of engine with special provision for exploiting the turbo-charger to obtain responsiveness.

It has already been shown that responsiveness involves increased mean effective pressure as engine speed is reduced. Figure 3 shows that increased mean effective pressure involves increased manifold pressure.

Turbo-charging results in a manifold pressure which reduces as engine speed and load is reduced and is typified by Fig. 5.

It follows that to secure any degree of responsiveness with a turbo-charged engine the natural tendency of the turbo-charger must be reversed, with manifold pressure increasing with reduced engine speed so that the air charge per cycle increases, permitting the combustion of an increased fuel supply. This can only be achieved via the turbine by supplying it with a greater quantity of energy in unit time; the quantity and energy of the exhaust gas from the engine must increase, which is an impossibility with reducing engine speed. It follows that other means must be incorporated to achieve this feature, e.g., a positive drive from the engine itself to take over automatically under the engine throttle operation.

Such a scheme is shown in Fig. 16, where a turbo-charger is connected to the engine itself through a variable hydraulic coupling "A" and a two-speed gear "B." Both of these units have been employed extensively in the past for engine operation in aircraft and thus involve no new principle. With the proposed arrangement there is actually some improvement of operating conditions compared with the aircraft applications, as will be seen.

The operation of the combined unit is as follows. Under normal vehicle conditions when moderate loads are involved the engine operates as a simple turbo-charged engine; the fluid coupling "A" is empty with just enough oil supplied to it to keep it cool and lubricated. The drag of the coupling will be small under such conditions since it will not be transmitting any power. The engine will be rotating at the desired speed and the turbo-charger at that speed determined by the exhaust gas energy as is normally the case; the speed of the turbo will be below that of the gear ratio of the drive; good fuel economy will be secured as well as present-day power output and engine bulk. The controls then call for power in excess of the normal maximum as the engine speed begins to fall under increased load; this change of speed is transmitted to the control governing the supply of oil to the fluid coupling; and the coupling begins to fill, transmitting an increased torque and speed to the compressor, resulting in an increasing manifold pressure supplying the engine. When the coupling is full and slip is at a minimum the first gear ratio of the box "B" is driving the supercharger at a speed giving increased air flow to the engine and increased mean pressure. However, the turbine is still generating power from the exhaust and the coupling only supplies the difference, if any, plus a definite, required design speed of the compressor and thus a manifold pressure capable of producing some increased maximum mean pressure at the lowest engine speed of operation desired. The engine can thus be responsive to such speed.

Further loading of the engine will produce if necessary a further gear-shift in the gearbox "B" resulting in an additional increase in the driven speed of the compressor and correspondingly in the maximum mean pressure that can be produced. The responsive range of the engine is thus further increased. As judged from aircraft experience the two units A and B will be small and compact and can be automatically controlled and connected into the throttle with little difficulty. The gearbox B could have just a single-speed drive, simplifying the system, but then the degree of responsiveness would be reduced. However, it may still provide a sufficiently wide range to reduce the required number of transmission gear changes by a reasonable amount, and may ease conditions for the operator.

In order to pin down the requirement, the following broad calculation has been made to give some idea of the ranges in the parameters that would be necessary.

It is necessary to estimate the maximum pressure ratios that can be expected in the near future from turbo-chargers if the maximum use is to be

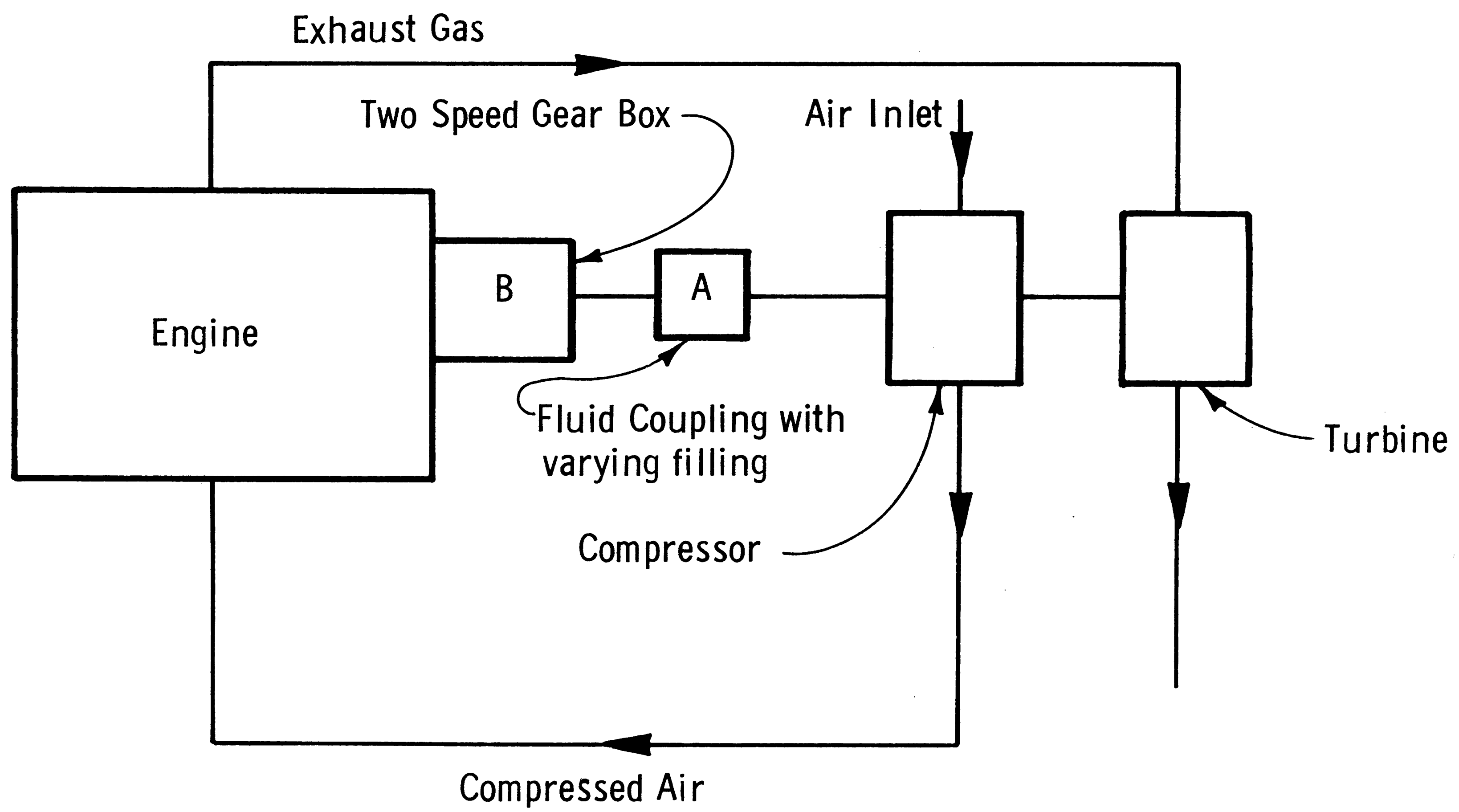

Fig. 16. Diagrammatic engine arrangement for responsiveness.

made of this principle. It is estimated that the following pressure ratios are not too optimistic:

Time	Pressure Ratio (Maximum)	Manifold Pressure (in. Hg abs)
Present	2.5-3.0	75-90 in.
1-2 years	3.0-4.0	90-120 in.
2-3 years	4.5	135 in.
3-4 years	5-6.0 (2 stage)	150-180 in.

It will probably take a longer period than the above to develop the engine to absorb the maximum ratio indicated. Compression and firing pressure will increase substantially, requiring a major engine development program. The need for a variable compression ratio will be emphasized if engine weight is to be kept down.

In order to keep such factors as engine size and weight competitive, assume an engine designed for normal full power and speed as given below:

Maximum rpm	3000
Maximum I.M.E.P.	250
Maximum B.H.P.	500
Number of cylinders	8
Stroke/bore ratio	1.075
Mechanical efficiency at 3000 rpm	85%

Calculation gives an engine size of 4-1/2 in. bore by 4-7/8 in. stroke as meeting these specifications and the adjusted parameters then become:

Cylinder diameter	4-1/2 in.
Stroke	4-7/8 in.
Stroke/bore ratio	1.082
I.M.E.P.	250 psi
B.M.E.P.	214 psi
Mechanical efficiency	85.5%

Using Fig. 3 the manifold pressure required for this performance is 73 in. Hg at a F/A ratio of 0.043

$$\text{Supercharger ratio} = 73/29.92$$

$$= 2.45 \text{ approx.}$$

This value is attainable with present machines.

In order to obtain responsiveness assume that the 4.5:1-ratio machine can be made available by the time it is needed; then a maximum manifold pressure of 135 in. Hg can be contemplated and Fig. 3 gives

Maximum I.M.E.P. at 135 in. Hg = 475 psi

Responsiveness to be expected = 2:1 approx.

In order to improve the above degree of responsiveness let it be assumed that some sacrifice is made in normal engine size and weight to secure increased responsiveness. Then a specification as given below would secure about 3:1 speed range for constant horsepower.

Manifold pressure	53 in. Hg
I.M.E.P.	180 psi
B.M.E.P.	145 psi
Cylinder diameter	5-1/8 in.
Stroke	5-1/2 in.
Responsive I.M.E.P.	475 psi
Responsive B.M.E.P. (Maximum)	425 psi

Responsiveness = 425/145 = 2.93 (say 3:1).

If these engines are assumed to be 90° Vee air cooled, then the sacrifice in size of the engine can be obtained from Fig. 12.

Engine volume 4-1/2 x 4-7/8 = 38.0 cu ft approx.

Engine volume 5-1/8 x 5-1/2 = 65.0 cu ft approx.

The sacrifice amounts to 27.0 cu ft and probably some 600 to 750 lb of engine weight. Some of this loss would be offset by the reduction in transmission ratio required by 3:1, but if allowance is made for the increased torque to be transmitted some of this gain would be lost. It will be neglected for the time being (see the Conclusions to Section II).

To estimate the performance of such an engine over the speed range from 3000 to 1000 rpm, the following calculations are made.

Let it be assumed that:

(a) The cylinder has a volumetric efficiency of 100%, neglecting the air flow which would reduce the above value somewhat.

(b) The engine operates at full power and 4.5:1 compressor ratio at a S.F.C. of 0.33 lb/I.H.P./hr with F/A = 0.0430.

(c) The S.F.C. gradually improves as the pressure ratio and mean pressure reduces and reaches the values shown in tables below, taken from Fig. 2.

(d) The frictional mean pressure at full speed is 36 psi without fan and generator.

The performance has been determined for the complete range of pressure ratios despite the fact that only part of this will be used at high speeds. The frictional and fan losses can be taken from Fig. 17 for typical values in this size range.

Case 1: Engine rpm = 3000; F/A = 0.043; F.M.E.P. = 36 psi

Pres. Ratio	I.M.E.P.	I.H.P.	B.M.E.P.	B.H.P.	Fuel lb/ I.H.P./hr	Fuel lb/ B.H.P./hr	Fuel Flow, lb/hr
4.5	475	1635	439	1510	0.33	0.357	540
4.0	425	1462	389	1340	0.325	0.355	476
3.0	330	1138	294	1012	0.305	0.343	347
2.0	240	826	204	702	0.27	0.318	223
1.6	205	706	169	582	0.25	0.304	176.8
1.0	130	447	94	323	0.25	0.346	111.9

Case 2: Engine rpm = 2400; F/A = 0.043; F.M.E.P. = 25 psi

Pres. Ratio	I.M.E.P.	I.H.P.	B.M.E.P.	B.H.P.	Fuel lb/ I.H.P./hr	Fuel lb/ B.H.P./hr	Fuel Flow, lb/hr
4.5	475	1308	450	1240	0.33	0.348	432
4.0	425	1170	400	1100	0.325	0.346	380
3.0	330	910	305	840	0.305	0.33	277
2.0	240	661	215	592	0.27	0.30	178
1.6	205	564	180	496	0.25	0.285	141
1.0	130	358	105	289	0.25	0.31	89.5

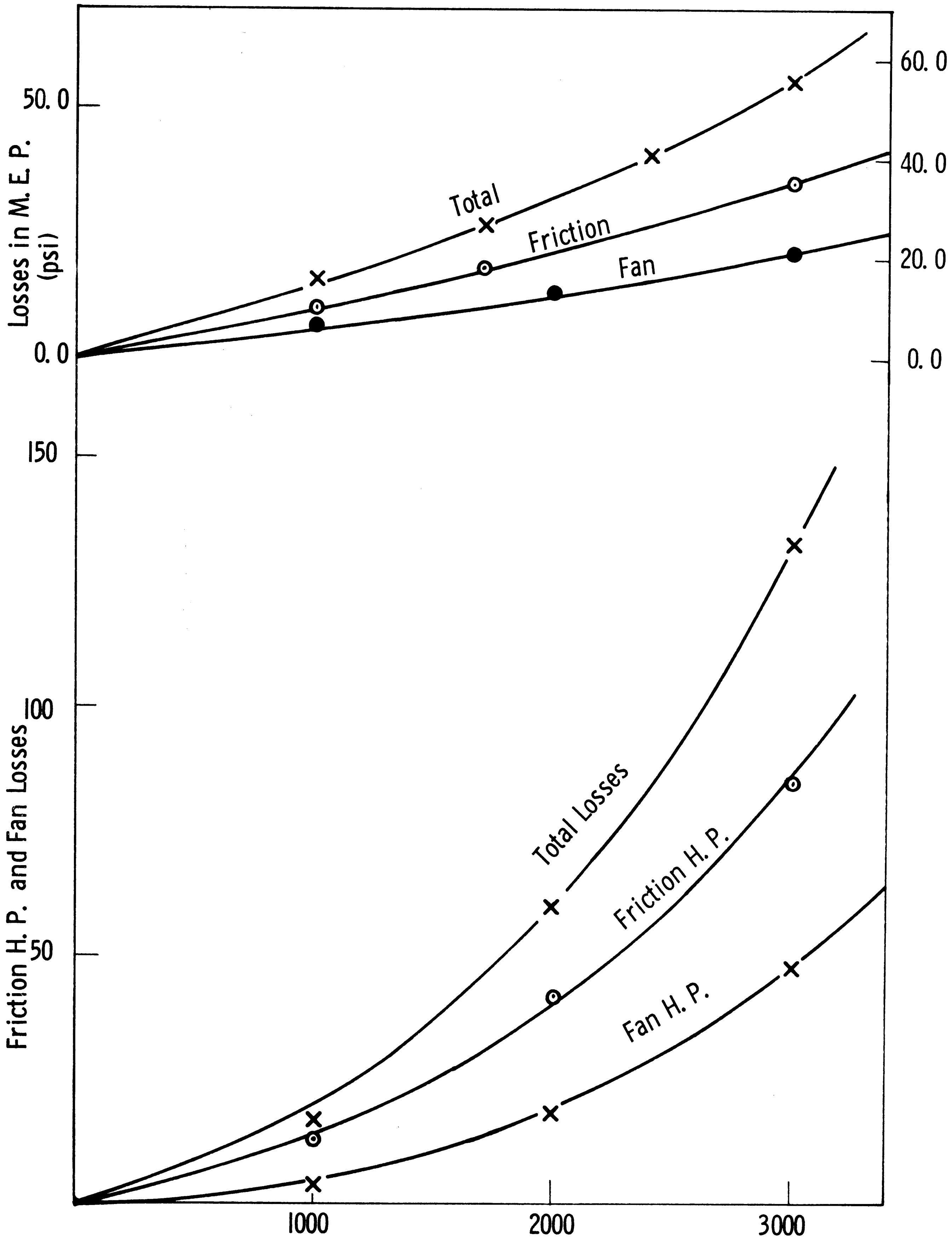

Fig. 17. Frictional and fan losses.

Case 3: Engine rpm = 1700; F/A = 0.043; F.M.E.P. = 18.0 psi

Pres. Ratio	I.M.E.P.	I.H.P.	B.M.E.P.	B.H.P.	Fuel lb/ I.H.P./hr	Fuel lb/ B.H.P./hr	Fuel Flow, lb/hr
4.5	475	926	457	891	0.33	0.343	306
4.0	425	828	407	794	0.325	0.339	269
3.0	330	644	312	609	0.305	0.322	196
2.0	240	468	222	433	0.27	0.292	126.5
1.6	205	399	187	364	0.25	0.275	100.0
1.0	130	253	112	218	0.25	0.290	63.4

Case 4: Engine rpm = 1000; F/A = 0.043; F.M.E.P. = 10.0 psi

Pres. Ratio	I.M.E.P.	I.H.P.	B.M.E.P.	B.H.P.	Fuel lb/ I.H.P./hr	Fuel lb/ B.H.P./hr	Fuel Flow, lb/hr
4.5	475	545	465	534	0.33	0.338	180
4.0	425	488	415	476	0.325	0.333	158.8
3.0	330	379	320	367	0.305	0.315	115.7
2.0	240	276	230	264	0.27	0.283	74.6
1.6	205	235	195	224	0.25	0.263	58.8
1.0	130	149	120	138	0.25	0.270	37.3

The above values for the fuel flow plotted on a B.H.P. basis are shown in Fig. 18; the diagonal lines through the origin are constant S.F.C. lines from 0.24 to 0.35 lb/B.H.P./hr respectively.

The vertical line drawn through the 500 B.H.P. point represents the operating line of a responsive engine. Reading off the various values, Table V is obtained.

TABLE V

RESPONSIVE ENGINE VEHICLE PERFORMANCE
(B.H.P. = 500; F/A = 0.043)

Speed, rpm	Pres. Ratio	Manifold Pressure, In. Hg	S.F.C., lb/B.H.P./hr	Fuel Flow, lb/hr	I.M.E.P., psi	B.M.E.P., psi
3000	1.4	42.0	0.30	150	181	145
2400	1.61	48.0	0.288	144	207	182
1700	2.3	69.0	0.302	151	275	257
1000	4.25	127.0	0.33	165	446	436

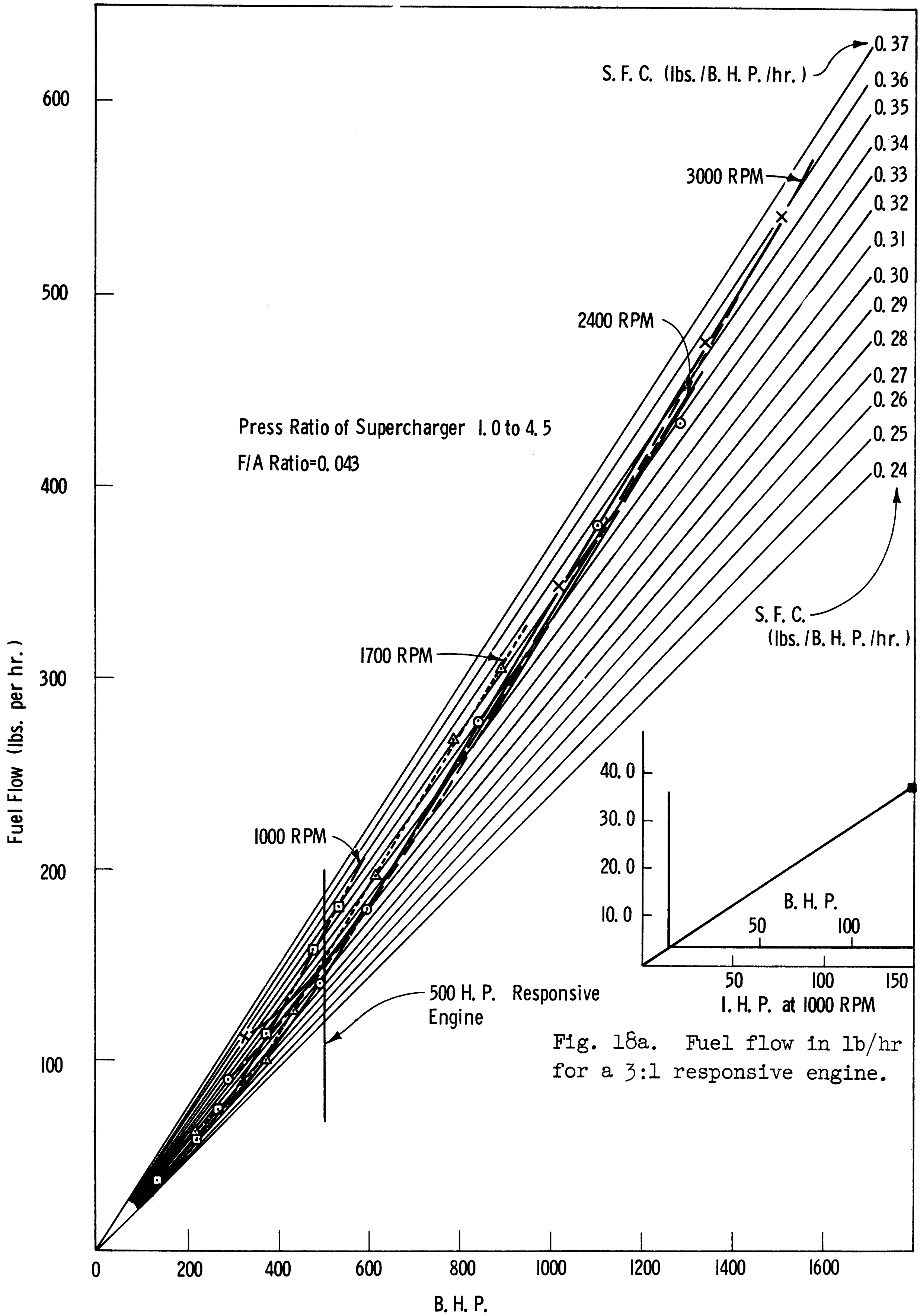

Fig. 18a. Fuel flow in lb/hr for a 3:1 responsive engine.

