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Technical Report

THE TURBO-SUPERCHARGED SPARK IGNITION ENGINE WITH VARIABLE COMPRESSION RATIO

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ABSTRACT

This report discusses the use of a turbocharger, rather than a throttle, to vary the power output of a spark-ignition engine using a variable-compression-ratio piston. Performance data are established for use in comparing the L-141 engine having 141 cu in. displacement (used in the M-151 vehicle) with a supercharged version having 80.6 cu in. and developing the same maximum net power, 61 BHP at 3600 rpm. Systems with and without aftercooling are considered.

Three methods of limiting knock are used. The results show fairly close agreement; hence they can be considered accurate enough for this preliminary examination.

The results indicate that fuel consumption at vehicle speeds of 35 mph or less can be reduced, at the expense of a small increase at higher speeds. As with all other turbocharged engines, there will be a lag between throttle operation and the speed-up of the turbine.

The problem of matching the turbocharger and the engine has not been investigated.

I. OBJECT

The object of the study reported here was to predict the performance characteristics of a turbocharged spark-ignition engine fitted with a piston having an automatically variable compression ratio. The power output is varied by changes in the degree of supercharge rather than by a throttle, so that pumping losses and engine size are reduced and economy is increased. The results obtained are applied to the L-141 engine, used in the M-151 vehicle, for comparative purposes.

The problem was proposed to the writers by Mr. Floyd Lux of the Power Plant Laboratory at the Army Tank Automotive Command.

II. INTRODUCTION

Most spark-ignition engines are controlled by a throttle valve which regulates the mass of mixture to suit the load on the engine. The procedure with which the present investigation is concerned is to vary the air mass flow over as wide a range as possible by using a variable-speed turbocharger to change the inlet manifold pressure, resorting to throttle operation only when the required manifold pressure falls below atmospheric pressure.

The output of the spark-ignition engine is limited primarily by the detonation characteristics of the fuel; any degree of supercharging of a normally designed engine demands that the usable compression ratio be reduced to limit detonation. It follows that any engine fitted with supercharging must lose some fuel economy as increased supercharge is applied, but gains in power output; these conditions are well known. As a result of the reduced compression ratio, light-load fuel performance is also adversely affected. Hence supercharging of normal engines is resorted to only when power output is the most important consideration, as with racing engines and aircraft.

The development of the Continental Aviation and Engineering Corporation's variable-compression-ratio (VCR) piston has created the possibility of modifying the operation of an engine according to the load of the vehicle. For a heavy load, the needed large power output would be provided by operating the engine with supercharging and low compression ratio; for light load, fuel economy would be increased by operating at a compression ratio higher than normal. Fuel consumption would then depend on the operating schedule. If the engine had long periods of idling and low-power operation, and only short periods of full-power operation, it would use less fuel than a standard one.

The range of ratios might be roughly as follows:

	Compression Ratio	
	Standard engine	Supercharged engine
Idle	7.5:1	15:1
Light load	7.5:1	15 to 10:1
Light to medium load	7.5:1	10 to 7.0
Medium to heavy load	7.5:1	7.0 to 4.0

With such a wide operating range, the engine should be adjusted to be at its optimum point for all loads. This is usually the condition in which the engine is at the point of detonation at all times.

For strictly comparative purposes, this proposed power plant for the M-151 vehicle should have a maximum output comparable with that of the standard engine, 61 BHP net at 3600 rpm. This output could be secured by using as high a supercharge ratio and compression ratio as possible without detonation; from these two ratios the mean effective pressure (MEP) for each cycle can be determined, and then the engine displacement can be calculated. The displacement will of course be much less than that of the present engine; therefore losses, etc., will also be less. Thus the increases in specific fuel consumption (SFC) resulting from the reduced compression ratio at heavy loads, when supercharging occurs, will be offset to some extent.

As the load is reduced, a condition will eventually be reached at which no supercharging is necessary. The engine will then behave as a naturally aspirated one, but will use a high compression ratio as a result of the VCR piston. With further reduction in load, a throttle will have to be provided to reduce manifold pressure still further, and thus air flow. Pumping will then cause some power loss, but much less than in a conventional engine throttled over the whole power range. This reduction in losses due to pumping will also result in some improvement in light-load operation, by reducing the fuel flow requirements. At the same time the piston will adjust the compression ratio at a considerably higher ratio than in the standard engine, and cycle efficiency will be higher.

The sum of these gains and losses could result in improved engine operating conditions under the most common Army operating conditions, the loss at heavy load being more than offset by the gain at light load.

If the scheme fulfills expectations, it will have the following advantages and disadvantages:

1. The engine will be smaller and lighter.
2. The range of operation for a given fuel supply will be greater.
3. Little maximum vehicle performance will be lost.
4. Acceleration will be slightly reduced, because of the longer time required to speed up the charger.
5. The manifolding of inlet and exhaust pipes will be some what complicated.
6. The reduced size and bulk will reduce basic engine costs.
7. The application of the turbocharger will increase the basic cost slightly.
8. A carburetor to handle the combined manifold conditions will need to be developed.
9. The material and structure of the exhaust valves might need some improvement.
10. The structure of the engine might need some strengthening if high peak

pressures are encountered. The calculations to be carried out will determine whether this change is necessary.

11. Since the maximum output will remain unchanged, the present cooling system will probably be adequate for such changes in heat rejection as will occur.
12. The use of a variable compression ratio, with its accompanying lubrication problems, etc., will cause some complications.

III. METHOD OF CALCULATION

The most important part of this problem is to determine a method of predicting the characteristics of an engine which will approach the conditions of detonation at all times. Such a prediction will make it possible to compare the turbocharged engine with the standard one; and, if the turbocharged engine shows enough promise to warrant further research, the prediction will provide a basis for that research.

The problem of detonation is quite complicated, and most methods of studying it are not generally applicable to all sizes and shapes of engines. The most favored one is that based on the fuel octane requirements of an engine; the engine is adjusted to the desired fuel, usually on the basis of the results of bench and road tests. In the case to be examined it seems that the engine's performance must be predicted on the assumption that the standard 83-octane military fuel, MIL-G-3056A, will be used. Therefore relative compression ratios must be determined for a series of manifold pressure ratios at each of which it is possible to approach detonation with equal closeness at all times, by automatic adjustment of the variable-compression-ratio (VCR) piston. In an actual engine test this is readily done by advancing the spark, adjusting the F/A ratio, etc., as appropriate for any given load and speed, since the VCR piston will respond to a near-detonating condition by adjusting the ratio downward until detonation is suppressed. Without such engine tests the conditions must be determined by calculation; it is here that more information is needed.

It is proposed to follow two broad methods, as follows:

1. Take as the limiting condition the state of the end-gas portion of charge just before detonation (Ref. 1).

2. Take as the limiting condition the ratio of air density to compression temperature (Ref. 2).

METHOD 1

The first method involves the assumption that the combustion of a few molecules of gas occurs slowly enough that, during normal burning,pressure is uniform throughout the combustion chamber. As a result, the molecules combining at any instant, being an extremely small fraction of the total in the cylinder, burn at almost constant pressure. As they expand, they compress the unburned mixture ahead of the flame front, increasing its pressure and temperature until it reaches the detonation point. Then all the rest of the mass undergoes instantaneous combustion, or else the flame front passes through it completely before detonating conditions are reached, giving normal engine combustion. If detonating conditions are reached before combustion is complete, the amount of charge that detonates burns at constant volume, since under detonating conditions the rate of combustion is many times faster than normal flame propagation and pressure equilibrium throughout the cylinder is not established. The result is an instantaneous increase in the pressure in the pocket of charge, which therefore detonates, producing a shock wave which travels across the chamber and is responsible for knock.

This process can be represented by a specific volume-pressure relationship as shown in Fig. 1,

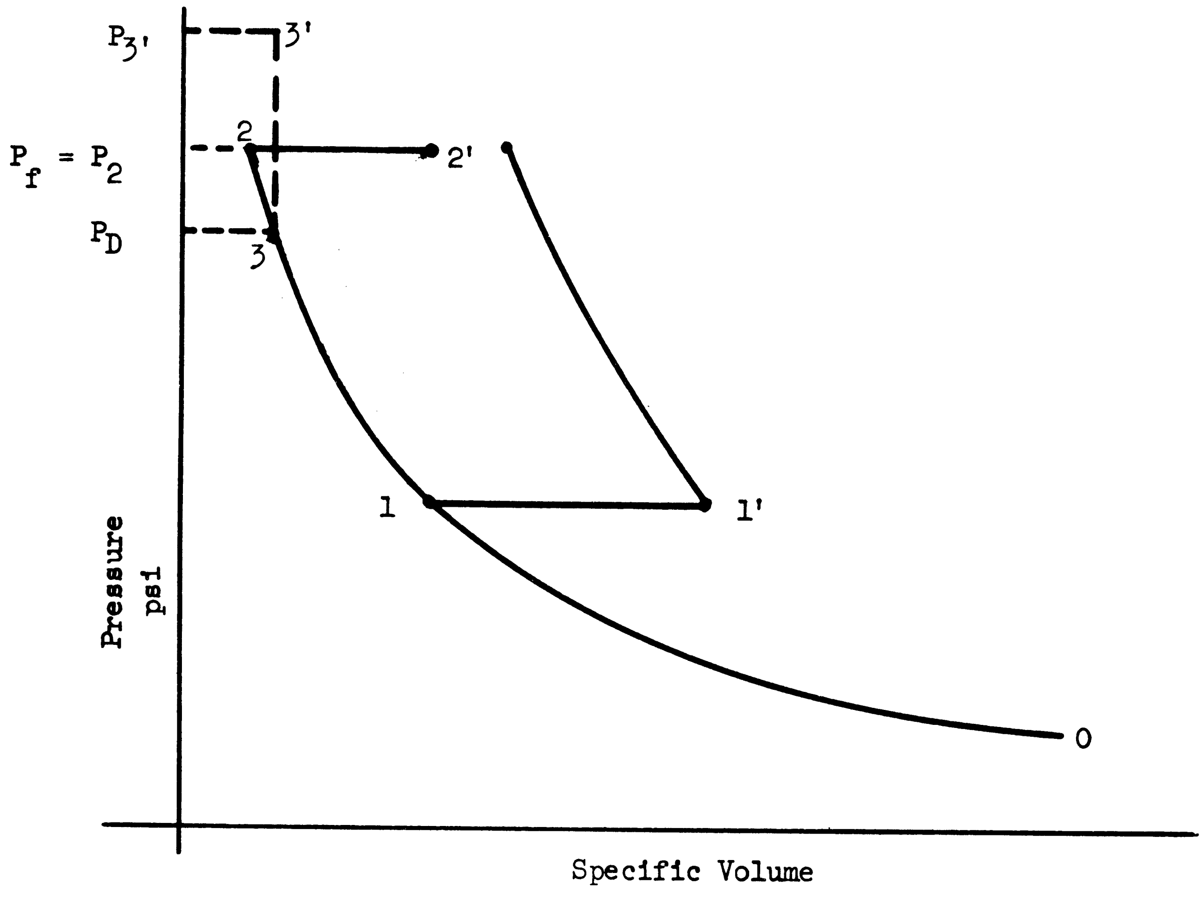


Fig. 1. Specific volume vs. pressure.

where 0 represents the specific volume of the charge at the beginning of compression, and 0-1 represents the compression process reaching the compression pressure P_c at 1, the TDC of the engine. The first few molecules that combine, releasing heat, do so at approximately constant pressure, the specific volume of the element changing along the line 1-1'. In the gradual combustion process, as heat is released and pressure is increased, the unburned gas is compressed ahead of the flame front. As a result, the unburned mixture continues to undergo the compression process along the line 1-2, reaching the maximum firing pressure P_f at 2; at this point, if combustion is normal, the last elements of the charge burn along the line 2-2'. But if detonation occurs when the pressure has reached P_D at 3; the whole remaining portion will ignite so rapidly that its volume will remain constant volume along the line 3-3', reaching a higher pressure $P_{3'}$. Detonation will therefore occur at some point such as 3, depending upon the state of the unburned mixture. It has been established (Ref. 1) that this process does not violate the usually accepted principles of thermodynamics, cycle efficiencies, mean pressures, etc.; combustion is considered to occur at TDC with the piston stationary, but even this limitation is not a necessary condition.

The problem under consideration can be solved by assuming that a small amount of charge, say 5%, reaches the state of detonation during the standard ideal cycle of the engine. Hottel's Charts can be used, within the range of their applicability, to determine the effects of various mixture ratios, chemical equilibrium, etc. If this last 5% is brought to the same state, or subjected to the necessary combination of factors, for each of the various cycles to be investigated, it should be possible to compare the cycles and thus determine the detonation point for each.

The calculations herein are based on two assumptions: (a) that detonation always occurs at the same density, and (b) that there is a relationship between the density of end gas and the temperature at which detonation will occur (see Ref. 2); calculations based on this assumption are considered below at Method 2b. On the basis of these assumptions, cycles having various supercharging ratios were studied to determine, for each one, the compression ratio at which the last 5% of the mixture would detonate. From the results the IMEP, ISFC, etc., were then obtained.

METHOD 2

In Calculations by the second method the work of Ref. 2 was applied directly, by constructing diagrams showing the reciprocal of the knock-limited manifold air pressure versus compression ratio for 83-octane fuel, as well as a plot of knock-limited compression density versus calculated compression temperature. In this application, Siegel considers that the pressure and temperature at the end of the compression stroke, rather than at the end of combustion, will define the detonation limits and determine the limiting performance of the engine. Since the actual portion of the charge that detonates is the last portion to burn, there is considerable doubt about whether this theory applies to all engines. Siegel, however, reports results obtained from his test unit which more than suggest that it does. We have therefore applied his theory in calculating engine performance, as an alternative procedure to show the combined effects of compression temperature and density of the charge on the limiting compression ratio.

Three slightly different approaches, then, have been employed to determine the possible operating compression ratio as the degree of supercharging changes. They are based, respectively, on the following criteria:

1. The last 5% of the end gas has a constant density at the point of detonation.

2. The end-gas density and temperature have a certain relationship.

3. The compression gas density and temperature have a certain relationship.

It is hoped that by this means the actual engine conditions have been bracketed closely enough to give a reliable picture of the process being investigated.

IV. PROCEDURE

METHOD 1

The end-gas method of calculation was applied to the L-141 engine in the following manner. First, the ideal standard engine cycle of the L-141 engine was examined on Hottel's rich mixture charts for compression ratios of 7.5:1 (standard) and 10:1. The latter ratio was predicted as a possible upper limiting condition for the highly supercharged engine, since that engine's displacement will be considerably less than that of the standard L-141. A redesigned head would be needed to obtain the maximum possible ratio. It is of considerable importance that the unsupercharged ratio be as high as possible; otherwise intolerable conditions will arise when the engine is supercharged at the reduced ratios resulting from use of the VCR piston.

When these ideal cycles were known, the pressure, temperature, density, etc., of the last 5% of the end gas in the cylinder were determined and used as the factors limiting the compression ratio for a turbocharged engine.

In the second step, the compression ratio that produces the same end-gas conditions as in the first step was determined for supercharge ratios of 1.5, 2.0, 2.5 and 3.0:1, to be supplied by the turbocharged with and without aftercooling.

The engine with aftercooler was taken to have an effectiveness E of 70%. The necessary solutions for the compression ratio were obtained by trial and error; information from the combustion charts was used in order to include chemical equilibrium, etc., in the solution.

Where the method of compression gas density-temperature relationship was used, it was found more convenient to employ the method described in Ref. 2.

In all cases the cycle was analyzed for the output per pound of air supplied to the engine. The IMEP, thermal efficiency, IHP/lb of air/sec, specific fuel consumption in lb/IHP/hr, etc., were obtained. In tests of an actual engine these results would have to be adjusted for the effects of volumetric efficiency, frictional losses, pressure losses between atmosphere and engine manifold, etc.

Since the pressure drop between the atmosphere and the manifold, and the temperature change caused by fuel addition, hot-spot heating, etc., are fairly stnadardized, the values referring to the inlet air were modified to give the analysis a closer approach to reality. The losses were estimated as follows:

Air cleaner and carburetor pressure drop at full speed (3600 rpm)	=	2.0 psi
Air cleaner and carburetor pressure drop at 800 rpm	=	0.75 psi
Temperature drop due to fuel addition	=	-40°F
Temperature increase due to hot spot	=	+80°F

Calculations using the above values result in a manifold pressure and temperature of 12.7 psi and 580° abs for an ambient condition of 14.7 psi and 80°F at full speed when naturally aspirated, and 19.95 and 580° abs at low speed. At intermediate speeds the manifold pressure for the naturally aspirated engine was taken as proportional to the speed change; it is represented by the conditions in Fig. 2.

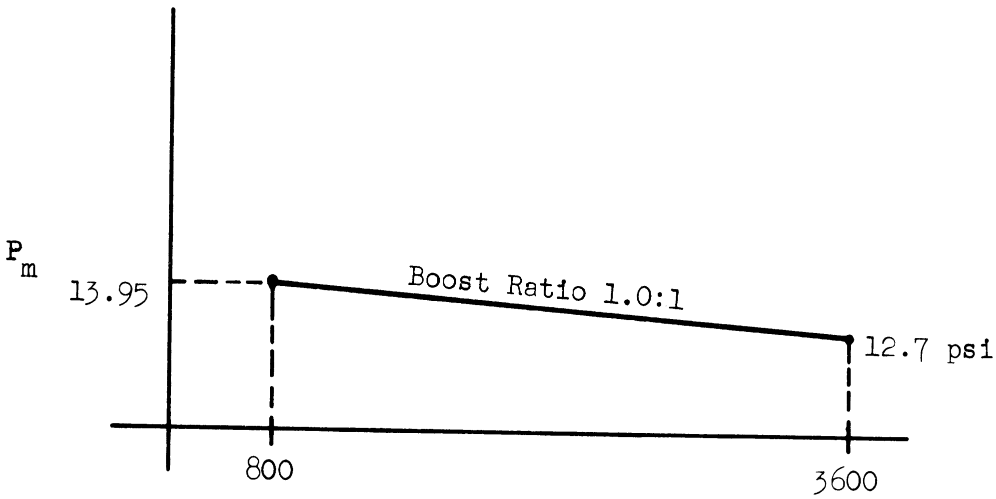


Fig. 2. Manifold pressure vs. rpm curves.

Other important assumptions about the engine conditions investigated are given in Table I.

TABLE I

ASSUMPTIONS EMPLOYED
Maximum Power A/F Ratio = 0.0782
Ambient Pressure and Temperature, 14.7 psia and 540° abs

Supercharger ratio	1:1	1.5:1	2.0:1	2.5:1	3.0:1
Fraction of exhaust gas (f) in new charge	0.035	0.032	0.03	0.029	0.028
Temperature of exhaust (°F at TDC)	1600	1650	1700	1750	1800
Inlet manifold pressure at full load and speed (psi)	12.7	20.1	27.4	34.8	42.2
Exhaust manifold pressure when turbocharged (psi)	14.7	17.1	23.3	29.6	35.9

For an engine operating with any form of charger, the temperature of the air supply to the carburetor was calculated by the following equation:

$$T_T = T_a\left[1 + \frac{1}{\eta_c}\left(R^{\frac{k-1}{k}} - 1\right)\right] \tag{1}$$

where
T_T = temperature of air leaving charger (°abs)
T_a = ambient air (°abs)
η_c = compressor efficiency
R = compressor ratio
k = 1.4

This delivery temperature T_T was then corrected for fuel addition and hot-spot heating, as already mentioned, and the inlet manifold conditions P_m, T_m obtained.

The next step in the operation of the engine is to mix the new charge with the f lb of exhaust gas remaining in the cylinder. During this step the exhaust gas is first compressed to the pressure of the air inlet manifold, changing its temperature slightly. Then the inlet charge is mixed with this gas under constant enthalpy conditions. The cylinder then contains f lb of exhaust gas, consisting of f lb of burnt air and (F/A) f lb of burnt fuel, plus (1 - f) lb of fresh air and (1 - f)F/A lb of unburnt fuel, giving an overall weight of (1 + F/A) lb per cycle at P_1 and T_1.

The final cylinder charge conditions P_1 and T_1 were calculated by use of the energy equation, and from the results state 1 of the cycle was located on the charts. From this point the whole cycle was determined.

The procedure of calculation can be illustrated by analysis of two examples. In the first analysis, the density at which the end gas detonates in a standard engine with a compression ratio of 10:1 will be determined. In the second, the compression ratio at which the same end-gas density is achieved in a turbo-charged engine with a boost ratio of 1.5:1 will be found.

