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RESEARCH MEMORANDUM

CALCULATIONS OF THE PERFORMANCE OF A COMPRESSION-IGNITION ENGINE-COMPRESSOR TURBINE COMBINATION

II - PERFORMANCE OF COMPLETE COMBINATION

By Alexander Mendelson

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WASHINGTON

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SUMMARY

Calculations based on test data taken on a single-cylinder compression-ignition engine with a compression ratio of 13.1 and an engine speed of 2200 rpm were made to determine the performance at sea-level conditions of a compression-ignition engine geared together with a compressor and a turbine. The maximum cylinder pressure was assumed constant at 1400 pounds per square inch and the effects of fuel-air ratio, compression ratio, exhaust back pressure, and engine speed on the performance of the combination were determined. The analysis indicated that the net specific power output increased with decreasing compression ratio and increasing fuel-air ratio and engine speed. At an engine speed of 2200 rpm and compressor and turbine efficiencies of 0.70 and 0.65, respectively, a minimum net specific fuel consumption of approximately 0.40 pound per net horsepower-hour was obtained. Increasing the compressor and turbine efficiencies to 0.85 decreased the minimum net specific fuel consumption to approximately 0.32 pound per net horsepower-hour. Decreasing the engine speed to 1200 rpm decreased the minimum net specific fuel consumption to 0.37 pound per net horsepower-hour when the compressor and turbine efficiencies were 0.70 and 0.65, respectively. Comparison with a compression-ignition engine using a turbosupercharger showed that little could be gained by gearing the turbine to the engine, provided the turbosupercharger could be stably operated with a closed waste gate.

INTRODUCTION

The use of jet-propelled aircraft has aroused great current interest in systems for aircraft propulsion that utilize gas turbines as the prime movers. (See references 1 and 2.) One of the principal limitations in the use of gas turbines in the energy-conversion process is the lack of materials that withstand gas temperatures above approximately 1600° F. In order to maintain the temperature below this limit, a large amount of excess air must be supplied, which decreases the energy per pound of gas that can be extracted by the turbine and consequently seriously reduces the net power output of the system.

A dual power unit consisting of a highly supercharged compression-ignition engine geared to a turbine has been proposed. (See references 1, 3, and 4.) The compression-ignition engine can operate at high pressures without the occurrence of knock and preignition and in the design of the engine some control of the temperature of the gas entering the turbine can be effected by varying the fuel-air ratio of the mixture burned in the engine. The design operating conditions of the system can be so adjusted that the turbine, supplied with the exhaust gas from the engine, will produce a large proportion of the power.

Test data on the performance characteristics of a highly supercharged compression-ignition engine, as well as information for the calculation of release temperatures and pressures, are presented in reference 3. At mixtures richer than stoichiometric the energy in the exhaust gas of a compression-ignition engine is shown to be in excess of that needed for supercharging. No attempt is made in reference 3 to estimate the performance of the engine-turbine combination.

The calculations presented herein were therefore made to determine the performance of a compression-ignition engine, a turbine, and a compressor geared together. The calculations are based on the data presented in reference 3. No data as to the effect of exhaust back pressure on volumetric efficiency were available; therefore a theoretical equation given in reference 5 was used. The effects of fuel-air ratio, compression ratio, engine back pressure, and engine speed on the performance of the combination were computed to find the operating conditions for minimum specific fuel consumption. The difference between the results of the theoretical analysis of the performance of a compression-ignition engine-compressor-turbine combination reported in reference 1 and the results obtained in the present investigation is discussed.

ANALYSIS

Calculations based on test data from reference 3 at a compression ratio of 15.1, at an engine speed of 2200 rpm, and at NACA standard sea-level conditions were made to determine the net specific power output of the compression-ignition engine and the turbine and the power input to the supercharger. The data taken at this compression ratio were extrapolated to other compression ratios. Efficiencies of 0.70 and 0.65 were assumed for the compressor and the turbine, respectively. These efficiencies take into account the gearing efficiency. For one set of calculations the compressor and the turbine efficiencies were raised to 0.85. An intercooler of 60 percent effectiveness was assumed in all calculations. The net specific power output nhp of the combination is given by the relation

$$\text{nhp} = \text{bhp} + \text{thp} - \text{shp}$$

where

bhp brake horsepower of compression-ignition engine per cubic inch of engine displacement

thp horsepower output of turbine per cubic inch of engine displacement

shp horsepower required by supercharger per cubic inch of engine displacement

The net specific fuel consumption nsfc can then be found by the equation

$$\text{nsfc} = \text{isfc}\ \frac{\text{ihp}}{\text{nhp}}$$

where

isfc indicated specific fuel consumption of compression-ignition engine

ihp indicated specific horsepower of compression-ignition engine

The maximum cylinder pressure in all calculations was assumed constant at 1400 pounds per square inch. Inlet-air pressures of 30, 60, 90, and 150 inches of mercury absolute were assumed, with the following compression ratios assumed in each case to give a maximum cylinder pressure of 1400 pounds per square inch: 29.2, 17.4, 13.0, and 8.8. At each of these compression ratios, four fuel-air ratios were assumed: 0.020, 0.035, 0.050, and 0.065. The effect of varying the ratio of exhaust back pressure to inlet-air pressure on the net specific power output and the net specific fuel consumption of the combination at each of these conditions was determined.

These calculations were made at four engine speeds: 1200, 1500, 1800, and 2200 rpm. In order to determine the effect of more efficient components on the performance of the combination, the calculations were all repeated at an engine speed of 2200 rpm with the compressor and turbine efficiencies raised to 0.85.

The performance of the compression-ignition engine and the data necessary to calculate the exhaust-gas temperature were taken from reference 3. The injection was assumed to be so retarded that an indicator card similar to figure 10 of reference 3 with almost constant-pressure combustion would be obtained. The combustion chamber used in the tests of reference 3 is shown in figure 1. As can be seen, the inlet and exhaust valves are directly opposite. When the inlet-air pressure is higher than the exhaust back pressure, any increase in valve overlap would cause a relatively high loss of fresh charge through the exhaust ports. This loss would increase the air flow and the power required by the compressor and would decrease the power output of the turbine per pound of air as a result of dilution of the hot gas. The better scavenging obtained, however, would tend to compensate for this decrease in turbine power output, particularly at low compression ratios. When the exhaust back pressure is higher than the inlet-air pressure, the air flow would be reduced and the scavenging would be very poor. It is assumed in these calculations that no effective valve overlap was used. Inasmuch as no reliable data on the effect of large exhaust back pressure on volumetric efficiency were available, this efficiency was theoretically calculated by means of equation (5) in the appendix. The effect of exhaust back pressure on volumetric efficiency thus obtained is probably too small even though no valve overlap was assumed. This error would tend to shift the point of minimum specific fuel consumption to a higher ratio of exhaust back pressure to inlet-air pressure than might actually be expected. This equation does not take into account heat transfer from the residual gases or the dynamic effects.

DISCUSSION OF RESULTS

The variation of net specific power output and net specific fuel consumption of the combination at sea level with ratio of exhaust back pressure to inlet-air pressure for various fuel-air ratios and compression ratios is shown in figure 2. The engine speed is 2200 rpm and the compressor and turbine efficiencies are 0.70 and 0.65, respectively. The net specific power output increases with decrease in compression ratio and with increase in fuel-air ratio because of high inlet-air pressures at the low compression ratios and high heat inputs at the high fuel-air ratios. At any given fuel-air ratio and compression ratio, the net specific power output increases to a maximum value with increase in the ratio of exhaust back pressure to inlet-air pressure and then decreases. The ratio of exhaust back pressure to inlet-air pressure at which the net specific power output is a maximum varies from approximately 1.0 to 1.5 as the compression ratio is increased from 8.8 to 29.2. When the ratio of exhaust back pressure to inlet-air pressure is less than approximately 1.0, the turbine power output increases with exhaust back pressure at a faster rate than the compression-ignition engine power output decreases; the net specific power therefore increases. With further increase in the ratio of exhaust back pressure to inlet-air pressure, the decrease in volumetric efficiency and the increase in engine friction causes the engine net specific power to decrease at a faster rate than the turbine power increases. The net specific power output therefore decreases.