Fig. 18. Fuel flow for a 3:1 responsive engine.

Figure 18a, shown in the right hand lower corner of Fig. 18, is a straight-line extension of the 1000 rpm curve drawn to zero hp on an I.H.P. basis with a B.H.P. axis placed through the F.H.P. of the unit at 1000 rpm; viz., 11.5 hp, a straight-line for this extension, is sufficiently accurate since the F/A ratio is decreasing rapidly over this portion as the throttle is closed. This portion of the curve will serve for the prediction of idling fuel flow.

In order to obtain the part-load conditions a curve such as Fig. 6 is employed but plotted for the corret F/A ratio now being employed for 100% load, viz., 0.043. Such a curve is given in Fig. 19 and can be plotted from Fig. 6 by increasing the F/A ratios proportionally at each manifold pressure. The starting point is:

$$\text{Full I.M.E.P.} = 100\%$$
$$\text{Full power} = 100\%$$
$$\text{Full power F/A} = 0.043$$

These values position the line AB and the % I.M.E.P. at 100% speed can be marked off; then the 40 and 80% speed values are given by

$$\text{F/A at 100\% mean pressure and 80\% speed} = 0.043 \times \frac{0.0425}{0.0375}$$
$$= 0.0487$$

$$\text{F/A at 100\% mean pressure and 40\% speed} = 0.045 \times \frac{0.725}{0.0375}$$
$$= 0.083$$

By proceeding in a similar manner for all points desired, Fig. 19 will result.

Part load can now be estimated in the following manner. Since air flow depends upon speed and manifold pressure for a constant supercharge temperature, Fig. 5 (for 80% I.M.E.P. and 50% speed, i.e., 40% power) gives a manifold pressure of 57% of full value.

Fuel flow at 500 B.H.P. or 625 I.H.P. and 3000 rpm	= 150 lb/hr (Table V)
F/A	= 0.043
Trapped air flow	= 3490 lb/hr

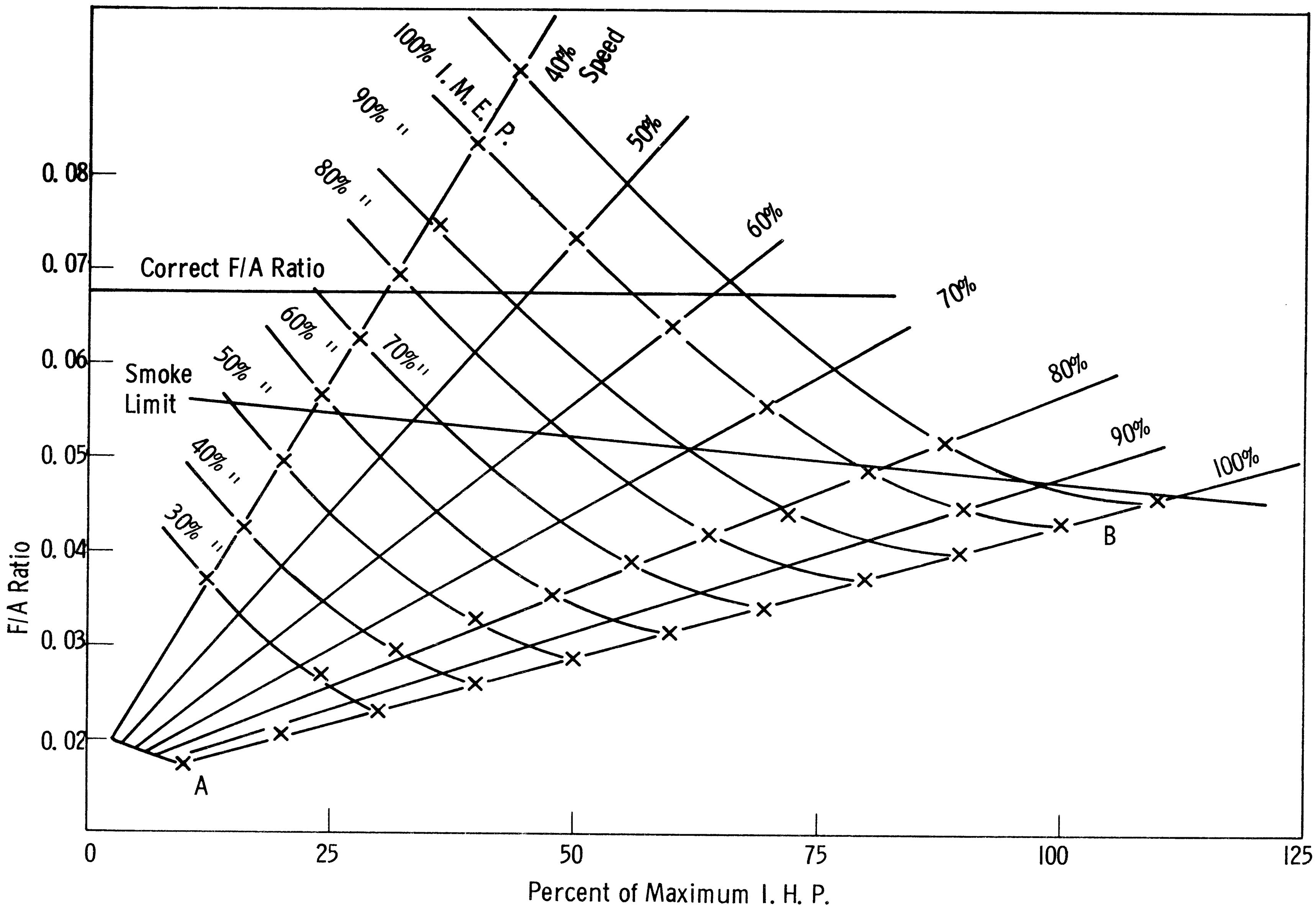

Fig. 19. Percentage performance plot for F/A = 0.043.

Manifold pressure at 50% speed and 80% I.M.E.P. = 145 psi (Fig. 5)	=	57%
Trapped air flow at 50% speed	=	3490 x 0.50 x 0.57
	=	995 lb/hr
From Fig. 19, expected F/A	=	0.061 at 50% speed and 80% I.M.E.P.
Fuel flow	=	0.061 x 995
	=	60.7 lb/hr
hp	=	% speed x % I.M.E.P. x 100% hp
	=	0.5 x 0.8 x 625
	=	250 I.H.P.
S.F.C.	=	$\frac{60.7}{250}$ = 0.243 lb/I.H.P./hr
F.M.E.P. at 1500 rpm	=	15 (Fig. 17).

Assuming an air-scavenge ratio of 5%, the supercharger air flow at full and half speed would then be 3680 lb/hr and 1045 lb/hr respectively, and the performance at these two points would be

500 B.H.P.	625 I.H.P.	Air flow 3680 lb/hr	S.F.C. = 0.30 lb/B.H.P./hr
224 B.H.P.	250 I.H.P.	Air flow 1045 lb/hr	S.F.C. = 0.27 lb/B.H.P./hr

Repeating the calculations, the complete fuel map of such an engine can be estimated.

In order to evaluate the effect of responsiveness employ the data supplied by the Detroit Tank Arsenal regarding the rolling resistance of the 43 ton vehicle viz.

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

At 30 mph

$$\text{Work done} = 70 \times 43 \times 30 \times 5280 \text{ ft lb/hr}$$

$$= \frac{70 \times 43 \times 30 \times 5280}{60 \times 33000} \text{ hp}$$

$$= 241 \text{ hp.}$$

Assuming an 82% transmission efficiency

$$\text{hp output of engine} = 241/0.82$$

$$= 294.0 \text{ hp}$$

The battlefield day is to involve 1.57 times the above for 15 to 20 mph for 20% of the day and 2.0 times the above for 40% of the day for speeds of 2 to 10 mph. It follows that the maximum hp involved will be 294 at 20 mph in the first instance and 234 hp in the second series. In other words the need for responsiveness at 500 hp will not be called upon except for emergencies such as steep hill climbing or obstacle negotiation. This makes the prediction of the effects of responsiveness difficult since it could be employed for long periods in some cases and not at all in others, depending on terrain.

In order to secure some feeling for this situation examine the case of the use of full horsepower under the normal gear transmission and responsive conditions. To do this examine the responsive case and estimate what overall efficiency of transmission is necessary to be equal to this condition, then judge if such a value can be achieved.

The engine being considered is to have a 3:1 responsiveness; thus with a maximum engine speed of 3000 rpm and maximum road speed of 35 mph with a 10% slip and a sprocket pitch diameter of 22.19 in. as at present the required final drive ratio would be

$$\text{Speed of sprocket} = \frac{35}{0.9} \times \frac{5280 \times 12}{\pi \times 22.19 \times 60} \text{ rpm}$$

$$= 589 \text{ rpm}$$

$$\text{Final drive ratio} = 3000/589$$

$$= 5.1:1$$

Such a gear ratio will involve definite losses even with responsiveness; assume that the losses in such a gear amount to only 5% and depend on the number of tooth contacts and torque transmitted rather than on speed. Then at any given speed the loss varies with torque.

At 3000 rpm and 500 B.H.P. the losses will be

Gear losses $= 0.05 \times 500$
$= 25$ hp

hp to sprocket $= 475$ hp

Fuel flow $= 150$ lb/hr

Speed of vehicle with 10% slip $= 0.9 \times \dfrac{\pi \times 22.19}{12} \times \dfrac{3000 \times 60}{5.1 \times 5280}$

$= 35$ mph

Maximum rolling resistance engine can exert $= \dfrac{\text{hp} \times 60 \times 33000}{\text{mph} \times 5280}$

$= \dfrac{475 \times 60 \times 33000}{35 \times 5280}$

$= 5090$ lb

$= \underline{\underline{118.5 \text{ lb/ton}}}$

This is about 1.7 times the amount for a first-class road. The fuel in ton-miles/gallon then becomes, if S.G. = 0.85,

Ton-miles/lb $= \dfrac{43 \times 35}{150}$

$= 10.0$

Ton-miles/gal $= \dfrac{43 \times 35 \times 7.16}{150}$

$= \underline{\underline{71.6 \text{ ton-miles/gal}}}$

Now assume that the maximum responsiveness of 3:1 is employed, with the B.H.P. remaining at 500.

Engine rpm	=	1000
Torque	=	3 times previous value
Losses	=	3 times but only 1/3 speed
	=	same as before
	=	25
hp at sprocket	=	475
Vehicle speed with 10% slip	=	$0.9 \times \frac{\pi \times 22.19}{12} \times \frac{1000 \times 60}{5.1 \times 5280}$
	=	11.66 mph
Fuel flow (Fig. 18)	=	165 lb/hr
Maximum rolling resistance engine can exert	=	$\frac{475 \times 60 \times 35000}{11.66 \times 5280}$
	=	15,270 lb
	=	$\underline{\underline{355.5 \text{ lb/ton}}}$
Maximum possible tractive effort with 0.8 coeff. of friction	=	$0.8 \times 43 \times 2000$
	=	68,800 lb
	=	$\underline{\underline{1600 \text{ lb/ton}}}$

It follows that the conditions encountered at 500 hp and 1000 rpm could be absorbed by the track without continuous slipping on an improved road surface. What would be the case in practice depends upon the state of the soil, slope of ground, type of obstacle, etc., and slipping may or may not occur in such cases.

$$\text{Ton-miles/gal} = \frac{43 \times 11.66 \times 7.16}{165}$$

$$= \underline{\underline{21.8 \text{ ton-miles/gal}}}$$

Repeating the above calculations the figures of Table VI are obtained.

TABLE VI

MAXIMUM ROLLING RESISTANCE

Engine, rpm	B.H.P.	Vehicle Speed With 10% Slip, mph	Fuel Flow, lb/hr	Ton-Miles per Gallon	hp at Track	Max. Rolling Resistance, lb/ton
3000	500	35.0	150	71.6	475	118.2
2400	500	28.0	144	59.8	475	148.0
1700	500	19.85	151	40.5	475	219.0
1000	500	11.67	165	21.8	475	355.5

In Table VI is shown the maximum rolling resistance in lb/ton that a responsive engine of 500 B.H.P. could exert at the assumed transmission efficiency of 95%. The assumed engine has a 3:1 responsiveness; thus the conditions from 30 to 10 mph could be obtained with one gear ratio, from 10 to 4 mph with a second and 4 to 2 mph with a third set of gears. It follows that a final drive of 5.1:1 would need a three-speed gearbox attached if responsiveness were to be secured over the whole speed range. Such a box would involve some losses in excess of the 95% assumed for the final drive ratio; therefore the values given for maximum rolling resistance should be re-evaluated for such losses.

If the engine were to be exploited over the maximum of 3:1 responsiveness, gear ratios of 1.16, 3.5, and 7:1 would be required as indicated in Table VII. The effects in low gear would be that the maximum tractive effort of 68800 lb or 1600 lb/ton could be exceeded; it is probable that a ratio of less than 7:1 would be satisfactory. The present transmission employed, XTG-411, is only fitted with a first-gear ratio of 4.68:1 when coupled to a non-responsive engine; it follows that the setup as indicated in Table VI to exploit the responsiveness would be more than ample for the needs, providing that the present gearset arrangement can take care of the requirements.

At the same time a gearbox with three ratios as indicated above is going to have about the same efficiency as the present four-speed, with ratios of 4.68, 3.17, 1.58, and 0.78:1 if the XTG-411 is not fitted with a torque converter. It follows that in the absence of torque converters an engine of 3:1 responsiveness will give about the same ton-miles/gallon as the conventional transmission when fitted to a conventional engine, provided the fuel flow to the two engines remains about the same. Examination of Table V and Fig. 18 shows that fuel flow first reduces and then increases when operating at constant hp in the responsive manner. For the conditions assumed, responsiveness from 3000 to 1700 rpm would perhaps pay off slightly as far as fuel flow is concerned when operating at maximum power. The case of the throttled condition can be seen by examining the trend of the cases shown plotted in Fig. 18. Here it is seen that as the hp is reduced below the maximum of 500 the trend is for the lower operating speeds to give the better fuel economy, which would, of course, be expected from the rapid reduction of frictional losses as speed is reduced. It follows that the greater the period of time in which the third gear could be used the better would be the over-all economy.

In order to complete the curves of Fig. 18 down to zero B.H.P. it should be noted that the last plotted point in each curve is one for the unsupercharged condition. If this plot is in terms of I.H.P. then a straight line joining this point with the origin would be sufficiently accurate for prediction purposes. In view of the scale of the diagram it is considered that such a straight line would still suffice, even for the B.H.P. scale used, and the conclusions drawn above still hold, viz., that operation of the engine at the slowest speed possible with a mean pressure approaching about 75 to 80% of the maximum for that speed would provide the best over-all economy. In the responsive condition when the full 500 hp is to be used at all speeds, maximum economy in terms of fuel used per hr occurs at about 2400 rpm with an increased fuel flow at lower speeds.

Under the conditions of Table VI it is assumed that the rolling resistance is the maximum possible for the absorption of 500 hp from the engine at the speed of the vehicle with 10% slip. This resistance varies from about 1.7 to 7.0 times the rolling resistance of a first-class road when in direct gear. The question then arises as to how often and for how long such resistance can be encountered; without such information it seems impossible to calculate with any degree of accuracy the probable over-all performance.

One other factor to be noted is that the gear ratios shown in Table VII are based upon the maximum engine speed of 3000 rpm plus an effort to include most of the full responsiveness of 3:1 for each gear. The maximum available tractive effort under such conditions then goes to such high values as 2070 lb/ton, which seems far higher than any tracks could provide. As a result it would be possible to change the 7.0:1 ratio without detriment to the performance. In view of this it seems possible that a responsive vehicle of the type being discussed could possibly make out with two gears, the 1.6 and 3.5:1, allowing the engine to depart slightly from responsiveness at the 2 and 3 mph conditions where the torque available is probably greater than can be absorbed by the track anyway.

So far the problem of gear ratio and steps required has been assumed to be the simple one of connecting the engine to the final drive. Unfortunately this is not the only requirement. If the ratios between steps is too great, difficulty in gear changing and vehicle reaction at change occurs. The ratio of the steps indicated in Table VII are believed to be far greater than those commonly allowable. In other words, the two- or three-step transmission would be unsatisfactory from an operator's viewpoint and an additional step or steps would be required. The problem is now back where it started, with at least a four-speed transmission required. In this case the need for the 3:1 speed

TABLE VII

RESPONSIVE PERFORMANCE TON-MILES/GALLON
(500 hp at 3000 rpm; 5-1/8 in. diam x 5-1/2 in. stroke; B.M.E.P. of 145 psi; responsiveness 3:1)

mph	2	3	4	5	6	7	8	9	10	15	16	17	18	19	20	30
Rolling resistance, lb/ton	67.0	67.5	68.0	68.0	68.0	68.5	69.0	70.0	73.5	72.0	69.0	68.0	67.0	67.0	67.0	70.0
hp at ground	15.3	23.2	30.6	38.2	46.0	54.9	63.0	72.1	84.1	124.0	126.0	130.0	138	146	153	241
Assumed transmission efficiency	0.65	0.65	0.65	0.65	0.72	0.72	0.72	0.72	0.72	0.78	0.78	0.78	0.78	0.78	0.82	0.82
B.H.P. at engine required	23.5	35.7	47.0	58.8	64.0	76.1	87.5	100.0	117.0	159.0	162.0	167.0	177.0	187.0	187.0	294.0
Maximum rolling resistance responsive, lb/ton	2070	1380	1037	828	690	592	518	460	414	276	258.5	243.2	230	218	207	138
Engine Speed 3rd ratio									1000	1500	1600	1700	1800	1900	2000	3000
Engine Speed 2nd ratio			1200	1500	1800	2100	2400	2700	3000							
Engine Speed 1st ratio	1200	1800	2400	3000												
Sprocket speed 10% slip, rpm	33.7	50.5	67.5	84.4	101.0	117.9	134.8	151.7	168.2	252.3	269.3	286.1	303.5	320.0	336.6	505.5
Gearbox ratio	7.0:1		3.5:1						1.16:1							

range with responsiveness seem unnecessary; perhaps a 1-1/2 or 2:1 would do all that is necessary, overlapping the steps to such an extent that all the benefits are available without the problem of such an extensive development program.

In the final analysis the best possible responsive engine of the near future (within, say, two years of development) would be one having no more than a 3:1 responsiveness, provided a small sacrifice in engine space and weight is made by derating the maximum mean pressure at high speed. This engine would need a 5.1:1 final drive ratio, about the same as at present; thus the weight, size, etc., of this final drive unit would not change appreciably. In addition there would be at least a two-speed gearbox between the engine and the final drive; whether this would result in any economy in space and weight cannot be judged without working out a complete layout of the engine and transmission, since the possible shape of the package could seriously affect the results. The resulting efficiency of the complete assembly might be some 3 to 5% higher at the most than present step transmissions.

The penalties to be paid for this small gain, apart from easier operation, would be about as follows:

(a) The engine would be somewhat larger and heavier for a given B.H.P. and rpm.

(b) It would be necessary to fit the engine with a turbo-charger which is coupled to the engine via a variable speed gear.

(c) High manifold pressure would be employed, a compressor capable of a pressure ratio of 4.5:1 being required.

(d) As a result of high manifold pressures, high compression pressures amounting to 3000 psi for a 17.5:1 engine compression ratio would be involved. This ratio might still be too low for real cold starting.

(e) Due to the high pressures noted in (d) it seems necessary that a variable compression engine be developed if firing pressures are to be kept under control.

(f) Even with variable compression ratio, some increase in peak cylinder pressures above those in common use today is to be expected. Such increase might contribute to a further increase in engine weight.

(g) The heat disposal from the cylinder would be increased when operating responsively. This would introduce a further development problem if air-cooling were still desired.

Improved responsiveness and the elimination of some of the above disadvantages could be obtained by derating the engine to modest mean pressures at maximum speed. This of course would result in an increase in engine space and weight requirements. The correct answer in this case seems to be a reestimation of the real requirements as regards power at peak speed, if a responsive engine of the type outlined is possible. A reduction from 500 to say 350 hp for the same size and weight could permit a considerable increase in responsiveness with perhaps little or no sacrifice in over-all performance of present equipment. Detailed examination of such an engine would be necessary to ascertain if such a change would eliminate the need for the second gear, and permit the use of a direct connected engine with a simple clutch only.

2.5. CONCLUSIONS

This examination permits of the following conclusions.

1. An engine with characteristics somewhat similar to those of present models as regards power, size, weight, etc., at full speed, can possibly be made responsive over a 3:1 range given a fairly extensive development program to achieve high mean pressure at low speed.