A. Standard Engine 10:1 Compression Ratio

Assumptions

Ambient air 14.7 psia and 80°F
Exhaust dilutant f = 0.035
Exhaust temperature $T_e°$ = 2060° abs
Manifold pressure P_m = 1.5 x 14.7 - 2.0
= 20.1 psia
Manifold temperature T_m = 540 - 40 + 80
= 580° abs

Assume that P_m and P_e, the pressure at the end of the exhaust stroke in the naturally aspirated engine, are equal. Then

Enthalpy of mixture = Enthalpy of new charge + Enthalpy of exhaust gases

$$(1 + F/A)\, C_{p_1} T_1 = (1 + F/A)(1 - f) C_{p_m} T_m + (1 + F/A) f\, C_{p_e} T_e$$

$$C_{p_1} T_1 = (1 - f) C_{p_m} T_m + f\, C_{p_e} T_e$$

Using gas tables for the values of the specific heats at the various temperatures gives

$$T_1 = \frac{(1 - 0.035) \times 0.242 \times 580 + 0.035 \times 0.295 \times 2060}{0.247}$$

$$= 628°R$$

Thus the starting point of the cycle is $P_1 = 12.7$ and $T_1 = 628°R$. The cycle is shown in Fig. 3. The chart for F/A = 0.782 gives the following values for points 1-5:

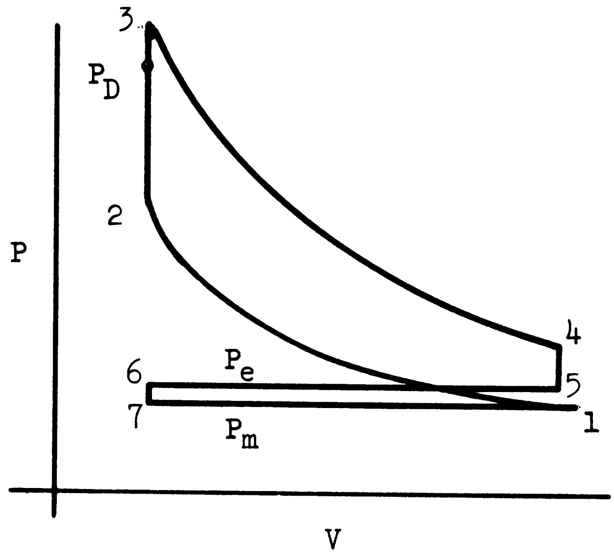


Fig. 3. Naturally aspirated cycle for 10:1 ratio.

Point 1

$P_1 = 12.7$, $T_1 = 628$, $V_1 = 18.5$, $E_1 = 23$, $S_1 = 0.1125$

where V_1 = volume in cu ft of (1 + 0.0782) lb of mixture

E_1 = Internal energy of mixture (Btu)

S_1 = Entropy of mixture

Point 2

$S_2 = 0.1125$, $V_2 = V_1/CR = 18.5/10 = 1.85$

$P_2 = 280$, $T_2 = 1315°$, $E_2 = 178$

Energy of Combustion $E_c = 1507\ (1 - f) + 300\ f$
$= 1465$ Btu

Point 3

$E_3 = E_2 + E_c = 1643$ Btu $V_3 = v_2 = 1.85$

$P_3 = 1180$, $s_3 = 0.525$, $T_3 = 5110°$ abs

Point 4

$V_4 = 18.5$, $S_4 = 0.525$

$P_4 = 66.0$, $E_4 = 925$, $T_4 = 2920$

$$\begin{aligned}\text{net work} &= (E_3 - E_4) - (E_2 - E_1)\\ &= 718 - 155\\ &= \underline{\underline{563 \text{ Btu}}}\end{aligned}$$

$$\begin{aligned}\text{cycle efficiency} &= \text{net work/heat added}\\ &= 563/1465 = 38.4\%\end{aligned}$$

$$\begin{aligned}\text{IMEP} &= \text{work done/change of volume}\\ &= 563 \times 778/(18.5 - 1.85) \times 144 \text{ psi}\\ &= \underline{\underline{182.6 \text{ psi}}}\end{aligned}$$

$$\begin{aligned}\text{Horsepower} &= \text{work} \times 778/550\ (1 - f)/\text{lb of air charge/sec}\\ &= 563 \times 778/550 \times 0.965\\ &= \underline{\underline{826 \text{ hp}}}/\text{lb air/sec}\end{aligned}$$

$$\begin{aligned}\text{Fuel flow} &= 0.782 \times 360\\ &= 282 \text{ lb/hr}\end{aligned}$$

$$\text{SFC} = 282/826 = 0.341 \text{ lb/IHP/hr}$$

The above values represent ideal performance for a given manifold pressure and temperature. A well developed actual engine operating under these conditions would develop an IMEP of about 165 gross mean pressure, indicating a relative cycle efficiency of about 0.90 and a BMEP of 135 psi at 81.5% mechanical efficiency. This compares with 109 psi for the L-141, indicating a maximum volumetric efficiency of about 80%; however, when heat losses are also included a value of 70% seems closer to the mark. Conversion can be made in this manner for actual output BHP, etc., when desired.

In this analysis it is proposed to employ the ideal cycle analysis since relative results will make it possible to evaluate the basic principle being examined.

It is now necessary to establish the state of the 5% of end gas which is to be considered at the detonating point. In Ref. 1, it is established that the pressure rise due to combustion at any instant, $P - P_2$, is proportional to the amount of charge burnt. It follows that P_D, the detonating point of the 5% (see Fig. 1), is given by

$$P_D = P_2 + 0.95\,(P_3 - P_2)$$

$$= 280 + 0.95\,(1180 - 280)$$

$$= 1135 \text{ psia}$$

The 5% of unburnt charge then, has been compressed from pressure P_1 to P_D, isentropically in the ideal case. This operation cannot be read off the mixture chart, because the range of the chart is too small. Hence the desired value was found by obtaining from the gas tables the average values of k for the process, and using them in the following equations:

$$P_1 V_1^{k} = P_D\, V_D^{h} \qquad (2)$$

$$T_D = T_1 \;(P_D/P_1)\;(k-1)/k \qquad (3)$$

The results obtained when k - 1.324 satisfied the temperature range from $T_1 = 630$ to $T_D = 1895$; solving then showed that V_D is

$$V_D = 0.623 \text{ cu ft}$$

When the density at detonation is considered constant, then,

$$\rho = P/RT = \text{constant}$$

Therefore,

$$P_D/T_D = \text{constant}$$

and

$$P_D/T_D = \frac{1135 \times 144}{1895} = 86.2$$

Thus the density at detonation must be held as close as possible to 86.2 at all times for all the other cycles having varying degrees of supercharge.

B. Turbocharged Engine with VCR Pistons and 1.5:1 Boost Ratio, without Aftercooling

Assumptions

Ambient Air 14.7 psia and 80°F
Exhaust dilutant f = 0.032
Exhaust temperature $T_e° = 1650°F$
$= 2110°R$
Manifold pressure = 1.5 x 14.7 - 2.0
= 20.1 psia

Temperature after compressor = T_r

$$= T_a \{1 + \frac{1}{\eta_c} [R\ (k\text{-}1/k)\ -\ 1]\}$$

T_a = Ambient air = 540° abs
η_c = Compressor efficiency = 0.70
P = Pressure ratio = 1.5:1
$T_T = 540\ \{1 + 1/0.70\ (1.5\ .286\ -\ 1)\}$
$= 635°R$
Manifold temperature $T_m = T_T - 40 + 80$
$= 675°R$

With supercharging at 20.1 psia and exhaust manifold at 17.1 psia (see Table I), the indicator diagram is as shown in Fig. 4, where it is seen that

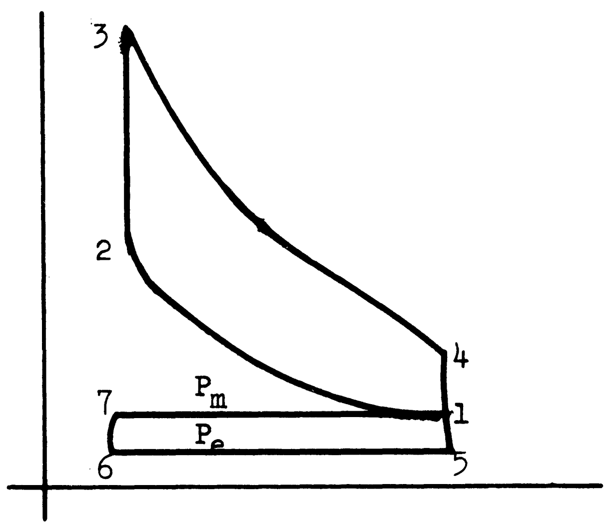


Fig. 4. Supercharged cycle for boost ratio of 1.5:1.

the exhaust dilutant is compressed from P_e to P_m, increasing its temperature and pressure by the addition of work. The mixture temperature is then given by

$$C_{P_1} T_1 = (1 - f) C_{P_m} T_m + f \left[C_{P_e} T_e - C_{V_e} (T_{e_7} - T_{e_6}) \right] \qquad (4)$$

$$T_{e_7} = T_{e_6} \times (P_m/P_e)^{k-1/k} \qquad (k = 1.35 \text{ for } T_6 = 2110)$$

$$= 2110 \times 1.043$$

$$= 220°R$$

From the gas tables, $C_{P_m} = 0.242$, $C_{P_e} = 0.312$, $C_{P_1} = 0.25$ and $C_{V_e} = 0.245$. Substituting in Eq. (2) gives

Temperature of charge $T_1 = 716°R$

Point 1

$P_1 = 20.1$, $T_1 = 716°$, $V_1 = 13.5$, $E_1 = 39.0$, $S_1 = 0.11$

Point 2

Since the compression ratio that will produce $P_D/T_D = 86.2$ is unknown, the only known fact about point 2 is that the heat of combustion released at this point is given by

$$E_c = 1507(1 - f) + 300f$$

$$= \underline{\underline{1468 \text{ Btu}}}$$

A trial-and-error solution is resorted to, a compression ratio is assumed, and the calculations are completed to obtain P_D/T_D. The compression is then charged until $P_D/T_D = 86.2$, as nearly as possible.

When a ratio of 7.1:1 was employed, the following data were obtained:

$$V_2 = \frac{13.5}{7.1} = 1.9 \text{ cu ft}, \quad P_2 = 270 \text{ psi}$$

$$T_2 = 1300°R, \quad E_2 = 174, \quad S_2 = 0.11$$

Point 3

$$V_3 = V_2 = 1.9, \quad E_3 = 174 + 1468 = 1642, \quad P_3 = 1160 \text{ psi}, \quad T_3 = 5120°R$$

Point 4

$$V_4 = 13.5, \quad S_4 = S_3, \quad P_4 = 100 \text{ psia}, \quad T_4 = 3220°R, \quad E_4 = 1013$$

$$P_D = P_2 + 0.95\,(P_3 - P_2)$$

$$= 0.95\,P_3 - 0.05\,P_2$$

$$= \frac{1114 \text{ psi}}{0.24}$$

$$T_D = T_1\,(P_D/P_1) = 1880$$

$$\therefore = P_D/T_D = 85.5, \text{ close to the desired } 86.2.$$

$$\text{Work done} = (E_3 - E_4) - (E_2 - E_1) + \frac{(P_m - P_e)(V_1 - V_2) \times 144}{778}$$

$$= (1642 - 1013) - (174 - 39) + \frac{3.01 \times 11.6 \times 144}{778}$$

$$= \underline{\underline{500.4 \text{ Btu}}}$$

$$\text{Thermal efficiency} = \frac{500.4}{1468} = 35.3\%$$

$$\text{IMEP} = \frac{500.4 \times 778}{(13.5 - 1.9) \times 144} = 235 \text{ psi}$$

$$\text{hp} = \frac{500.4 \times 778}{550 \times 0.968} = 731 \text{ hp/lb air/sec}$$

$$\text{Fuel flow} = 282 \text{ lb/hr}$$

$$\text{SFC} = \frac{282}{731} = 0.386 \text{ lb/IHP/hr}$$

It is true that the value of P_D/T_D is not exactly the desired value, but it was impossible to read the charts accurately enough to obtain exact equality.

The actual difference in ratio between 85.5 and 86.2 is small and will of course be meaned out by plotting the results obtained in the subsequent diagrams.

By proceeding in the above manner for the various supercharging ratios a series of results was obtained for the constant density relationship.

In the case in which aftercooling was employed, it was assumed that a heat exchanger with an effectiveness of 0.7 was employed. That heat exchanger is defined by

$$\text{Exchanger effectiveness } \epsilon = \frac{T_T - T_a}{T_T - T_0}$$

where T_a = temperature after exchanger and before carburetor

T_0 = ambient air temperature

$$\therefore T_a = T_T - 0.7\,(T_T - T_0)$$

$$= 0.3\,T_T + T_0$$

With the fuel addition and hot spot effects, T_a changes as already defined and the manifold temperature becomes

$$T_m = T_a - 40 + 80$$

$$= T_a + 40$$

V. RESULTS

The data obtained when the value of P_D/T_D was held approximately constant are shown in Tables II-V for the various cases indicated by the table headings.

TABLE II

SPECIFIC PERFORMANCE OF TURBOCHARGED ENGINE WITHOUT AFTERCOOLER, FOR CONSTANT END-GAS DENSITY

($A/F = 0.0782$; $T_0 = 540°R$; $P_0 = 14.7$)

Boost Ratio	L:1	1.5:1	2.0:1	2.5:1	3.0:1
Work (Btu/cycle)	563.0	500.4	452	440	417
Thermal efficiency (%)	38.4	35.3	30.8	29.9	28.3
IMEP (psi)	182.6	235	267	300	335
IHP/lb air/sec	826	731	660	641	607
SFC lb/IHP/hr	0.341	0.386	0.414	0.44	0.465
Compression ratio	10:1	7.1:1	6.0:1	5.5:1	4.8:1
P_D/T_D	86.2	85.5	85.9	86.3	86.6
P_m	12.7	20.1	27.4	34.8	42.2
T_m	628	675	748	812	877

TABLE III

SPECIFIC PERFORMANCE OF TURBOCHARGED ENGINE WITH AFTERCOOLER, FOR CONSTANT END-GAS DENSITY

($A/F = 0.0782$; $P_0 = 14.7$; $T_0 = 540°R$; $\epsilon = 0.70$)

Boost Ratio	1.0:1	1.5:1	2.0:1	2.5:1	3.0:1
Work (Btu/cycle)	563.0	467	4.4	367	274
Thermal efficiency (%)	38.4	31.8	28.1	24.9	18.6
IMEP (psi)	182.6	251	307	352	347
IHP/lb air/sec	826	683	603	535	399
SFC lb/IHP/hr	0.341	0.413	0.467	0.528	0.707
Compression ratio	10:1	6.1:1	4.3	3.6	3.0
P_D/T_D	86.2	86.2	86.0	87.0	87.0
P_m	12.7	20.1	27.4	34.8	42.2
T_m	628	608	630	649	666

TABLE IV

SPECIFIC PERFORMANCE OF DIRECTLY DRIVEN SUPERCHARGER WITHOUT AFTERCOOLER, FOR CONSTANT END-GAS DENSITY

($F/A = 0.0782$; $P_0 = 14.7$; $T_0 = 540°R$; $\eta_c = 0.70$; $\epsilon = 0.70$)

Boost Ratio	1.0:1	1.5:1	2.0:1	2.5:1	3.0:1
Work (Btu/cycle)	563	483.0	426.7	416	375
Thermal efficiency (%)	38.4	32.9	29.0	28.3	25.5
IMEP (psi)	182.6	225	252	284	302
IHP/lb air/sec	826	705	622	607	546
SFC lb/IHP/hr	0.341	0.40	0.453	0.465	0.516
Compression ratio	10:1	7.1:1	6.0:1	5.5:1	4.8:1
P_D/T_D	86.2	85.5	85.9	86.3	86.6
P_m	12.7	20.1	27.4	34.8	42.2
T_m	628	675	748	812	877

TABLE V

SPECIFIC PERFORMANCE OF DIRECTLY DRIVEN SUPERCHARGER WITH AFTERCOOLER, FOR CONSTANT END-GAS DENSITY

($F/A = 0.0782$; $P_0 = 14.7$; $T_0 = 540°R$; $\eta_c = 0.70$; $\epsilon = 0.70$)

Boost Ratio	1.0:1	1.5:1	2.0:1	2.5:1	3.0:1
Work (Btu/cycle)	563	448	385	328	225
Thermal efficiency (%)	38.4	30.5	26.1	22.3	15.3
IMEP (psi)	182.6	241	286	315	285
IHP/lb air/sec	826	655	561	477	328
SFC lb/IHP/hr	0.341	0.431	0.502	0.591	0.851
Compression ratio	10:1	6.1:1	4.3:1	3.6:1	3.0
P_D/T_D	86.2	86.2	86.0	87.0	87.0
P_m	12.7	20.1	27.4	34.8	42.2
T_m	628	608	630	649	666

When a direct-drive charger was used, the necessary adjustments for the power required for its operation, plus a 2% loss for the gear set, were made. The isentropic efficiency of the compressor was taken at 70%, a value capable of being achieved by a compression type of blower, but too high for one of the Rootes type. In addition, the engine mean effective pressure was adjusted to allow for the fact that the exhaust manifold pressure will be approximately atmospheric when such a compressor is used.

The above results are plotted in Figs. 5 to 8, and Fig. 9 shows a comparison of compression ratio, maximum pressure, and detonation temperature for the turbocharged engine with and without aftercooling.

All of the results presented give the maximum performance data at full speed and load for each boost ratio with F/A = 0.0782. As the engine speed decreases the boost ratio will decrease, because of the reduction in mass flow of gas through the charger. Let us assume that the manifold pressure will decrease along a straight line until the engine is idling at 800 rpm. The manifold pressure will then be 13.95 psia, and this value will remain constant at the speed no matter what the speed control of the charger is in use. That is, at 800 rpm the charger will give no boost, but the pressure drop, due to flow through the air cleaner, the charger, the carburetor, etc., will be 0.75 psi (at 3600 rpm it is 2.0 psi), then the diagram shown in Fig. 10 can be constructed to represent the approximate manifold pressure relationship versus rpm for the engine. If a constant F/A ratio of 0.0782 is employed as the full throttle mixture ratio at each point, the cycle analysis gives the maximum power outputs that can be obtained over the range of manifold pressures shown in Fig. 10; the results are given in Table VI and Fig. 11, from which one can determine the performance that will be obtained by using a VCR piston at any manifold pressure and speed for a F/A ratio of 0.0782 and constant end-gas density.

TABLE VI

SPECIFIC PERFORMANCE AND MANIFOLD PRESSURE

F/A = 0.0782; T_0 = 504°R; P_0 = 14.7

Manifold pressure, psia	42.1	34.75	33.1	27.4	23.1	20.1	18.1	16.0	12.7
Work (Btu/cycle)	417	440	442	452	484	500.4	521	532	563
Thermal efficiency (%)	28.3	29.9	30.0	30.8	32.9	35.3	35.5	36.2	38.4
IMEP (psi)	355	306	307	267	247	235	217	206	182.6
IHP/lb air/sec	607	641	643	660	708	731	761	777	826
SFC/lb/IHP/hr	0.465	0.44	0.439	0.414	0.399	0.386	0.371	0.363	0.341
Compression ratio	4.8:1	5.5:1	5.3:1	6.0:1	6.6:1	7.1:1	7.9:1	8.4:1	10.0:1

In addition the throttled operation of the unsupercharged engine was determined and added to Fig. 11 for manifold pressures of 10.0 and 7.5 psia at a F/A ratio of 0.0605; and the compression ratio was determined for the same constant end-gas density. At pressures below 7.5 psia the compression ratio increased to such high values that a practical engine would not be possible; hence the condition giving a maximum ratio of 19:1 was the highest calculated. This ratio is probably higher than could be conveniently designed into the engine. The data for throttled operation are given in Table VII.

From the results of Tables VI and VII, as plotted in Fig. 11, it is possible to obtain peak power for all possible manifold pressures up to 42.1 psi.

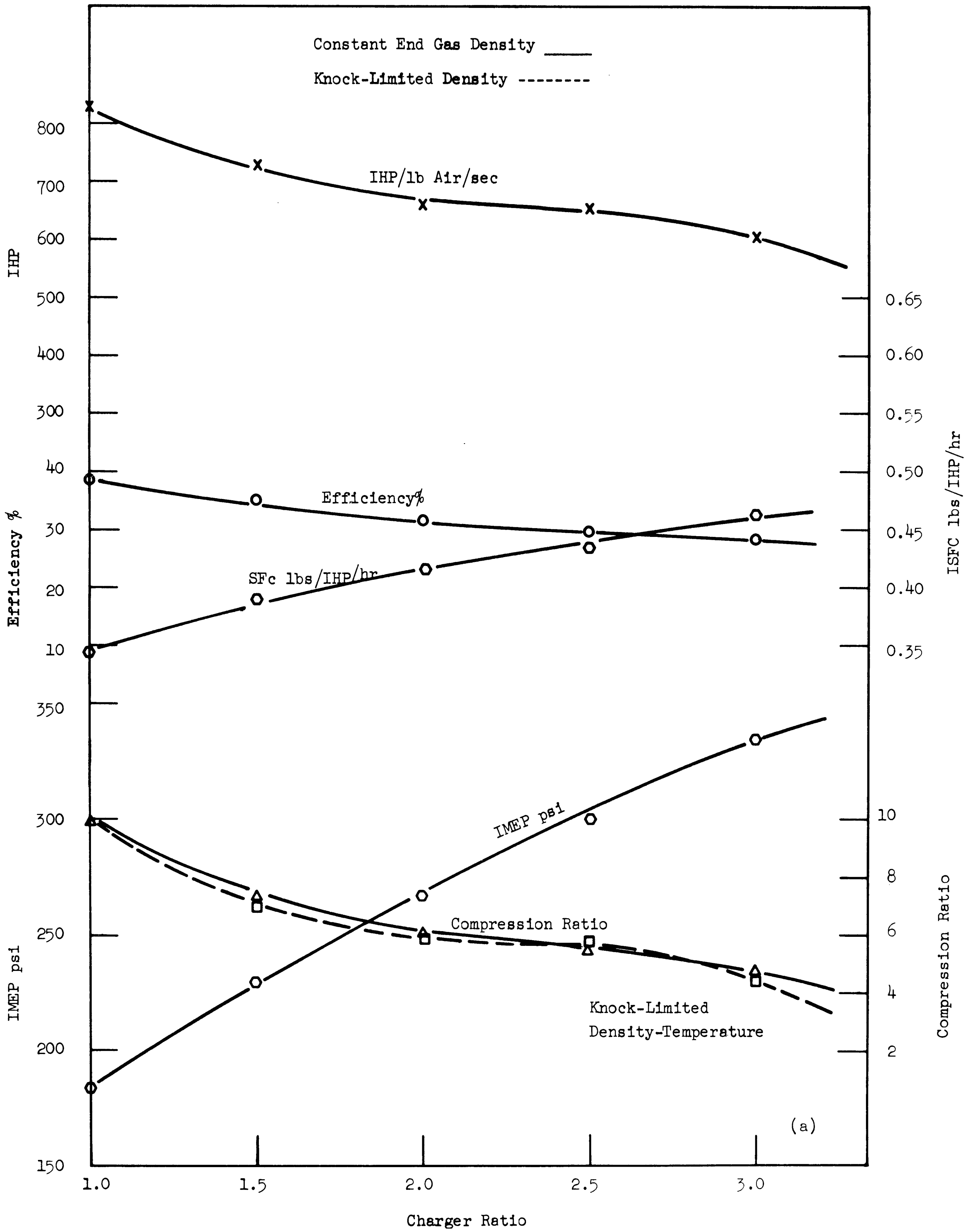


Fig. 5. Specific performance of turbocharged system without aftercooler.