The net specific fuel consumption shown in figure 2 reaches a minimum value at a ratio of exhaust back pressure to inlet-air pressure that varies with fuel-air ratio and compression ratio. The variation of this minimum with fuel-air ratio for the various compression ratios is shown in figure 3(a) and its variation with compression ratio for various fuel-air ratios is shown in figure 3(b). The corresponding net specific power outputs are also plotted. The optimum operating condition for low fuel consumption occurs at a fuel-air ratio of approximately 0.035 and a compression ratio of approximately 14.0. The minimum net specific fuel consumption obtained is approximately 0.40 pound per net horsepower-hour and the corresponding net specific power output is approximately 0.50 horsepower per cubic inch. When the fuel-air ratio is increased to 0.050 and the compression ratio decreased to 10.0, the net specific fuel consumption is increased approximately 6 percent but the net specific output is increased approximately 80 percent. Very little gain in fuel consumption would therefore be obtained by operating at the point of minimum fuel consumption, whereas the engine weight would be greatly increased.

The variation of minimum net specific fuel consumption and corresponding net specific power output with fuel-air ratio at an engine speed of 1200 rpm is shown in figure 4. The fuel-air ratio at which the net specific fuel consumption is a minimum has decreased from 0.035 to 0.030 (compression ratio, 8.8). The minimum net specific fuel consumption at this fuel-air ratio is approximately 0.37 pound per net horsepower-hour. Curves similar to figure 4 were drawn for engine speeds of 1500 and 1800 rpm and the points of minimum net specific fuel consumption were then plotted as a function of engine speed in figure 5.

The effect of increasing the efficiencies of the turbine and compressor to 0.85 is shown in figure 6. Comparison with figure 3 shows that the minimum net specific fuel consumption at an engine speed of 2200 rpm has decreased to approximately 0.32. The corresponding net specific output has increased approximately 75 percent. The compression ratio at which this minimum is attained has decreased to 8.8.

The power output of each of the component units of the combination is shown in figure 7, as well as the net power output for the values plotted in figure 3. The difference between the net power output and the output of the compression-ignition engine varies linearly with fuel-air ratio. The maximum net power output is approximately 53 percent higher than the compression-ignition engine power alone at a fuel-air ratio of 0.065 and a compression ratio of 8.8.

A comparison of the best performance obtainable with the compression-ignition engine geared together with a compressor and a turbine within the range of conditions used in figures 2 and 3 and the best performance obtainable with a compression-ignition engine using a turbosupercharger is shown in figure 8. The fuel consumption and the net specific power output of the compression-ignition engine with a turbosupercharger were calculated, assuming an exhaust back pressure sufficiently high that the turbine and compressor powers were equal. It was also assumed that the turbine could operate with a closed waste gate. At the best net specific fuel consumption obtainable, which corresponds to a fuel-air ratio of approximately 0.035, the decrease in net specific fuel consumption obtained by gearing the turbine to the compression-ignition engine is only approximately 3 percent at a compression ratio of 13.0. If, therefore, operation at the lowest possible fuel consumption is desired, apparently little economy can be gained by gearing the turbine to the compression-ignition engine. At a fuel-air ratio of 0.065, where the maximum power is obtained, the net specific fuel consumption is, however, reduced as much as 12 percent.

The investigation covered in reference 4 shows the theoretically attainable performance of a compression-ignition engine-compressor-turbine combination. A modified spark-ignition engine operating on a Diesel cycle was assumed. Tests on such an engine presented in reference 6 showed that the predicted results are as yet unattainable. The work presented herein is based on test data taken on a high-speed, high-turbulence compression-ignition engine. The optimistic results obtained in reference 4 compared with the results presented herein and in reference 6 are probably due to the high combustion efficiency indicated by the cycle used in reference 4, which gives a very low value for the indicated specific fuel consumption. The test results of reference 3 upon which the present calculations are based, as well as the data of reference 6, show rather high values of indicated specific fuel consumption because of the poor combustion efficiencies actually obtained.

The results of the present analysis, as well as those of reference 4, seem to indicate that a large increase in power with a small decrease in fuel consumption can be obtained by using a combination consisting of a Diesel engine, a compressor, and a turbine, particularly if the component units are highly efficient. The desirability of gearing the turbine and compressor to the Diesel engine is doubtful, inasmuch as the system becomes more complicated and the decrease in fuel consumption is small.

SUMMARY OF RESULTS

Computations of the performance at sea-level conditions of a compression-ignition engine-compressor-turbine combination based on test data obtained from a single-cylinder compression-ignition engine at a compression ratio of 13.1 and an engine speed of 2200 rpm with the maximum cylinder pressure assumed constant at 1400 pounds per square inch yielded the following results:

1. The net specific power output increased with decreasing compression ratio and with increasing fuel-air ratio and engine speed.

2. At an engine speed of 2200 rpm, with compressor and turbine efficiencies of 0.70 and 0.65, respectively, the net specific fuel consumption reached a minimum value of 0.40 pound per net horsepower-hour at a fuel-air ratio of approximately 0.035 and a compression ratio of approximately 14.0. When the compressor and turbine efficiencies were raised to 0.85, the net specific fuel consumption reached a minimum of 0.32 at a compression ratio of 9.8.

3. With compressor and turbine efficiencies of 0.70 and 0.65, respectively, decreasing the engine speed from 2200 to 1200 rpm decreased the minimum net specific fuel consumption from 0.40 to 0.37 pound per net horsepower-hour.

4. At sea level with compressor and turbine efficiencies of 0.70 and 0.65, respectively, an engine speed of 2200 rpm, and a compression ratio of 13.0, the minimum fuel consumption obtained by

gearing the turbine to the compression-ignition engine was only approximately 3 percent less than that obtained by operating the compression-ignition engine with a turbosupercharger using a closed waste gate.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, October 10, 1946.

APPENDIX - METHOD OF CALCULATION

Symbols and Abbreviations

The following symbols and abbreviations are used in the calculations:

c_p	specific heat at constant pressure, Btu/lb °F
c_v	specific heat at constant volume, Btu/lb °F
n_c	compression exponent in equation $pV^n = c$
n_e	expansion exponent in equation $pV^n = c$
N	engine speed, rpm
p	pressure at various points in cycle, in. Hg absolute
p_0	atmospheric pressure, in. Hg absolute
p_e	exhaust back pressure, in. Hg absolute
Q	net heat added per pound of air, Btu/lb
r	compression ratio
r_c	cut-off ratio
R	gas constant for air, ft-lb/lb °F
R_e	gas constant for exhaust gas, ft-lb/lb °F
T	temperature at various points in cycle, °R
T_e	temperature of exhaust gas, °R
W_s	work input to supercharger, ft-lb/lb air
W_t	work output of turbine, ft-lb/lb exhaust gas
γ	ratio of specific heats for air
γ_e	effective ratio of specific heats for exhaust gas

η_{ad} adiabatic efficiency of supercharger

η_{adt} adiabatic efficiency of turbine

η_t thermal efficiency

η_v volumetric efficiency

ρ density of inlet air, lb/cu ft

Subscripts:

0 ambient conditions

1 inlet to compression-ignition engine

2 end of compression stroke

3 cut-off

4 release

Abbreviations:

bhp brake horsepower of compression-ignition engine per cubic inch displacement

ihp indicated horsepower of compression-ignition engine per cubic inch displacement

nhp net specific output of combination, hp/cu in. engine displacement

shp power required by supercharger, hp/cu in. engine displacement

thp output of turbine, hp/cu in. engine displacement

fmep friction mean effective pressure, lb/sq in.

imep indicated mean effective pressure, lb/sq in.