2. In order to achieve this objective, turbo-chargers with pressure ratios of 4.5:1 would have to be developed so that they can be geared to the engine via a two-speed drive.

3. With 4.5:1 turbo-chargers the engine would need complete evaluation as regards bearing sizes, stresses, etc., to handle the peak pressures to be encountered.

4. Unless responsiveness in excess of 3:1 is secured, a three-speed transmission will still be required together with a 5.1:1 final ratio in order to provide the vehicle speed range required with responsiveness over the complete range.

5. The use of a two-speed set would permit responsiveness over almost all of the operating range. The range not covered is the 2 to 4 mph range where the engine, even with two speeds, would probably provide a greater tractive effort than the tracks could absorb.

6. The gains in fuel economy with responsiveness would be small when compared with a normal multi-step transmission.

7. Gains in fuel economy would be possible if torque converter types of transmissions were replaced with responsive engines.

8. To evaluate the gains completely, some reasonably accurate schedule of engine operation in the full-power responsive condition would have to be setup. The present schedule of operation over a Battlefield Day does not involve the responsive feature at any time so far as the example employed in this analysis is concerned.

9. The main advantage seen resulting from the responsiveness would be the reduction in number of gear changes during operation and reduction in operator fatigue.

SECTION 3

AIR-BOOSTED CYCLE ANALYSES

It is seen from the previous work that a responsive engine is one which digests an approximately constant weight of air at any speed of rotation, to which is supplied the appropriate fuel to keep the F/A ratio also constant.

This requirement was carried out in the work previously described by increasing the charge density in the engine manifold. A second method which would secure the same results would be to supply additional air to the cylinder after the intake valve has closed. It was seen that increased air charge via supercharge involved greatly increased cylinder pressures throughout the cycle, which adversely affect engine weight. By supplying the excess air at the end of a normal compression stroke without increase of cylinder pressure, i.e., during the early stages of the expansion stroke, an increase in charge weight and thus fuel supply is possible without increase of cylinder pressures.

Such an analysis was submitted in February, 1962 under the terms of this contract as Progress Report 04612-2-P. The complete report will not be repeated but the highlights and the calculated performance obtained will be given.

As the vehicle resistance increases and the engine slows down, additional air and fuel is supplied during the early portion of the expansion stroke to keep the cylinder pressure approximately constant at its desired maximum value, the change in the indicator diagram being as shown in Fig. 20.

The advantages of such a procedure are:

1. The engine behaves and is rated as a normal engine of present-day design, without additional loads, etc., so long as normal operating conditions are applied.

2. The engine is capable of being operated at increased mean pressures for temporary overloads (period of overload can be long if suitably designed).

3. Increased load achieved without increase in maximum cylinder pressure and thus constant engine stress.

4. Increase of load achieved without increase (probably a decrease) in maximum combustion temperature.

5. Greater energy available for recovery by compounding.

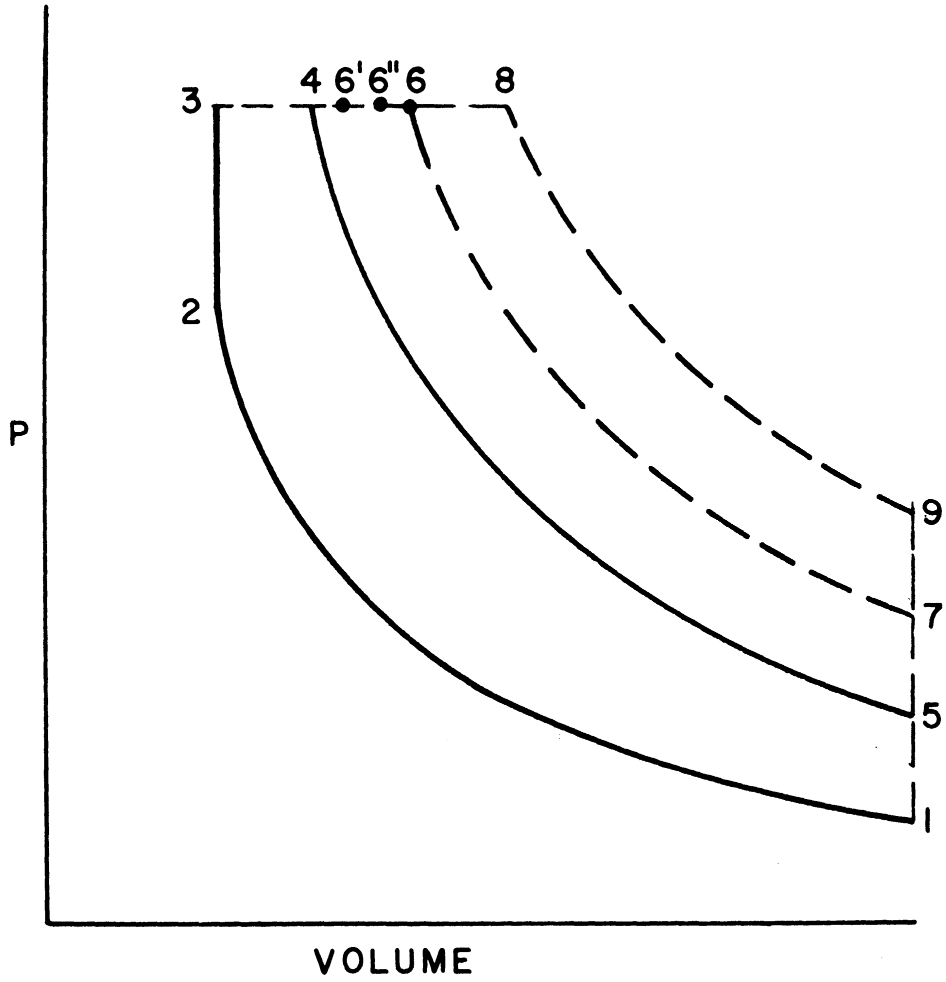

Fig. 20. Indicator diagram of air-booster cycle.

6. Equipment to achieve this purpose only operates when required.

7. The increase in weight, space and costs involved can be kept small by designing the engine for its normal output requirements, say 300 hp, and employ the proposed principle to meet the emergency needs of, say, 500-600 hp only when needed; by this means little change in weight, space, etc., is seen. A typical design would need to be worked out to establish the requirements accurately.

3.1. ENGINE ARRANGEMENT

The proposed cycle of operation would be achieved by providing an air compressor capable of delivering air at, say, 1600 psi to a small receiver, to damp out oscillations and provide a small reserve. This receiver is connected to an engine supply manifold, and flow to the cylinders is provided by a mechanically controlled valve in the cylinder head capable of being varied as regards period or lift or both.

When need arises for the use of responsiveness—as the engine stalls on high load—the compressor is coupled to the engine and the air supply is fed to the receiver and then to the cylinder as in Fig. 21. The appropriate increase in fuel is also added to the air supply. If high load is required at maximum speed for any reason the air supply is also available for this purpose. Thus a typical design could be as follows:

Normal power at max rpm	400 hp
Maximum emergency power at max rpm	600 hp
Normal power at min rpm	200 hp
Maximum power at min rpm	400 hp

Engine responsiveness from rpm_{max} to $rpm_{max}/2$ (constant 400 hp).

Other combinations of hp and speed could also be achieved. In the above example, a 400-hp engine can produce 600 when required, and also be responsive at 400 hp over a 2:1 speed range. Alternatively, the full 600-hp normal engine could be designed and made responsive at 600 hp down to, for example, half maximum rpm; this unit would of course be larger than the former. A typical arrangement would be as shown in Fig. 21; the turbo-charger would be geared to the engine if compounding was to be employed.

3.2. PERFORMANCE

The performance of this type of power plant was obtained in the manner given in the Progress Report under discussion.

The cycle employed was:

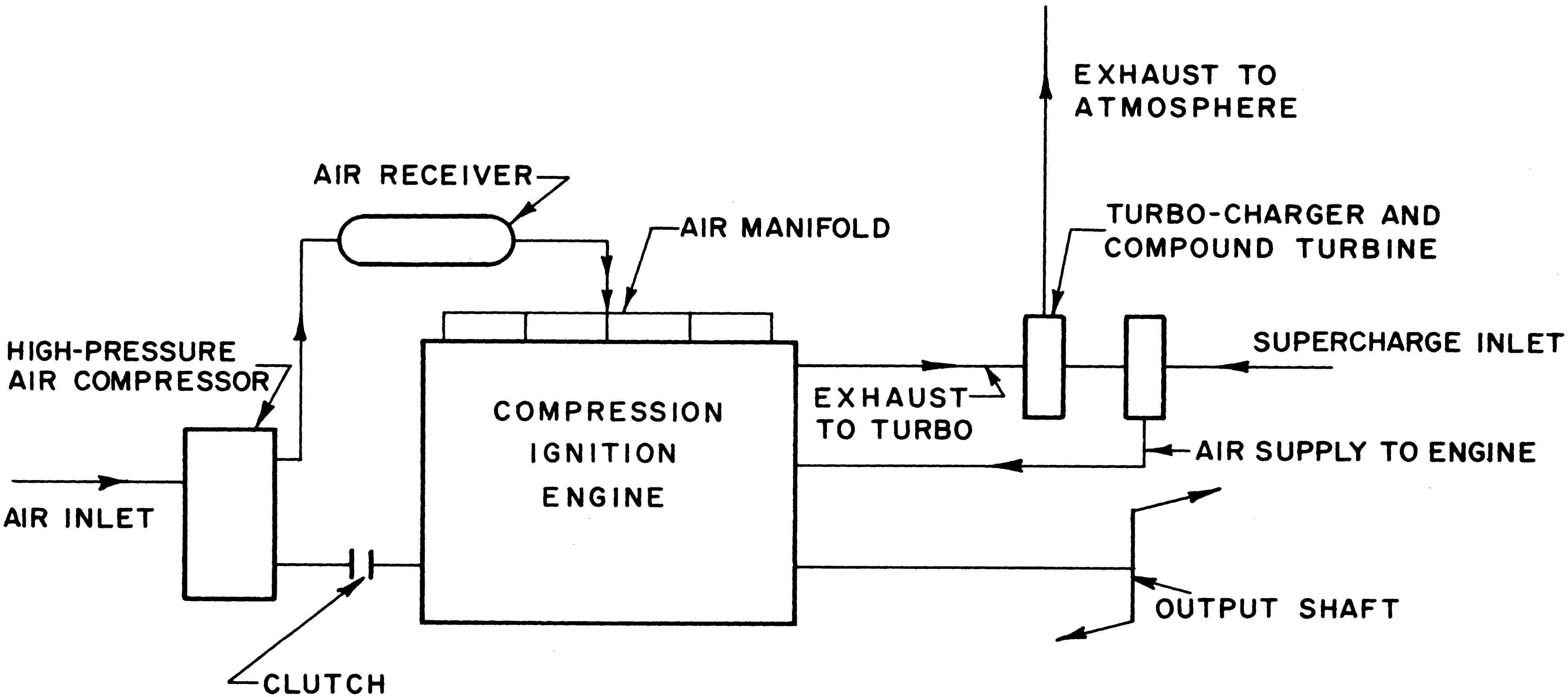

Fig. 21. Arrangement of air-boosted engine.

Compression ratio	15:1
Supercharge ratio	2:1
F/A ratio	0.05
Calorific value of fuel	18500 Btu/lb
Turbo-compressor efficiency	0.70
Turbine efficiency	0.86
Ambient air	P_a = 14.7 psi and T_a = 125°F
Total heat loss	21%
Heat-loss compression	0.4%
Heat-loss combustion	1.7%
Incomplete combustion	2.0%
Heat-loss expansion	5.0%
Heat-loss exhaust	5.5%
Heat-loss to oil	1.8%
Miscellaneous	4.6%

By evaluation of the four strokes of the cycle as given in report, the work summary in Table VIII was obtained.

Considering these results and plotting the I.M.E.P. on a diagram such as Fig. 14 of this report, it is seen that if the engine with no aftercooler was rated at 312 I.M.E.P. at 2800 rpm, responsiveness would exist down to a speed of 1475 rpm for an additional air flow equal to that of the basic engine air flow: a not unattractive performance, particularly since the S.F.C. only changes by about 10%, as a maximum, in the process on an indicated basis. Since the engine speed would decrease with responsiveness, the mechanical efficiency will generally improve under such conditions, with the result that the change in S.F.C. will generally be less than the 10% calculated.

3.3. COMPOUNDING

In the cycle under consideration, it is apparent that as the auxiliary air flow (Δw) increases, the energy available at the end of the expansion stroke is also increased, e.g., the pressure P_7 changes from 114 psi at $\Delta w = 0$ to 216 psi when $\Delta w = 1.0$; greater blow-down energy is thus available. To examine what improvement can be obtained from such conditions by means of compounding, the following cases were investigated.

Case 1. Where it is assumed that conventional present-day turbocharger efficiencies are employed.

Case 2. Where a reasonable increase in turbine and compressor efficiencies are employed, increases that can be easily justified with the best of modern design; however, such design will probably add something to their cost, unless produced in large numbers.

Examination of Case 1 resulted in the figures of Tables IX and X.

TABLE VIII

WORK SUMMARY OF AIR-BOOSTED CYCLE

Item	No Aftercooling						Aftercooling, ϵ = 0.60					
	0.0	0.2	0.4	0.6	0.8	1.0	0.0	0.2	0.4	0.6	0.8	1.0
Suction loop (positive Btu)	6.58	6.58	6.58	6.58	6.58	6.58	5.36	5.36	5.36	5.36	5.36	5.36
Compression work (negative Btu)	-230.2						-192.2					
Constant pressure exp. Btu	191.9	243.2	292.5	342.5	393.0	443.5	184.3	234.8	284.8	335.5	386.0	435.5
Expansion Btu	556	603	650	663	760	790	482	524	558	604	639	659
Net work of cycle Btu	524.3	622.6	718.9	781.9	929.4	1009.9	479.5	572	656.0	752.7	838.2	907.7
I.H.P./lb of basic air/sec	742	881	1018	1107	1313	1442	679	809	928	1066	1187	1283
Total air flow, lb/sec	1.0	1.20	1.40	1.60	1.80	2.00	1.0	1.20	1.40	1.60	1.80	2.00
Fuel flow, lb/sec	.05	.06	.07	.08	.09	0.10	.05	.06	.07	.08	.09	0.10
Total gas flow/lb of basic air/sec	1.05	1.26	1.47	1.68	1.89	2.10	1.05	1.26	1.47	1.68	1.89	2.10
Fuel per hr, lb/lb of basic air	180	216	252	288	324	360	180	216	252	288	324	360
Fuel lb per I.H.P./hr	.243	.245	.248	.260	.247	.250	.265	.267	.271	.270	.273	0.280
I.M.E.P., psi	312	370	426	465	552	599	344	416	477	547	610	660

TABLE IX

COMPOUND ENGINE OUTPUT, NO AFTERCOOLING

Δw	0.0	0.2	0.4	0.6	0.8	1.0
Net cycle work	524.3	622.6	718.9	781.9	929.4	1009.4
Turbine work	91.9	105.0	122.0	143.5	162.3	182.0
Total output	616.2	727.6	840.9	925.4	1091.7	1191.4
Input to comp.	42.7	42.7	42.7	42.7	42.7	42.7
Net outpu	573.5	684.9	798.2	882.7	1049.0	1148.7
I.H.P./lb of basic air/sec	811	969	1130	1250	1486	1627
Fuel flow lb/hr/lb basic air	180	216	252	288	324	360
S.F.C. lb/I.H.P./hr	.222	.223	.223	.230	.218	.222
I.M.E.P. psi	341.0	406.0	474.0	525.0	624.0	682.0

TABLE X

COMPOUND ENGINE OUTPUT WITH 60% AFTERCOOLING

Net cycle work	479.5	572.0	656.0	752.7	838.2	907.7
Turbine work	87.2	104.7	122.2	143.9	161.5	181.9
Total output	566.7	676.7	778.2	896.6	999.7	1089.6
Input to comp.	42.7	42.7	42.7	42.7	42.7	42.7
Net output	524.0	634.0	735.5	853.9	957.0	1046.9
I.H.P./lb of basic air/sec	741.0	896	1041	1208	1358	1480.0
Fuel flow lb/hr/lb basic air	180	216	252	288	324	360
S.F.C. lb/I.H.P./hr	.243	.241	.242	.239	.239	.243
I.M.E.P. psi	376	462	536	620	696	761

Examination of these compounded cycles reveals that there is a change in output of roughly 2:1 for a practically constant S.F.C. for both cases. It follows that a responsiveness of 2:1 is possible for the conditions assumed.

The details obtained for Case 2 were determined for compressor efficiency of 0.75 in place of 0.70, with a turbine efficiency of 0.88 in place of 0.86. These values can be substantiated by modern small gas turbine developments; the units will probably increase somewhat in costs however.

Using the above values the figures of Table XI and XII were obtained.

TABLE XI

COMPOUND ENGINE OUTPUT,
NO AFTERCOOLING—IMPROVED EFFICIENCY TURBO

Δw	0.0	0.2	0.4	0.6	0.8	1.0
Net output Btu	578.5	690.4	804.1	889.0	1065.8	1155.8
I.H.P./lb basic air/sec	819.0	978.0	1139.0	1259	1509	1658
I.M.E.P. psi	344	410	478	529	634	687.0
S.F.C. lb/I.H.P./hr	0.22	0.221	0.222	0.228	0.215	0.22

TABLE XII

COMPOUND ENGINE OUTPUT
WITH 60% AFTERCOOLING—IMPROVED EFFICIENCY TURBO

Δw	0.0	0.2	0.4	0.6	0.8	1.0
Net output Btu	529.0	639.6	741.4	860	964.6	1053.9
I.H.P./lb basic air/sec	749.0	905	1049.0	1218.0	1364.0	1491.0
I.M.E.P. psi	380.0	466.0	541.0	625.0	699.0	768.0
S.F.C. lb/I.H.P./hr	0.241	0.239	0.240	0.237	0.237	0.241

Comparison of these results with those of Case 1 shows that little has been gained by a research effort to improve the turbo-charger efficiencies alone, other things remaining the same. However, it will be found that a reasonable gain in power is possible if the improved efficiencies are employed to increase the turbo-charger pressure ratio instead of holding it constant as was the case employed.

3.4. HIGH-PRESSURE AIR COMPRESSOR

So far no allowance has been made for the power required to supply the high-pressure air employed. This power must be subtracted from the engine output and is calculated on the following basis.

It is assumed that a three-stage air compressor is employed which is to deliver at a pressure of 1600 psi for injection of the air into the cylinder where the pressure is to be maintained at 1500 psi. The previous calculations have assumed a maximum of 1 lb of compressed air to be supplied by the compressor per lb asperated into the engine. On this basis the power for the compression of 1 lb of air in a three-stage compressor can be obtained.

The conventional relationship for maximum efficiency of a multi-staged compressor will be employed.

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} \cdots = \frac{P_n}{P_{n-1}}$$

where

P_1 = initial pressure
P_2 = pressure in 2nd stage
P_3 = pressure in 3rd stage
P_n = pressure in nth stage.

Then with no losses between stages, but with intercooling to the original temperature, it can be shown, for a three-stage machine, that

$$W = \frac{RT_1}{J}\frac{3n}{n-1}\left[\left(\frac{P_4}{P_1}\right)^{\frac{n-1}{3n}} - 1\right] \text{ Btu/lb}$$

$$= \frac{53.34 \times 585}{778}\,\frac{3 \times 1.38}{0.38}\left[\left(\frac{1600}{14.7}\right)^{\frac{0.38}{3 \times 1.38}} - 1\right]$$

$$= 234.5 \text{ Btu/lb of air.}$$

This assumes that the inlet temperature T_1 is 125°F and that n in $PV^n = C$ for the compressor is 1.38. In this case the efficiency of the process is given by

$$\eta = \frac{\log_e r}{\frac{3n}{n-1}\left[r^{\frac{n-1}{3n}} - 1\right]}$$

where

$r = P_4/P_1$ the pressure ratio.

Then

η = 80%.

These values are close enough for present purpose (they may be a little optimistic). It follows that the net performance of an engine fitted with such a compressor will be as follows.

3.5. CONTINUOUS PERFORMANCE AT MAXIMUM POWER

It is to be assumed that the air compressor is designed to be capable of supplying the desired mass of air for operation at the required condition continuously. It follows that as Δw is varied from 0 to 1.0 the engine net performance when compounded will be as shown in Tables XIII and XIV.