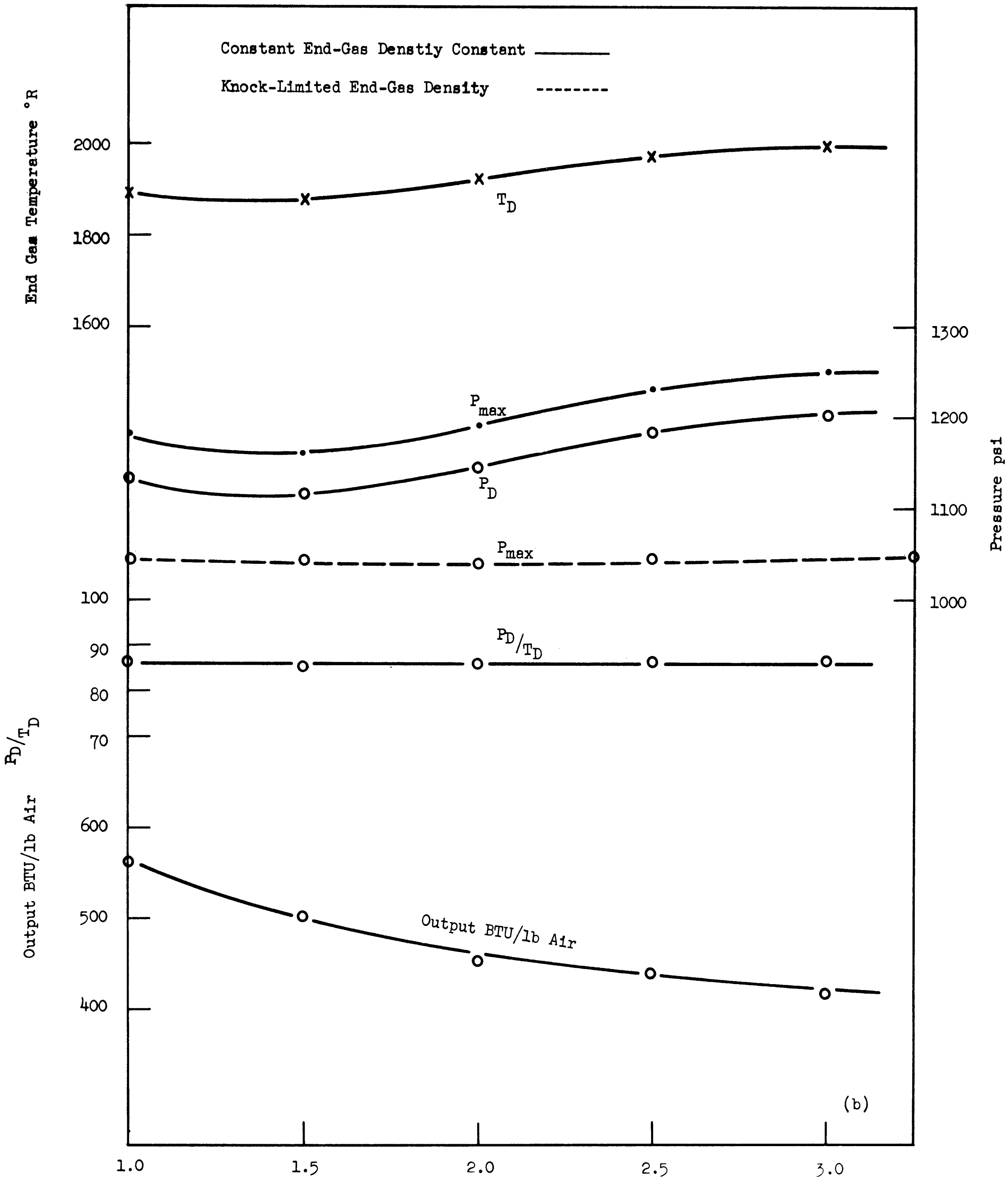


Fig. 5. (Concluded)

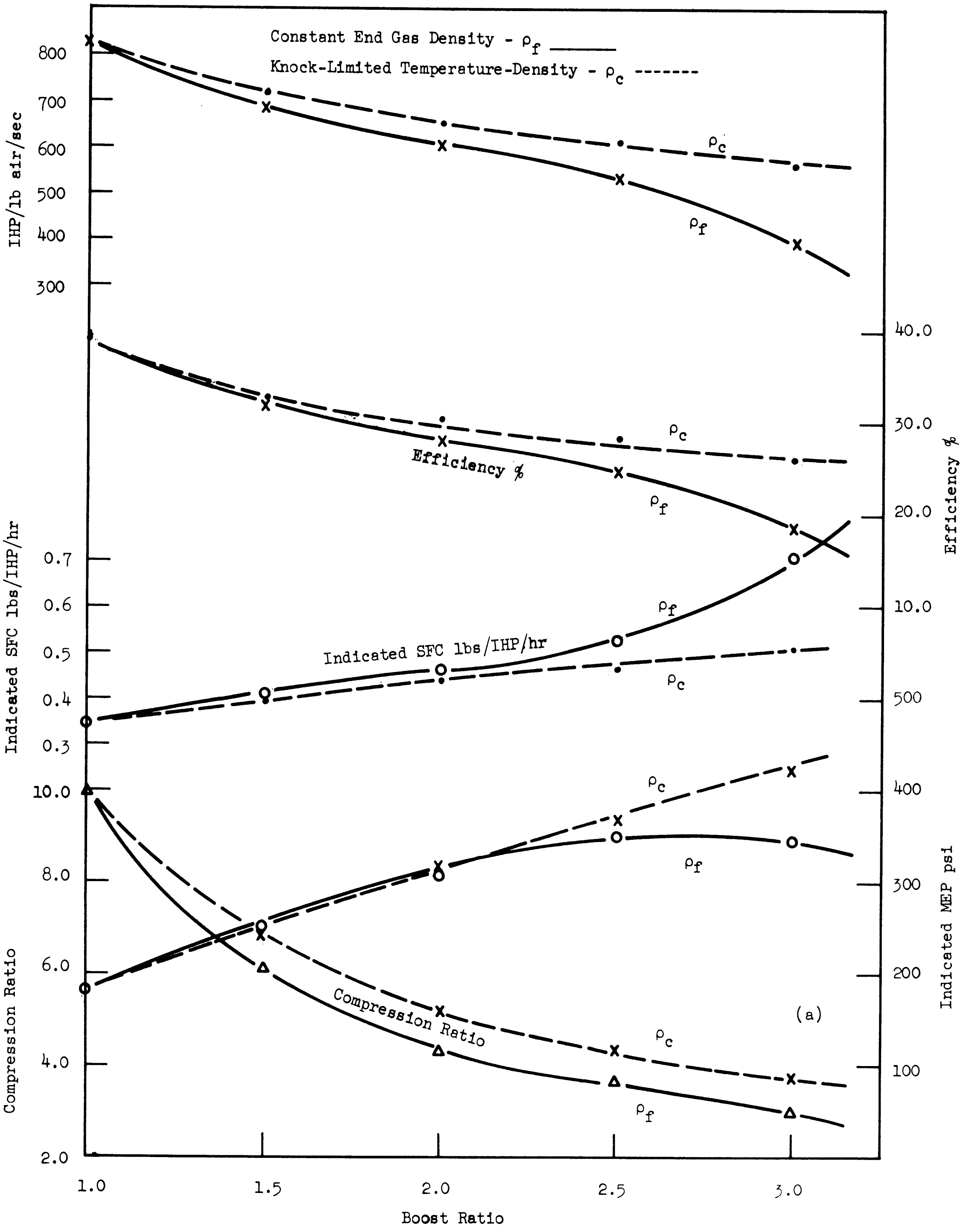


Fig. 6. Specific performance of turbocharged system with aftercooler.

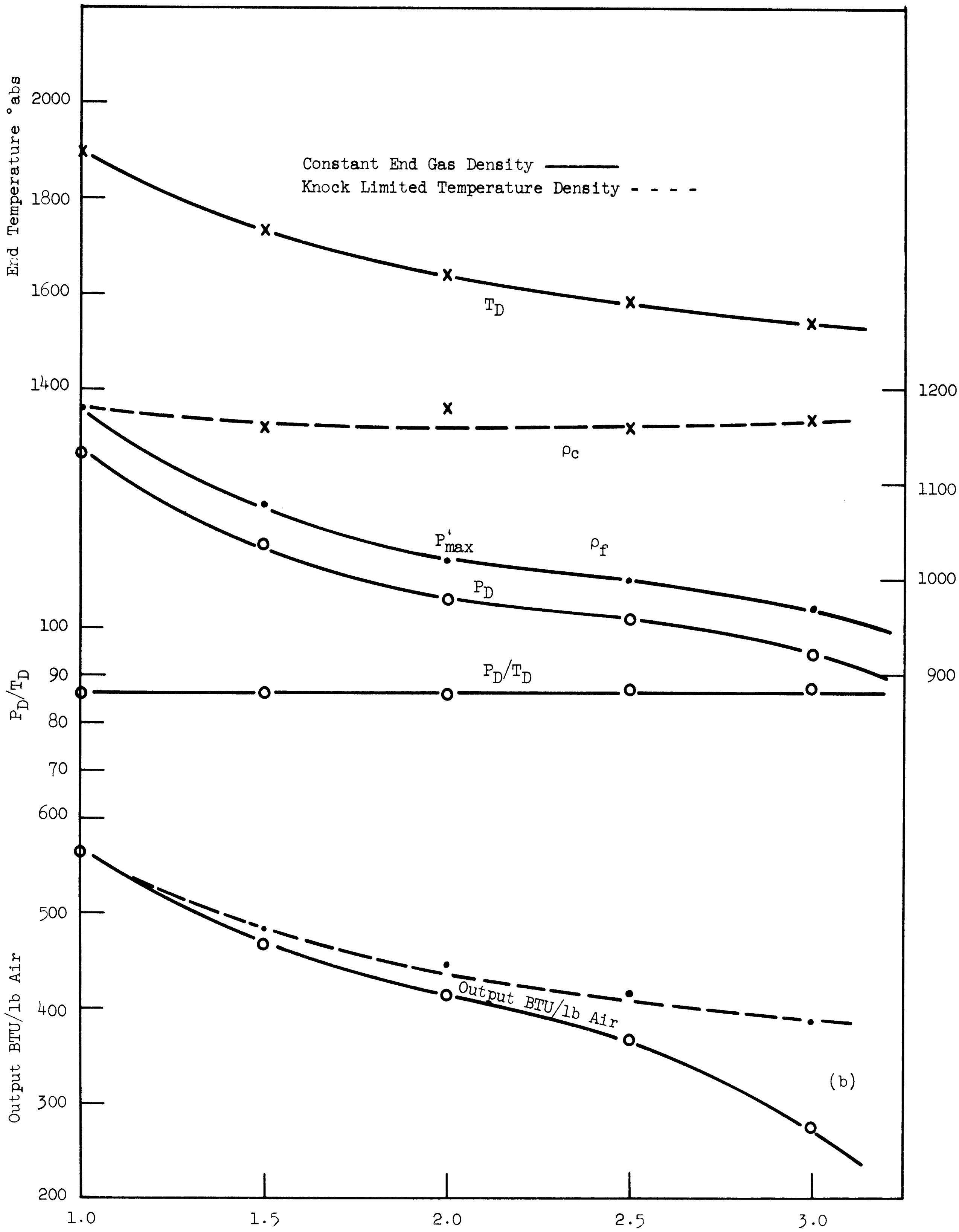


Fig. 6. (Concluded)

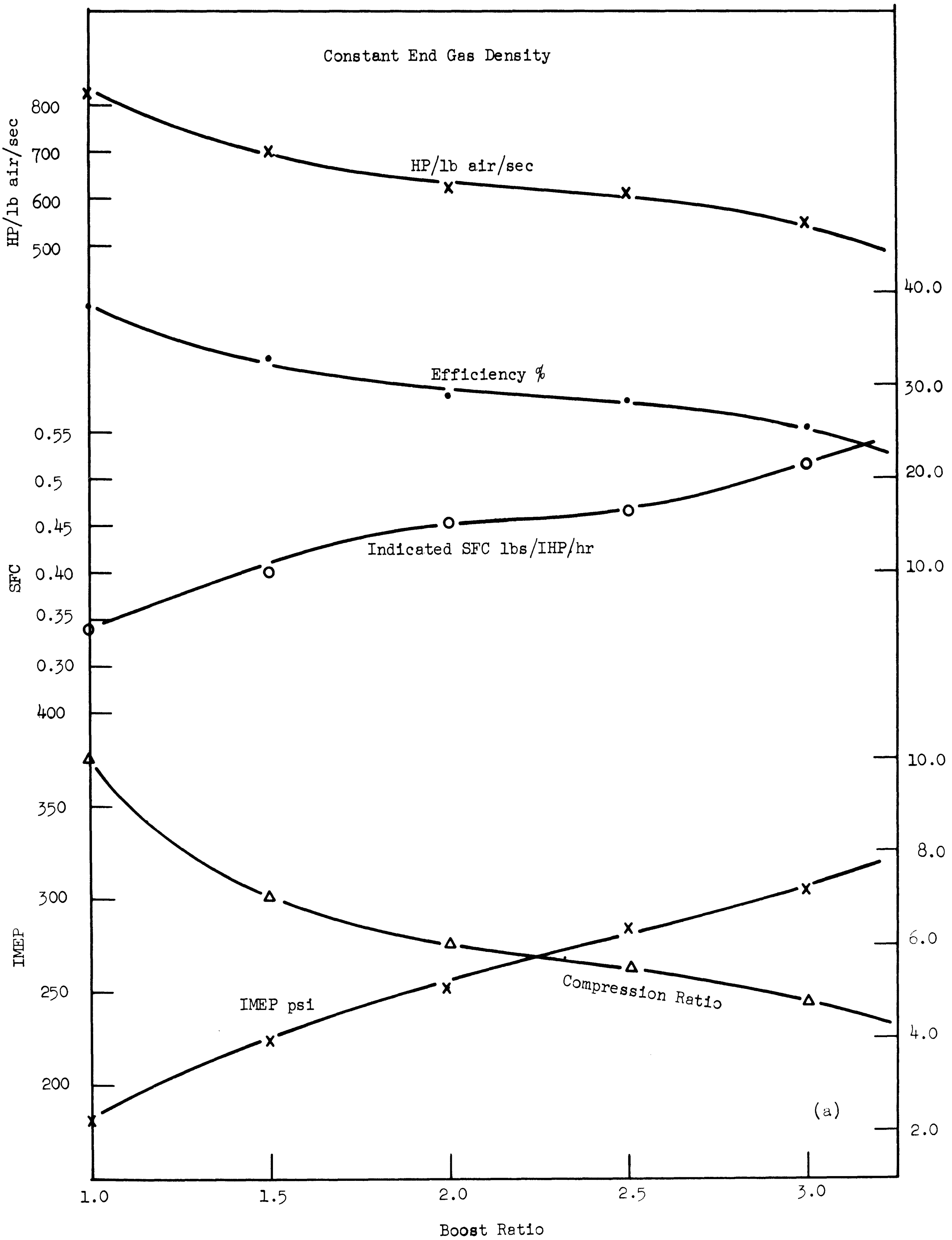


Fig. 7. Specific performance of directly driven charger without aftercooler.

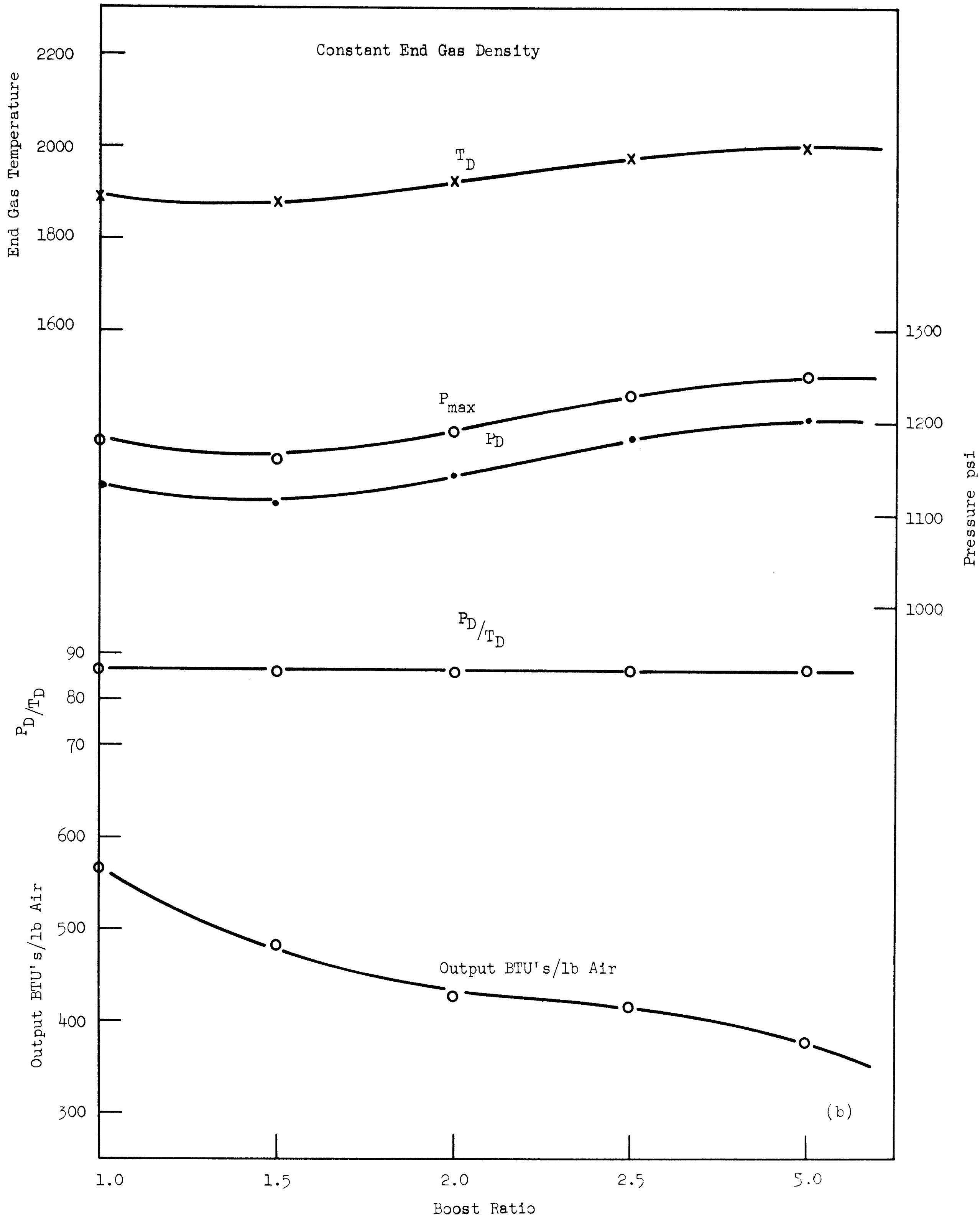


Fig. 7. (Concluded)

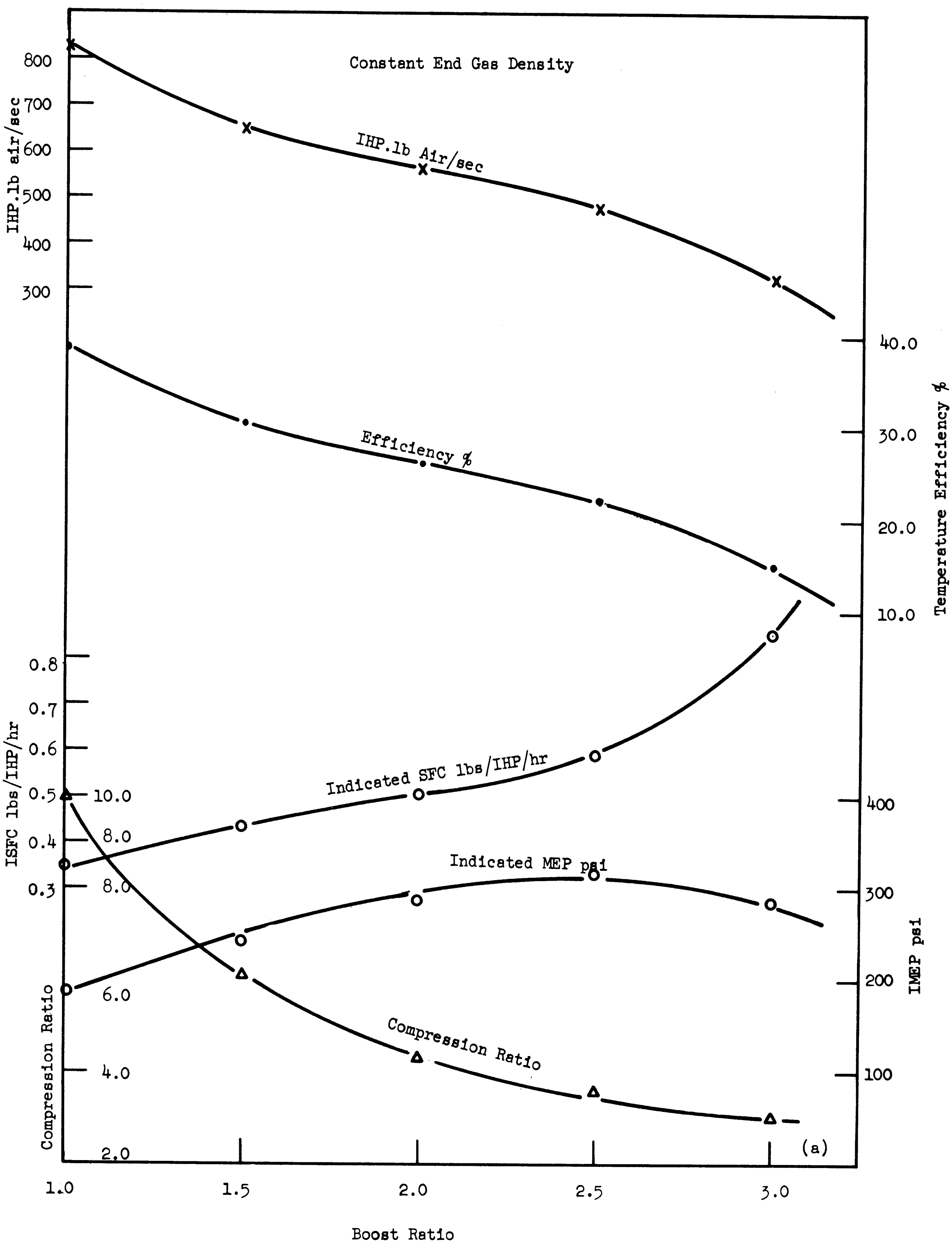


Fig. 8. Specific performance of directly driven charger with aftercooler.