isfc indicated specific fuel consumption of engine, lb/hp-hr

nsfc net specific fuel consumption of combination, lb/nhp-hr

F/A fuel-air ratio

Net Specific Output and Net Specific Fuel Consumption

Power output of compression-ignition engine. - The indicated horsepower per cubic inch displacement of the compression-ignition engine is given by the following two equations:

$$\text{ihp} = \frac{\text{imep } N}{792{,}000} \tag{1}$$

and

$$\text{ihp} = \frac{F/A}{\text{isfc}} \frac{\rho N}{57.6} \eta_v \tag{2}$$

where $\frac{\rho N}{57.6} \eta_v$ is the air flow in pounds per hour per cubic inch. The volumetric efficiency was approximated by assuming it is affected only by the inlet and the exhaust pressures. The ratio of volumetric efficiency under any two conditions represent d by subscripts 1 and 2 may then be expressed as in reference 5, but using the notation of this paper

$$\frac{\eta_{v1}}{\eta_{v2}} = \frac{r - \left(\frac{p_e}{p_1}\right)^{\frac{1}{\gamma_e}}}{r - \left(\frac{p_e}{p_1}\right)_2^{\frac{1}{\gamma_e}}} \tag{3}$$

If the volumetric efficiency is assumed to be 0.86 when the exhaust back pressure is equal to the inlet-air pressure, then

$$\eta_v = 0.86 \frac{r - \left(\frac{p_e}{p_1}\right)^{\frac{1}{\gamma_e}}}{r - 1} \tag{4}$$

Figure 3 shows the volumetric efficiency plotted against the ratio of exhaust back pressure to inlet-air pressure for a compression ratio of 13.0.

The inlet-air density is given by

$$\rho_1 = 1.327 \frac{p_1}{T_1} \tag{5}$$

When equations (1), (2), (4), and (5) are combined and $\gamma_e = 1.35$

$$\text{imep} = 15{,}690\ \frac{F/A}{\text{isfc}}\ \frac{p_1}{T_1}\ \frac{r - \left(\frac{p_e}{p_1}\right)^{0.741}}{r - 1} \tag{6}$$

The friction mean effective pressure was obtained by taking the friction mean effective pressure of a multicylinder crankcase at an engine speed of 2200 rpm, adding 5 pounds per square inch for the displacer action of the compression-ignition engine, and using an empirical formula obtained from an analysis of data on several single-cylinder engines. The equation obtained is

$$\text{fmep} = 31 + 0.32\ (p_e - p_1) \tag{7}$$

The first term on the right side of equation (7) is due to the mechanical friction and the second term is due to the pumping loss. It was assumed that the pumping loss is independent of the speed and the mechanical friction is proportional to the speed. From equation (7) the constant of proportionality was found to equal 14.1×10^{-3}. The friction mean effective pressure then becomes

$$\text{fmep} = 14.1 \times 10^{-3}\ N + 0.32\ (p_e - p_1) \tag{8}$$

The power output of the compression-ignition engine then becomes

$$\text{bhp} = (\text{imep} - \text{fmep})\ \frac{N}{792{,}000} \tag{9}$$

Power output of turbine. - The power output of the turbine was calculated from the equation

$$W_t = \frac{\gamma_e\ \eta_{adt}}{\gamma_e - 1}\ R_e T_e \left[1 - \left(\frac{p_0}{p_e}\right)^{\frac{\gamma_e - 1}{\gamma_e}}\right] \tag{10}$$

The turbine horsepower per cubic inch of engine displacement becomes

$$\text{thp} = \frac{W_t\,(1+F/A)}{33{,}000 \times 60}\,\frac{\rho N}{57.6}\,\eta_v$$

$$= 1.0 \times 10^{-8}\,\eta_{adt}\,\frac{\gamma_e}{\gamma_e - 1}\,R_e T_e\,\frac{p_1 N}{T_1}\left[1 - \left(\frac{p_0}{p_e}\right)^{\frac{\gamma_e - 1}{\gamma_e}}\right]\left[\frac{r - \left(\frac{p_e}{p_1}\right)^{\frac{1}{\gamma_e}}}{r - 1}\right](1 + F/A) \qquad (11)$$

The temperature of the exhaust gas T_e was calculated by the equation given in reference 3, which, in the notation of the present paper, is

$$T_e = \frac{T_4}{\gamma_c}\left[1 + (\gamma_e - 1)\,\frac{p_e}{p_4}\right] \qquad (12)$$

If the following two relations are substituted in equation (12)

$$T_4 = T_1 r_c^{\,n_e}\, r^{(n_c - n_e)} \qquad (13)$$

$$p_4 = p_1 r_c^{\,n_e}\, r^{(n_c - n_e)} \qquad (14)$$

the exhaust-gas temperature becomes

$$T_e = \frac{T_1}{\gamma_e}\left[r^{(n_c - n_e)}\, r_c^{\,n_e} + (\gamma_e - 1)\,\frac{p_e}{p_1}\right] \qquad (15)$$

The values of n_c and n_e were obtained from figure 14 of reference 3 and the values of γ_e and R_e for the exhaust gas from reference 7. Values of exhaust-gas temperature T_e plotted against ratio of exhaust back pressure to inlet-air pressure are shown in figure 10.

Power required by supercharger. - The work input of the supercharger per pound of air supercharged is given by the equation

$$W_s = \frac{\gamma}{\gamma - 1}\,\frac{RT_0}{\eta_{ad}}\left[\left(\frac{p_1}{p_a}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right] \qquad (16)$$

The horsepower per cubic inch is then

$$\text{shp} = \frac{W_s}{33{,}000 \times 60}\,\frac{\rho N}{57.6}\,\eta_v \qquad (17)$$

When $\eta_{ad} = 0.7$, $R = 53.3$, and $\gamma = 1.4$

$$shp = 2.67 \times 10^{-6}\, N p_1 \frac{r - \left(\frac{p_e}{p_1}\right)^{0.741}}{r - 1} \left[\left(\frac{p_1}{p_0}\right)^{0.286} - 1\right] \qquad (18)$$

Net output and fuel consumption. - If the power outputs of the compression-ignition engine and of the turbine and the power required by the supercharger are known, the net power can be found from the relation

$$nhp = bhp + thp - shp \qquad (19)$$

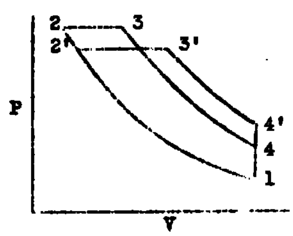
The net specific fuel consumption can then be calculated

$$nsfc = \left(\frac{ihp}{bhp + thp - shp}\right) isfc \qquad (20)$$

The indicated specific fuel consumption for a compression ratio of 13.1 is given in figure 5 of reference 3.

Effect of Varying Compression Ratio

Cut-off ratio. - In order to find the effect of varying the compression ratio on the cut-off ratio, the amount of heat added per pound of air is assumed to be constant at a given fuel-air ratio. From the following sketch



$$Q = c_p (T_3 - T_2) + \frac{\gamma_e - n_e}{n_e - 1} c_v (T_3 - T_4)$$

$$= c_p (T_3{}' - T_2{}') + \frac{\gamma_e - n_e}{n_e - 1} c_v (T_3{}' - T_4{}') \qquad (21)$$

$$\gamma_e\, r^{n_c-1}\,(r_c-1)+\frac{\gamma_e-n_e}{n_e-1}\,r_1{}^{n_c}\left[\frac{r_{c1}}{r_1}-\left(\frac{r_{c1}}{r_1}\right)^{n_e}\right]=\gamma_e\, r_2{}^{n_c-1}\,(r_{c2}-1)$$

$$+\frac{\gamma_e-n_e}{n_e-1}\,r_2\, n_c\left[\frac{r_{c2}}{r_2}-\left(\frac{r_{c2}}{r_2}\right)^{n_e}\right] \qquad (22)$$

where the subscripts 1 and 2 used with r and r_c refer to any two values of compression ratio. For $r_1 = 13.1$, values of r_{c1} taken from figure 14 of reference 3 can be used to calculate r_{c2} for any desired compression ratio. Q is here defined as the net heat added to the air when the heat loss to the coolant is neglected.