TABLE XIII

CONTINUOUS PERFORMANCE CHARACTERISTICS, NO AFTERCOOLING

Δw	0.0	0.2	0.4	0.6	0.8	1.0
Engine output Btu	573.5	684.9	798.2	882.7	1049.0	1148.7
Compressor input Btu	0.0	46.9	93.9	140.8	187.6	234.5
Net output engine and comp. Btu	573.5	638.0	704.3	741.9	861.4	914.2
Fuel lb/hr	180	216	252	288	324	360
S.F.C.	0.222	0.327	0.253	0.274	0.266	0.279
Over-all net I.M.E.P.	341.0	379.0	418	441	512	543

TABLE XIV

CONTINUOUS PERFORMANCE CHARACTERISTICS WITH 60% AFTERCOOLING

Δw	0.0	0.2	0.4	0.6	0.8	1.0
Engine output Btu	524.0	634.0	735.5	853.9	957.0	1046.9
Compressor input Btu	0.0	46.9	93.9	140.8	187.6	234.5
Net output engine and comp. Btu	524.0	587.1	641.6	713.1	770.4	812.4
Fuel lb/hr	180	216	252	288	324	360
S.F.C.	0.243	0.260	0.277	0.285	0.297	0.513
Over-all net I.M.E.P.	376	428	467	518	560	691

The above ideal results would be reduced to a brake horsepower performance approximately as given below when corrected for normal losses.

B.H.P. Performance

Δw		0.0	0.2	0.4	0.6	0.8	1.0
B.H.P.	No cooling	700	788	877	926	1036	1158
	Aftercooling	639	725	800	895	974	1028
S.F.C.	No cooling	.257	.274	.287	.31	.322	.347
lb/B.H.P./hr	Aftercooling	.282	.298	.315	.322	.333	.357

3.6. INTERMITTENT PERFORMANCE AT MAXIMUM POWER

The results given above represent the engine fitted with a compressor capable of maintaining the air supply indefinitely. At the opposite end of the scale, if the maximum output was maintained for a limited time only then the responsive effect could be improved since the input to the compressor would be reduced proportionally.

Assume that the peak power is required for a period of 5 min/hr. There would be no change in the maximum power available except that resulting from the small change in compressor power, but the average compressor input would be reduced by 1:12 and the performance would be as shown in Table XV.

TABLE XV

INTERMITTENT PERFORMANCE

Δw for 5 min		0.0	0.2	0.4	0.6	0.8	1.0
Max net output	No cooling	700	831	963	1055	1208	1373
B.H.P.	With cooling	639	768	886	924	1046	1243
S.F.C.	No cooling	.257	.26	.262	.272	.276	.293
lb/B.H.P./hr	With cooling	.282	.282	.284	.312	.310	.295

The above figures are based upon a storage vessel capable of holding sufficient air for the 5-min burst of power, plus a compressor capable of re-charging the vessel in the remaining 55 min.

Two such bursts of power per hour could be obtained with little change in the above values. The longer the time required, the closer the results approach the continuous values.

3.7. AIR COMPRESSOR

To visualize the final power plant, some idea of the compressor displacement necessary for the purpose is required.

In the case of continuous operation at 500 hp with aftercooler it was shown in the Progress Report that the low-pressure stage of a single cylinder compressor suitable for the purpose would be as follows:

rpm = 4000

Cylinder Diam = 5.26 in.

Piston Stroke = 5.26 in.

This size is not out of proportion when it is considered that, by its application, a 500 hp engine can develop 800 hp continuously without change of pressures, temperatures, etc.

Looking at the case for the 5-min burst of power, then the 500 hp increases to 974 hp for 5 min for a double acting compressor as follows:

rpm = 4000

Cylinder Diam = 2.3 in.

Piston Stroke = 2.3 in.

This is a comparatively small unit when compared with the 500 hp engine.

A receiver must be added to provide air storage; a vessel 2 ft in diameter and 7-1/2 ft long would contain all the air for one boost. This appears large but the volume of a 500 hp engine plus compressor is figured at 50 to 60 cu ft less than an engine which would develop the 974 hp without the additional air supply.

In the case of continuous output the 500 hp engine plus compressor would occupy about 55 cu ft and give an output of 800 hp; a 800 hp engine alone would occupy about 80 cu ft, which represents a saving of 25 cu ft in power plant bulk.

3.8. CALCULATED PERFORMANCE

The plotted data for these calculations are shown in Figs. 22, 23, and 24.

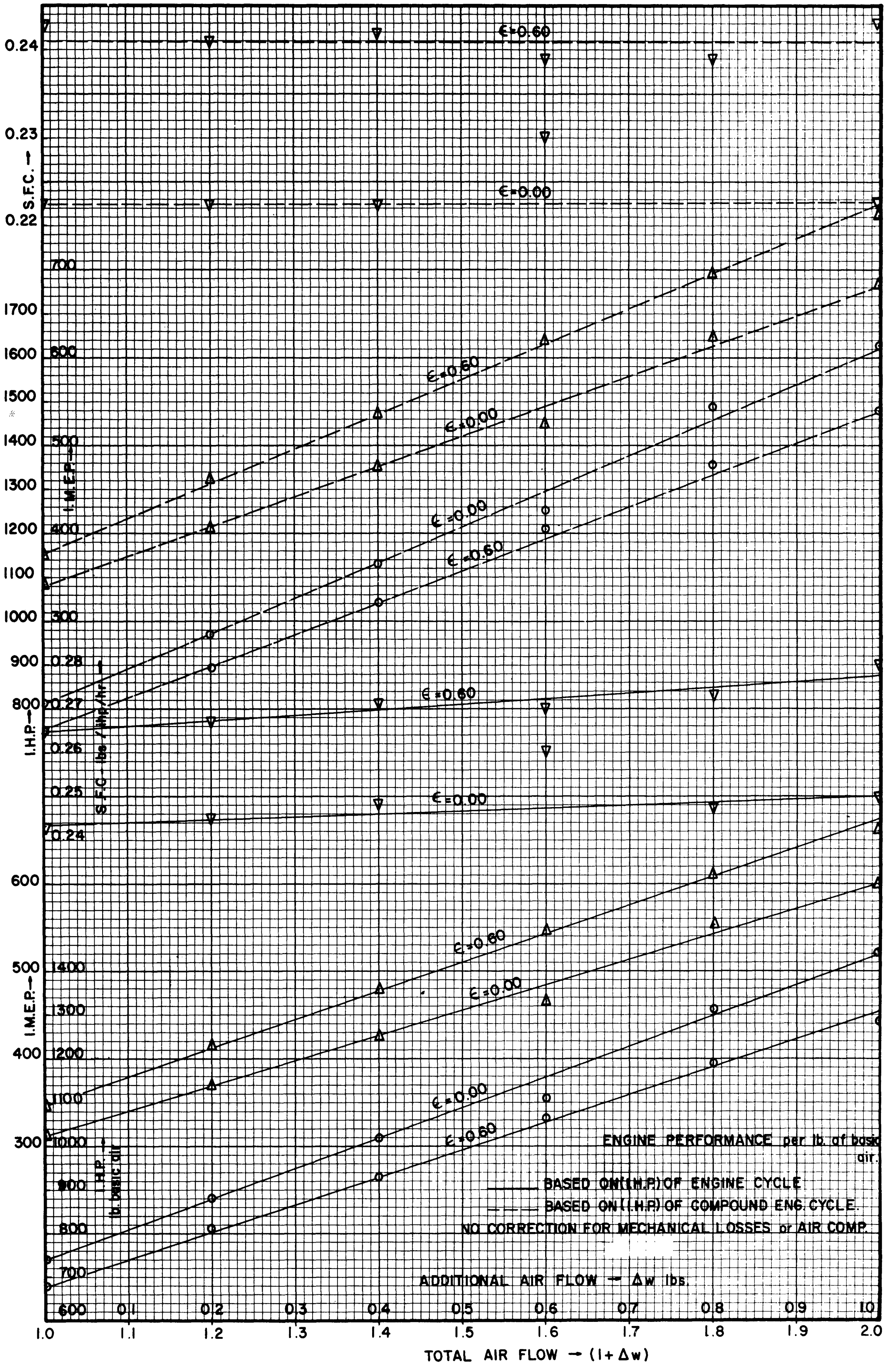

Fig. 22. Air-boosted engine performance.

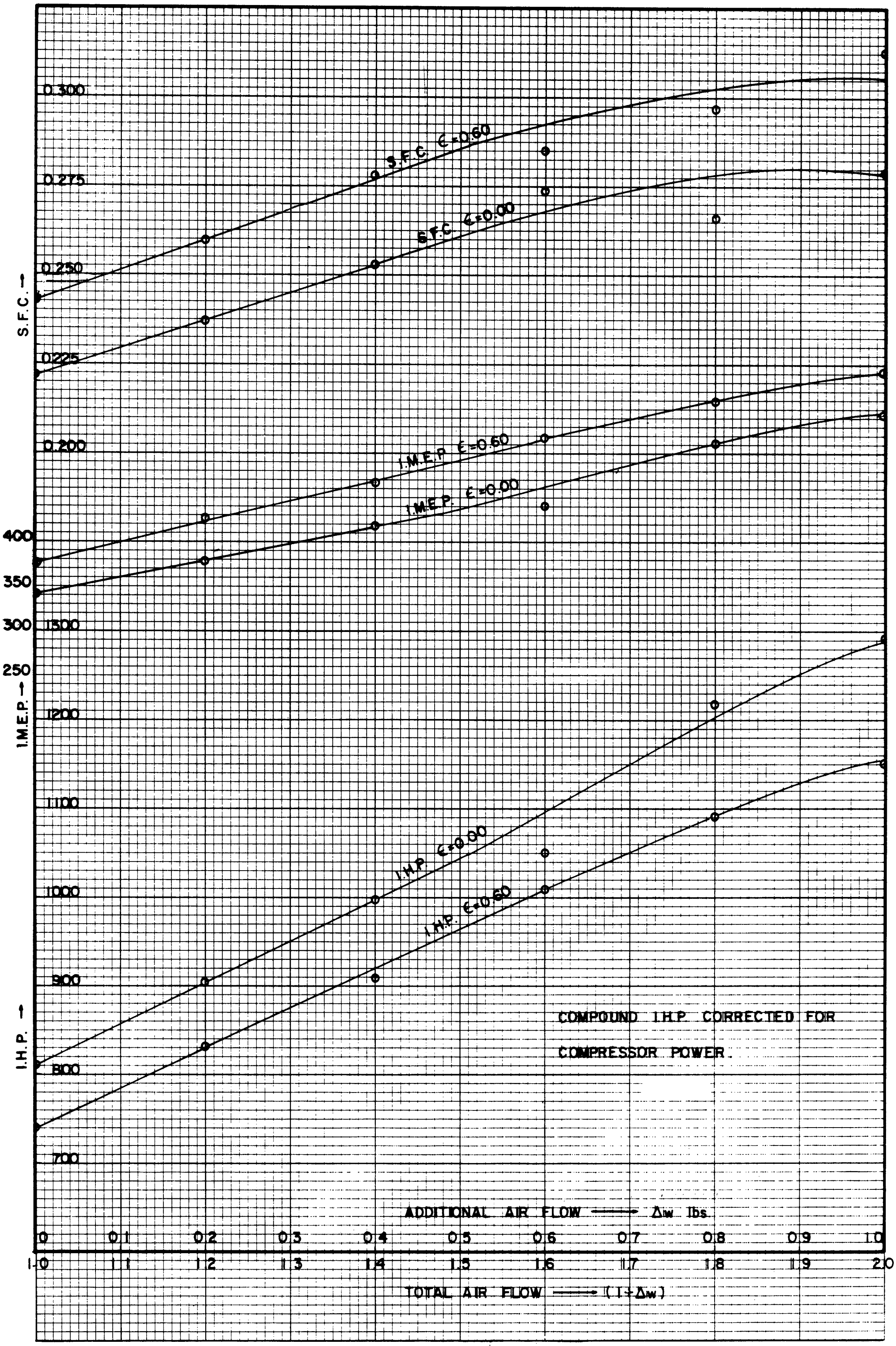

Fig. 23. Air-boosted engine performance corrected for compressor.

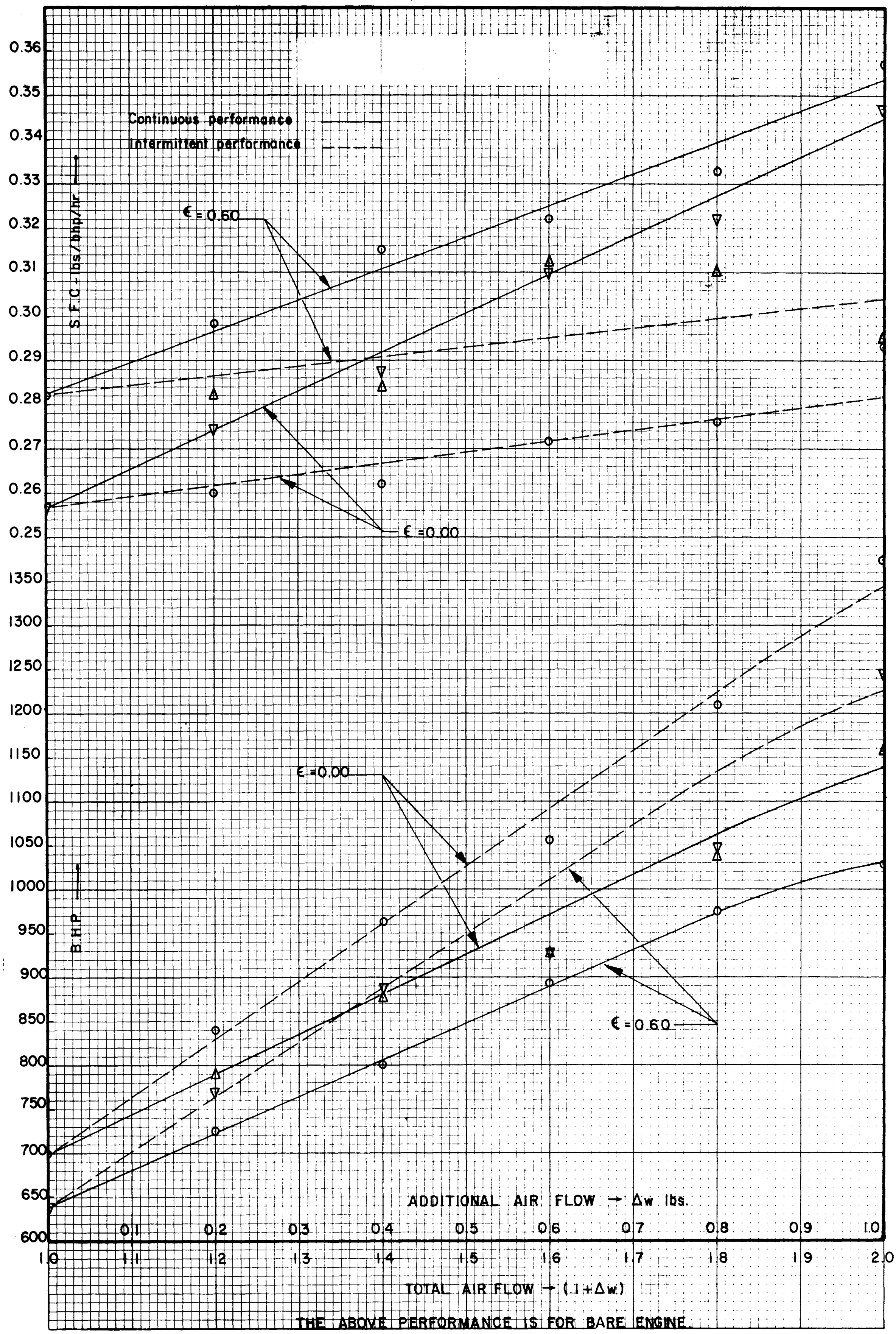

Fig. 24. Net B.H.P. performance of air-boosted engine.

The specific fuel consumptions given represent the minimum attainable on the assumption of perfect combustion of the fuel with the air. In practice, particularly at the high F/A ratio of 0.05, some departure from these values is to be expected, since at such a ratio as the above some smoke is usual and a realistic value of S.F.C. would be some 15% greater than that shown. In addition, the B.H.P. given is that of the bare engine and must be reduced by the accessories driven, such as cooling fan, generator, etc. An allowance for these items cannot be made at the present time since the cooling fan power will depend not only on the engine but the efficiency of the transmission attached to it.

3.9. CONCLUSIONS

The method of additional air supply at the TDC of the compression stroke has advantages for the purpose of military vehicle propulsion where high power is needed for a small fraction of the time. In addition, a degree of responsiveness can also be obtained at the same time by the use of this principle (roughly a 2:1 speed change at constant horsepower).

If the engine and compressor are properly proportioned, it appears that a normal 350-hp compression ignition engine could take the place of a 600-hp one, resulting in a reduction of engine bulk and possibly weight. There would be an increase in the fuel used at the maximum load condition, but a decrease at the 350 hp and less. Since the engine operates most of the time at part load, with 40% of the time at idle, it is believed that an actual reduction in the fuel rate in lb/hr will be achieved when this engine is coupled to the transmission and the operational performance finally established.

The penalty to be paid would be the additional complication of the air-supply system. However, the increased power is obtained without increase of stress levels.

SECTION 4

TURBO-CHARGED COMPRESSION IGNITION ENGINE AND TURBINE CONVERSION

In this case it is proposed to examine the possibilities of an under-powered vehicle using a low-hp, turbo-charged compression ignition engine in order to secure good part-load economy under the majority of operating conditions, with the turbo-charger converted to a gas turbine when required in order to give high load for the desired periods but the engine exhaust still used for gas supply. This arrangement will also provide a high-powered turbo starter capable of continuous cranking of the engine under cold conditions.

The engine arrangement will be as shown diagrammatically in Fig. 25, where it is seen to consist of the compression ignition engine "A" supplied with air by the compressor "B," which is driven by the turbine "C" or through the engine gearing "D," which may also contain a fluid coupling. The exhaust pipe is provided with an afterburner combustor "E," in which additional fuel can be burnt as necessary to secure high-power output from the turbo when it acts as a turbine engine. There is a by-pass "F" which permits completely separate operation of the compressor and turbine units as a gas turbine for starting and heating purposes and also as an augmentor of engine output, as described later.

The complete operation of the unit will be as follows:

1. Starting—with the by-pass valve F open, the turbine can operate as a separate power plant by burning fuel in the combustor.

The main engine can then be cranked with high power, giving high engine speed and easy starting of the compression ignition cycle.

Under these conditions the gearing D will have an over-running clutch or a variable-fill fluid coupling which will permit operating of the turbine and can at will be set to engage the turbine unit as a starter.

2. With the engine running the compressor and turbine will automatically behave as a turbo-charger, but since it is geared via the overrun clutch it can also be designed as a compound unit feeding back to the engine any excess power developed by the turbine improving the fuel economy.

3. The air flow delivered to the engine will be larger than normally employed; i.e., the engine will be overblown as are many marine and railroad engines where good economy is desired. This produces the following results, viz.

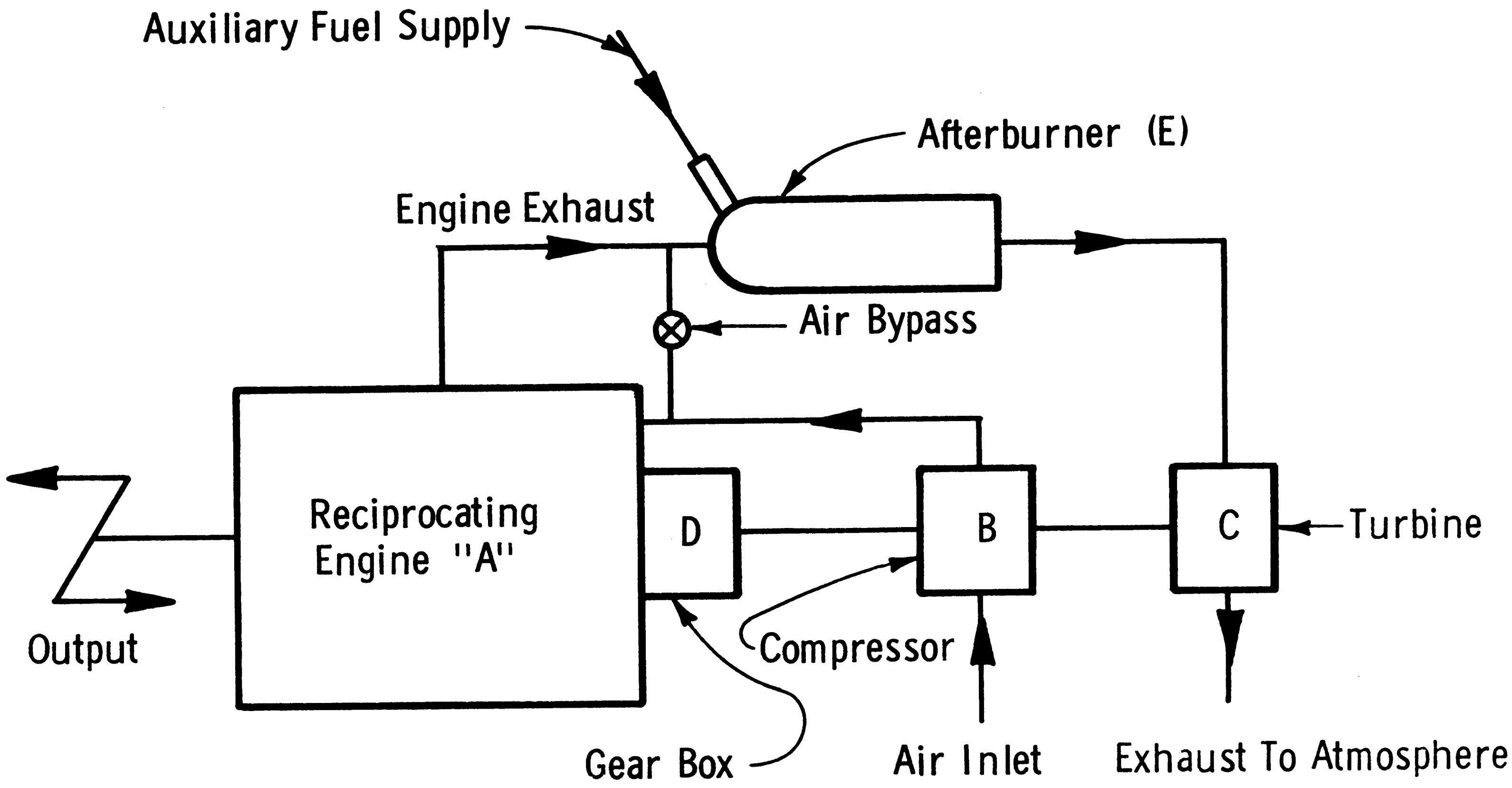

Fig. 25. Turbo-changer turbine conversion.