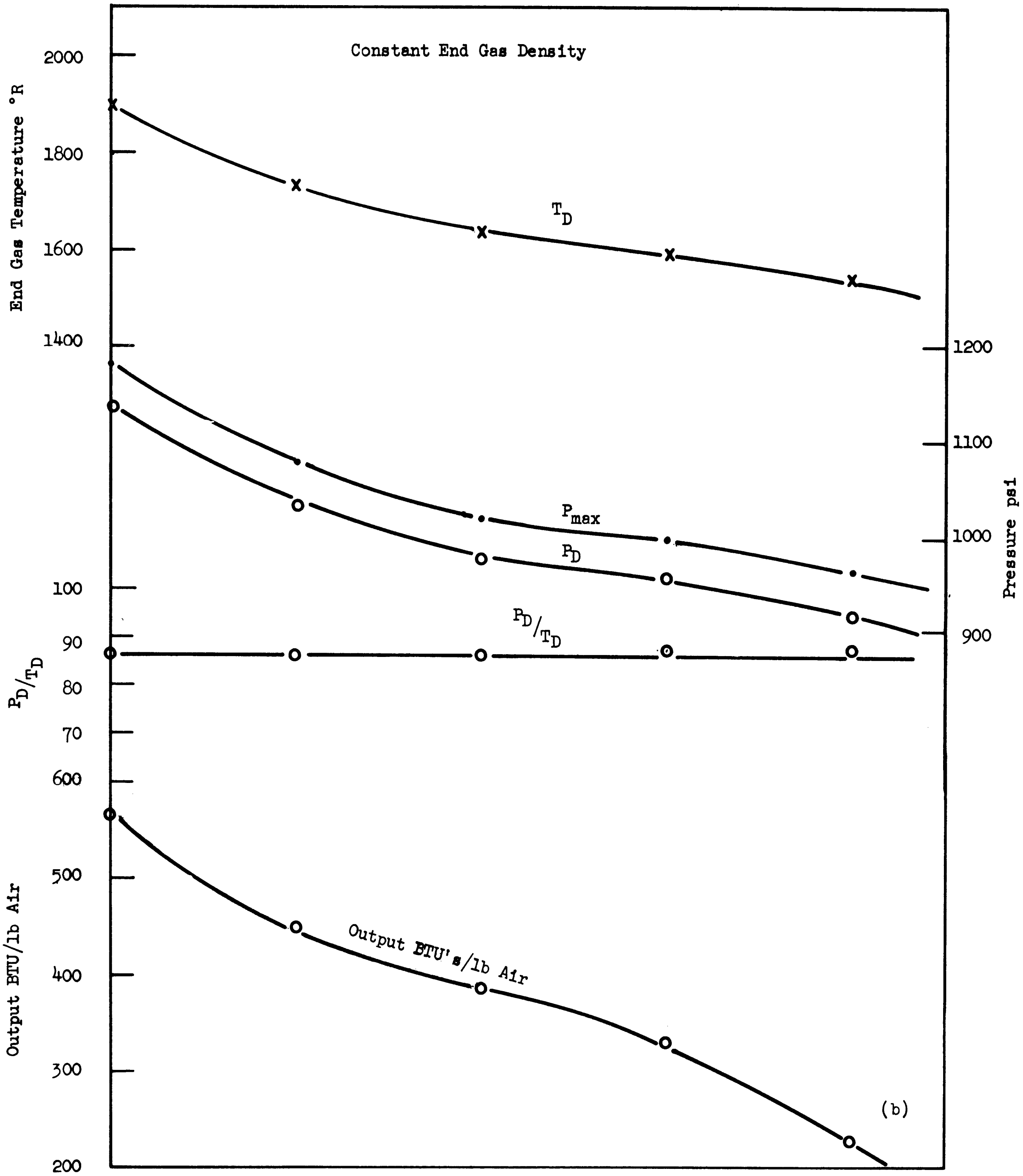


Fig. 8. (Concluded)

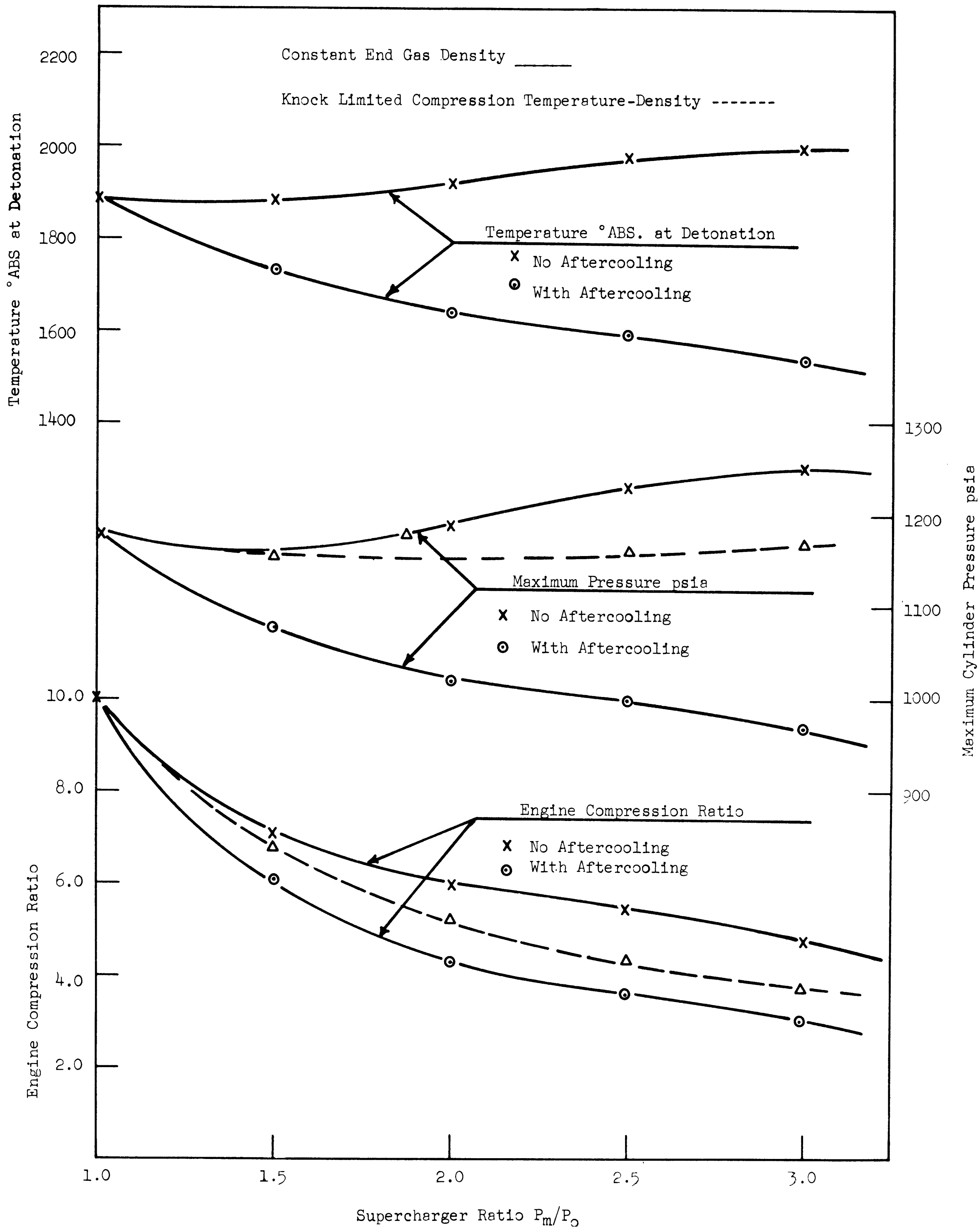


Fig. 9. Compression ratio and maximum pressure without aftercooling.

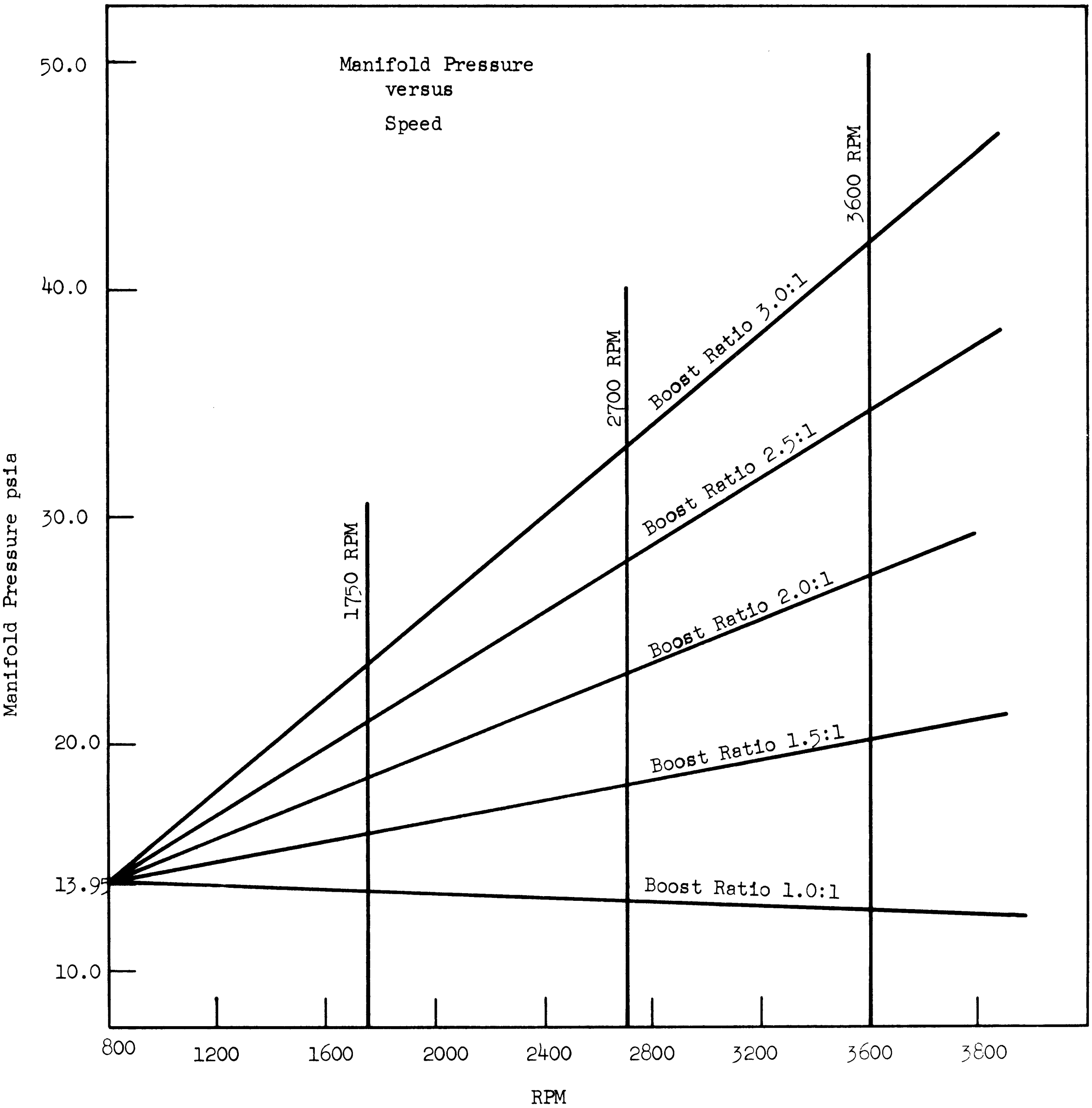


Fig. 10. Manifold pressure vs. speed data.

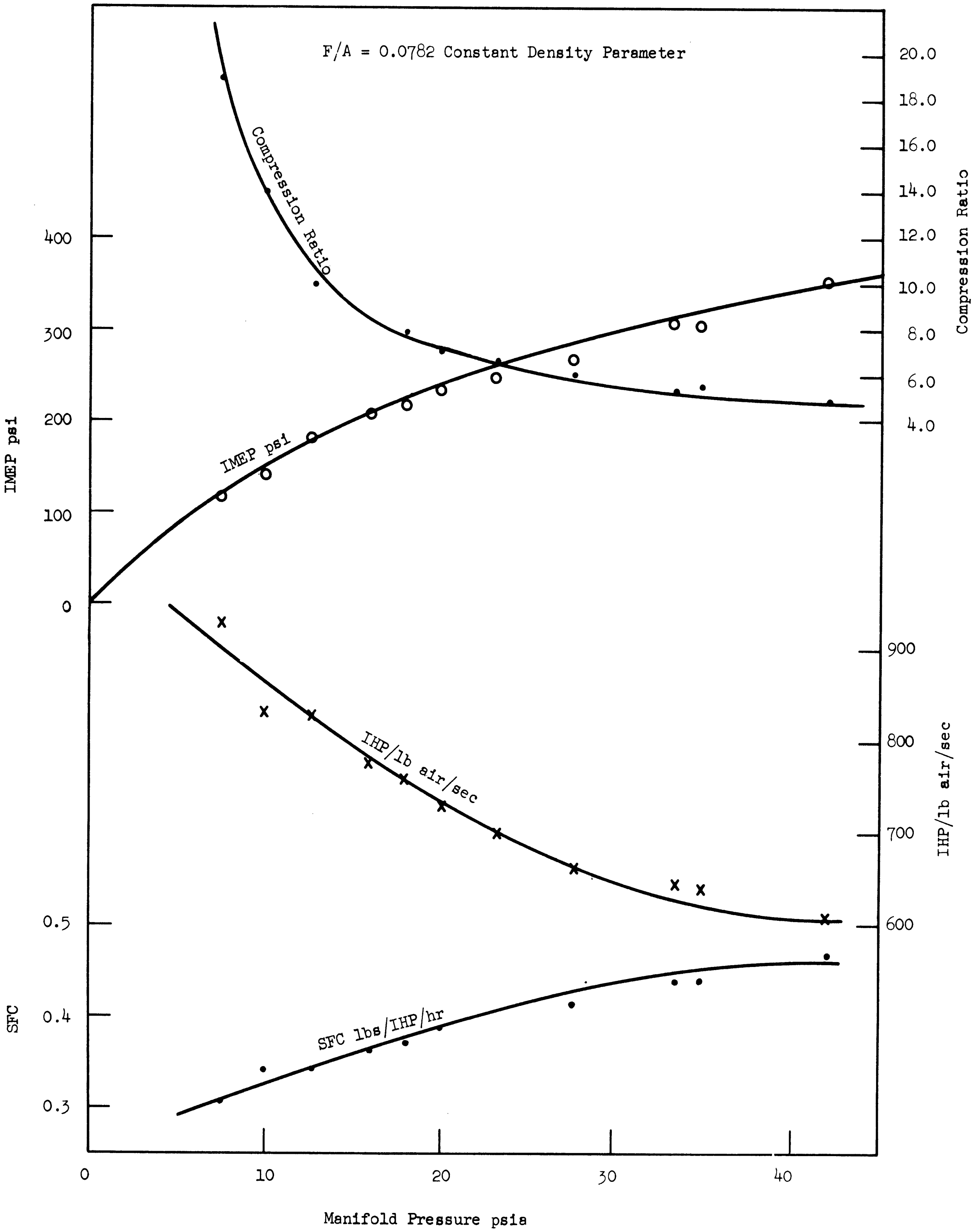


Fig. 11. Specific performance at full power operation, for F/A = 0.0782.

TABLE VII

THROTTLED OPERATION OF ENGINE
$F/A = 0.0605$; $T_0 = 540°A$; $P_0 = 14.7$ psia

Manifold pressure (psia)	10.0	7.5
Work (Btu/cycle)	565	630
Thermal efficiency (%)	50.3	51.1
IMEP (psi)	140.1	116
IHP/lb air/sec	829	927
SFC lb/IHP/hr	0.341	0.304
Compression ratio	14.0	19.0

METHOD 2

In Method 1 the effects of both pressure and temperature upon detonation are taken to have roughly equal influence. It is generally believed, however, that temperature has more influence than pressure does. After intensive test work on a cooperative fuel-research engine, recorded in Ref. 2, Siegel suggested that a plot of compression temperature and compression density could give coordinating data over a wide range of conditions.

When this approach is viewed in its broad aspects, these pressures and temperatures seem to have no direct bearing upon the state of the gas at detonation, since between the end of engine compression and the detonation point the end-gas density and temperature undergo a change roughly of the order of 4 to 10:1. However, the compression state was shown to have some significance in the engine tested in Ref. 2; as a result, the principle will be employed in a second approach to the problem under review.

The results presented in Ref. 2 cannot be applied directly to the problem in hand, since they apply to a very different system. They do include, however, a range of knock-limited manifold air pressure values that can be employed for fuel of any given octane value. If, then, any one actual test point representing the state of the charge inside the cylinder undergoing the cyclical process is available for the L-141 engine, the magnitudes of this pressure at all other points can be estimated. To employ this method it will be assumed again that the naturally aspirated engine at a compression ratio of 10:1 represents the limiting condition for detonation of the MIL-G3056-A, 83-octane fuel to be used in engine tests. The description of the standard engine performance of the 10:1 version, given on page 12, records that, at the end of the compression stroke, $P_2 = 280$, $V_2 = 1.85$, and $T_2 = 1315°$ and that the mass of the charge is 1.0782 lb. It follows that the compression density is $1.0782/1.85 = 0.582$ lb/cu ft and the temperature is 1315°R. Thus the coordinates of one point of the desired knock-limiting curve are known; also the reciprocal of the knock-limited manifold pressure is 1/12.7 = 0.0788. This value permits the construction of Fig. 12 for 83-octane fuel, which is similar to Fig. 7 of Ref. 2.

Now the performance has already been examined for additional manifold pressures of 20.1, 27.4, 34.8, and 42.2 psi or reciprocal values of 0.0495, 0.0365, 0.02875, and 0.0237. Obtaining the corresponding compression ratios from Fig. 12, we get 6.65, 5.15, 4.25, and 3.7, whereas the constant end-gas density relationship gave 7.1, 6.0, 5.5, and 4.8; thus this appears to be a reasonable check of the method despite the varying temperature in the system being considered. However, the same manifold pressures were used for the system using an aftercooler; here the temperature of compression will again change considerably, but Fig. 12 does not take this into account. It therefore becomes necessary to calculate the equivalent of Fig. 13 of Ref. 2 from the one known point, namely a knock-limited compression density of 0.582 at a compression ratio of 10:1. Plotting a curve in Fig. 13 through this value with the corresponding values determined from the relationship gives

$$\text{K - L Manifold Pressure} = \frac{W}{\text{Disp}} (\text{CR} - 1)$$

$$= \text{Const} (\text{CR} - 1)$$

where W = lb of air/hr

Disp = displacement in cu ft/hr

Then, since P_m = 12.7 psi at 10:1 ratio, the constant becomes 114.3 and ρ_c = 0.582, if the chart values for displacement are used, whereas if the relations $\rho_c = P_1 \, (CR)^{1.332}$, $T_c = T_1 \times CR^{0.332}$ and $P_1 V_1 = wRT_1$ (where R = 53.34, w = wt of air [lb]) are employed for ρ_c its value becomes 0.542. This latter method of calculation will be employed, since this is the method of Ref. 2; and the data of Table VIII are calculated.