Indicated specific fuel consumption. - The indicated specific fuel consumption is assumed to be inversely proportional to the thermal efficiency. The thermal efficiency of conversion of the net heat added to the cycle, when $\gamma = n_c$ is assumed for the compression stroke, is obtained by

$$\eta_t = 1 - \frac{r_c{}^{n_e}\, r^{n_c-n_e} - 1}{\gamma_e\, r^{n_c-1}\,(r_c-1)+\dfrac{\gamma_e-n_e}{n_e-1}\,r^{n_c}\left[\dfrac{r_c}{r}-\left(\dfrac{r_c}{r}\right)^{n_e}\right]} \qquad (23)$$

At a compression ratio of 13.1, the values of r_c, n_c, n_e, and isfc for different fuel-air ratios were obtained from figure 14 of reference 3 and η_t was calculated. The constant k in the equation

$$\frac{1}{\text{isfc}} = k\,\eta_t$$

was then evaluated for the different fuel-air ratios. For any compression ratio, therefore

$$\frac{1}{\text{isfc}} = k\left\{1 - \frac{r_c{}^{n_e}\, r^{n_c-n_e} - 1}{\gamma_e\, r^{n_c-1}\,(r_c-1)+\dfrac{\gamma_e-n_e}{n_e-1}\,r^{n_c}\left[\dfrac{r_c}{r}-\left(\dfrac{r_c}{r}\right)^{n_e}\right]}\right\}$$

Curves of the reciprocal of the indicated specific fuel consumption for different fuel-air ratios are plotted in figure 11.

The justification for the procedure can be found in the fact that the actual curve of thermal efficiency plotted against compression ratio is parallel to the calculated curve (fig. 1, reference 9). The empirical constant k therefore corrects the theoretical curve to the actual curve.

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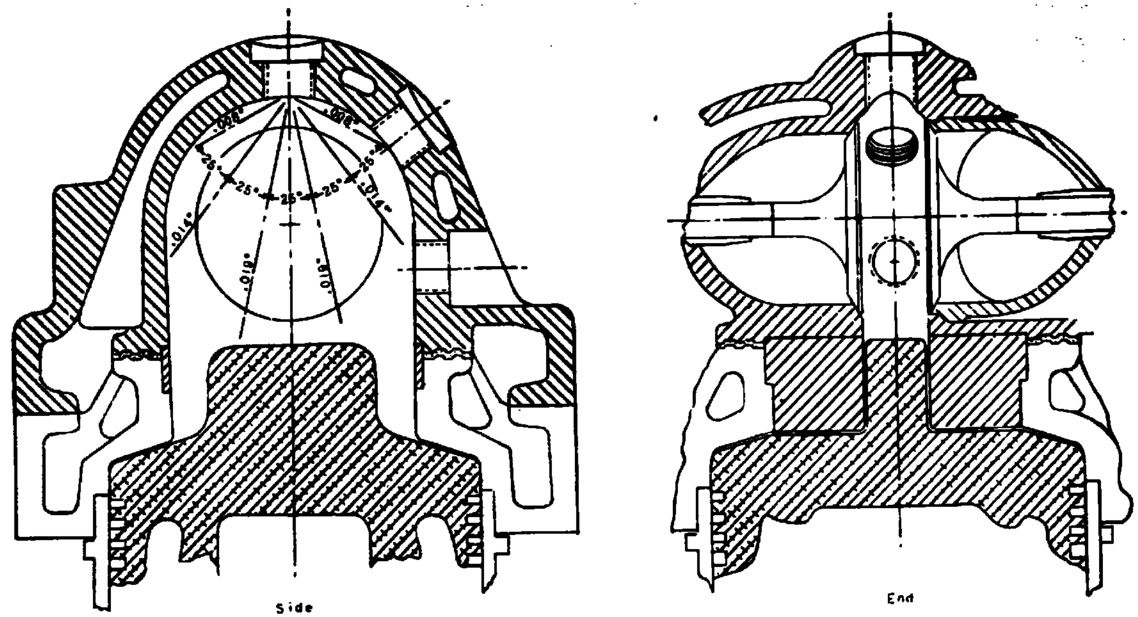

Figure 1. - Combustion chamber used in investigation of reference 3. (Reproduced from reference 3.)

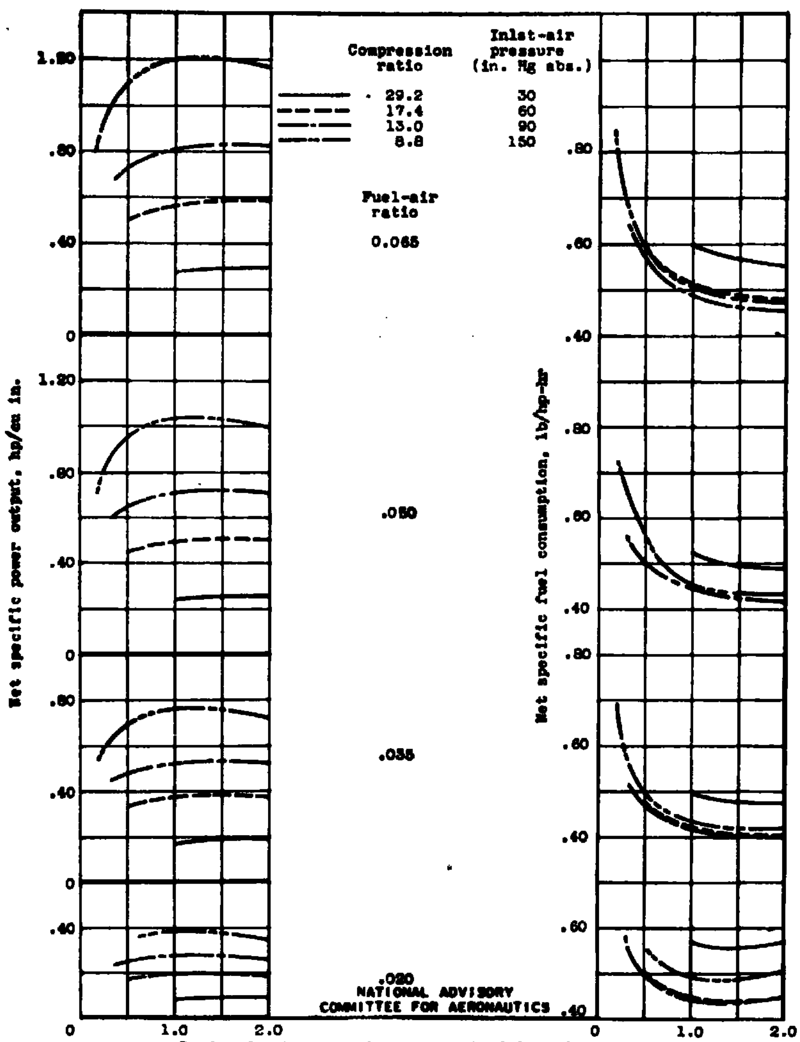

Figure 2. - Variation of net specific power output and corresponding net specific fuel consumption with ratio of exhaust back pressure to inlet-air pressure for various fuel-air ratios and compression ratios. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm; compressor efficiency, 0.70; turbine efficiency, 0.65.