(a) For a given power output of the engine the F/A ratio will be low, the gas temperatures reduced, and the heat flow to the jacket also reduced; that is, the wall temperature will be lower, assisting in low maintenance, etc.

(b) Due to the use of a low fuel-air ratio the S.F.C. of the engine will improve up to some definite magnitude of the excess air flow.

(c) A large percentage of oxygen will be present in the exhaust gases from the engine. Of course the temperature will also be somewhat lower than normal but turbo-charging will still be possible in most conditions of operation. Where the energy recoverable from the exhaust is slightly deficient for driving the compressor at the desired speed, the gear D will automatically supply the difference.

4. When the engine reaches its capacity and additional load is desired, the increased oxygen content of the exhaust gas due to low F/A makes it possible to burn considerable fuel in the afterchamber—increasing the gas temperature and thus the power and speed of the turbine. As a result, the manifold pressure of the engine is also increased without an abnormal increase in S.F.C., since the auxiliary fuel supply has to raise the gas temperature from the engine exhaust temperature to the maximum desired only.

5. When maximum load is desired, the by-pass valve can be opened and additional air fed directly to the combustor from the compressor, resulting in increased horsepower.

6. The main disadvantage will be the need for a variable nozzle turbine; however this nozzle has already been employed with success in some turbine designs.

4.1. PERFORMANCE CALCULATION

In order to obtain some estimate of the manner in which the performance of such an engine would vary under the different operating conditions the following calculations were made.

In order to keep within present known bounds as far as possible let it be assumed that a moderate responsiveness of, say, 2:1 will be achieved. It will also be assumed that the normal maximum load and speed of the compression ignition engine, when operating alone, is at values consistent with good practice today (say a supercharger pressure ratio of 2.5:1), corresponding to a manifold pressure of 75 in. Hg abs, and fitted with an aftercooler to give an air temperature of 200°F. Under such conditions an I.M.E.P. of 250 psi can be expected with a F/A ratio of about 0.04.

By investigating this engine for a 2:1 responsiveness the following is obtained:

$$\text{Manifold density} = P/RT$$

$$= \frac{144 \times 75 \times 14.7}{29.92 \times 53.34 \times 660}$$

$$= 0.1509 \text{ lb/cu ft.}$$

For an air flow of 1 lb/sec

$$\text{Volume flow} = \frac{3600}{0.1509} = 23900 \text{ cu ft/hr.}$$

Assuming a volumetric efficiency of 93%

$$\text{Engine displacement} = \frac{23900}{0.93} \times 2$$

$$= 51400 \text{ cu ft/hr.}$$

If it is assumed that there is some blow-through during valve overlap, the total air flow by the compressor would be approximately 3800 lb/hr. This total air flow will be available to the turbo, but only 3600 lb will be available for combustion in the cylinder.

$$\text{Indicated engine hp} = \frac{\text{mean pressure} \times \text{displacement cu in./min}}{792000}$$

$$= \frac{250 \times 51400 \times 1728}{792000 \times 60}$$

$$= 467.0 \text{ hp/lb of air/sec}$$

$$\text{Engine displacement} = \frac{51400 \times 1728}{60}$$

$$= 1480000 \text{ cu in./min}$$

$$= \text{cylinder displacement} \times n \times \text{rpm}$$

$$\text{Where } n = \text{number of cylinders} = 6 \text{ say}$$

$$\text{Cylinder displacement} = \frac{1480000}{6 \times 3000}$$

$$= 82.4 \text{ cu in.}$$

This value seems reasonable for the assumed conditions; thus the engine specification for handling 1 lb/sec would be as follows:

Cylinder displacement	82.4 cu in.
Manifold pressure	75 in. Hg
F/A ratio	0.04
I.M.E.P.	250 psi
I.H.P.	467.0 per lb of air/sec
rpm	3000
Air flow	3600 lb/hr
Fuel flow	144 lb/hr/lb of air/sec
S.F.C.	0.308 lb/I.H.P./hr

Now examine what is needed for a 2:1 responsiveness without afterburning. The engine has to develop 467.0 hp/lb of air/sec at 1500 rpm.

When coupled to a turbo-charger the manifold pressure at 50% speed will by Fig. 5 be 0.62% of the 75 in. or 46.5 in. Hg if the I.M.E.P. is maintained at 250 psi. As shown in Fig. 3, such a condition cannot be met. It follows that the turbo speed will have to be increased via the fluid coupling and gearbox D if necessary. When the speed is reduced to 50% the initial I.M.E.P. of 250 has to change to 500 psi for responsiveness; Fig. 3 shows that for a F/A of 0.04 a manifold pressure of 150 in. Hg would be required when

$$\text{Air density} = \frac{144 \times 150 \times 14.7}{29.92 \times 53.34 \times 660}$$

$$= 0.3018 \text{ lb/cu ft.}$$

For an engine suction displacement of 25700 cu ft/hr at 93% volumetric efficiency, i.e., the same engine as previously,

$$\text{Air flow} = 0.3018 \times 25700 \times 0.93$$

$$= 7200 \text{ lb/hr}$$

Fuel flow = 288 lb/hr

hp = 934

S.F.C. = 0.308

These values check approximately those shown in Section 2. It can be seen that conditions are very severe.

Now examine what can be done to achieve a 2:1 responsiveness with afterburning.

4.2. AFTERBURNING

The above calculations have indicated that a 2:1 responsiveness in a conventional engine under the conditions assumed involves extremely high load. It is now proposed to see what if anything can be accomplished by afterburning.

In order to be relatively competitive with present engines and at the same time exploit responsiveness and afterburning, the basic engine obtained above will be somewhat adjusted to reduce the loadings. The basic engine will be assumed to be as follows:

Manifold pressure	70 in. Hg
I.M.E.P.	200 psi
Fuel/air	0.032
Air density	= P/RT
	$= \dfrac{144 \times 70 \times 14.7}{29.92 \times 53.34 \times 660}$
	= 0.1408 lb/cu ft
Volume flow/lb/sec	= 25600 cu ft/hr
Engine suction displacement at 93% volume efficiency	= 27500 cu ft/hr
Engine displacement (4 cycle)	= 2 x 27500
	= 55000 cu ft/hr

Engine hp $= \dfrac{200 \times 55000 \times 1728}{792000 \times 60}$

$= 400$ hp/lb of air/sec

Cylinder displacement/min $= \dfrac{55000 \times 1728}{60}$

$= 1582000$ cu in./min

For a 6-cylinder engine as above at 3000 rpm

Cylinder displacement $= \dfrac{1582000}{6 \times 3000}$

$= 88.0$ cu in./cylinder

The basic engine specification thus becomes

I.H.P.	=	400/lb of air/sec
I.M.E.P.	=	200 psi
Manifold pressure	=	70 in. Hg abs
F/A	=	0.032
rpm	=	3000
Displacement/cylinder	=	88.0 cu in.
Fuel flow	=	115.2 lb/hr
S.F.C.	=	0.288 lb/I.H.P./hr

After several calculations it was decided that, to approximate the required degree of responsiveness (2:1) it would be necessary to employ a manifold pressure of 120 in. Hg at low speed, even when by-passing air. The results obtained under these conditions are as follows

Air density $= \dfrac{144 \times 120 \times 14.7}{29.92 \times 53.34 \times 660}$

$= 0.24101$ lb/cu ft

Volume flow for some displacement $= 25600$ cu ft/hr at 3000 rpm

$= 12800$ cu ft/hr at 1500 rpm

Air flow $= 12800 \times 0.24101$

$= 3080$ lb/hr

When a F/A ratio at this high manifold condition of 0.025 is used to provide high oxygen content, Fig. 3 shows that an I.M.E.P. of 260 is available; thus:

$$\text{Output of engine} = \frac{\text{I.M.E.P.} \times \text{displacement} \times \text{rpm}}{792000}$$

$$= \frac{260 \times 88.0 \times 6 \times 1500}{792000}$$

$= 260$ I.H.P.

Fuel flow $= 0.025 \times 3080$

$= 77.0$ lb/hr

$$\text{S.F.C.} = \frac{77.0}{260} = 0.296 \text{ lb/I.H.P./hr}$$

Allow a 5% blow through air

Total air flow $= 1.05 \times 3080$

$= 3240$ lb/hr.

For the assumed conditions, with a F/A = 0.025, an exhaust temperature of about 1000°F = 1460° abs will exist (Fig. 7).

To evaluate the effect of afterburning, some maximum exhaust gas temperature must be assumed. This will be determined by the materials of the turbine. In this case assume that a peak temperature of 1700°F = 2160° abs can be employed. Using the Keenan and Kay tables for a 200% theoretical air mixture, then:

Enthalpy of exhaust gas from engine $= 10729.3$ Btus/mol

$= 371$ Btus/lb

Enthalpy of gas leaving the combustor $= 16557.7$ Btus/mol

$= 572$ Btus/lb

Assuming a combustion efficiency of 95% and a calorific value of the fuel of 18500 Btus/lb, then:

Fuel required in afterburner $= \dfrac{572-371}{0.95 \times 18500} = 0.01142$ lb/lb of gas

Total gas flow $= (3240 + 77.0)(1 + 0.01142)$

$= 3347$ lb/hr

Assume engine back pressure at 87% of the manifold pressure

Back pressure $= 0.87 \times 120$

$= 105$ in. Hg

Expansion ratio $= 105/29.92$

$= 3.5$

4.3. TURBINE WORK

Inlet temperature to turbine = 2160° abs

Using gas tables,

Enthalpy = 16557.7 $\quad P_{r_1} = 294.8$

At turbine exhaust $\quad P_{r_2} = 294.8/3.5$

$= 84.0$

∴ Isentropic gas temp. = 1604° and enthalpy = 11892.3

Change of enthalpy = 4665.4 Btus/mol

= 161.2 Btus/lb

For an 88% efficient turbine

Work output of turbine $= 0.88 \times 161.2$

= 142.0 Btus

$$= 200 \text{ hp/lb of gas/sec}$$

$$= 200 \times \frac{3347}{3600} \text{ hp for engine}$$

$$= 186 \text{ hp}$$

4.4. COMPRESSOR WORK

Pressure ratio of compressor $= \dfrac{120}{29.92} = 4.005$

Assuming an inlet temperature of 120°F

State of gas at entry $T_1 = 580°$

$h_1 = 138.66$

$P_{r_1} = 1.78$

Isentropic state of gas at exit $P_{r_2} = 4.005 \times 1.78 = 7.14$

$T_2 = 860°$ abs

$h_2 = 206.5$

With a 78% efficient compressor

Work of compression $= \dfrac{206.5 - 138.6}{0.78} = \dfrac{67.9}{0.78}$

$= 87.0$ Btus/lb

Air flow $= 3240$ lb/hr

hp of compressor $= 87.0 \times \dfrac{3240}{3600} \times \dfrac{778}{550}$

$= 110.8$ hp

Net engine output $= \text{B.H.P.} + hp_{turb} - hp_{comp}$

$= 260 + 186 - 110.8$

$= \underline{\underline{335.2 \text{ hp}}}$

Fuel flow (engine + turbine) $= 77.0 + 37.9$

$= 114.9$ lb/hr

Specific fuel consumption $= \frac{114.9}{335.2}$

$= 0.342$ lb/hp/hr

Now this same engine alone operating with a turbo-charger only will develop about 40% of its full load rating (see Fig. 5) at 50% speed or 80 hp. Thus the use of afterburning plus speed control of the turbo-charger has increased this to a total of 335 hp—which is almost equal to the initial engine rating of 400. At the same time the S.F.C. has only increased from 0.288 lb/I.H.P./hr to 0.342 lb, a moderate increase for a hp increase of about 4:1 from 80 to 335. Without the afterburner the performance, at the same manifold pressures, would be 260 hp at a S.F.C. of 0.296. The afterburner alone has added 29% power for a 16% increase in fuel flow.

The turbo-charged basic engine when operating at full load and speed developed 400 I.H.P.; with afterburner the hp at 50% speed is estimated at 335, which is still somewhat short of 2:1 responsiveness. To achieve the required degree of response the by-pass valve between the compressor outlet and the afterburner inlet can be opened to supply a greater air flow for the turbine. There is a pressure difference of $\Delta P = 120\text{-}105 = 15$ in. Hg between these pipes, so flow will occur. Calculate what this flow must be to achieve the response required.

Additional hp required $= 400 - 335$

$= 65$ hp

There is no need for the additional air flow to go through the after-cooler since it is to be directly heated to the maximum temperature of 1700°F.

Temperature of air leaving compressor = temp. for an enthalpy of (138.66 + 87.0) = 225.66

= 938 abs

This air must be heated from 938° to 2160° abs

Fuel required at 95% combustion efficiency	=	$\dfrac{549.35 - 225.66}{0.95 \times 18500}$
	=	$\dfrac{323.69}{0.95 \times 18500}$
	=	0.0184 lb/lb of air
Additional gas flow through turbine/lb of by-pass air	=	1.0184
hp of compression/lb air/sec	=	$87.0 \times \dfrac{778}{550}$
	=	123.0 hp
hp of turbine/lb of air/sec	=	200 x 1.0184
Increase in power/lb of by-pass air	=	203.5 - 123
	=	80.5 hp
Additional power required	=	65
By-pass air	=	65/80.5
	=	0.765 lb/sec
	=	2750 lb/hr
Additional fuel	=	0.0184 x 0765
	=	0.01408 lb/sec
	=	50.7 lb/hr
Over-all fuel consumption	=	77.0 + 37.9 + 50.7
	=	165.6 lb/hr
Total output	=	400 hp
S.F.C.	=	165.6/400
	=	0.413 lb/hp/hr .

It must be remembered in considering these figures that the reciprocating engine is on an I.H.P. basis, while the turbine is actual output. There would be about 28 hp loss in friction at half speed, thus on a B.H.P. basis the performance would become

rpm	=	1500
Total B.H.P. output	=	372
Total fuel flow	=	165.6
S.F.C.	=	0.445 lb/B.H.P./hr

4.5. RECIPROCATING ENGINE ONLY

rpm	=	1500
B.H.P.	=	232
Fuel flow	=	77.0
S.F.C.	=	0.335

The low S.F.C. is due to the use of the low F/A ratio of 0.025 at the half-speed condition. Without the afterburner, so low a value for F/A would not be employed and the same hp could be secured with a lower manifold pressure than the 120 in Hg.

The above results are presented in Table XVI which is based on 1 lb of air/sec for case (a) actually trapped in the cylinder with a total flow of 3800 lb hr.

Considering these results it is seen that a constant horsepower of 400 is capable of being maintained over a speed range of 2:1 with the aid of a turbo-charger converted to a gas-turbine when necessary. The penalty in this case is still a high manifold pressure and thus high peak cylinder pressures; there is also some moderate increase in fuel consumption but only when exceptional conditions make conversion to a turbine necessary.

A second alternative, not examined in this report, is to employ the same system in a nonresponsive engine. In the case the compression ignition engine is designed for about 0.6 to 0.7 of the required maximum output and the normal full power at full speed is reached by the turbine conversion. In this case it will be found that moderate manifold pressures result and that a smaller engine displacement is needed, giving higher economy in fuel and

TABLE XVI

AFTERBURNING ENGINE COMBINATION

Condition	Total Air Flow, lb/hr	Fuel Flow, lb/hr	Over-all F/A in Cylinder	Total F/A	Total hp	S.F.C. lb per I.H.P./hr	Manifold Pres. in. Hg	Manifold Temp. ° abs	Engine I.M.E.P.	Relative Speed of Turbo
Engine Only										
(a) Full speed	3800	115.2	.032	.0304	400	.288	70	660	200	1.00
(b) Half speed	3240	77.0	.025	.0237	260	.296	120	660	260	1.32
Engine + Turbine										
Half speed	3240	114.9	.025	.0354	335	.342	120	660	260	1.32
Half speed plus by-pass	5990	165.6	.025	.0276	400	.413	120	660	260	1.32

space. Such an engine will supply the great majority of the operating schedule and a moderate penalty in increased fuel will be paid for the peak power only.

One other problem to be considered in this instance is the range of air and gas flow to be handled by the compressor and turbine.

Table XVI shows the air flow to be 3240, 3800, and 5990 lb/hr. The first range, 3240 to 3800, is not disturbing and could readily be met in a centrifugal compressor if the manifold pressure did not change. In the case in question this pressure goes from 70 in. to 120 in. Hg at the same time; to do this a speed increase of the turbo of 1.32:1 is required and a new part of the compressor field of flow would be involved. Judging from the general shapes of such compressor maps even this is not an impossible condition, but a complete compressor analysis would be necessary to establish this.

In the case of a flow increase to 5990 lb/hr some difficulties would be encountered since this increase is also required with the manifold pressure of 120 in. Hg. A variable nozzle turbine could be employed. Such items are becoming common in some fields of gas turbine practice, but for the compressor it is not so easy to maintain such a high flow while maintaining efficiency as well. True, the low-flow, low-speed condition does not involve high unit efficiency since there is power to spare in the exhaust gases under these conditions and the maximum efficiency conditions could be pushed in the direction of the high-speed case. Again only a thorough investigation of the compressor conditions will establish the practicability of such a change of air flow.

The change of area involved is not as great as the flow rates alone indicate; it must be remembered that a change of pressure occurs at the same time. A rough estimate indicates the relative areas for the turbine nozzles would be 1.0, 0.51, and 0.92 for the three conditions; this condition does not look impossible to meet.

4.6. CONCLUSIONS

The method examined in Section 4 shows that a 2:1 responsiveness could perhaps be obtained by the proposed scheme. It would, however, involve the following problems:

1. The development of a wide range of air flows in a two-speed compressor.

2. The development of a variable-area nozzle for the turbine.

3. Relatively high manifold pressures under the peak loading at low speeds and thus high combustion pressures.

The normal engine operation during the great majority of its life would be at other than responsive conditions, in which case normal manifold and combustion pressures would be involved most of the time.

As was concluded in Section 2, since this design does not resolve the need for a multi-step transmission, any gains of fuel economy will be of negligible proportions; and the value of the scheme must be based on other performance factors.

SECTION 5

PUMPING CYCLE

The pumping cycle is defined as one in which the engine, in addition to developing power, is acting as a pump. This means that the pressure in the exhaust pipe will exceed that in the intake manifold and the engine must pump the mixture from one side to the other.

The object of this examination is to determine if the exhaust turbine will increase in power much faster than the reciprocating engine fall in power as the exhaust manifold pressure increases for a constant inlet pressure. If this difference in rate of power change is sufficiently great, an approximation to a responsive engine results. Sufficient power increase for a given speed change may result so that a large part of the benefits of responsiveness can be secured without too much complication and difficulty.

In order to achieve this pumping cycle it is believed that a considerable change in valve timing would be involved so that overlap is eliminated to prevent exhaust gas blow through into the intake manifold and diluting the intake air. Such a change of timing would adversely affect the basic engine output by some percentage.

In analyzing this cycle the difference from the conventional will be the fact that the clearance space will be left filled with exhaust gas at exhaust manifold pressure P_e, which must first be expanded to intake manifold pressure P_m before the inlet valve opens to admit fresh air. This expansion can occur by opening the inlet early, allowing the gas to expand through the valve, and then sucking it back into the engine cylinder; theoretically the results would be the same whichever method was employed. From a practical standpoint the expansion of the gas by piston movement seems preferable since, as P_e is $> P_a$, work is done on the piston by this process; though to meet the varying value of P_e with load in a simple manner, this expansion process need not be complete for all conditions.

5.1. PERFORMANCE ESTIMATION

In order to obtain some concept of what the resultant engine performance will be, the following methods and assumptions were employed.

Turbo-charger ratio	=	3.0:1
Maximum F/A ratio	=	0.045

Manifold temperature (by aftercooler)	=	200°F
Inlet manifold pressure	=	90 in. Hg abs
I.M.E.P. expected (normal engine)(see Fig. 3)	=	325 psi
Expected S.F.C. (bare engine)	=	0.30 lb/B.H.P./hr
Compression ratio	=	17:1
Average exhaust gas temperature in cylinder	=	1400°F

The above data would be expected from a normal oil engine under the assumed conditions.