TABLE VIII

KNOCK-LIMITED DENSITY-TEMPERATURE FOR 83-OCTANE FUEL

Compression Ratio	K-L Manifold Pressure	T_1	T_c	P_c	ρ_c
16	7.62	620	1560	305	0.528
14	8.79	624	1500	295	0.531
12	10.39	628	1435	285	0.536
10	12.7	632	1360	273	0.524
8	16.32	636	1269	260	0.554
6	22.85	640	1160	248	0.578
4	38.1	644	1020	241	0.638
2	114.3	648	814	288	0.955

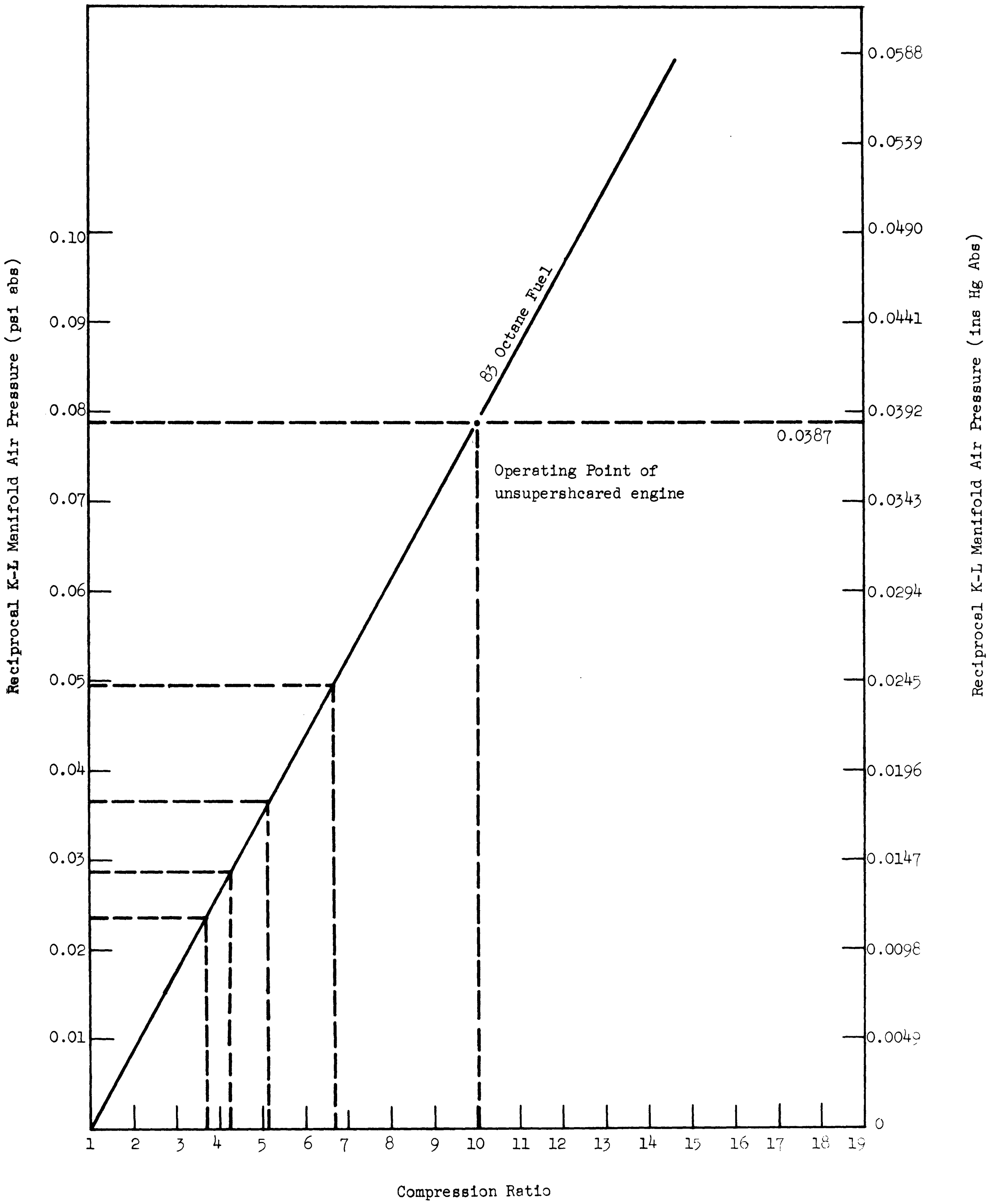


Fig. 12. Reciprocal of knock-limited manifold pressure vs. compression ratio, for 83-octane fuel.

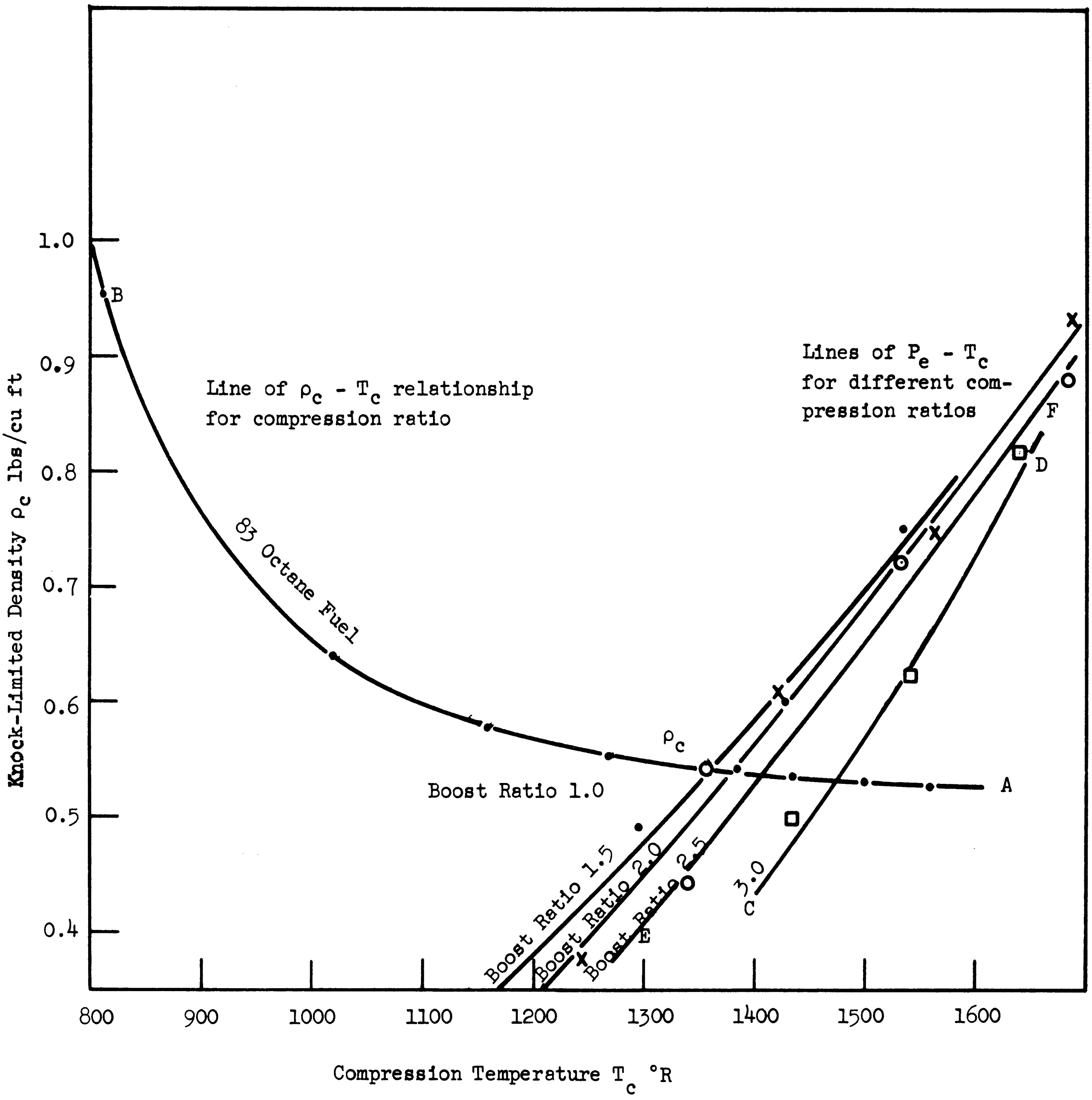


Fig. 13. Knock-limited temperature vs. density relationship without after-cooling.

Plotting the ρ_c - T_c relationship gives curve AB of Fig. 13. It now remains to obtain density-temperature data for each of the various boost ratios, in order to obtain a common intersection from which the desired compression ratio satisfying ρ_c - T_c can be obtained.

Now the knock-limited density ρ_c for the boosted conditions is given by

$$K - L \rho_c = \frac{144 P_c}{R T_c}$$

where P_c = compression pressure (psi)

T_c = compression temperature

R = gas constant = 53.99

$p_c = P_1 \ (V_1/V_2)^{1.332}$

$T_c = T_1 \ (V_1/V_2)^{0.332}$

$V_1 = wRT_1/P_1$

From these relations the data of Table IX were calculated.

Plotting this information on Fig. 13 gives the curves CD, EF, etc. Reading the point of intersection of the boost ratio lines with the ρ_c - T_c line for 83-octane fuel then gives the data in Table X.

The compression ratio is obtained from Eq. 1 of Ref. 2:

$$(CR - 1) = \frac{K - L \rho_c \text{ x displacement}}{\text{weight of air}} \tag{5}$$

where displacement = $V_1 \ (1 - \frac{1}{CR})$. Substituting in Eq. (5), we get

$$(CR - 1) = \frac{K - L\rho_c}{W} V_1(1 = \frac{1}{CR})$$

$$= \frac{K - L\rho_c}{W} \text{ x } \frac{V_1(CR - 1)}{CR}$$

$$CR = \frac{(K - L\rho_c)V_1}{W}$$

TABLE IX

KNOCK-LIMITED DENSITY VERSUS BOOST RATIO, WITHOUT AFTERCOOLER

Boost Ratio	Compression Ratio	$\frac{V_1}{V_c}^{1.332}$	P_c	$(CR)^{0.332}$	T_1	T_c	$\rho_c = 2.67\ P_c/T_c$
1.5	10.0:1	21.5	432	2.148	716	1536	0.751
	8.0	15.99	321	1.995	716	1429	0.600
	6.0	11.87	238.5	1.812	716	1297	0.491
	4.0	6.35	127.7	1.585	716	1135	0.300
	2.0	2.52	50.7	1.258	716	900	0.1505
2.0	10.0	21.5	589	2.148	784	1685	0.934
	8.0	15.99	438	1.995	784	1565	0.747
	6.0	11.87	325	1.812	784	1421	0.610
	4.0	6.35	173.5	1.585	784	1243	0.375
	2.0	2.52	69.1	1.258	784	986	0.187
2.5	8.0	15.99	556	1.995	845	1687	0.88
	6.0	11.86	413	1.812	845	1531	0.72
	4.0	6.35	221	1.585	845	1340	0.44
	2.0	2.52	87.7	1.258	845	1064	0.22
3.0	8.0	15.99	675	1.995	904	1804	1.00
	6.0	11.86	501	1.812	904	1638	0.816
	5.0	8.52	360	1.705	904	1541	0.624
	4.0	6.35	268	1.585	904	1434	0.499
	2.0	2.52	106.5	1.258	904	1138	0.250

TABLE X

KNOCK-LIMITED COMPRESSION RATIO, WITHOUT AFTERCOOLER

Boost ratio	1.0	1.5	2.0	2.5	3.0
K-L density ρ_c	0.542	0.54	0.5375	0.535	0.53
lb air/cycle	0.965	0.968	0.970	0.971	0.972
Total cylinder volume V_1 cu ft/cycle	17.75	12.8	10.3	8.73	7.71
Compression ratio	10.0	7.14	5.7	4.81	4.2

V_1, the total volume of the cylinder, and W, the air charge are known for each inlet condition. Thus the compression ratio can be calculated.

This process is now repeated for an engine with an aftercooler having an effectiveness of 0.7. The results are shown in Tables XI and XII and in Fig. 14.

TABLE XI

KNOCK-LIMITED DENSITY VERSUS BOOST RATIO, WITH AFTERCOOLER

Boost Ratio	Compression Ratio	$\left(\frac{V_1}{V_2}\right)^{1.332}$	P_c	$(CR)^{0.332}$	T_1	T_c	$\rho_c = 2.67 \frac{P_c}{T_c}$	Total Charge Volume (cu ft)
1.5	10.0	21.5	432	2.148	655	1407	0.82	12.0
	8.0	15.99	321	1.995	655	1307	0.656	
	6.0	11.87	239	1.812	655	1187	0.538	
	4.0	6.35	128	1.585	655	1038	0.329	
	2.0	2.52	50.7	1.258	655	824	0.1645	
2.0	9.0	18.65	510.5	2.074	672	1392	0.98	9.4
	8.0	15.99	438	1.995	672	1340	0.873	
	6.0	11.87	325	1.812	672	1218	0.713	
	4.0	6.35	174	1.585	672	1065	0.436	
	2.0	2.52	69.1	1.258	672	845	0.218	
2.5	7.0	13.35	463	1.909	693	1323	0.935	7.8
	6.0	11.87	412	1.812	693	1255	0.877	
	5.0	8.52	297	1.706	693	1183	0.670	
	4.0	6.35	221	1.585	693	1098	0.537	
	3.0	4.32	151	1.439	693	996	0.405	
3.0	6.0	11.87	501	1.812	708	1284	1.043	6.4
	5.0	8.52	360	1.706	708	1208	0.795	
	4.0	6.35	268	1.585	708	1122	0.638	
	3.0	4.32	182	1.439	708	1019	0.477	
	2.0	2.52	107	1.258	708	890	0.321	

TABLE XII

KNOCK-LIMITED COMPRESSION RATIO, WITH AFTERCOOLER

Boost Ratio	1.0	1.5	2.0	2.5	3.0
K-1 density ρ_c	0.542	0.561	0.580	0.589	0.598
lbs air cycle	0.965	0.968	0.97	0.971	0.972
Total cylinder volume, V_1 cu ft/cycle	17.75	11.70	8.81	7.16	6.04
Compression ratio	10.0	6.78	5.27	4.34	3.72

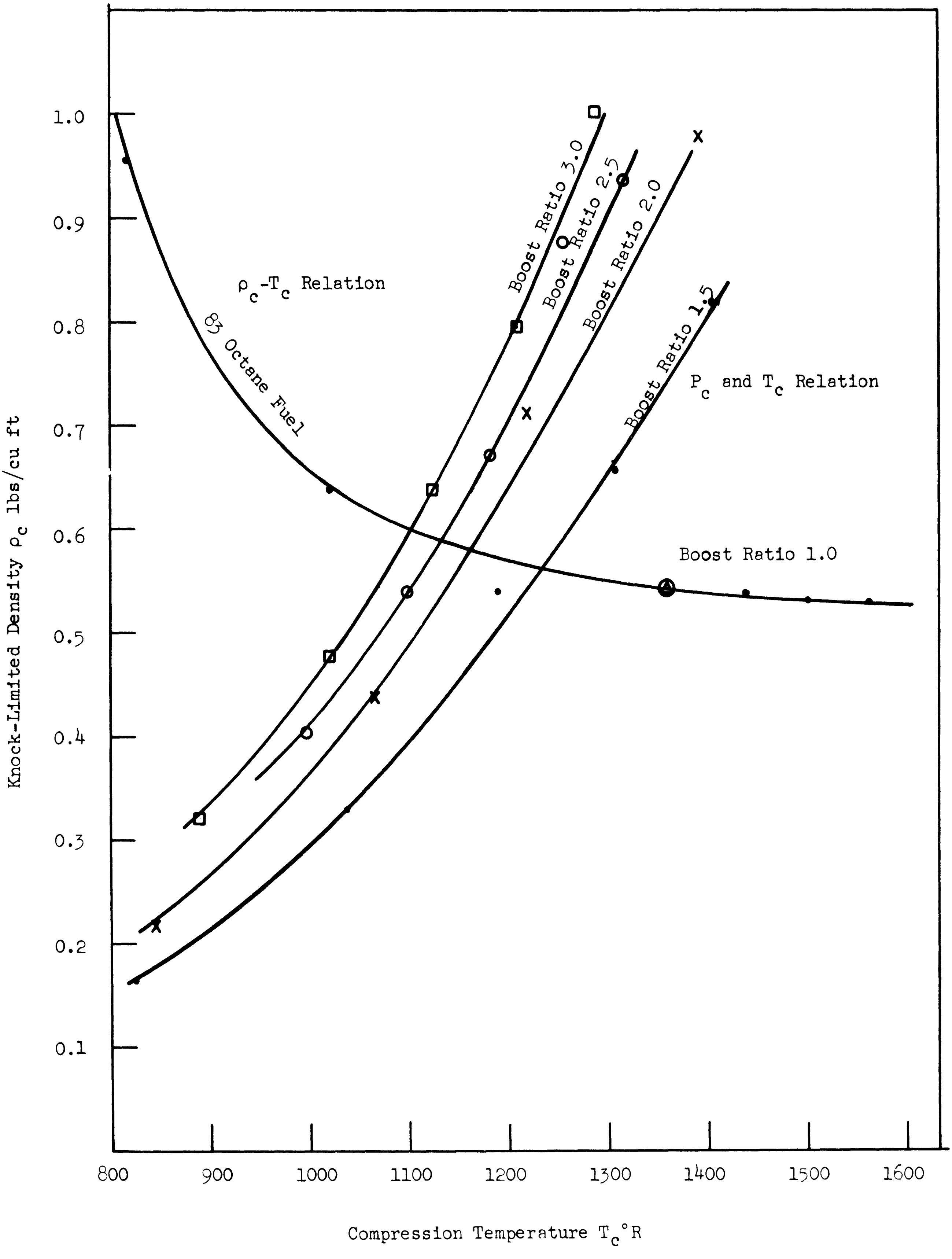


Fig. 14. Knock-limited temperature vs. density relationship with aftercooling.

The points of intersection of the P_c - T_c curves of 83-octane fuel with the K - $L\rho_c$ at different boost are obtained from Fig. 14, and the accompanying compression ratios are recorded in Table XII.

The results obtained by the different methods plotted in Fig. 15 reveal that the greatest difference between systems with and without aftercooling occurs when the end-gas density relationship is employed. The knock-limited condition gives a much smaller spread between the two curves. These effects will be discussed later.

Effect of Fuel-Air Ratio

To provide a better understanding of the overall picture of engine performance, a series of calculations were made for those F/A ratios for which combustion and compression charts were available, and for the manifold pressures given in Fig. 10. The F/A ratios used were F/A = 0.0782 at 3.0:1 boost ratio; 0.0665 at 2.5:1; and 0.0605 at 2.0, 1.5, and 1.0:1 boost ratios. These values represent fairly well the typical operating conditions for an engine at full power and economy performance. The results obtained are listed in Table XIII for constant end-gas density.

TABLE XIII

CONSTANT END-GAS DENSITY FOR PART LOAD PERFORMANCE

Manifold pressure, psia	42.1	34.75	32.71	27.4	22.9	20.1	18.0	15.98	12.7
F/A Ratio	0.0782	0.0665	0.0665	0.0605	0.0605	0.0605	0.0605	0.0605	0.0605
T_1	904	823	809	766	726	697	674	649	626
P_1	42.2	34.75	32.71	27.4	22.9	20.1	18.0	15.98	12.7
Work, Btu/cycle	417	448	416	447	473	475	480	484	527
Thermal Efficiency (%)	28.3	36.0	33.5	395	41.9	42.0	42.6	43.0	46.8
IMEP (psi)	355	325	292	260	249	228	221	198	175
IHP/lb air/sec	607	651	605	652	691	695	703	708	773
SFC/lb/IHP/hr	0.465	0.368	0.375	0.310	0.293	0.29	0.287	0.286	0.261
Compression ratio	4.8	5.25	5.4	6.4	6.9	7.4	7.7	8.4	10.4
P_{max}	1250	1220	1200	1195	1170	1160	1150	1135	1170

With the material now available it is possible to approach data on the performance of an actual engine.