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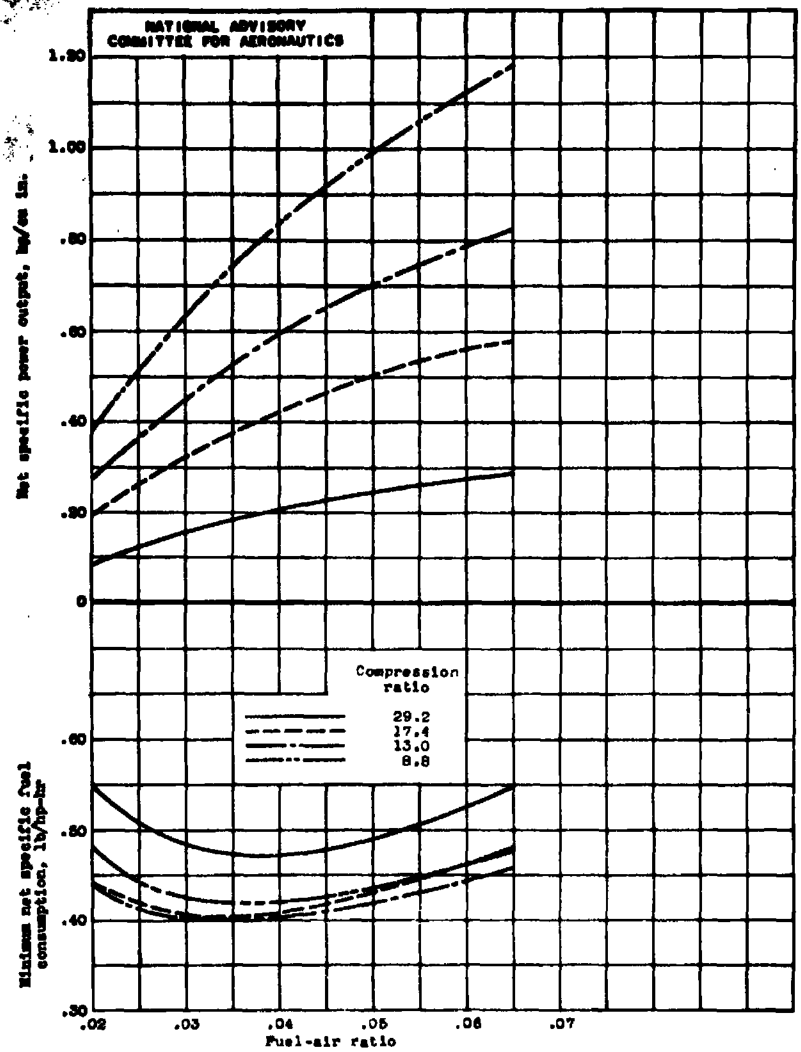
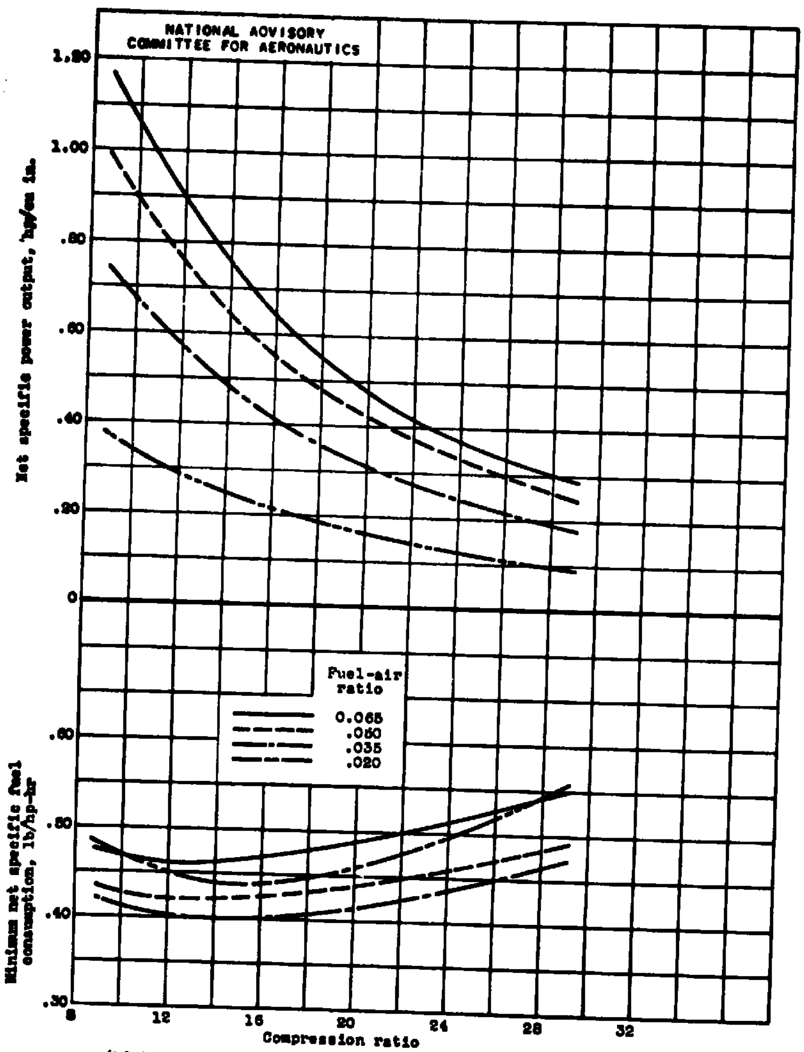


(a) Variation with fuel-air ratio for various compression ratios.

Figure 3. - Variation of minimum net specific fuel consumption and corresponding net specific power output with fuel-air ratio and compression ratio. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm; compressor efficiency, 0.70; turbine efficiency, 0.65.



(b) Variation with compression ratio for various fuel-air ratios.

Figure 3. - Concluded. Variation of minimum net specific fuel consumption and corresponding net specific power output with compression ratio and fuel-air ratio. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm; compressor efficiency, 0.70; turbine efficiency, 0.65.

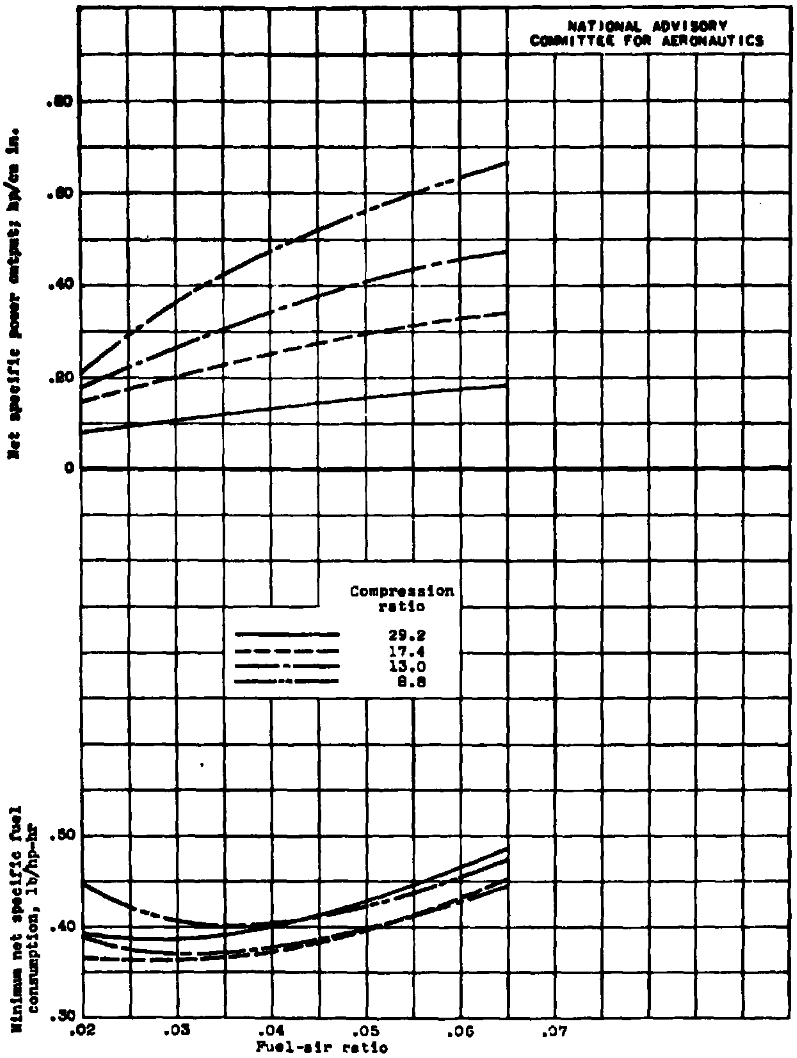


Figure 4. - Variation of minimum net specific fuel consumption and corresponding net specific power output with fuel-air ratio for various compression ratios at engine speed of 1200 rpm. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; compressor efficiency, 0.70; turbine efficiency, 0.85.