The performance expected will be evaluated by taking the normal dual combustion cycle and evaluating the work of compression, expansion, exhaust, and suction strokes, together with the fuel supply at the assumed F/A ratio. The value of the exponent "n" in PV^n = constant is taken as the average found for the large number of cases, with variable heat losses employed to develop Fig. 1, expansion of exhaust gases being $n = 1.33$ and compression of the new charge having a variable "n" depending upon the temperature of the mixture of exhaust gas and air trapped in the cylinder due to the back pressure conditions.

The calculations will be based upon the flow of 1 lb of air/sec under all conditions, which means that the engines for the different back pressures will be of slightly different displacements resulting from the expansion of the trapped exhaust gas. It is believed that the results so obtained will permit an evaluation of the assumed process despite this difference in engine sizes. If the results are encouraging a more detailed study to eliminate this difficulty can be made.

5.2. CALCULATIONS OF EXHAUST GAS IN CYCLE

The clearance space at TDC is the starting point. With this volume filled with exhaust gas at pressure P_e, expansion occurs to inlet pressure P_m; and using the diagram of Fig. 26.

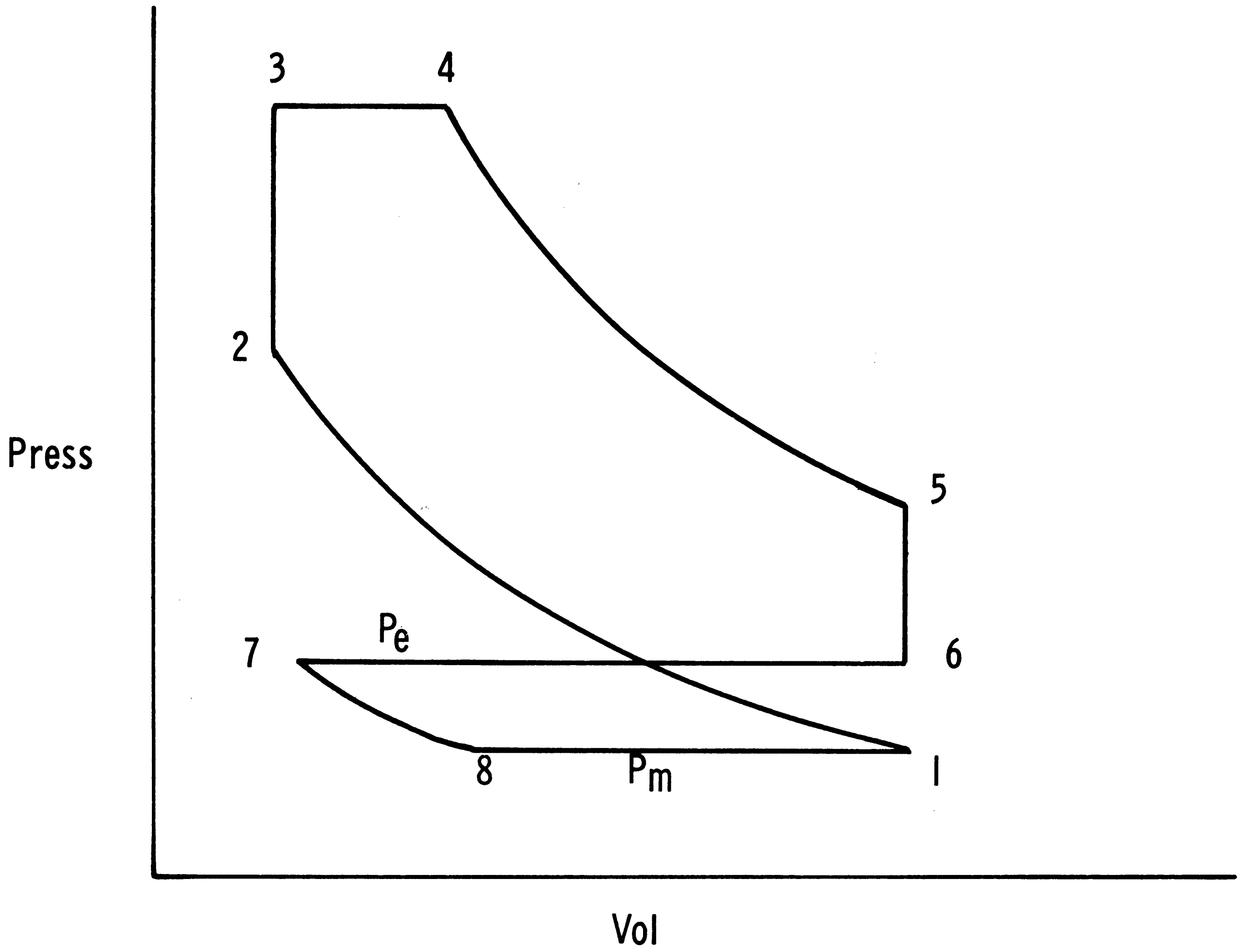

Fig. 26. Indicator diagram for pumping cycle.

Assumed expansion curve 8 - 7 $=$ $PV^{1.33}$

$$P_e V_7^{1.33} = P_m V_8^{1.33}$$

$$V_8 = \left(\frac{P_e}{P_m}\right)^{\frac{1}{1.33}} (V_7) .$$

With the manifold pressure at 90 in. Hg, assume that the back pressure P_e varies from 80 in. Hg (a normal engine design) in steps up to 200 in. Hg then the exhaust expansion will proceed as follows:

P_e in. Hg abs	80	120	160	200
$\left(\frac{P_e}{P_m}\right)^{1/1.33}$	0.915	1.238	1.531	1.813
V_8 if $V_7 = 1.0$	0.915	1.238	1.531	1.813
$V_8 - V_7$	-.085	+.238	+.531	+.813

In the above table V_7 has been assumed as 1 unit; this does not mean 1 cu ft, rather that with a compression ratio of 17:1 the 1 unit will be the volume of 1 lb of inlet air plus the exhaust gas when compressed to TDC.

The 1 lb of new air charge then occupies volume V_1-V_8; thus

$$V_1 - V_8 = \frac{wRT}{P} = \frac{1.0 \times 53.34 \times 660 \times 29.92}{144 \times 90 \times 14.7}$$

$$= 5.53 \text{ cu ft}$$

$$\text{Compression ratio} = \frac{V_1}{V_7} = 17 .$$

Since V_7 has been made 1.0 it follows that

$$V_1 - V_7 = 16$$

$$(V_1 - V_8) + (V_8 - V_7) = 16$$

$$V_1 - V_8 = 16 - (V_8 - V_7)$$

and since

$$V_1 - V_8 = 5.53 \text{ cu ft}$$

we obtain the following values for the unit employed:

P_e =	80	120	160	200
$V_8 - V_7$	-0.085	+.238	+.531	+.813
$V_1 - V_8$ (units) = 5.53 in ft	16.085	15.762	15.469	15.187
1 unit = cu ft	0.344	0.3505	0.3575	0.364
V_1 cu ft = 5.53+1 unit	5.85	5.96	6.08	6.20
V_8 + 1 unit	0.32	0.43	0.55	0.67
Relative displacement to 80 in. P_e	1.0	1.019	1.04	1.06
Percentage change in displacement	0.0	1.9	4.0	6.0

It is seen that a 6% change in displacement is the maximum involved. It will be taken into account in evaluating the cycle but will be neglected as far as its effect on evaluating the resultant engines in this first approximation to the problem.

The exhaust gas at (7) has been assumed to have a temperature of 1400°F; its temperature at 8 and the work done from 7 to 8 is as follows.

$$T_8 = T_7 \left(\frac{P_m}{P_e}\right)^{n-1/n}$$

$$\text{Work done} = \frac{P_7V_7 - P_8V_8}{n - 1}$$

$$n = 1.33 .$$

Evaluating these magnitudes the following results together with the temperature of the mixture at point 1.

P_e =	80	120	160	200
T_8	1918	1730	1618	1528
Work V_7-V_8, Btus	0.0	0.672	2.33	4.34
Wt of exht. w_r = PV/RT	0.0178	0.0396	0.0724	0.1057
Wt of air	1.00	1.00	1.00	1.00
Temperature air	660	660	660	660
Specific heat of products	0.275	0.275	0.275	0.275
Specific heat air	0.241	0.241	0.241	0.241
Enthalpy of products = wC_pT	9.38	18.82	32.2	44.4
Enthalpy of air = wC_pT	159.0	159.0	159.0	159.0
Enthalpy of mixture	168.38	177.82	191.2	203.4
$w_aCp_a+w_rCp_r = w_mCp_m$	0.246	0.252	0.260	0.270
Temp. of mixture = T_1	685	705	736	753

The temperature at point 1 is evaluated by

Enthalpy before mixing = enthalpy after mixing

$$h_{exht} + h_{air} = h_{mix}$$

$$w_r Cp_r T_r + w_a Cp_a T_a = w_{mix} Cp_{mix} T_{mix}$$

$$T_1 = T_{mix} = \frac{w_r Cp_r T_r + w_a Cp_a T_a}{w_a Cp_a + w_r Cp_r}$$

where

w_r and w_a = wt of exhaust and air

C_{P_r} and C_{P_a} = specific heat of exhaust and air

T_r and T_a = temperature of exhaust and air.

The state of the gas at point 1 is now known for the conditions being evaluated.

5.3. THE COMPRESSION PROCESS

It was assumed that the following heat loss schedule represented the conditions during the cycle.

Heat lost during compression	= 1.5% of calorific value of fuel injection
	= 13 Btus
Heat lost during constant volume combustion	= 2.5% = 21 Btus
Heat lost during constant pressure combustion	= 3% = 25 Btus
Heat lost during expansion stroke	= 5% = 42 Btus
Incomplete combustion	= 2% = 17 Btus.

With the above value of heat losses and the method shown in Ref. 1, page 56, the following table is constructed for the compression and expansion strokes.

P_e in. Hg	80	120	160	200
T_1	685	705	736	753
"n" in PV^n = constant	1.34	1.339	1.338	1.336

Work of compression, Btus	223.0	229.8	239.5	244.0
km	1.3615	1.3595	1.357	1.355
P_2	1945	1940	1932	1924
T_2	1780	1833	1915	1960
V_4/V_3	2.5	2.46	2.39	2.35
V_3	0.344	0.3505	0.3575	0.364
V_4	0.861	0.863	0.855	0.855
T_4	4450	4493	4575	4560
V_5/V_4	6.68	6.88	7.10	7.29
P_5	175.0	168.0	160.8	155.0
V_5	5.85	5.96	6.08	6.20
T_5	2665	2670	2695	2670

The state of the gas at each point of the diagram is thus obtained and it is possible to calculate the work of the various phases from

Work of constant pressure combustion $= \frac{144}{788} P_2(V_4 - V_3)$

Work of expansion $= \frac{P_4V_4 - P_5V_5}{(n - 1)} \times \frac{144}{778}$

Work of suction $= \text{work}_{7-8} + \frac{144}{778} P_m(V_1 - V_8)$

Work of exhaust $= \frac{144}{778} P_e(V_6 - V_7)$

giving the following table.

P_e =	80	120	160	200
Combustion (1)	185.9	184.0	177.8	174.8
Expansion (2)	402	413	415	423
Exhaust (3)	40.1	60.2	80.3	100.7
Suction (4) (7-8)+(8-1)	45.2	45.87	47.5	49.5
Compression (5)	223.0	229.8	239.5	244.0
Work of cycle (1+2+4)-(3+5), Btus	370.1	352.9	320.5	302.6
I.H.P./lb of air/sec	524	499	454	429
I.M.E.P. psi	364	340	303	281
Fuel flow lb/hr	162	162	162	162
S.F.C. lb/I.H.P./hr	0.309	0.324	0.356	0.378

The above gives the engine performance alone. If the turbo-charger is compounded to the engine to pickup the additional power available in the exhaust which isnow considerable, then the proposal will result in the following compressor and turbine performance. It is necessary to know the expected exhaust gas temperature in the manifold. This is given by equating the internal energy in cylinder at point 5 to the energy in the exhaust pipe, making due allowance for heat losses.

$$\text{Exhaust gas temperature } T_e = \frac{C_v}{C_p} T_5 + \frac{0.185\ P_e V_5 - dQ}{(1 + f) C_p}$$

where C_v and C_p are specific heats

dQ = heat lost during operation

f = F/A

Using $C_v = 0.24$ and $C_p = 0.33$ for the temperatures involved, and $dQ = 50$ Btus, the following table is calculated for a turbine efficiency of 0.87 and $C_p = 0.284$.

$$\text{Work of turbine} = 0.87\ wC_pT_e \left[1 - \left(\frac{P_o}{P_e}\right)^{k-1/k}\right]$$

P_e	80	120	160	200
T_e	1918	1983	2050	2121
Turbine press ratio = P_e/P_o	2.67	4.01	5.34	6.69
Work of turbine, Btus	101.0	145.0	182.0	214.0
Work of compressor Btus, η_c = 0.70	73.4	73.4	73.4	73.4
Net output of turbo-charger, Btus	28.6	71.6	108.6	140.6
Total output engine + turbo, Btus	398.7	424.5	429.1	443.2
Total hp	565	601	608	627
Effective I.M.E.P. psi	392	408	406	411
Effective S.F.C., lb/hp/hr	0.287	0.27	0.267	0.258

The two results, engine alone with back pressure exceeding the inlet pressure and engine plus turbo geared together under the same condition, are plotted in Fig. 27 on a base of P_e/P_m.

Considering these results on the basis of responsiveness alone, it is seen that for a very considerable back pressure, which would involve a multi-stage turbo, the scheme would not be particularly responsive—the compound hp only increasing some 20% from normal turbo-charging to compounding with rather severe exhaust conditions. The main advantage shown by this scheme is the improved specific fuel consumption arising from a compound cycle, the economy improving by about 20% as the power increased by 20% approximately.

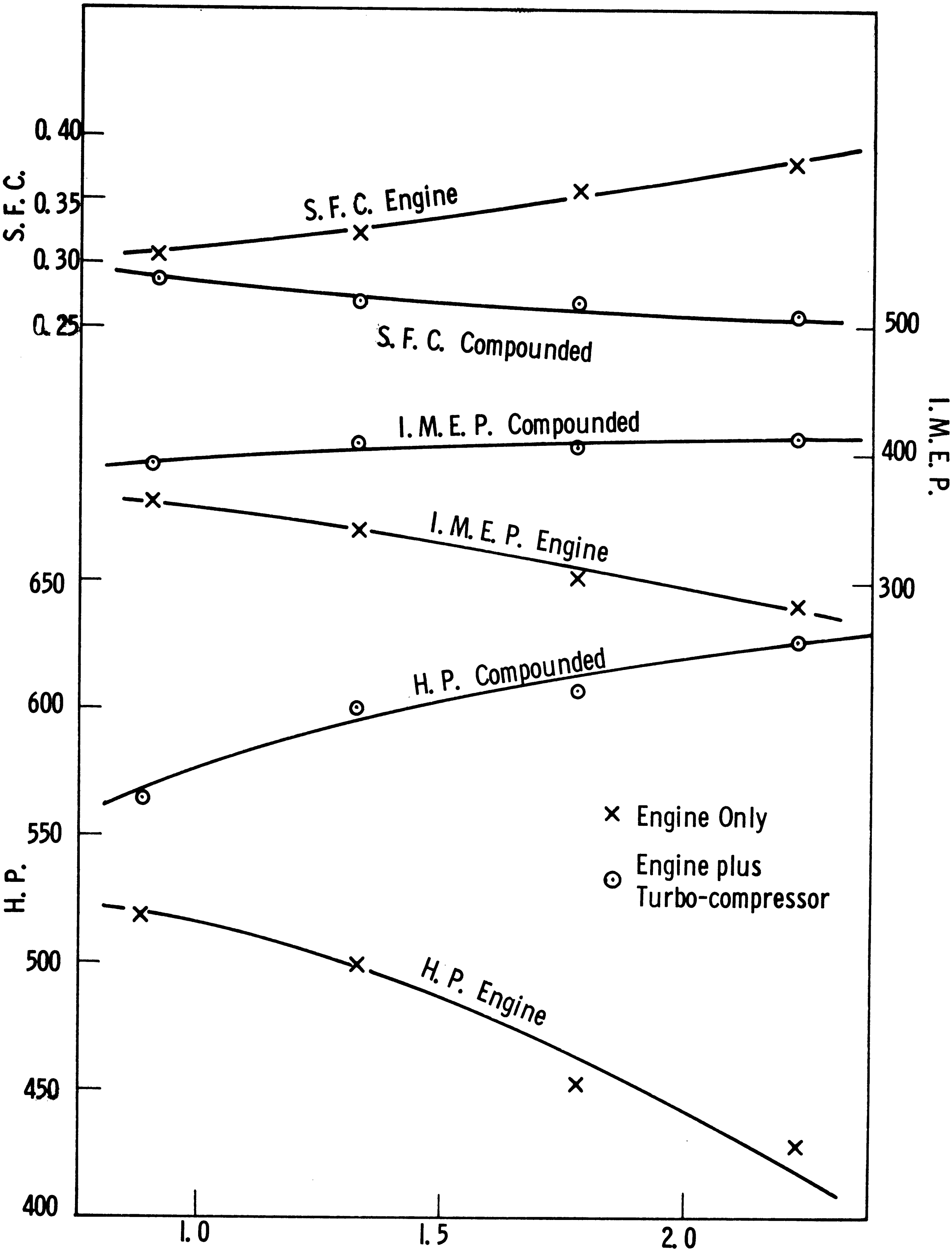

Fig. 27. Pumping cycle performance.

5.4. CONCLUSIONS

1. A pumping cycle results in an improvement in responsiveness, but only to a small degree. It might be sufficient to assist in improved handling of the engine by the operator.

2. Such a cycle can improve the fuel consumption, at least under the responsive conditions investigated.

3. To judge the over-all performance, when most of the operation would be other than responsive conditions, additional calculations would be necessary.

SECTION 6

TURBINE POWER PLANT

It is not proposed to examine all the combinations of turbines and compressors possible, since that would be a very major operation. It is proposed to examine a simple cycle turbine with and without a regenerator. The purpose of this is to enable some estimates to be made of a possible dual power plant—that is, a compression ignition engine and a gas turbine which are independent units capable of operation separately, and are not compounded in any way. The definition of a compound engine employed in this report specifies that the same gases generated in one unit pass in turn through the two or more units on their way to the atmosphere. The term dual power plant seems to fit the arrangements to be examined in this section.

The method employed in this instance will be that of the Keenan and Kay Gas Tables, to which reference can be made for details.[4] The cycle examined is as illustrated in Fig. 28. The pertinent gas states are as numbered at the various stations.

The following assumptions apply to the various cases examined:

Inlet state of air	P_1 =	14.2 psi
	T_1 =	115°F = 575°R
Compressor efficiencies		78 and 83%
Combustion chamber pressure drop		5% of P_3 with no regenerator 6% of P_3 with no regenerator
Combustion efficiency		96%
Turbine efficiency		86 and 88%
Regenerator effectiveness		0.75
Exhaust back pressure (no regenerator)		15 psi
Exhaust back pressure (with regenerator)		16 psi
Combustion temperature		1750°F = 2210°R

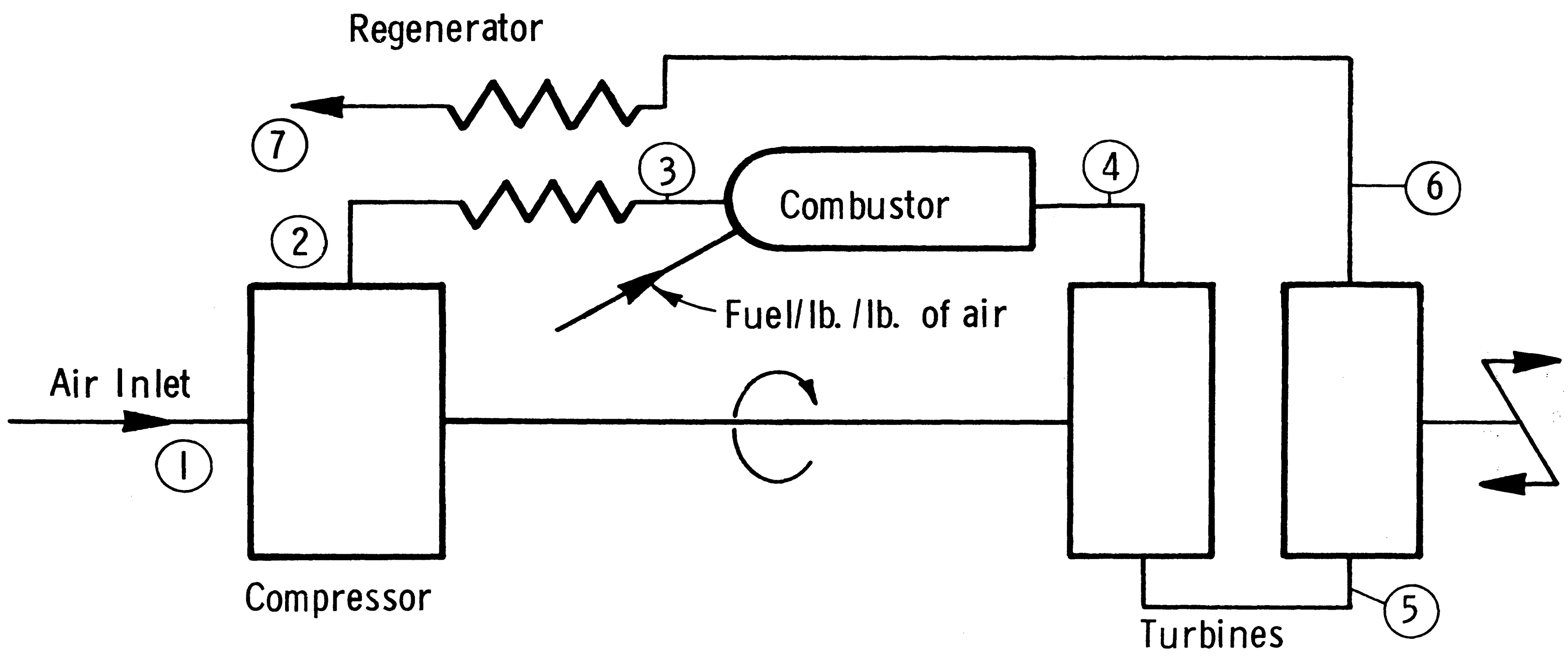

Fig. 28. Turbine power plant.