Equivalent L-141 Engine Performance

So far this report has been concerned with the performance of an ideal engine consuming air at the rate of 1 lb/sec. It is now proposed to convert

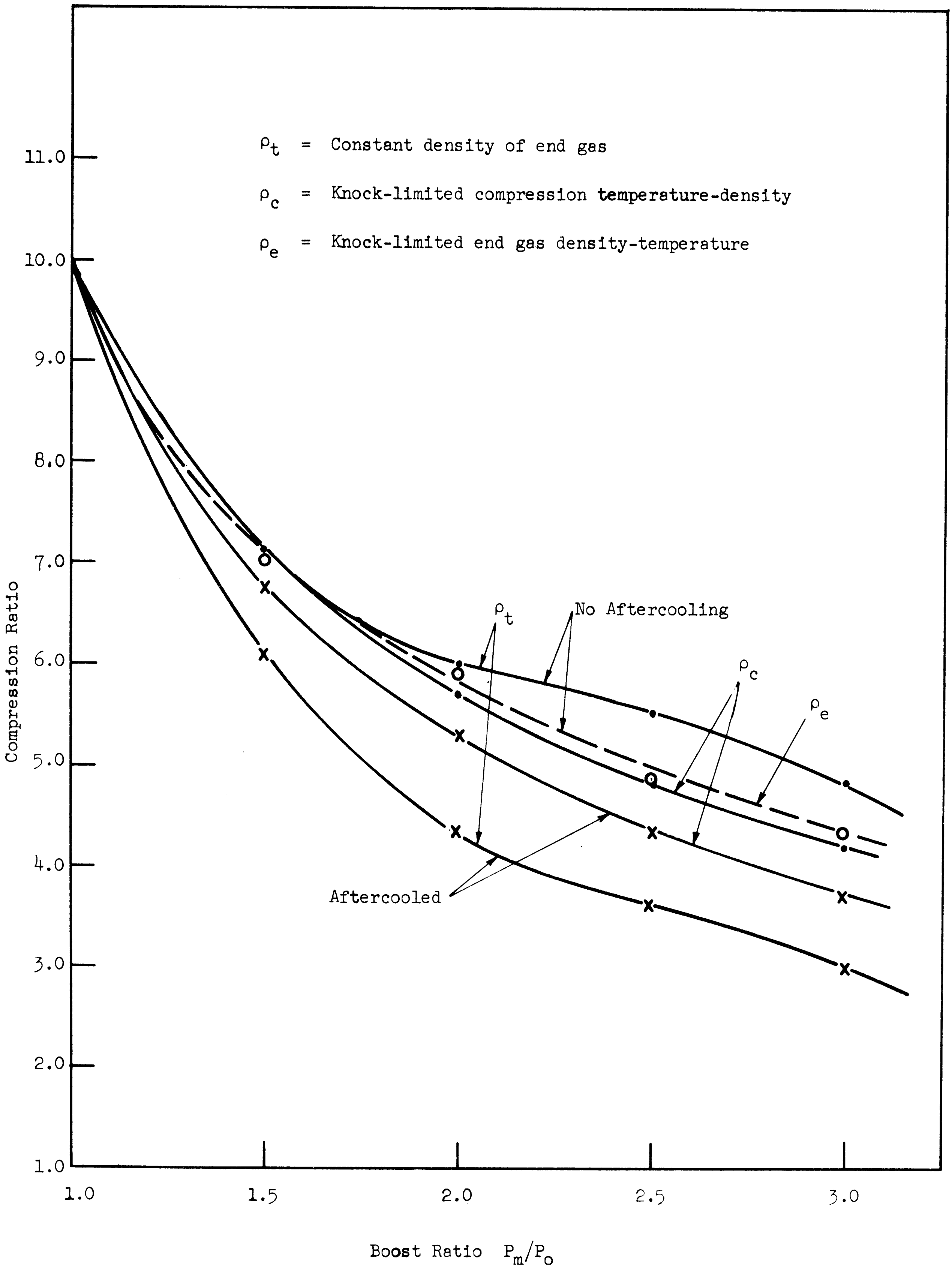


Fig. 15. Compression vs. boost ratio obtained with and without aftercooling.

the data so that the resulting net output is that of the L-141 engine, 61.0 BHP. This can be done by including suitable volumetric efficiency and efficiency ratios in order to obtain a close approximation to the output likely to be developed if an engine equivalent to the L-141 is operated in the proposed manner.

The official Engine Performance Curves for the L-141, issued by the Ford Motor Co. and dated May 13, 1958, give the following data:

rpm	= 3600
gross BHP	= 70.3
Friction hp	- 17.0
IHP	= 87.0
Mech. Effy.	= 80.5%
SFC	= 0.495 lb/gross hp/hr
Compression ratio	= 7.5:1

An ideal cycle analysis with the above compression ratio results in the following data, if the same air cleaner and carburetor resistance as for the other ideal cycles, 2.0 psi, is assumed.

p_1 = 12.7 psia, T_1 = 630°R, V_1 = 18.7 cu ft, E_1 = 22

Ratio = 7.5 ∴ V_2 = 2.49 cu ft, p_2 = 185, T_2 = 1200°R, E_2 = 149

P_3 = 840 psi, T_3 = 4932°R, V_3 = 2.49, E_3 = 1563.5

P_4 = 66, T_4 = 3066, V_4 = 18.7, E_4 = 940

Work of negative loop = -5.98 Btu

Work of cycle = (1563.5 - 940) - (149 - 22) - 5.98

= 490.52 Btu/0.965 lb of air

Heat added = 1415.5 Btu

$$\text{Cycle efficiency} = \frac{490.52 \times 100}{1415.5} = 34.7\%$$

$$\text{IMEP} = \frac{490.52 \times 778}{16.2 \times 144} = 164.0 \text{ psi}$$

$$\text{hp/lb of air/sec} = \frac{490.52 \times 778}{550 \times 0.965} = 719 \text{ hp}$$

$$\text{Fuel flow} = 3600 \times 0.0782$$

$$= 282 \text{ lb/hr}$$

$$\text{SFC} = 282/719 = 0.392 \text{ lb/IHP/hr}$$

When these values are compared with actual engine performance, it is seen that there is a cycle ratio of 0.392/0.495 = 0.792 or 79.2%; this factor includes the effects of volumetric efficiency, heat loss, etc. This value is somewhat lower than that usually expected when the ratio is calculated by the above methods; the difference could be due to poor volumetric efficiency, abnormal heat flow to the jacket due to combustion-surface-to-volume ratio, etc. However, if the value is used in converting the ideal values obtained so far into values for an actual engine system, it is believed that the result will be comparable to the performance of the present L-141 engine, though performance might still be improved.

The aim will now be to predict the size of an engine without aftercooler, which develops 87 IHP at 3600 rpm when supplied with a turbocharger having a boost ratio which appears to be roughly optimum for calculated overall performance. True, the engine will be smaller if aftercooling is employed, but the compression ratio is then so low that the SFC is very large, being 0.892 lb/IHP/hr in place of 0.589 lb/IHP/hr for the straight turbocharged unit. It is believed that the relative value of the proposed control system can be judged from this one set of data without aftercooling. If useful improvements seem apparent, any one or more of the other solutions could then be carried through to a final performance map.

For the engine defined above, Table XIV can be generated from the data in Tables VI and XIII. This conversion is based on the assumption that a 10:1 compression can be achieved in the engine by adjustments of the shape of the combustion chamber, the valve location, the ignition and valve timing, etc., and by the great reduction in the size of the cylinder.

The relative engine size is determined on the basis of two assumptions: that a boost ratio of 2.5:1 is about the highest that can be used without too great a sacrifice in fuel economy, and that the efficiency ratio remains constant at 79.2% over the range of operation.

The ideal cycle at full load, then, is (see Table II)

$$\text{Boost ratio} = 2.5:1$$

$$\text{Thermal efficiency} = 29.9\%$$

$$\text{IMEP} = 300 \text{ psi}$$

TABLE XIV

ESTIMATED ENGINE PERFORMANCE

(Displacement = 80.6 cu in.)

Ratio Boost	Ratio Comp.	rpm	P_m, psi	T_m, °R	F/A	IMEP, psi	SFC, lb/IHP/hr	IHP	FHP	BHP, Gross	SFC, lb/BHP/hr
3.0	4.8	3600	42.2	877	0.0782	265	0.587	97.2	10.5	86.7	0.685
	5.3	2700	33.1	781	0.0782	243	0.554	66.8	6.2	60.6	0.610
	6.6	1750	23.5	686	0.0782	196	0.504	34.9	3.1	31.8	0.553
	10.0	800	13.9	580	0.0782	150	0.435	12.2	1.1	11.1	0.478
	5.2	2700	32.7	810	0.0665	250	0.464	68.8	6.2	62.6	0.509
	6.9	1750	23.2	685	0.0605	198	0.370	35.3	3.2	32.1	0.408
2.5	5.5	3600	34.8	812	0.0782	238	0.555	87.3	10.1	77.2	0.627
	6.0	2700	28.1	706	0.0782	214	0.520	58.9	6.3	52.6	0.583
	6.6	1750	21.5	194	0.0782	194	0.505	34.6	3.1	31.5	0.555
	10.0	800	13.9	580	0.0782	150	0.435	12.2	1.1	11.1	0.478
	5.3	3600	34.8	810	0.0665	250	0.464	91.6	10.1	81.5	0.521
	6.4	2700	27.8	692	0.0665	216	0.435	59.4	6.4	53.0	0.486
	6.9	1750	21.0	685	0.0605	194	0.375	34.6	3.2	31.4	0.413
2.0	6.0	3600	27.4	708	0.0782	212	0.522	77.7	10.2	67.5	0.600
	6.6	2700	23.1	686	0.0782	195	0.504	53.7	6.45	47.2	0.574
	7.9	1750	18.5	623	0.0782	172	0.469	30.7	3.2	27.5	0.523
	10.0	800	13.9	580	0.0782	150	0.435	12.2	1.1	11.1	0.478
	6.4	3600	27.4	730	0.0605	206	0.393	75.5	10.3	65.2	0.454
	6.9	2700	22.9	685	0.0605	198	0.307	54.5	6.5	48.0	0.419
	7.7	1750	18.4	626	0.0605	176	0.363	31.3	3.2	28.1	0.406
1.5	7.1	3600	20.1	675	0.0782	185	0.487	67.8	10.5	57.3	0.575
	7.9	2700	18.1	623	0.0782	171	0.470	47.1	6.6	40.5	0.546
	8.4	1750	16.0	599	0.0782	163	0.458	29.0	3.3	25.7	0.516
	10.0	800	13.9	580	0.0782	150	0.435	12.2	1.1	11.1	0.478
	7.4	3600	20.1	652	0.0605	180	0.366	66.0	10.6	55.4	0.435
	7.7	2700	18.0	626	0.0605	175	0.362	48.3	6.4	42.9	0.407
	8.4	1750	16.0	599	0.0605	157	0.355	28.0	3.3	24.7	0.403
1.0	10.0	3600	12.7	580	0.0782	144	0.432	52.8	10.9	41.9	0.545
	10.0	2700	13.1	580	0.0782	146	0.433	40.3	6.7	33.6	0.519
	10.0	1750	13.5	580	0.0782	148	0.434	26.4	3.4	23.0	0.497
	10.0	800	13.9	580	0.0782	150	0.435	12.2	1.1	11.1	0.478
	10.4	3600	12.7	580	0.0605	138	0.33	50.6	11.0	39.6	0.422
	10.4	2700	13.1	580	0.0605	140	0.335	38.5	6.7	31.8	0.406
	10.4	1750	13.5	580	0.0605	142	0.336	25.3	3.4	21.9	0.388
Throttled	14.0	3600	10.0	580	0.0605	111	0.332	40.6	11.5	29.1	0.463
	14.0	2700	10.0	580	0.0605	111	0.332	30.5	7.0	23.5	0.431
	14.0	1750	10.0	580	0.0605	111	0.332	19.8	3.5	16.3	0.404
	14.0	800	10.0	580	0.0605	111	0.332	9.0	1.2	7.8	0.383
Throttled	19.0	3600	7.5	580	0.0665	91.8	0.328	33.6	12.3	21.3	0.517
	19.0	2700	7.5	580	0.0665	91.8	0.328	25.2	7.5	17.7	0.466
	19.0	1750	7.5	580	0.0665	91.8	0.328	16.4	3.7	12.7	0.422
	19.0	800	7.5	580	0.0665	91.8	0.328	7.5	1.25	6.2	0.397

IHP/lb of air/sec = 641

SFC = 0.44 lb/IHP/hr

Compression ratio = 5.5:1

Actual engine performance expected would then be 79.2% of the above.

Thermal efficiency = 23.7%

IMEP = 237.5 psi

IHP/lb air/sec = 508.0

SFC = 0.555 lb/IHP/hr

Compression ratio = 5.5:1.0

$$\text{IHP} = 87.0 = \frac{\text{Mean pressure x displacement x rpm}}{792000}$$

$$\text{Displacement} = \frac{87.0 \times 792000}{237.5 \times 3600}$$

$$= \underline{\underline{80.6 \text{ cu in.}}}$$

$$\text{Air flow} = \frac{87.0}{508.0} = 0.1712 \text{ lb/sec}$$

$$= \underline{\underline{616.0 \text{ lb/hr}}}$$

Total cylinder volume (4 cylinders) =

= Displacement + clearance volume

$$= 80.6 + \frac{80.6}{5.5-1} = 80.6 + 17.9$$

$$\doteq \underline{\underline{98.5 \text{ cu in.}}}$$

$$\frac{\text{Check on cylinder volume}}{\text{Total charge/hr}} = \frac{616}{0.971}(1 + 0.0782) \text{ lb/hr}$$

$$= 684 \text{ lb/hr}$$

$$\text{Total cylinder volume/hr} = \frac{98.5 \times 1800 \times 60}{1728}$$

$$= 6160 \text{ cu ft/hr}$$

$$\text{Weight of charge/hr} = \frac{P_1V_1}{RT}$$

$$= \frac{34.7 \times 144 \times 6160}{53.34 \times 845}$$

$$= \underline{\underline{683 \text{ lb/hr}}}$$

It is seen that the proposed displacement will handle a total mass flow within 1 lb/hr of that estimated. It is now possible to obtain the IHP, air flow, fuel flow, etc., from the ideal cycle, IMEP, for each of the manifold conditions shown in Fig. 10. These data are given in Table XIV.

The frictional horsepower employed was estimated from Fig. 16, where the FMEP of the standard engine is plotted on an rpm base. It is assumed that the same FMEP will be lost in a similar engine at the same speed regardless of any change of displacement, i.e., that the FHP is proportional to displacement. Then with the VCR pistons there is also an accompanying change of ratio and thus cycle pressures, plus of course the effect of supercharging raising the pressures throughout the cycle. These effects have been covered by the correction factor shown in Fig. 16, plotted on a ratio basis.

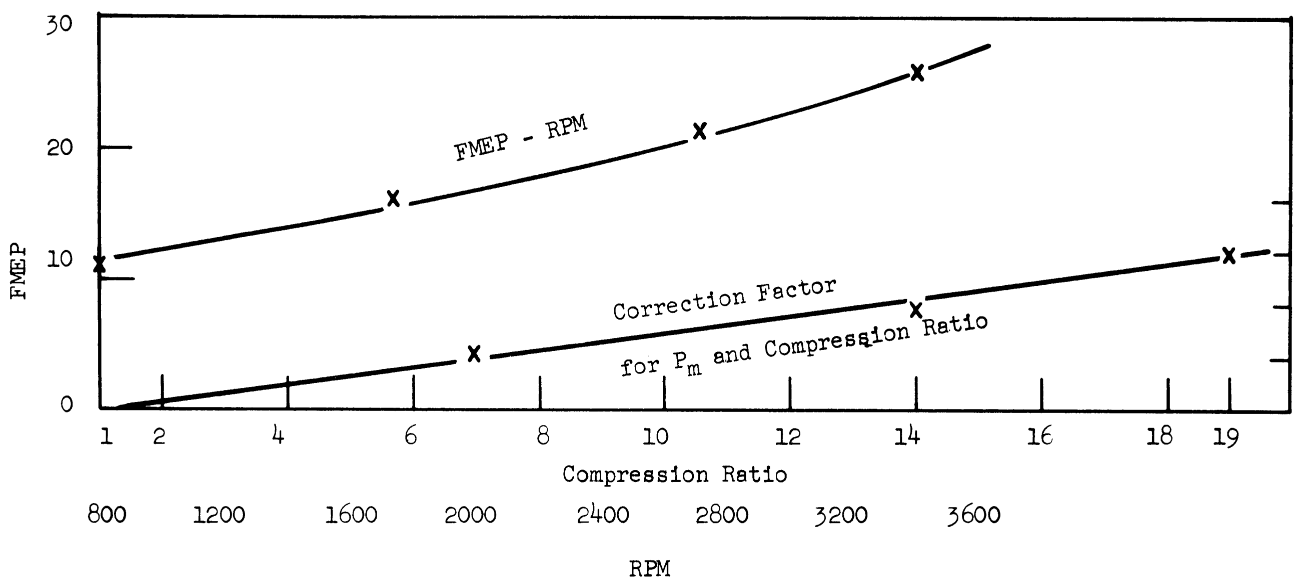


Fig. 16. Frictional losses and correction factors for 80.6 cu in. engine.

As an example, let us calculate the FHP of the 80.6 cu in. engine at 2700 rpm and 33.1 psi manifold pressure. Here

Standard engine FMEP = 21.1 psi at 7.5:1 ratio

Compression ratio of supercharged engine = 5.3:1

Correction factor (Fig. 16) = 1.06

FMEP required = 1.06 x 21.1 = 22.4 psi

$$\text{FHP} = \frac{\text{FMEP x Disp x rpm}}{792000}$$

= 6.12 hp

FHP (Fig. 16) of 141.5 cc engine = 10.2 hp

By this procedure the estimated gross BHP, SFC, and lb/BHP/hr for the proposed 80.6 cu in. engine have been calculated. They are shown in Table XIV, and plotted in Figs. 17 to 19. Figure 17 shows BHP and compression ratio versus boost ratio for various rpm. The BHP curves represent an averaging of the points in Table XIV; i.e., the various F/A ratios are included. The result is that the BHP curves shown represent a typical performance of an engine fitted with a carburetor which delivers an F/A ratio of 0.0782 at full load and speed, and leans out to 0.0665 at about 2700 and to 0.0605 at 1750; then the mixture is enriched to 0.0782 at 800 rpm. The compression ratio curves in Fig. 17 indicate this ratio by solid lines for F/A of 0.0782 and by broken lines for the leaner mixture.

When the boost ratio of 1.0:1 is reached, output must be reduced by throttling. This condition is represented in Fig. 17 by the broken line labeled "throttled" with its scale on the left-hand side; compression ratio is seen to increase rapidly up to 19:1. In view of the wide range of compression ratios involved, from 4.8 to 19.0, it is believed that the practical problem of the design of the VCR piston will limit the range that can be used to perhaps 4.8 to 15.0:1.0 approximately; this factor will be neglected in this report.

In Figs. 18 and 19, the solid line shows the values for a constant F/A ratio of 0.0782 (rich mixture) from 3600 to 800 rpm, and the broken one shows how the SFC will change if the carburetor gives an F/A varying from 0.0782 at 3600 rpm to 0.0665 at 2700 to 0.0605 at 1750 and back to the rich mixture of 0.0782 at idle at 800 rpm. Since the engine is fitted with a VCR piston, the compression ratio also varies as the F/A ratio; thus the broken lines in these diagrams give the combined effect of these two variables.

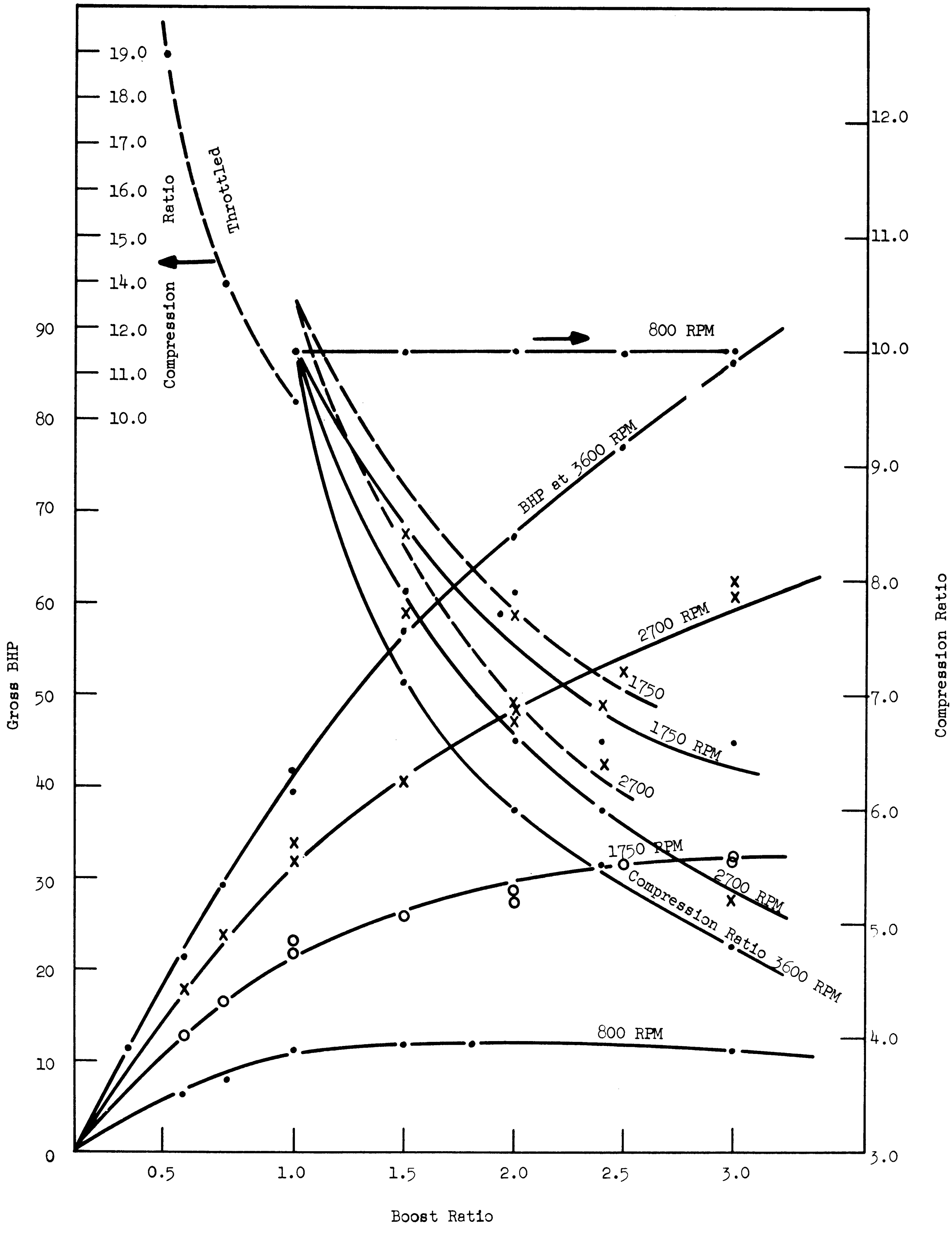


Fig. 17. Gross BHP for 80.6 cu in. engine.