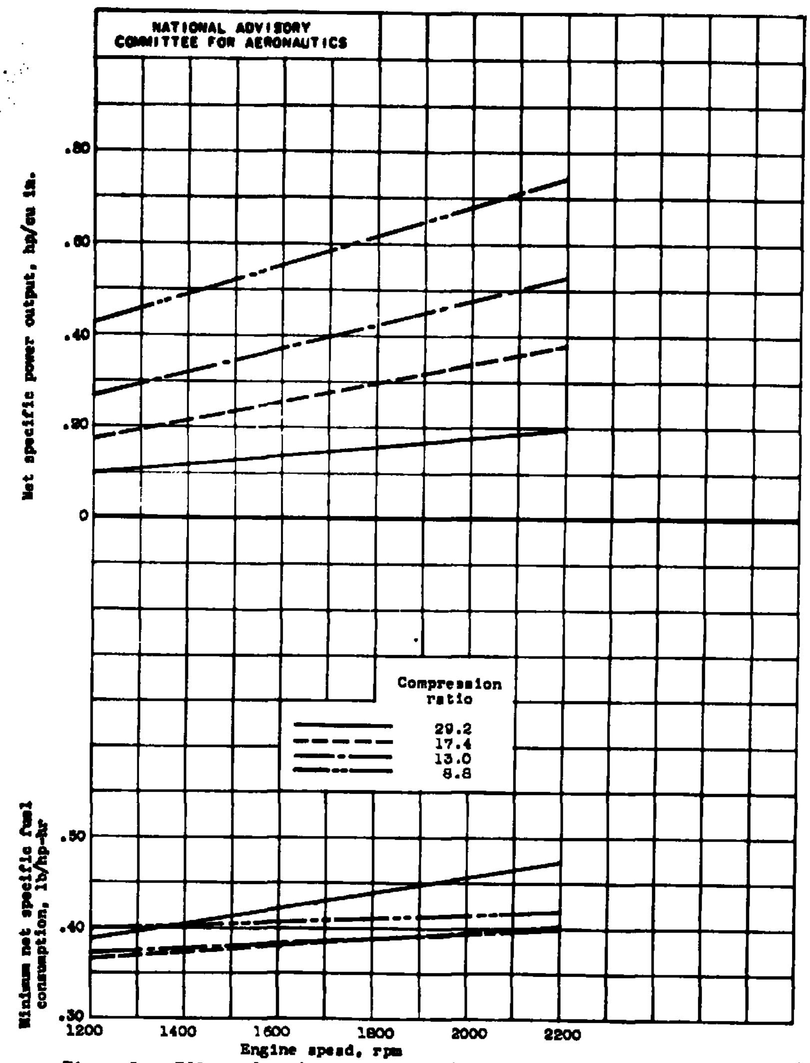
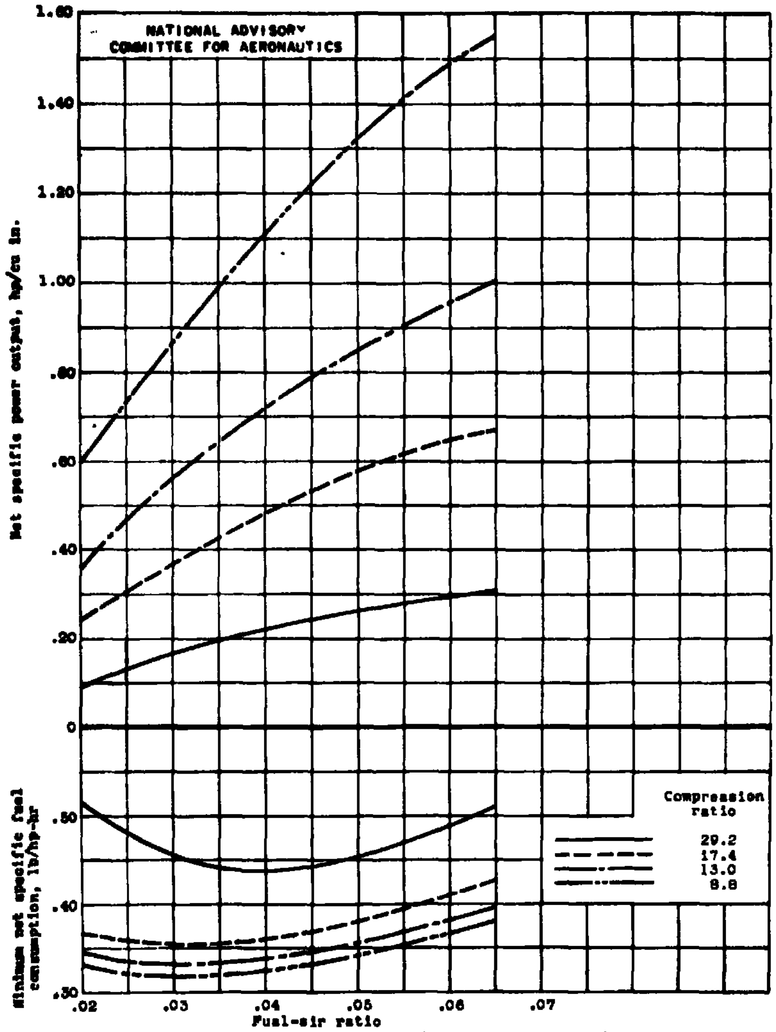
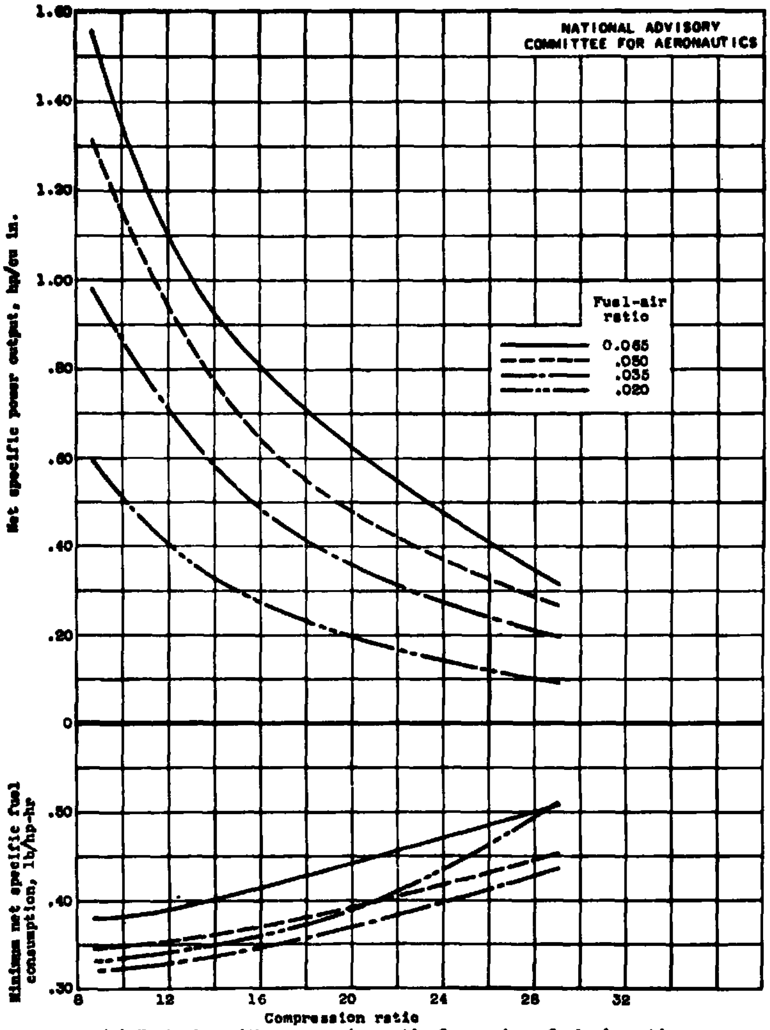


Figure 5. - Effect of engine speed on minimum net specific fuel consumption and corresponding net specific power output at optimum fuel-air ratio. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; compressor efficiency, 0.70; turbine efficiency, 0.85.