In order to bring out some of the results desired, the compressor pressure ratio will be varied from 3.0 to 15.0:1 for both the non- and regenerated cases. In the following table these symbols are used.

h = enthalpy at station 1, 2, etc.

P_r = pressure ratio from tables

$T_{2'}$ = isentropic temperature at 2, etc.

T_2 = actual temperature at 2, etc.

ϵ_0 = case with no regeneration

ϵ_{75} = case with 0.75 regeneration

6.1. COMPRESSOR

Press ratio P_2/P_1	3.0	6.0	9.0	12.0	15.0
H_1	137.5	137.5	137.5	137.5	137.5
P_{r_1}	1.727	1.727	1.727	1.727	1.727
P_{r_2}	5.18	10.36	15.52	20.7	25.87
$T_{2'}$	786	954	1066	1153	122.4
$h_{2'}$	188.4	229.57	257.46	279.37	297.43
$\Delta h'$	50.9	92.1	120.0	141.9	159.9
Δh comp. effy. = .78	65.3	118.1	153.9	182.0	205.0
h_2	202.8	255.6	291.4	319.5	342.5
T_2	845	1058	1200	1310	1398
Δh comp. effy. = .83	61.4	111.0	144.6	171.7	192.5
h_2	198.9	248.5	282.1	308.5	329.9
T_2	829	1030	1164	1267	1350

6.2. COMBUSTION

With an outlet temperature of 2210°R the state of the gas at station 4, 5, and 6 becomes

Ratio	3.0	6.0	9.0	12.0	15.0
h_4	563.4	563.4	563.4	563.4	563.4
P_{r_4}	261.4	261.4	261.4	261.4	261.4
P_2	42.6	85.3	128.0	170.5	213.0
$P_{4\epsilon_0}$	40.5	81.0	121.6	162.0	202.3
$P_{4\epsilon_{75}}$	40.0	80.2	120.2	160.2	200.2
Expansion ratio ϵ_0	2.698	5.4	8.1	10.9	13.49
Expansion ratio ϵ_{75}	2.67	5.35	8.01	10.68	13.35
$P_{r_5}-\epsilon_0$	97.0	48.4	32.3	24.2	19.4
$P_{r_5}-\epsilon_{75}$	98.0	48.9	32.6	24.5	19.6
$T_{5'\epsilon_0}$	1728	1445	1299	1203	1133
$h_{5'\epsilon_0}$	430.15	354.69	316.68	292.07	274.31
$T_{5'\epsilon_{75}}$	1733	1449	1302	1207	1136
$h_{5'\epsilon_{75}}$	431.5	355.7	317.5	293.1	275.1
$\Delta h'_{\epsilon_0}$	133.26	208.7	246.7	271.3	289.1
$\Delta h'_{\epsilon_{75}}$	131.9	207.7	245.95	270.3	288.3
Turb. Effy. = .86					
Δh_{ϵ_0}	114.7	179.5	212.0	233.4	249.0
$\Delta h_{\epsilon_{75}}$	113.3	178.6	211.5	232.7	248.0
Turb. Effy. = .88					
Δh_{ϵ_0}	117.2	183.7	217.0	238.9	254.6
$\Delta h_{\epsilon_{75}}$	116.0	182.8	216.2	238.0	253.8
Turb. Effy. = 0.86					
$h_{6\epsilon_0}$	448.7	383.9	351.4	330.0	314.4
$T_{6\epsilon_0}$	1796	1555	1433	1351	1290
$h_{6\epsilon_{75}}$	450.11	384.8	351.9	330.7	315.4
$T_{6\epsilon_{75}}$	1802	1559	1434	1353	1294
Turb. Effy. = 0.88					
$h_{6\epsilon_0}$	446.2	379.7	346.4	324.5	308.8
$T_{6\epsilon_0}$	1787	1540	1413	1329	1269
$h_{6\epsilon_{75}}$	447.4	380.6	347.2	325.4	309.6
$T_{6\epsilon_{75}}$	1792	1543	1417	1330	1272

In order to proceed with the calculation for fuel supply the temperature T_3 must be obtained via the regnerator.

6.3. REGENERATOR

$$\epsilon_{75} = 0.75 = \frac{T_6 - T_7}{T_6 - T_2}$$

$$T_7 = T_6 - 0.75(T_6 - T_2) .$$

For the cases where $P_2/P_1 \geqq 12.0$ it is found that T_6-T_2 is about zero or negative; in such cases a regenerator will only do harm to the output and S.F.C. due to the pressure drops it introduces. Thus only the cases for ratios of 3.0, 6.0, and 9.0 are calculated for regeneration.

P_2/P_1	3.0	6.0	9.0
T_7			
Comp. Effy. = 0.78 Turb. Effy. = 0.86	1085	1184	1259
Comp. Effy. = 0.83 Turb. Effy. = 0.86	1073	1163	1232
Comp. Effy. = 0.78 Turb. Effy. = 0.88	1082	1180	1253
Comp. Effy. = 0.83 Turb. Effy. = 0.88	1071	1159	1227

On the air side of the generator

$$\epsilon = 0.75 = \frac{T_3 - T_2}{T_6 - T_2} .$$

From this the value of T_3 for all cases can be found.

P_2/P_1	3.0	6.0	9.0
T_3			
Comp. Effy. = 0.78 Turb. Effy. = 0.86	1561	1434	1375
Comp. Effy. = 0.83 Turb. Effy. = 0.86	1558	1420	1366
Comp. Effy. = 0.78 Turb. Effy. = 0.88	1555	1421	1363
Comp. Effy. = 0.83 Turb. Effy. = 0.88	1550	1414	1354

6.4. COMBUSTION CHAMBER

With a combustion efficiency of 0.96 the enthalpy h_3 and h_4 are related by

$$h_3 + f(0.5T - 287 + \eta h_c) = (1 + f)h_4$$

where

f = fuel/lb of air

T = temperature of fuel assumed = 100°F

T_4 = 2210°R thus h_4 = 563.41 (tables)

η = combustion efficiency = 0.96

h_c = calorific value of fuel = 18500 Btus/lb

$$f = \frac{563.41 - h_3}{17179.6}$$

When ϵ_0 then $h_3 = h_2$

P_2/P_1	3.0	6.0	9.0	12.0	15.0
f					
When ϵ_0					
Comp. Effy. = 0.78	0.021	0.0179	0.0159	0.0142	0.0129
Comp. Effy. = 0.83	0.0212	0.0184	0.0164	0.01483	0.0136
When ϵ_{75}					
Comp. Effy. = 0.78, Turb. Effy. = 0.86	0.0104	0.01238	0.01323		
Comp. Effy. = 0.83, Turb. Effy. = 0.86	0.01041	0.01254	0.01336		
Comp. Effy. = 0.78, Turb. Effy. = 0.88	0.01049	0.01253	0.01341		
Comp. Effy. = 0.83, Turb. Effy. = 0.88	0.01053	0.01261	0.01353		

6.5. SUMMARY

Assuming that the accessories are driven by the compressor and that they amount to 3% of the work of compression, then input to the compressor is

P_2/P_1	3.0	6.0	9.0	12.0	15.0
η = 0.78	67.2	121.5	158.0	187.0	210.8
η = 0.83	63.2	114.1	148.8	175.9	198.0

The output of the turbine will be that for (1+f) lb of gas; assuming that the mechanical efficiency is 0.98, then output of the turbine is

P_2/P_1		3.0	6.0	9.0	12.0	15.0
ϵ = 0	η_T = 86	114.0	178.0	210.8	232.0	247.2
	η_T = 88	116.6	182.2	215.7	237.5	252.8
ϵ = 75	η_T = 86	112.9	177.2	210.2	231.2	246.2
	η_T = 88	115.3	181.2	215.0	236.3	251.8

The combined power plant will then give the net output shown in Table XVII.

TABLE XVII

NET TURBINE PERFORMANCE

	ϵ	Comp. Effy., η_c	Turb. Effy.	3.0	6.0	9.0	12.0	15.0
B.H.P./lb/sec	0	78	86	66.3	80.0	74.8	63.8	51.6
Fuel/hr				75.7	64.6	57.1	51.2	46.3
S.F.C.				1.14	0.807	0.763	0.802	0.897
B.H.P./lb/sec	0.75	78	86	64.7	78.8	73.9		
Fuel/hr				37.4	44.5	47.6		
S.F.C.				0.58	0.564	0.645		
B.H.P./lb/sec	0	78	88	70.0	86.0	81.6	71.5	59.5
Fuel/hr				75.7	64.6	57.1	51.2	46.3
S.F.C.				1.08	0.751	0.70	0.716	0.779
B.H.P./lb/sec	0.75	78	88	68.1	84.5	80.7		
Fuel/hr				37.8	45.1	48.3		
S.F.C.				0.556	0.534	0.599		
B.H.P./lb/sec	0	83	86	71.9	90.5	87.8	79.4	69.6
Fuel/hr				76.4	66.0	56.9	53.4	49.0
S.F.C.				1.062	0.73	0.672	0.673	0.718
B.H.P./lb/sec	0.75	83	86	70.3	89.3	86.9		
Fuel/hr				37.5	45.2	48.1		
S.F.C.				0.534	0.506	0.554		
B.H.P./lb/sec	0	83	88	75.5	96.4	94.6	87.2	77.5
Fuel/hr				76.4	66.0	58.95	53.4	49.0
S.F.C.				1.01	0.685	0.622	0.613	0.597
B.H.P./lb/sec	0.75	83	88	73.7	95.0	93.6		
Fuel/hr				37.9	45.4	48.8		
S.F.C.				0.514	0.478	0.521		

The results of these calculations are shown graphically in Fig. 29, where it is seen that the hp/lb of air peaks at about 6 to 8:1 pressure ratio in all cases, while the S.F.C. is a minimum for a regenerated engine at about 5:1 ratio and for an unregenerated engine at about 8:1 or better. It should also be observed that regeneration is little or no use at high-pressure ratios.

A most important phase of turbine application is the part load performance. The accurate analysis of this phase is quite complicated and involves almost the complete design of the power plant itself; the benefits to be derived from such a complete investigation seem somewhat doubtful at this stage of the problem. In order to obtain an approximate solution of the problem an investigation was made of existing results in this field, and it was found that by converting the published curves to a base of percent power the results appeared to group themselves in a reasonable manner. These results are believed to be accurate enough for the sort of preliminary systems analysis proposed here. The results are given in Fig. 30, which will be employed to predict part load performance when necessary.

One other well-known turbine characteristic is its torque multiplying ability when built as a free turbine unit capable of being fed with gas from the associated gas generator. In such cases a torque multiplication of from about 1.5 to 1.8:1 is possible without a complicated cycle. If multiple turbines, compressors, and combustors are used greater torque multiplication can be obtained. However, in such cases simplicity, small space, etc., are sacrificed so that its combination with a reciprocating engine is not too advantageous. It might better be the complete power plant.

6.6. CONCLUSIONS

The simple gas turbine cycle evaluated in this report using unit efficiencies that are of high order for the size of the machine contemplated (200-300 hp) shows that when operated near its optimum conditions:

1. The gain in performance and fuel economy for a reasonable change in unit efficiencies is not great.

2. A high compression simple turbine is not too far removed from a low pressure regenerated one for the purpose involved in this analysis.

3. The part load economy of such units is very poor unless a complicated cycle is involved.

4. At the present time it does not appear possible to achieve a considerable gain over the data given in this report. Losses, etc., have been kept to a minimum and compressor and turbine efficiencies at a maximum for the size and type of unit contemplated.

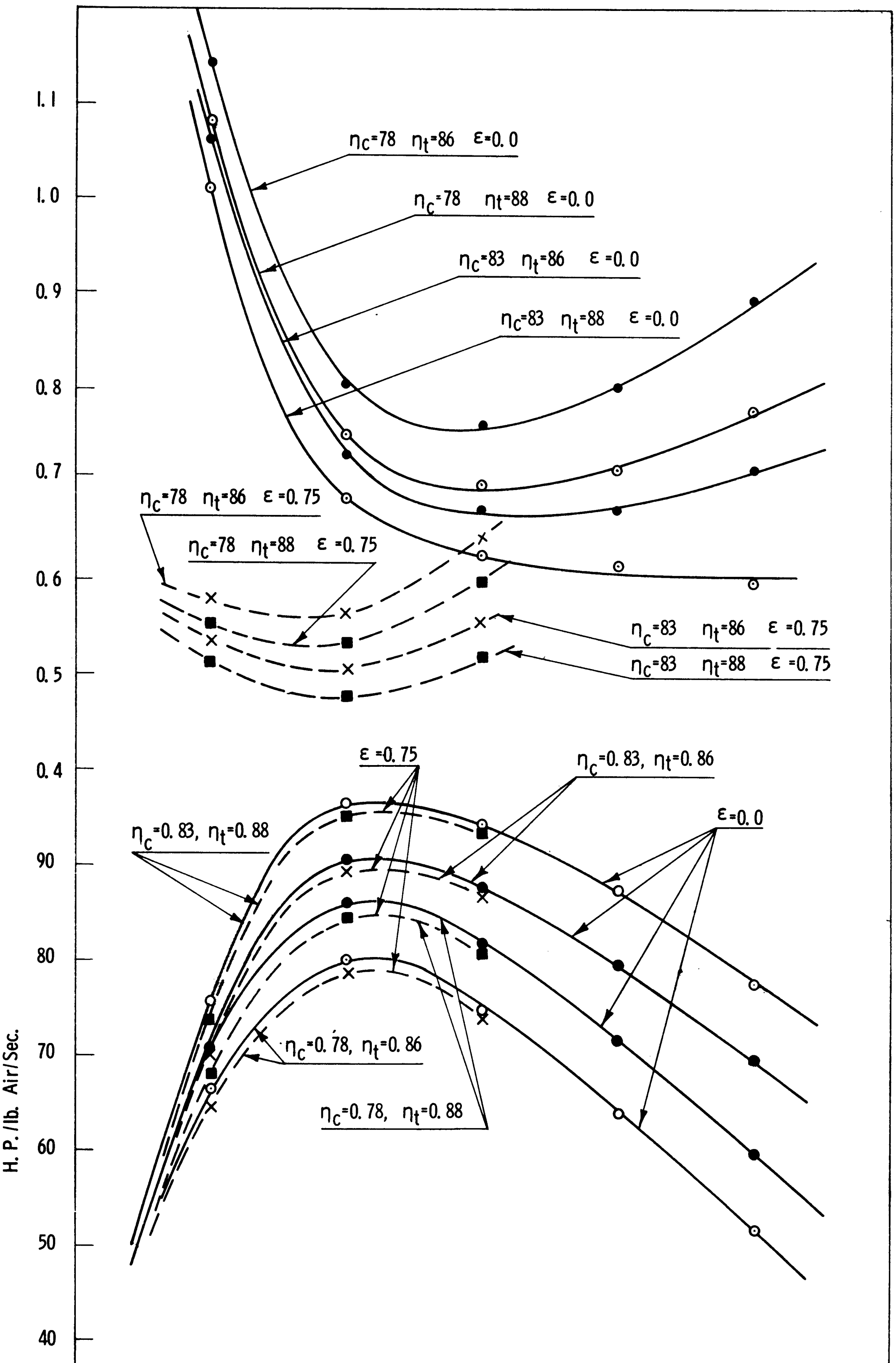

Fig. 29. Turbine performance.

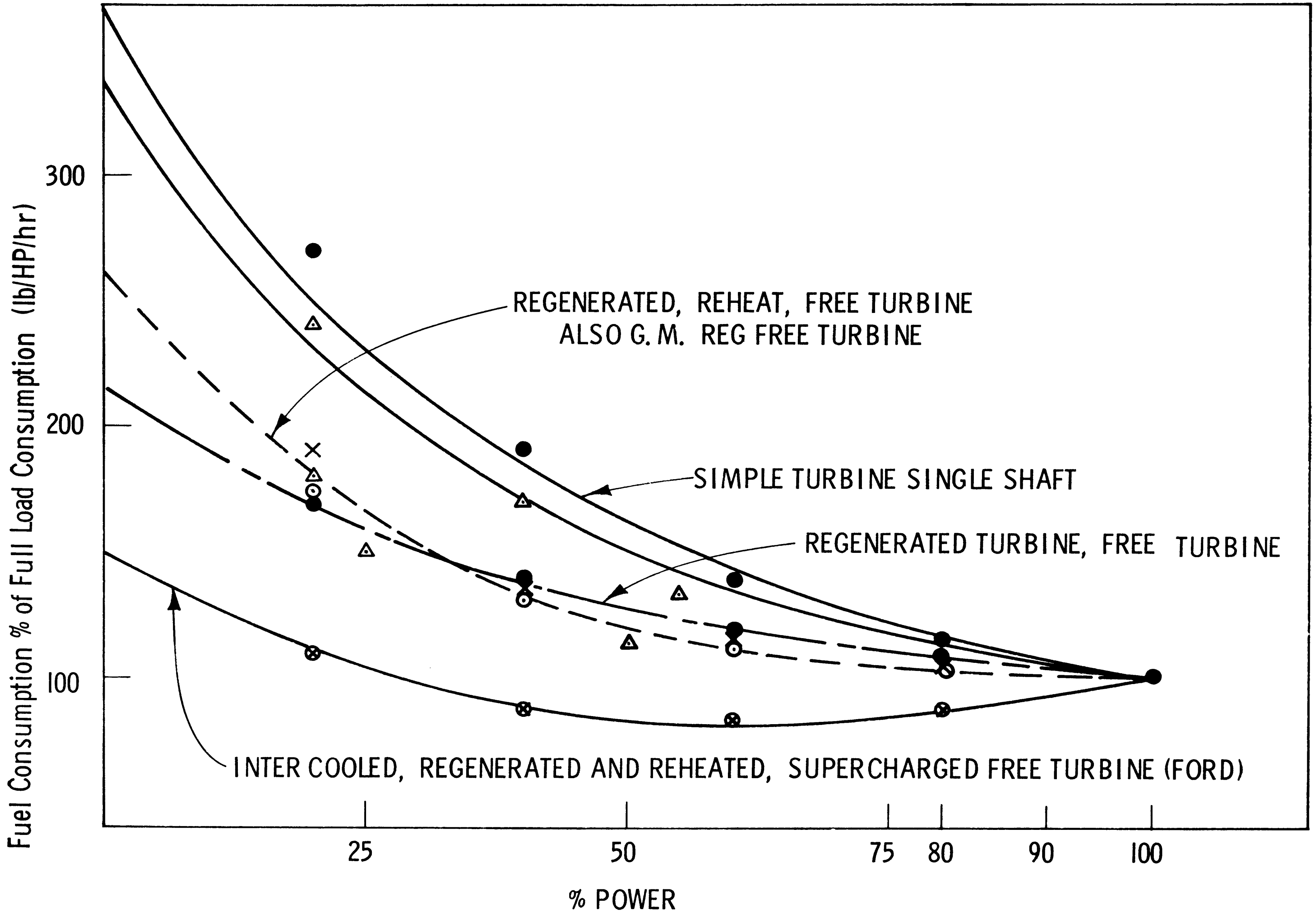

Fig. 30. Part load turbine performance.

SECTION 7

COMBINED COMPRESSION IGNITION ENGINE AND TURBINE POWER PLANT

The object of this section is to obtain some measure of the effect of a dual-engine power plant—a compression ignition engine associated with a gas turbine. A complete analysis of this problem would involve a great amount of work, but it is believed that the general advantages and possibilities of the system can be evaluated from the following material and a detailed approach then used, if necessary.

In order to approach this problem in a correct manner the distribution of power between the two types of units should be optimized. Since this would involve more work than seems justified in this report, an arbitrary selection of the power distribution will be made.