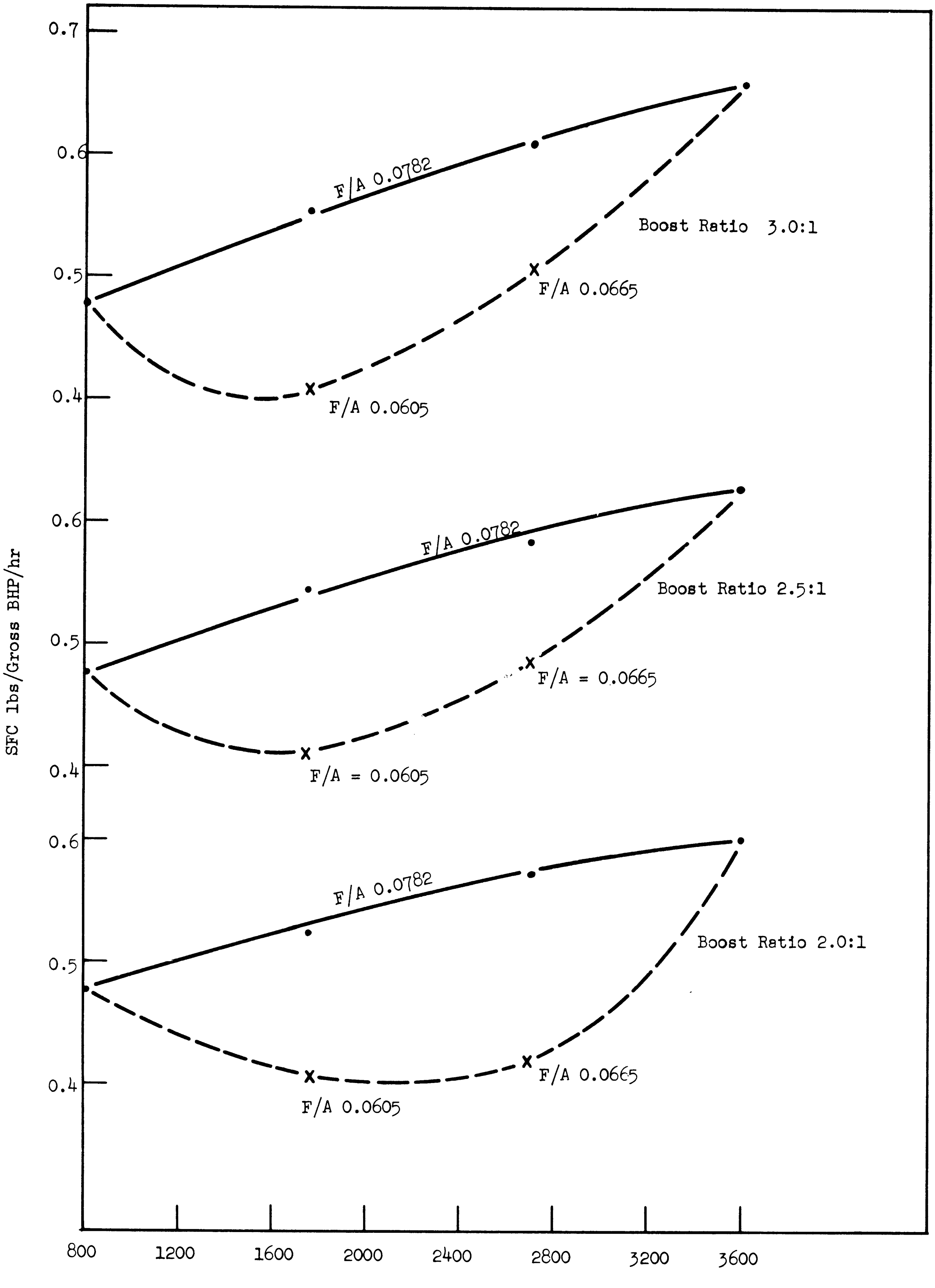


Fig. 18. SFC curves for 80.6 cu in. engine at different F/A.

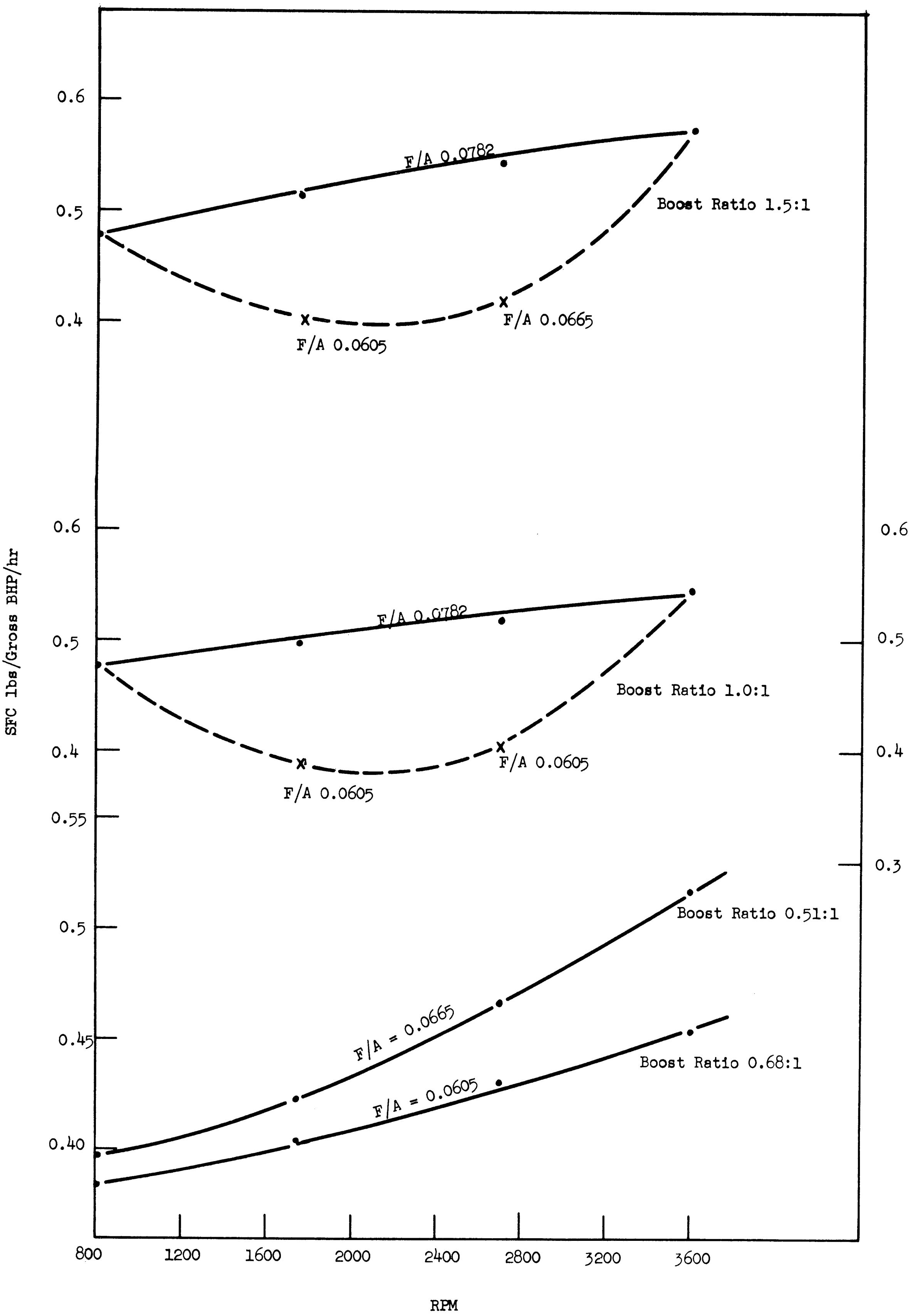


Fig. 19. SFC curves for 80.6 cu in. engine at different F/A.

When the engine has to be throttled below the boost ratio of 1.0:1, F/A has been assumed to be constant at 0.0605 for a boost ratio of 0.68:1, and at 0.0665 for a ratio of 0.51:1. Hence only one line is given for these two conditions.

Turbocharger With Aftercooling

For the system with aftercooling, as Fig. 15 shows, the permissible compression ratio is less than in the system considered above regardless of the method used. Also, Figs. 5b and 6b show how the maximum cylinder pressure changes with the boost ratio.

From the data in Fig. 5b it is seen that the maximum cylinder pressure increases from 1180 to 1250 psi as the boost ratio changes from 1.0 to 3.0:1. By the VCR principle this seems to be a fairly good characteristic for satisfactory operation of the variable-ratio piston. The piston motion would be under control at all times, so that an increase in load would make it tend to shift to a lower ratio at which it would be stable.

The broken line on the same diagram represents P_{max} for the knock-limited end-gas density assumption. Here again conditions are stable, provided that the range of possible ratios designed into the piston extends from 4.0 to 10.0:1, and from about 4.0 to 15.0:1, if at all possible, for throttled operation.

Examination of Fig. 6b shows that, when an aftercooler is used, maximum pressure decreases as load increases when the constant end-gas density method is used, and remains roughly constant when knock-limited temperature-density is the deciding factor. Reduction in pressure would produce a definitely unstable condition, since the VCR piston produces an essentially constant pressure. Hence the movable head would be forced down to a lower ratio by a reduced pressure rather than by an increased one. Under the effects of inertia, the ratio would increase to the point of detonation, which would force the piston down to the ratio desired for elimination of knock. But under these conditions the pressure is less than that for which the piston must be set for the naturally aspirated engine with a compression ratio of 10.0:1. The compression ratio will therefore immediately increase and induce detonation again, so that there will be a continual hunting between the ratios of 10:1.0 and 3.0:1. In addition, as already pointed out, the operating ratio with aftercooling is about 3.0:1 in any case, so that fuel economy is very poor.

The maximum pressure with aftercooling, determined by the knock-limited temperature-density condition and shown as a broken line in Fig. 6b, would be a satisfactory stable operating condition. It follows that the correct conditions determining detonation are essential if an aftercooler is used.

For the above reasons the aftercooling condition has not been examined in complete detail.

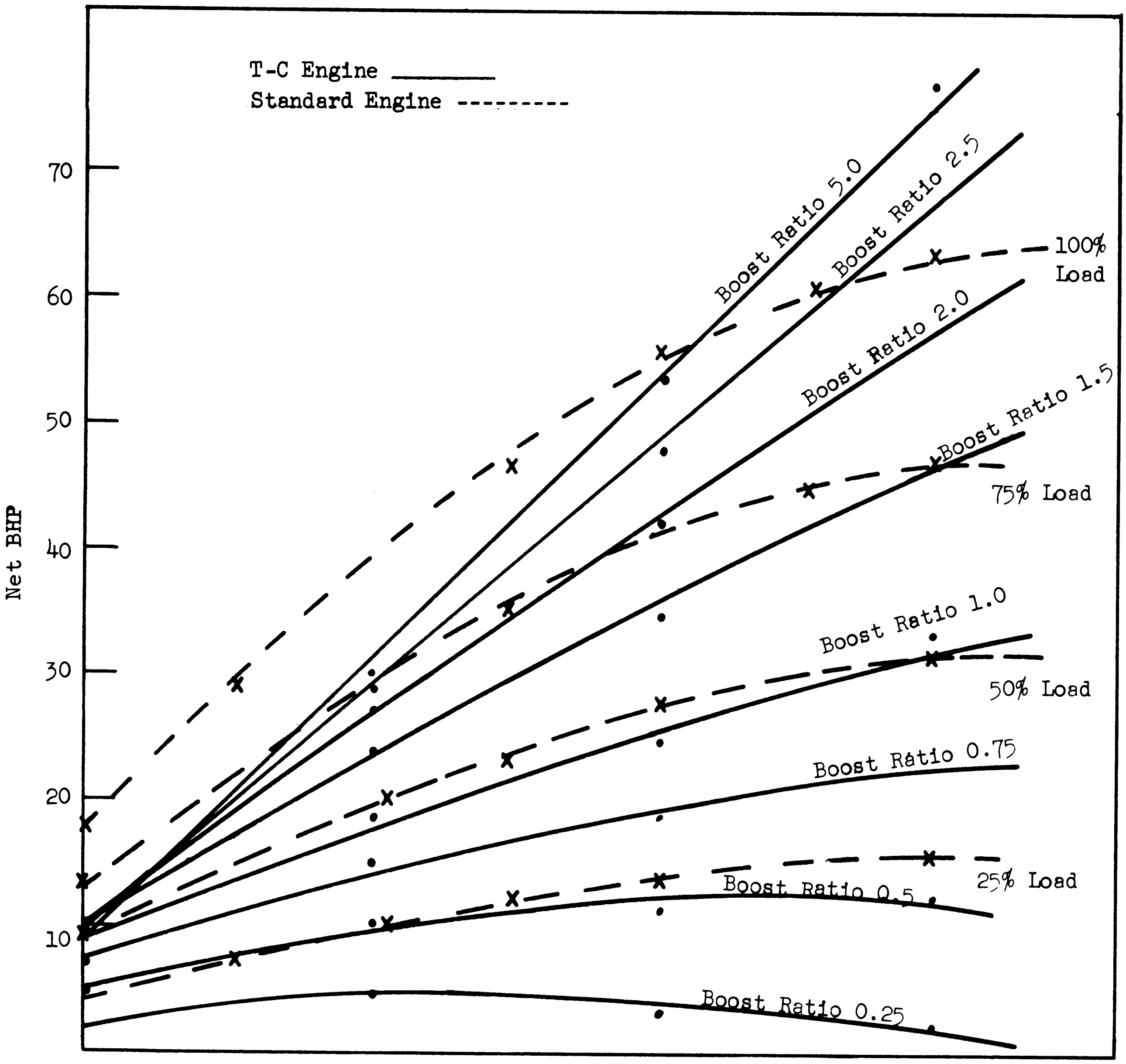
Comparison With Standard Engine

The performance curves available for the standard engine are based upon net output, which includes the power used by the fan, the generator, and exhaust system. To compare the present calculations with the available data, the losses resulting from these items are obtained from the official engine performance curves of the Ford Motor Company, dated May 13, 1958; they are as follows:

rpm	3600	3200	2800	2400	2000	1600	1200	800
Horsepower loss due to accessories	9.2	8.0	6.5	4.7	3.0	2.5	2.1	2.0

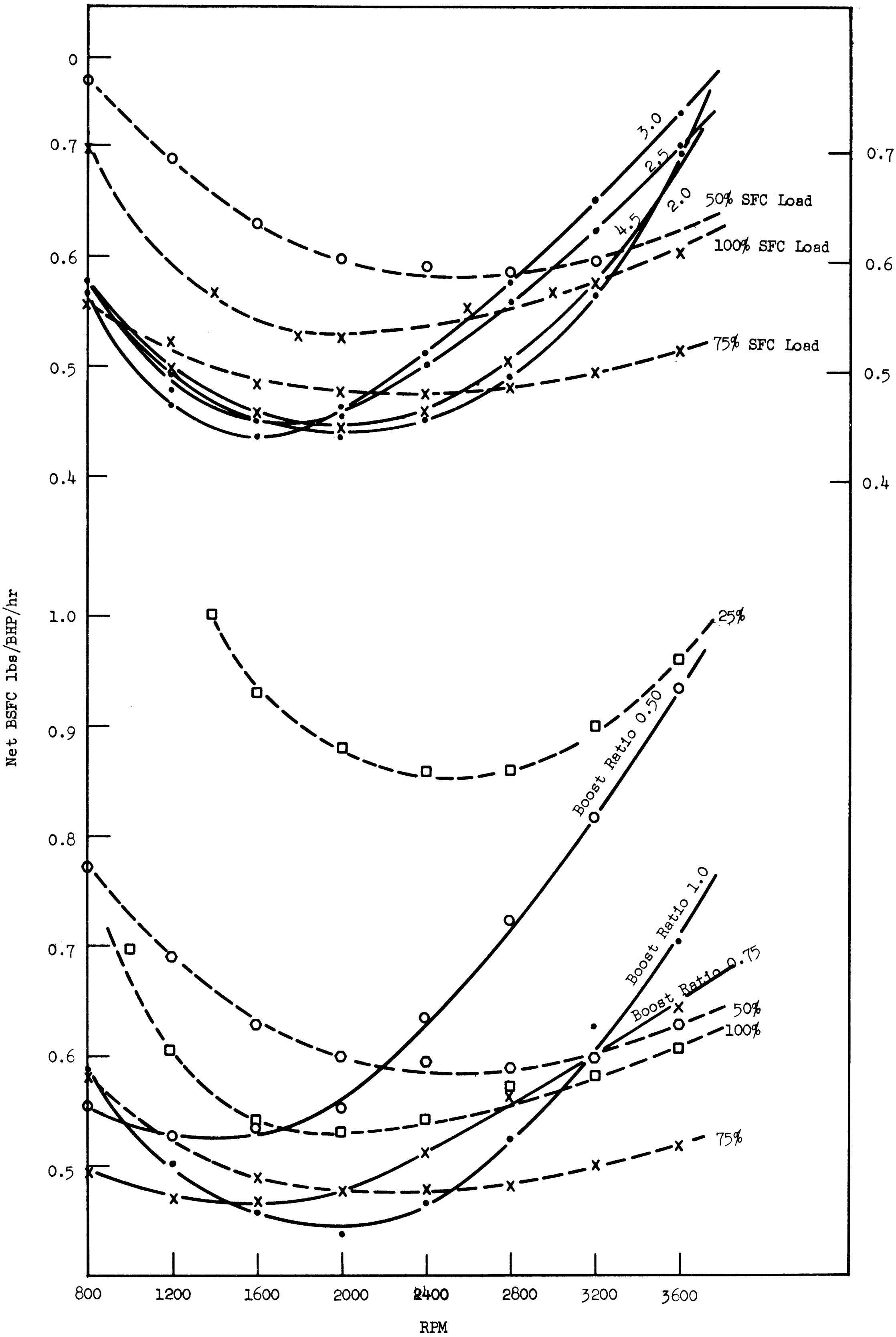
When the estimated gross hp preformance is corrected accordingly, the results are as shown plotted in Fig. 20 to which have been added, as broken lines, the 100%, 75%, 50%, and 25% load curves of the standard engine, taken from Ref. 3. By using these curves, the data were converted into a fuel flow versus net hp diagram, given in Fig. 21, on which have been shown, again as broken lines, similar curves from Ref. 3, for 3600, 2200, and 1000 rpm for the standard engine. It is at once apparent that performance for a given speed and horsepower has been improved at all low speeds, with some small sacrifice at full load and speed. This is to be expected, since the turbocharging for power variation with fuel of a given octane value has reduced the compression ratio at heavy loads and increased it at light loads. Another major improvement is that the substitution of a 80.6 cu in. engine having the same output as the 141.5 cu in. standard one reduces the frictional losses, which have a greater influence upon the SFC at light loads than at heavy ones.

This overall reduction in fuel flow can be seen from Table XV, which gives figures for standard and turbocharged engines installed in M-151 vehicles with the same transmission and operated on pavement in fourth gear and also on soft ground with twice the resistance of a paved surface. It is plain that for a vehicle operating during the 48-hr battlefield day of Ref. 3, when the engine idles during 10% of its operating time moves at 30 mph or less during 80% of that time, and moves at speeds above 30 mph during only 11% of the time, fuel flow will be materially reduced.



(a) BHP

Fig. 20. Net performance. (a) BHP. (b) SFC.



(b) SFC

Fig. 20. (Concluded)

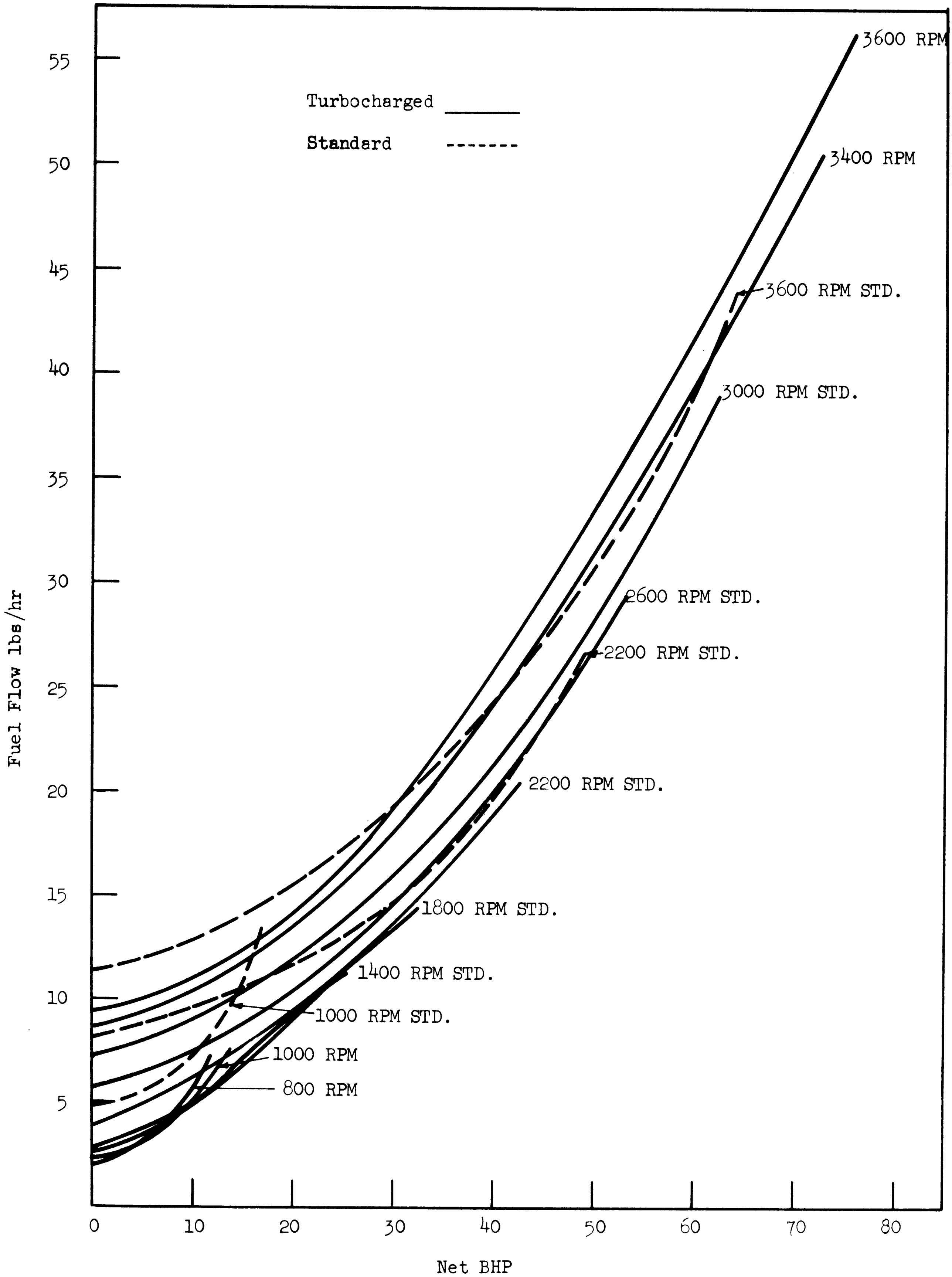


Fig. 21. Fuel flow vs. net BHP for turbocharged engine without aftercooler.