(a) Variation with fuel-air ratio for various compression ratios.

Figure 6. - Effect of raising compressor and turbine efficiencies to 0.85 on variation of minimum net specific fuel consumption and corresponding net specific power output with fuel-air ratio and compression ratio. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm.



(b) Variation with compression ratio for various fuel-air ratios.

Figure 6. - Concluded. Effect of raising compressor and turbine efficiencies to 0.85 on variation of minimum net specific fuel consumption and corresponding net specific power output with compression ratio and fuel-air ratio. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm.

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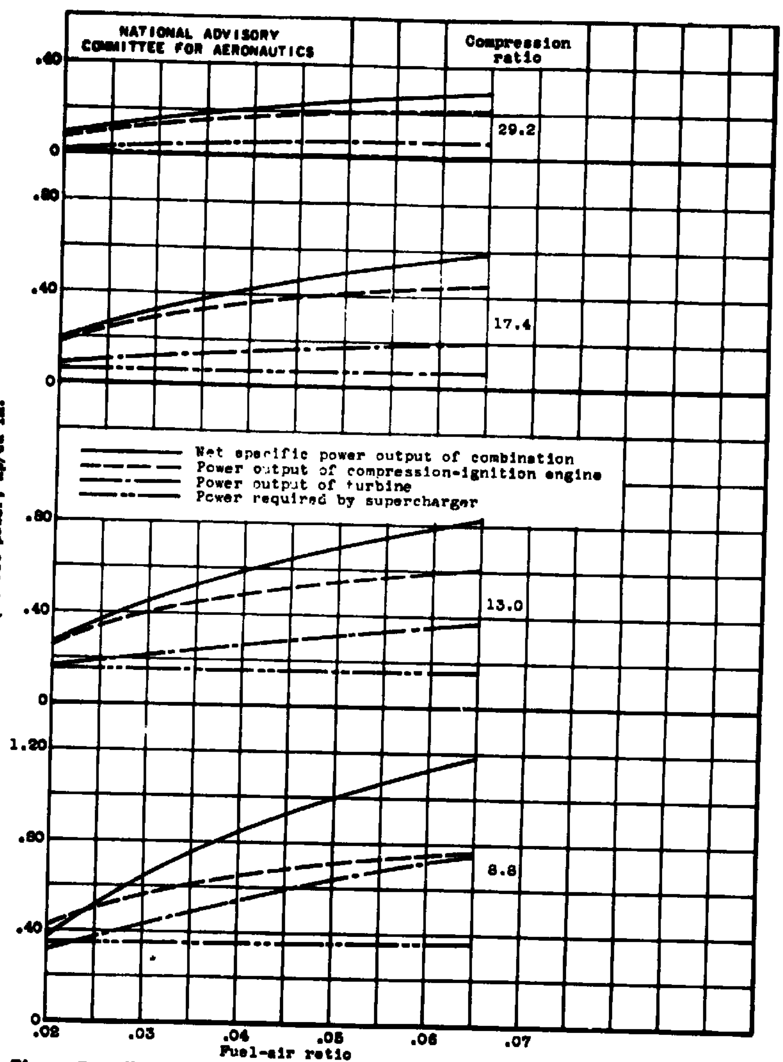

Figure 7. - Variation with fuel-air ratio of specific powers of components of combination at exhaust back pressures for minimum specific fuel consumption. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm; compressor efficiency, 0.70; turbine efficiency, 0.65.

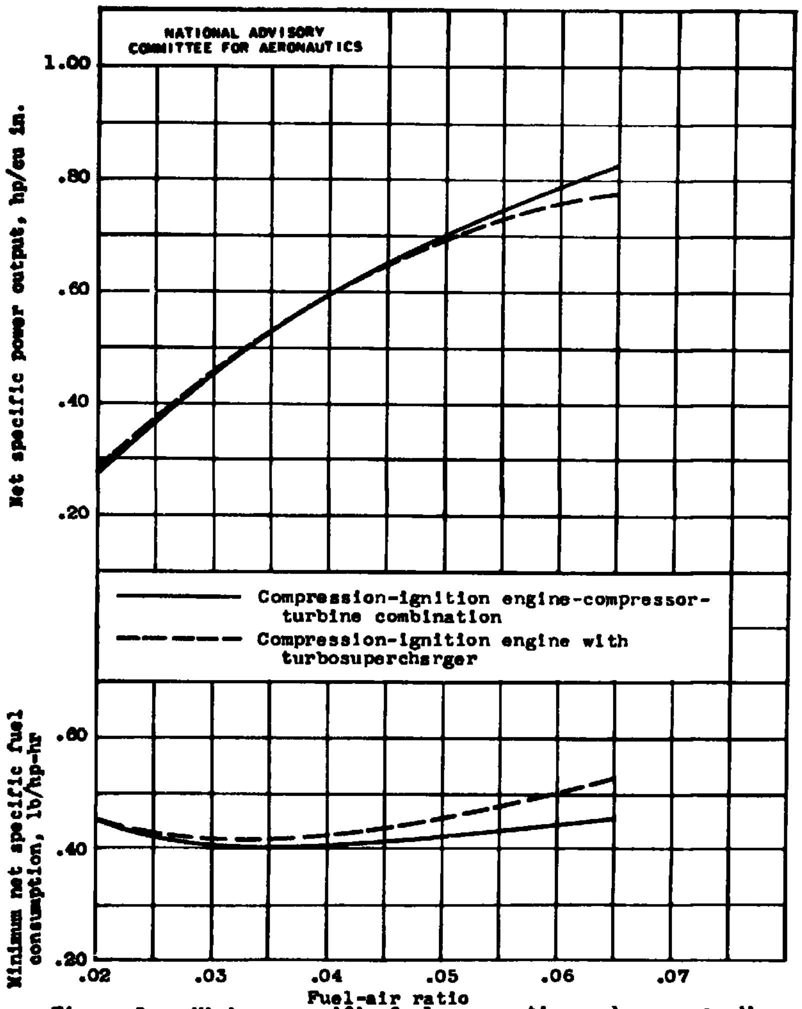

Figure 8. - Minimum specific fuel consumption and corresponding net specific power output of compression-ignition engine-compressor-turbine combination compared with compression-ignition engine with turbosupercharger. Maximum cylinder pressure maintained constant at 1400 pounds per square inch; engine speed, 2200 rpm; compressor efficiency, 0.70; turbine efficiency, 0.65; compression ratio, 13.0.