In addition to the power division, the size and weight of the combined unit will also be affected by the relative horsepower of the two units. It is fairly safe to assume that the greater the horsepower output of the turbine, the smaller and lighter will the resultant combined unit, but the greater the fuel consumed.

The turbine power plant will be most attractive in the large powered vehicles and least so in the small sizes, where high unit efficiencies are difficult to achieve.

The type of vehicle being considered has the important characteristic that the engine plus fuel for a given mission is more important than the engine alone.

Now any combination of these two units will need the development of a reciprocating engine as well as one or more turbines. Consider the case of solving the problem by the use of a compression ignition engine of a power suitable for the smallest of a family of vehicles which require three horsepower. Assume the total powers are 375, 500, and 750 hp. If the reciprocating engine was 275 hp coupled with a 100 hp turbine, then to employ the same turbine in all three vehicles, reciprocating engines of 275, 400, and 650 hp would be involved. Keeping in mind the size of a 650 hp compression ignition engine it seems that the addition of a 100 hp turbine would mean little if any advantage from an over-all point of view regarding performance, fuel economy, space weight, etc.

Examine this problem from the point of view of the variable turbine rather than that of the variable reciprocating engine. If the latter was kept at 275 hp, the three turbines would be of 100, 225, and 475 hp. The relative sizes of these three units would not vary greatly since the air handling ability and thus the power of the turbines can be changed radically with but small changes in over-all sizes.

Now suppose that the reciprocating unit developed 375 hp for the small vehicle (i.e., that no turbine is required for it), then two turbines of 125 and 375 hp would have to be coupled for the two larger units and only three power plants would be needed in place of four.

Another alternative would be to use a given reciprocating engine well blown for, say, 500 hp coupled with a 250 hp turbine for the 750 hp requirements; the engine would be derated by a smaller turbo-charger to 375 hp without a turbine for the small unit. In this case only two basic engines would be required, one reciprocating and one rotating; the largest horsepower requirements would be the only unit involving the dual set-up with its increased complication of gears, controls, etc.

The above comments indicate the great number of possibilities and the need for an extensive analysis if this subject is to be covered in detail. In order to obtain some kind of a picture let the following cases be examined.

7.1. RECIPROCATING ENGINE

This unit is to be designed as a 500 B.H.P. supercharged engine with high speed and mean pressure to keep bulk and weight to a minimum. It is proposed that the engine be of the 8-cylinder Vee type.

The 375 hp unit can be: (a) the same engine supercharged to a lower degree and thus of approximately the same bulk and weight as the 500 hp one; however there can be provided, by this means, some gain in fuel economy which may make the over-all bulk (engine plus fuel) an acceptable value; or (b) a 6-cylinder Vee or in-line engine with about the same rating as the 500 hp unit, with smaller bulk, etc., than in case (a).

7.2. TURBINE ENGINE

The turbine unit is to be designed to develop 250 hp at full power, to be geared to the engine transmission as necessary, and developed as a medium pressure ratio regenerated unit; secondly it is to be a simple open cycle engine but with high pressure ratio for efficiency.

7.3. 500 HORSEPOWER RECIPROCATING ENGINE

The specification proposed is as follows:

B.H.P.	500
I.H.P.	635
No. of cylinders	8 at 90°V
Maximum rpm	3000
I.M.E.P.	275 psi
Friction M.E.P.	37 psi
Fan Equiv. M.E.P.	21 psi
B.M.E.P.	217 psi
Cylinder bore	4-1/2 in.
Cylinder stroke	4-3/4 in.

The above represent about maximum ratings for present-day engine predictions for the near future. In order to operate with high fuel economy under the above conditions an inlet manifold pressure of 2.8:1 is considered to be required for the turbo-charger when:

Inlet manifold pressure = 84 in. Hg

F/A ratio = 0.0375

S.F.C. at full load = 0.33 lb/I.H.P./hr

= 0.418 lb/B.H.P./hr

Using the data supplied by OTAC regarding the rolling resistance of the 43-ton vehicle, then at 30 mph:

Rolling resistance = 70 lb/ton

= 3010 lb

Work done at 30 mph = 3010 x 5280 x 30 ft lb/hr

$$= \frac{3010 \times 5280 \times 30}{60 \times 33000} \text{ hp}$$

= 241 hp.

The above horsepower is that actually performed against the rolling resistance. The engine B.H.P. will be obtained via the use of a transmission efficiency which will include slip losses. Assume that this over-all efficiency at the above condition is 80%; then

$$\text{Transmission input hp} = \text{B.H.P. of engine}$$

$$= 241/0.80$$

$$= 300 \text{ B.H.P.}$$

Estimated cylinder air charge at 500 hp and 3000 rpm

$$= \frac{\pi \times 4.5^2}{4} \times 4.75 \times \frac{8 \times 3000}{2 \times 1728}$$

$$= 525 \text{ cu ft/min}$$

$$= 31600 \text{ cu ft/hr}$$

Allow a 5% through flow; then

$$\text{Compressor air flow} = 33200 \text{ cu ft/hr}$$

At 3000 rpm and 300 hp the B.M.E.P. becomes 130 and this I.M.E.P. = B.M.E.P. + friction + fan = 188 psi.

$$\text{\% of engine I.H.P.} = \frac{188}{275} = 68.4\% \text{ of } 3000 \text{ rpm.}$$

Using Fig. 6 for the F/A ratio under these conditions, then at 300 B.H.P.,

$$\text{Expected F/A} = 0.305$$

Manifold density at full load and speed

$$= \frac{P}{RT} = \frac{144 \times 41.15}{53.34 \times 660}$$

$$= 0.1685 \text{ lb/cu ft}$$

$$\text{Total air flow} = 0.1685 \times 33200$$

$$= 5600 \text{ lb/hr}$$

Air retained = 5330 lb/hr

Fuel flow = 0.0375 x 5330

= 200 lb/hr

S.F.C. = $\frac{200}{500}$ = 0.40 lb/B.H.P./hr

In the case of the 30 mph condition where the speed is 100% and the load 68.4%, Fig. 6 indicates that the manifold pressure will be 77%. Thus the air flow becomes

Air charge at 100% speed and 68.4% load = 5330 x 0.77

= 4100 lb/hr

Fuel flow = 0.305 x 4100

= 125.0 lb/hr

S.F.C. at 300 hp = 125.0/300

= 0.417 lb/B.H.P./hr

Examining the over-all performance on the basis of the rolling resistance given for the battlefield day, the following tables can be calculated for the various conditions specified and the assumed transmission efficiencies.

Vehicle Speed, mph	Rolling Resistance Off-Road, lb	Ground hp	Trans. Effy., %	B.H.P. of Engine	Minimum rpm% for B.H.P.	% of B.H.P., max
15	4860	195	75.0	260	70.0	52.0
16	4660	200	76.0	263	72.5	52.6
17	4590	209	77.0	272	75.0	54.4
18	4520	217	78.0	278	80.0	55.6
19	4520	229	79.0	290	80.0	58.0
20	4520	242	80.0	302	80.0	60.5

Vehicle Speed, mph	Rolling Resistance Off-Road, lb	Ground hp	Trans. Effy., %	B.H.P. of Engine	Minimum rpm% for B.H.P.	% of B.H.P., max
2	5760	30.7	70.0	43.8	40.0	8.8
3	5800	46.5	71.0	65.5	40.0	13.1
4	5850	62.5	72.0	86.8	40.0	17.4
5	5850	78.0	73.0	107.0	40.0	21.4
6	5850	93.8	74.0	127.0	50.0	25.4
7	5880	110.0	75.0	147.0	50.0	29.4
8	5930	127.0	76.0	167.0	60.0	33.4
9	6020	144.8	77.0	188.0	60.0	37.6
10	6320	169.0	78.0	217.0	70.0	43.4

In the above table the minimum rpm% for the B.H.P. represents the minimum speed at which the horsepower involved can be carried on the basis that such operation will be most economical at about 75 to 85% of the ratio I.M.E.P.

If possible, the maximum usable F/A ratio should be limited to 0.04 in order to keep a clean exhaust. The I.H.P. being developed can be obtained by the use of Fig. 6 and the S.F.C., etc., determined. The rpm for the first three speeds, 15, 16, and 17 mph were rounded out to 75% for convenience.

Table XVIII is constructed from Fig. 5 obtaining the value of P_m% and I.M.E.P.% via the % I.H.P. scale. Then since T_m, the manifold temperature, is a constant,

$$\% \text{ air flow} = \% P_m \times \% \text{ rpm}$$

$$\text{air flow} = \text{flow at full speed and power} \times \% \text{ air flow}$$

$$= 5330 \times \% \text{ air flow.}$$

The F/A ratio is then obtained from Fig. 6 for the % power and speed involved and the remainder of the table can be completed, the fuel flow being calculated from:

$$\text{Fuel flow} = \text{air charge} \times F/A \text{ lb/hr.}$$

The engine performance over the battle field day is thus available. The performance figures for 20% of the day at 15 to 20 mph and 1.57 times the rolling resistance on first-class roads makeup the first section of Table XVIII and this averages a fuel rate of 94 lb/hr; the second group for 2 to

TABLE XVIII

ESTIMATED ENGINE PERFORMANCE

rpm (%)	Load B.H.P. (%)	Losses Fan and Friction, hp	I.H.P. hp	I.H.P. (%)	Cylinder Air Charge lb/hr	Cylinder Air Charge (%)	F/A Ratio	P_m (%)	Fuel Flow, lb/hr	S.F.C. lb per I.H.P./hr	I.M.E.P. psi	I.M.E.P. (%)
75	50.4	75	335	52.8	2420	45.4	.0355	65	86.0	.256	192	70
75	52.8	75	338	53.3	2530	47.5	.0365	66	92.5	.273	198	72
75	55.6	75	347	54.8	2610	48.9	.037	67	97.7	.281	201	73
80	58.2	84	362	57.0	2610	48.9	.034	68	88.8	.245	198	72
80	60.6	84	374	59.0	2760	51.8	.035	70	96.7	.259	204	74
80	63.0	84	386	60.9	2880	54.0	.0355	71	102.2	.265	209	76
40	8.8	26	69.8	11.0	538	10.1	.03	36	16.2	.232	77	28
40	13.1	26	91.5	14.4	730	13.7	.035	38	25.6	.280	99	36
40	17.4	26	112.8	17.8	960	18.0	.040	41	38.4	.34	121	44
40	21.4	26	133	21.0	1210	22.8	.0475	43	57.5	.432	146	53
50	25.4	37	164	25.8	1300	24.4	.039	47	50.8	.31	143	52
50	29.4	37	184	29.0	1510	28.4	.0425	49	64.2	.348	160	58
60	33.4	50	217	34.2	1580	29.6	.0375	52	59.2	.273	157	57
60	37.6	50	238	37.5	1820	34.1	.040	55	72.8	.306	171	62
70	43.4	65	282	44.2	1980	37.1	.355	59	70.4	.25	174	63

10 mph averages at 50.6 lb/hr. Using Fig. 18a for an approximate idling fuel flow, 4.0 lb/hr is obtained, and the fuel requirements for a battlefield day become

Condition	Hours	Fuel/hr	Total lb
20% at 15 to 20 mph	4.8	94.0	450.2
40% at 2 to 10 mph	9.6	50.6	486.0
40% idle	9.6	4.0	38.4
	Grand total, lb/24 hr		974.6

Average hp for 20% time = 357 I.H.P.

Average hp for 40% time = 165.8 I.H.P.

The columns of I.M.E.P. values in the above tables show that the highest percentage is 76%, corresponding to an I.M.E.P. of 209 psi or a B.M.E.P. of 169 psi at 80% speed. That is, the assumed engine under the battlefield-day conditions is always operating at 78% or less of its rated B.M.E.P.; the 78% is about the most economical point in most engines, thus conditions appear good from this analysis. It also points up that the reciprocating engine proposed for the engine-turbine combination will perform the battlefield-day requirements without the use of the turbine. This leaves the latter power plant for emergency requirements such as steep hill climbing, obstacle crossing, etc.

The evaluation of how such use will affect the 24-hr fuel requirements would of course depend upon the terrain being crossed. Assume for a hypothetical case that the group for 2 to 10 mph is changed to 35% of the time in place of 40% and that the 5% balance is for emergency operation, with an average load of 125 hp on the turbine and 95% I.M.E.P. at 40% speed on the reciprocating engine; then the following assumptions can be made.

Simple Turbine Assumptions

Total hp required	250
Air flow	3.0 lb/sec
Pressure ratio	8:1
Comp. Effy.	78%

Turb. Effy.	88%
S.F.C. at full load	0.70 lb/hp/hr
Gas temperature	1750°F
Expected S.F.C. at 50% load (Fig. 30)	1.6 x 0.7 = 1.32 lb/hp/hr
Fuel flow at 125 hp	165 lb/hr
Fuel flow for 1.2 hr	198 lb

Regenerated Turbine Assumptions

Total hp required	250
Pressure ratio	5:1
Air flow	3.06 lb/sec
Regenerator effectiveness	0.75
Comp. Effy.	78%
Turb. Effy.	88%
S.F.C. at full load	0.52 lb/hp/hr
Gas temperature	1750°F
Expected S.F.C. at 50% load (Fig. 30)	1.20 x 0.52 = 0.624 lb/hp/hr
Fuel flow at 125 hp	77.9 lb/hr
Fuel flow for 1.2 hr	93.5 lb.

7.4. SPACE AND WEIGHT REQUIREMENTS; SIMPLE TURBINE

7.4.1. Turbine Volume

Using Eq. (21) for the engine volume:

Engine volume/hp $= 8.27/hp^{0.92}$

$= 8.27/250^{0.92}$

$= 0.0518$ cu ft/hp

Volume of turbine $= \underline{\underline{12.9 \text{ cu ft}}}$

Assume simple turbine of 250 hp = 12.0 cu ft.

7.4.2. Turbine Weight

Use Eqs. (23) and (24):

Turbine weight $= 6.58\ hp^{0.2}$ lb/cu ft

$= 6.58 \times 250^{0.2}$

$= 6.58 \times 3.02$

$= 20.7$ lb/cu ft

Turbine weight $= 15.2 \times 20.7$

$= \underline{\underline{314 \text{ lb}}}$

Turbine weight $= \dfrac{35.05}{hp^{0.644}}$ lb/hp

$= 35.05/35.0$

$= \underline{\underline{1.0 \text{ lb/hp}}}$

Turbine weight $= \underline{\underline{250 \text{ lb}}}$.

Averaging these estimates,

Turbine weight $= \underline{\underline{282 \text{ lb.}}}$

7.5. SPACE AND WEIGHT REQUIREMENTS; REGENERATED TURBINE

7.5.1. Turbine Volume

Using Eq. (22):

$$\text{Volume/hp} = 79.8/\text{hp}^{1.27}$$

$$= 79.8/1100$$

$$= 0.0726 \text{ cu ft/hp}$$

$$\text{Engine volume} = \underline{\underline{18.2 \text{ cu ft}}} \ .$$

This volume is on the basis of a regenerator built into the unit as in the Chrysler and G.M. designs; with an external regenerator the volume would increase greatly.

7.5.2. Turbine Weight

Using Eq. (25):

$$\text{Weight} = 11.8/\text{hp}^{0.71}$$

$$= 111.8/50$$

$$= \underline{\underline{2.24 \text{ lb/hp}}}$$

$$\text{Estimated turbine weight} = 560 \text{ lb.}$$

This estimated weight seems somewhat excessive. As was pointed out in Section 1, there is considerable scatter in the turbine data currently available, and this scatter is most predominant in range in which this engine occurs. To be a little optimistic, assume that the weight can be improved from the estimated weight given above to:

$$\text{Assumed turbine weight} = 400 \text{ lb.}$$

7.6. DUAL-POWERED UNIT

Combine the 500 hp compression ignition engine with the 250 hp turbine in the manner outlined above; then compare this combined performance with that of the 500 reciprocating engine alone, operating on battlefield-day condictions. The following data results, using Fig. 18 for the oil engine data.

DUAL POWER PLANT

500 hp Oil Engine and 250 hp Turbine

Condition	Power Plant		Fuel Flow/24 hr,
	Volume, cu ft	Weight, lb	lb
Battlefield day 500 hp oil engine only	40.0	2500	974.6
Battlefield day as re-arranged for emergency power, oil engine plus simple turbine	52.0	2782	1172.6
Battlefield day as re-arranged for emergency power, oil engine plus regenerated turbine	58.2	2900	1068.1
Battlefield day 750 hp oil engine, 12 cylinder only	55.0	3750	985.0

There are two comparisons provided above, viz., a 500 hp and 750 hp compression ignition oil engine, which can be compared with the 500 hp oil engine plus either a simple turbine of 250 hp or a 250 hp regenerated turbine.

The 500 hp engine alone will perform the task of the battlefield day but will not give the obstacle crossing or hill climbing ability of the 750 engine or combination.

The above example does not bring out the responsiveness of this system but it is believed that the comparison will enable investigators to determine if such a system is worth pursuing further.

7.7. CONCLUSIONS

The compression ignition engine plus turbine combination presents the following conclusions for the cases examined:

1. For a given total horsepower the combined unit of 500 hp oil engine plus 250 hp simple turbine permits a reduction in space and weight of the unit of about 6% and 25% respectively at the expense of a 20% increase in fuel consumption for the assumed conditions.

2. A more exact determination can be made, given more information regarding the required time for the use of the turbine, such as hill climbing, obstacle crossing, etc.

3. Normal operation off-the-road for the B.F.D. as supplied by OTAC does not need more than the 500 hp for such requirements and the assumed engine provides all necessary power.

SECTION 8

GENERAL CONCLUSIONS

A number of methods have been investigated by which various degrees of responsive engine operation can be secured; examination of the general overall results permit the following observations and conclusions.

1. The maximum degree of responsiveness examined and believed to be practicable at present was 3:1, using a turbo-charged engine fitted with means to maintain air flow as engine speed was reduced.

2. Such an engine would need to be developed, both engine and turbo, since it would involve mean pressures far in excess of modern practice accompanied with roughly twice the present-day firing pressures.

3. The fuel economy of such an engine shows no marked improvement over that of existing engines, and is in fact somewhat inferior under certain load conditions.

4. The need for a transmission would still exist unless the range of responsiveness could be increased from 3 to at least ≧ 12:1. This would involve prohibitive mean and maximum pressures unless a wide variation in compression ratio during operation could be achieved successfully.

5. To produce such an engine there would be the need for an aftercooler behind the compressor, adding to the weight and particularly to the over-all volume of the power package. Without such a cooler the predicted power outputs could not be achieved.

6. All other methods of producing responsiveness resulted in a degree less than 3:1 varying from a small value to as much as 2:1.

7. All methods showing any reasonable magnitude of responsiveness except one resulted in increased pressures, etc., throughout the cycle.

8. The Air-Boosted cycle indicated that a responsiveness of 2:1 could be practicable, without any serious additional strain on the engine parts, at some small increase in fuel flow during the period of responsive operation only.

9. Compounding under certain conditions gave both power increase and improved fuel flow simultaneously, though the responsive range was small.

10. In order to exactly evaluate the results of this analysis, additional rules for engine operation and requirements should be made available, such as:

(a) The exact horsepower range in which future engine design will be concentrated and vehicle weight and rolling resistance likely to be encountered.

(b) The time period during a battlefield day in which the responsive condition will be needed.

(c) The type of hill climbing and obstacle crossing considered typical.

11. If the responsive engine does meet a need, then an investigation into the fuel injector requirements for such operation must be made. Presently used equipment could not meet this wide range of conditions.

12. Research would also be needed in the fields of (1) increased manifold pressures, (2) the turbos to produce such pressures, and (3) a single-cylinder research engine to digest such pressures.

13. Responsiveness will add to the need of producing a variable compression ratio engine.

14. An immediate problem would be the analysis of a complete transmission to match the proposed responsive engine selected. This involves relating the engine and transmission phases of this contract to a concrete problem in order to produce a combined unit.

15. An investigation of an engine with a small degree of responsiveness coupled to a conventional step speed transmission should be made. It is possible that almost all the advantages of a highly responsive engine could be secured by this means without solving the difficult problem of very high mean effective pressures.

16. The combination of a simple turbine and reciprocating engine has some attractive features, provided fuel quantity per day is not an important item. The complexity of the controls may also limit its usefulness.

17. The time required for turbine warm-up could be quite important, if the turbine is used for the emergency powers only. If the unit were idled continuously to keep it hot, the amount of fuel supply consumed per 24 hr would be increased enormously.

18. The use of a regenerated turbine in combination with an oil engine does not seem too attractive.

19. A regenerated gas turbine with a high effectiveness of regeneration might prove competitive if the bulk could be kept down and a reliable regenerator which would stand up under the service conditions encountered could be produced.

20. Time did not permit much consideration of a flexible engine arrangement. It appears at first glance that this arrangement would not involve the serious problems of a responsive engine. It would transfer the problems from the engine to the transmission.

21. If fuel economy is of great concern, greater improvements in the use of the fuel appear possible by research on the cycle to improve the combustion phase (resulting in reduced fuel flow for a given power, or increased power for a given fuel flow) than can be made by the use of responsiveness.

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