TABLE XV

COMPARISON OF STANDARD AND TURBOCHARGED ENGINES ON PAVEMENT
IN FOURTH GEAR AND ON SOFT GROUND

Speed: Vehicle, mph	Speed: Engine, rpm	Net hp: Pavement	Net hp: 2 x Pavement	Fuel Flow, lb/hr: Pavement Std.	Fuel Flow, lb/hr: Pavement T.C.	Fuel Flow, lb/hr: 2 x Pavement Std.	Fuel Flow, lb/hr: 2 x Pavement T.C.
20	1210	3.2	6.4	5.8	2.8	6.5	3.9
30	1820	8.3	15.9	8.7	4.5	10.3	7.3
40	2420	16.2	32.4	12.0	8.4	16.4	15.2
50	3030	30.0	59.6	17.1	11.7	38.0	37.0
60	3630	48.0	95.6	27.4	32.0	--	--
Idle	800	0	0	4.0	2.0	4.0	2.0

METHOD 2b

It remains to determine whether the method based on the density-temperature relationship of the end gas will give results significantly different from those obtained by the method based on the relationship between compression density and compression temperature. This question will be examined, first, with respect to the system with no aftercooling, since in the cycle of this system the large range of inlet temperatures can be expected to give the maximum variation in compression temperatures.

The method can be represented by the diagrams in Figs 22 and 23. The

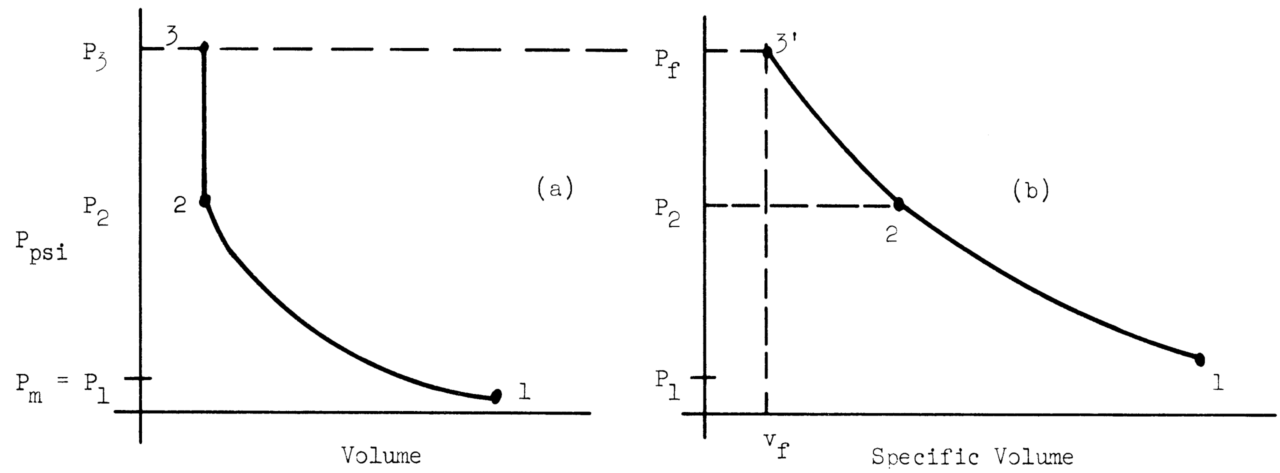


Fig. 22. End gas temperature vs. density relationship.

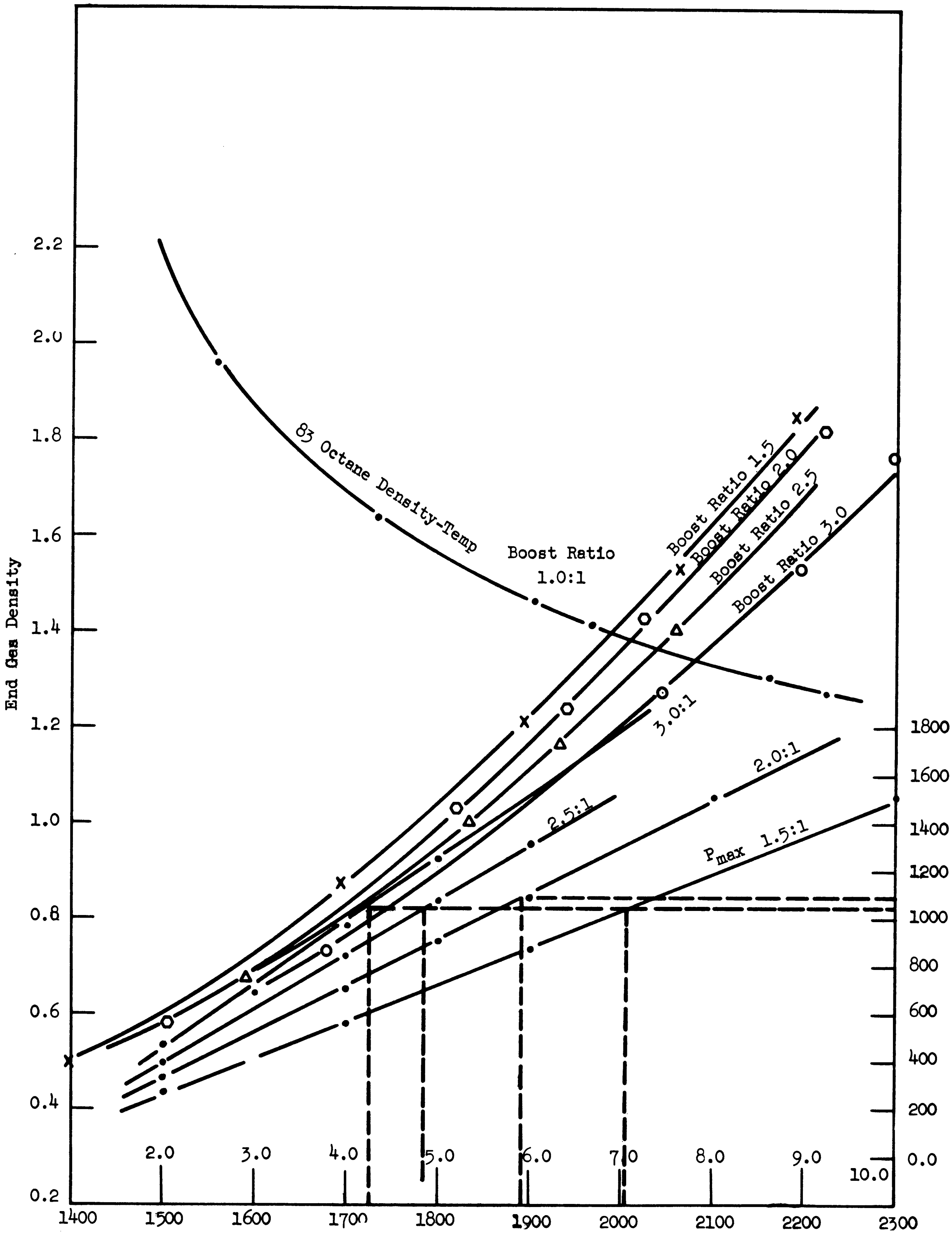


Fig. 23. Knock-limited end-gas temperature vs. density for 83-octane fuel.

former represents the P - V curve of compression and combustion plotted on a normal displacement basis, and the latter is a specific volume plot on which end-gas conditions can be shown to advantage. In these diagrams P_1 is the pressure at point 1, which is also equal to the manifold pressure, P_2 is the compression pressure due to the engine compression ratio; P_3 is the maximum firing pressure due to combustion; and $P_f (= P_3)$, V_f, are the pressure and the specific volume respectively, of the end gas at temperature T_f which has been compressed by the combustion process along the line 2-3' of Fig. 22b and in which no combustion has taken place.

The problem of the density-temperature conditions of the end gas at state P_f T_f can be solved by using the perfect gas laws as in Ref. 2. Average values for the ratio of the specific heats k are determined from the data extracted from the combustion charts for the various cycles solved by the chart method. By this means, if k = 1.35, the following equations are obtained:

$$P_2 = P_1(R)^k \tag{6}$$

$$P_f = P_3 = P_1(R_f)^k \tag{7}$$

where R_f = Final ratio of compression of the last element to burn

R = Compression ratio of engine = V_1/V

Heat of combustion = $H = wC_r(T_3 - T_2)$

$$T_3 = \frac{H}{wCv} + T_2 \tag{8}$$

Equation (7) can be written as

$$P_f = P_1(R)^k \times T_3/T_2$$

Substitute for T_3 from Eq. (8):

$$P_f = P_3 = P_1(R)^k \frac{\left(\frac{H}{wCv} + T_2\right)}{T_2}$$

Now the variation of the value of H/wCv, as determined by the combustion charts for all of the cycles with F/A = 0.0780, is from 3770 to 3790. Using a constant value of 3780 results in an error of only ±0.4%, which is negligible in relation to the other assumptions made. With this value for H we get

$$P_f = P_3 = P_1(R)^k \frac{3780 + T_2}{T_2}$$

But

$$T_2 = T_1(R)^{k-1}$$

Substituting in the above gives

$$P_f = P_3 = P_1(R)^k \left\{ \frac{T_1(R)^{k-1} + 3780}{T_1(R)^{k-1}} \right\}$$

Substituting from Eq. (7) for P_f gives

$$P_1(R_f)^k = \frac{P_1}{T_1}(R) \left\{ T_1(R)^{k-1} + 3780 \right\}$$

$$(R_f)^{1.35} = \frac{R}{T_1} \left\{ T_1(R)^{0.35} + 3780 \right\} \tag{10}$$

By Eqs. (6), (9), and (10), the data of Table XVI can be used in calculating the cycle of the basic standard engine having 10:1 ratio and using 83-octane fuel. As in Method 2, the density-temperature curve along which the engine must operate, shown in Fig. 23 and Table XVI, can be determined from Fig. 12.

TABLE XVI

KNOCK-LIMITED CURVE FOR 83-OCTANE FUEL

Compression Ratio	Knock-Limited Manifold Pressure	T_1	T_2	T_3	R_f	T_f	P_f	ρ_f
16	7.62	620	1637	5417	38.7	2230	1046	1.274
14	8.79	624	1571	5351	34.7	2160	1053	1.302
12	10.39	628	1500	5280	30.5	2080	1049	1.345
10	12.70	632	1417	5197	26.3	1985	1045	1.406
8	16.32	636	1319	5099	22.9	1903	1046	1.467
6	22.85	640	1198	4978	17.2	1732	1065	1.64
4	38.1	644	1047	4827	12.4	1556	1144	1.96
2	114.3	648	826	4606	7.15	1290	1626	3.365

To locate the actual operating points with the varying boost ratios and initial temperatures, a series of compression ratio calculations similar to the example below was made for each boost ratio, and the results are recorded in Table XVII. Plotting these points on Fig. 23 gives the common point of operation satisfying the two conditions; from this point the compression ratio can be calculated.

TABLE XVII

KNOCK-LIMITED END-GAS DENSITY FOR VARIOUS BOOST RATIOS

Ratio Boost	Ratio Comp.	T_1	T_2	T_3	P_2	P_3	R_f	T_f	ρ_f
1.0	10.0	632	1417	5197	284	1045	26.3	1985	1.406
	10.0	716	1604	5384	450	1512	24.5	2190	1.854
	8.0	716	1484	5264	333.5	1185	20.4	2062	1.534
1.5	6.0	716	1340	5120	222.5	862	16.08	1895	1.215
	4.0	716	1144	4924	129.5	558	11.7	1695	0.878
	2.0	716	912	4692	51.22	263.5	6.73	1397	0.504
	8.0	784	1625	5405	454.5	1511	19.5	2220	1.817
	6.0	784	1468	5248	307.0	1098	15.5	2045	1.434
2.0	5.0	784	1377	5157	241.0	903	13.3	1940	1.245
	4.0	784	1274	5054	177.8	705	11.1	1820	1.035
	2.0	784	999	4779	69.8	334	6.38	1505	0.592
	6.0	845	1580	5360	389	1320	14.8	2170	1.625
	5.0	845	1485	5265	305	1082	12.76	2058	1.405
2.5	4.0	845	1374	5154	225	844	10.6	1931	1.167
	3.0	845	1240	5020	170.5	691	9.18	1833	1.006
	2.0	845	1077	4857	88.6	399	6.10	1590	0.672
	6.0	904	1690	5470	472	1529	14.3	2296	1.777
	5.0	904	1585	5360	371	1257	12.4	2195	1.53
3.0	4.0	904	1467	5247	273.5	978	10.25	2042	1.279
	3.0	904	1326	5106	207.0	798	8.8	1935	1.102
	2.0	904	1150	4930	107.5	461	5.86	1680	0.732

Example 1

Calculate the temperature and density for the basic engine cycle having a compression ratio of 8.0:1 and an inlet temperature of 636°R.

Figure 12 gives K - L manifold air pressure as 16.32 psi. Therefore

$$T_2 = T_1(R)^{k-1} = 636 \times 8^{0.35}$$
$$= 636 \times 2.07$$
$$= 1319°R$$

$$T_3 = T_2 + 3780$$
$$= 1319 + 3780$$
$$= 5099°R$$

From Eq. (10),

$$R_f^{1.35} = \frac{R}{T^1}\left[T_1(R)^{0.35} + 3780)\right]$$

$$= \frac{5099 \times 8.0}{636} = 64.05$$

$$R_f = 22.9$$

$$T_f = T_1(R_f)^{0.35} = 636 \times 2.99$$
$$= 1903$$

$$P_f = P_1(R_f)^{1.35} = 16.32 \times 64.05$$
$$= \underline{\underline{1046}}$$

$$\text{End-gas density} = \frac{144 \times P_f}{53.99 \times T_f}$$

$$= 2.67 \times \frac{1046}{1903}$$

$$= \underline{\underline{1.467}}$$

Example 2

Calculate the end-gas density and pressure for the case in which the boost ratio is 1.5:1, the inlet temperature is 716°, and the compression ratio is 8.0:1.

$$P_1 = 14.7 \times 1.5 - 2.0$$
$$= 20.1 \text{ psi}$$

$$T_2 = T_1(R)^{0.35} = 716 \times 8^{0.35}$$
$$= 716 \times 2.07$$
$$= \underline{\underline{1484°R}}$$

$$T_3 = T_2 + 3780$$
$$= \underline{\underline{5264°R}}$$

$$P_2 = P_1(8)^{1.35}$$
$$= 20.1 \times 16.6$$
$$= \underline{\underline{333.5 \text{ psi}}}$$

$$P_3 = P_f = P_2 \times T_3/T_2$$
$$= 333.5 \times \frac{5264}{1484}$$
$$= \underline{\underline{1185 \text{ psi}}}$$

$$P_f = P_1(R_f)^{1.35}$$

$$R_f = (P_f/P_1)\ 1/1.35$$
$$= (1185/20.1)\ 1/1.35 = (58.9)\ 1/1.35$$
$$= \underline{\underline{20.4}}$$

$$T_f = T_1(R_f)^{0.35}$$
$$= 716 \times 2.88$$
$$= \underline{\underline{2062°R}}$$

$$\text{End-gas density} = 2.67 \times P_f/T_f$$
$$= \underline{\underline{1.534}}$$

$$T_2 = T_1(R)^{k-1} = 636 \times 8^{0.35}$$
$$= 636 \times 2.07$$
$$= 1319°R$$

$$T_3 = T_2 + 3780$$
$$= 1319 + 3780$$
$$= \underline{\underline{5099°R}}$$

By Eq. (10),

$$(R_f)^{1.35} = \frac{R}{T_1}\left[T_1(R)^{0.35} + 3780\right]$$

$$= \frac{5099 \times 8.0}{636} = 64.05$$

$$R_f = \underline{\underline{22.9}}$$

The boost-ratio curves intersect with the 83-octane one at the values given in Table XVIII, where the desired values of ρ_f and T_f are obtained. Then, the value of ρ_f can be obtained by

TABLE XVIII

COMPRESSION RATIO FOR END-GAS TEMPERATURE-DENSITY CONDITION WITH NO AFTERCOOLER

Boost Ratio	ρ_f	T_f	P_f	Compression Ratio
1.0	1.406	1985	1045	10.0
1.5	1.4	1987	1043	7.05
2.0	1.385	2010	1043	5.92
2.5	1.37	2040	1046	4.85
3.0	1.345	2082	1049	4.3

$$P_f = \frac{\rho_f \; T_f}{2.67} \tag{11}$$

By plotting P_f versus R for each boost and compression ratio, as shown in Fig. 23, the desired compression ratio for the knock-limited density of the cycle is obtained; it is shown in the diagram, and the important data are recorded in Table XIX.

TABLE XIX

COMPRESSION RATIO AND BOOST RATIO FOR END-GAS KNOCK-LIMITED CONDITION WITH NO AFTERCOOLER

Boost Ratio	P_m	T_m	P_{max}	ρ_e	Compression Ratio
1.0	12.7	630	1045	1.406	10.0
1.5	20.1	716	1043	1.4	7.05
2.0	27.4	784	1043	1.385	5.93
2.5	34.7	845	1046	1.37	4.85
3.0	42.2	904	1049	1.345	4.25

This compression ratio is plotted as a broken line in Fig. 5a. It is seen to be almost identical with the results produced by the constant density method.

VI. DISCUSSION

In this discussion the results derived when using a mechanically driven blower will be neglected, because of the reduction in power and economy resulting from the power absorbed by the blower.

A comparison of the data in Tables II and IV shows that at a boost ratio of 2.5:1, with and without aftercooling, the ratios of the outputs are 641 hp/lb air/sec at $P_m = 34.7$ psi and $T_m = 812°R$ without cooler, and 535 hp/lb air/sec at 34.7 psi and 693°R with cooler.

$$\text{Specific volume without cooler} = RT_m/P_m$$
$$= \underline{\underline{8.67}} \text{ cu ft/lb}$$

$$\text{Specific volume with cooler} = 53.34 \times 693/144 \times 347$$
$$= 7.4 \text{ cu ft/lb}$$

It follows that the 80.6 cu in. engine without aftercooling would have to be reduced to approximately 69.0 cu in. to develop the desired net BHP of 61.0, or 17.25 cu in. per cylinder in a four-cylinder engine. This displacement would require a cylinder diameter of approximately 2-3/4 in. if a square engine was employed. The size is becoming very small, but the fuel consumption is increasing rapidly because of the reduction in compression ratio, there being a 20% increase in the fuel required in the ideal engine cycles which would mean a brake specific fuel consumption approaching 0.8 lb/BHP/hr at full load. The turbocharged engine without cooler was taken as the reference point.

Figure 21 presents a comparison of the standard engine and the proposed turbocharged one. It is seen that the major economics are made at low vehicle speeds and low power, a condition which is appropriate to the type of operation involved. At the same time the loss in economy at high power and speed is rather small; however, the gains more than offset the losses. From Fig. 20a it will be observed that in the range from 800 to about 2400 rpm the standard engine will develop a few more hp at full throttle than will the supercharged one. This is the result of the charge pressure falling with engine speed because of reduction of gas flow. As a result the time of acceleration of a fully loaded vehicle will be somewhat longer with a turbocharger. It is possible that the use of variable geometry in the charger, which will be needed anyway, can offset this deficiency to a major extent.

The compression ratios calculated for the different engines studied are diagrammed in Fig. 15. It is seen that the general trends are similar but that the individual curves show differences amounting to about 0.5 to 0.75 of a ratio, depending on the methods. In view of the widely different assumptions made, this is considered to be good agreement between results; however, a difference of say 0.5 in the ratio brings about a major change in the thermodynamic performance.

Most of the work was devoted to the system without aftercooling because of the higher ratio it requires which gives better performance at high boost and because it procuces pressures suitable for the use of a VCR piston.

VII. RECOMMENDATIONS

The work that produced the above results took longer than was expected. As a result, some short cuts have been employed for the sake of a coherent description.

This work is presented for use by the staff of the Power Plant Laboratory in deciding whether any of the schemes discussed would be useful to the Army. If so, one or more of the analyses herein could be extended to give predictions of the performance of all the components of the system—engine, turbo, and controls. It would then be possible to determine the range over which the turbo could be operated successfully, and to make performance maps of power output, fuel consumption, and other properties as needed for a complete and detailed description of the performance of the system.

VIII. REFERENCES

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