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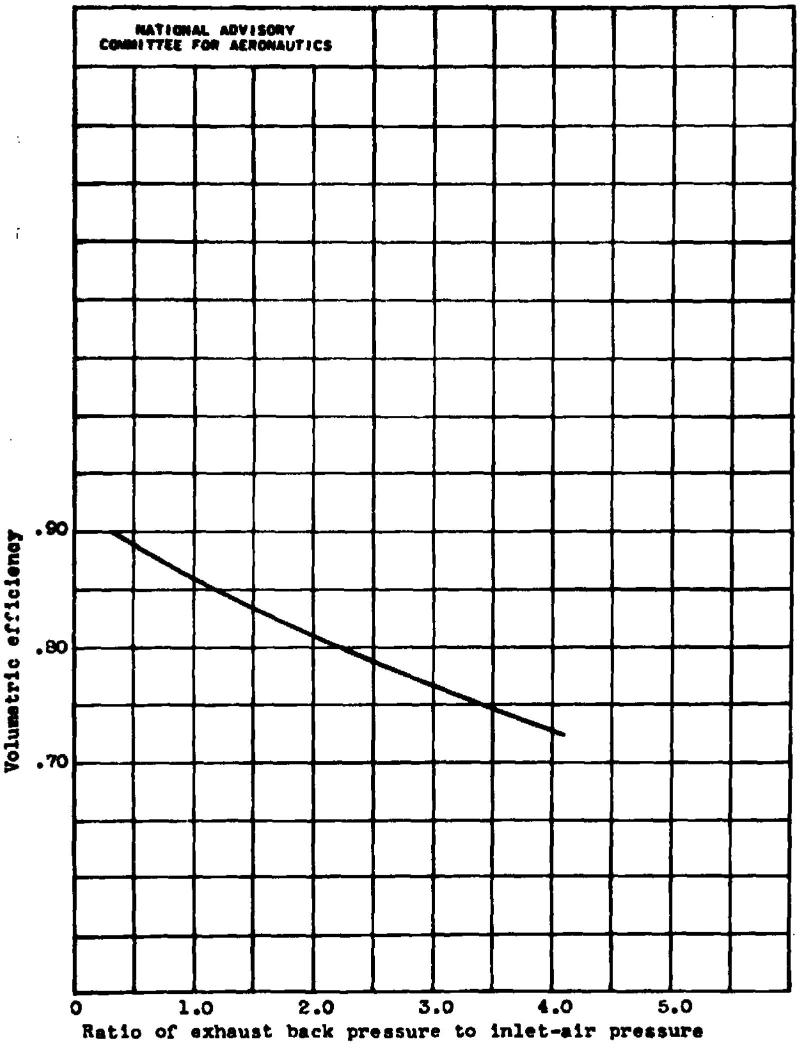

Figure 9. - Effect of ratio of exhaust back pressure to inlet-air pressure on volumetric efficiency of compression-ignition engine with no valve overlap. Engine speed, 2200 rpm; compression ratio, 13.0.

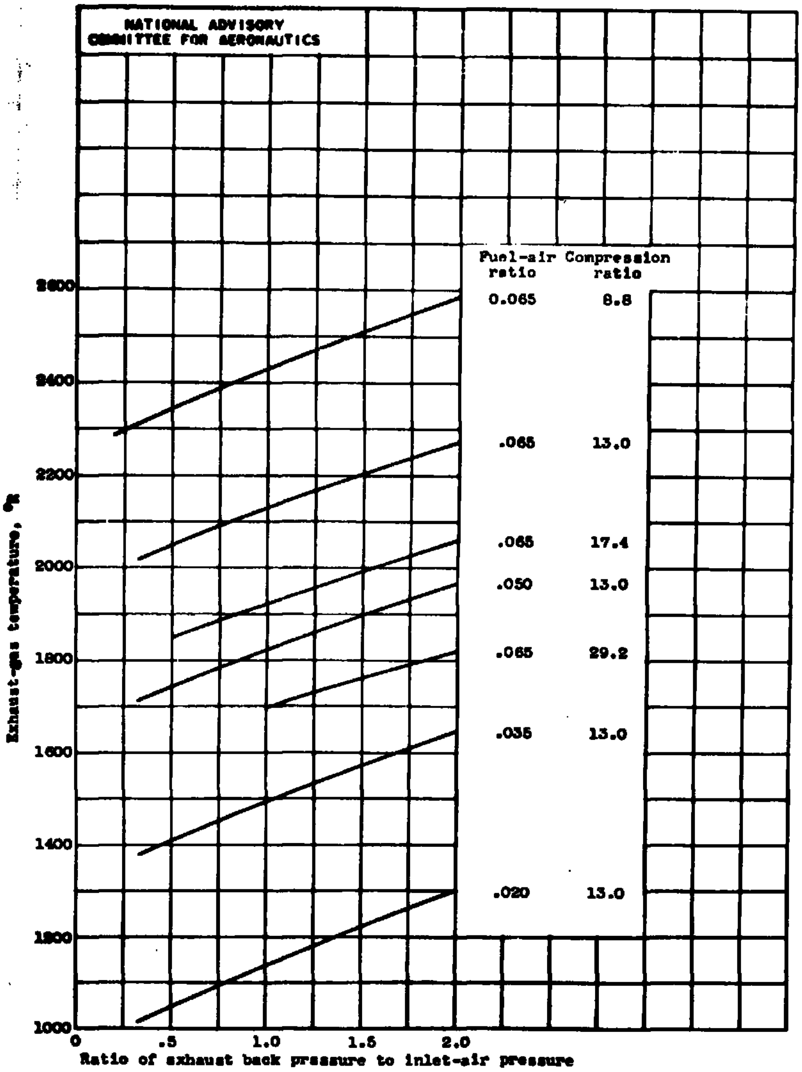

Figure 10. - Variation of exhaust-gas temperature with ratio of exhaust back pressure to inlet-air pressure for various fuel-air ratios and compression ratios.

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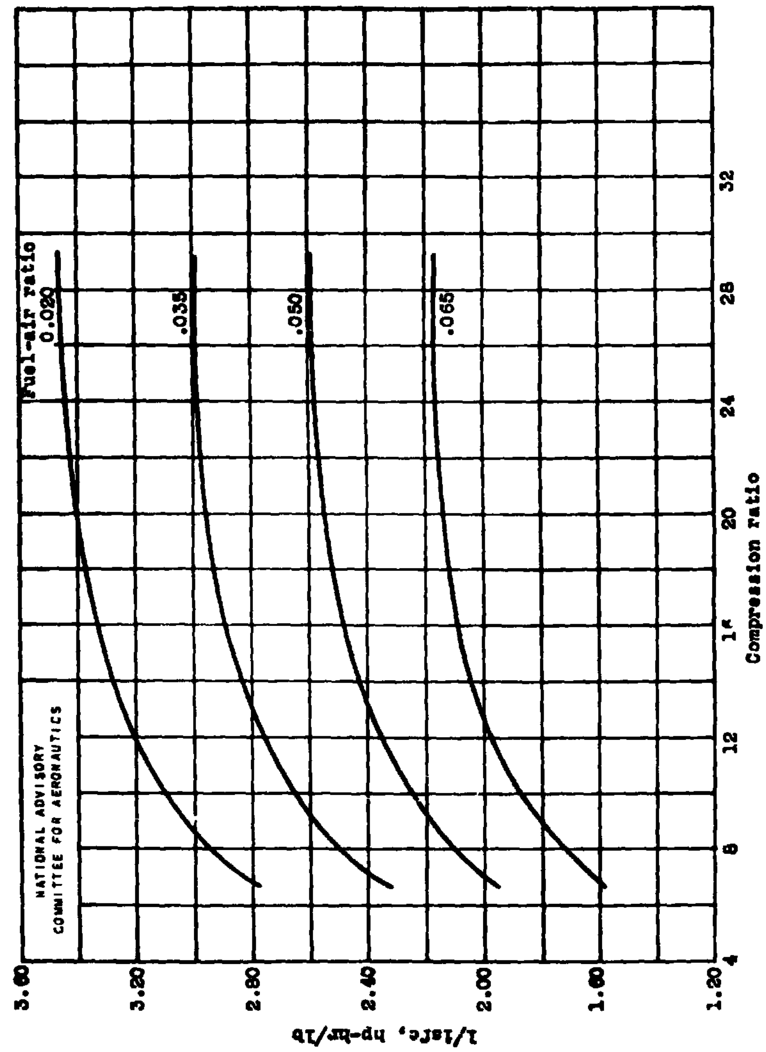

Figure 11. - Variation of indicated specific fuel consumption with compression ratio for various fuel-air ratios.

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APR 22 1949

TSDIN FORM 69 A (13 MAR 47)

Mendelson, A.

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SECTION: Performance (13)
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ABSTRACT

Tests were conducted on compression-ignition engine to determine effects of fuel air ratio, compression ratio, exhaust back pressure, and engine speed on performance. Maximum cylinder pressure was assumed at 1400 psi constant. Calculation methods and results are discussed. At 2200 rpm, compressor efficiency of 0.70, and turbine efficiency of 0.65, a minimum net specific fuel consumption of 0.40 lb/hp-hr was obtained. Gearing turbine to engine has only slight improvement over engine using supercharger with closed waste gate.

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