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

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SUMMARY

Small high-speed sing -cylinder compression-ignition engines were tested to determine their performance characteristics under high supercharging. Calculations were made on the energy available in the exhaust gas of the compression-ignition engines. The maximum power at any given maximum cylinder pressure was obtained when the compression pressure was equal to the maximum cylinder pressure. Constant-pressure combustion was found possible at an engine speed of 2200 rpm. Exhaust pressures and temperatures were determined from an analysis of indicator cards. The analysis showed that, at rich mixtures with the exhaust back pressure equal to the inlet-air pressure, there is excess energy available for driving a turbine over that required for supercharging. The presence of this excess energy indicates that a highly supercharged compression-ignition engine might be desirable as a compressor and combustion chamber for a turbine.

INTRODUCTION

In view of the current interest in aircraft-propulsion systems that utilize gas turbines in the energy-conversion process (reference 1), the development of a means of supplying gas at pressures and temperatures suitable for turbine operation is desirable.

The physical properties of materials available for turbine blades limit the maximum permissible gas temperature to approximately 1600° F. In a combustion-gas-turbine system consisting of a compressor, a combustion chamber, and a gas turbine, the temperature is maintained below the specified limit by supplying excess air. This limited temperature reduces the available energy in the gas that can be extracted by the turbine and for this reason highly efficient turbines and compressors are required to obtain over-all performance approaching that of the reciprocating engine (reference 2).

A dual power unit consisting of a highly supercharged compression-ignition engine geared to a turbine is proposed. The conditions of the working cycle would be so adjusted that the turbine, supplied with the exhaust gas from the engine, would produce a large proportion of the power. The compression-ignition engine is proposed for use in this engine-turbine combination because it can operate at high pressures without the occurrence of knock and preignition. 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. This plan for a dual power unit, which is not new, is mentioned in reference 1.

A preliminary evaluation of the desirability of a compression-ignition engine-turbine combination requires knowledge of the engine performance with high inlet-air pressures and requires information on exhaust-gas temperatures and pressures. Information on exhaust-gas temperatures and pressures and a discussion of the performance characteristics of a highly supercharged compression-ignition engine are presented herein. No test data on performance at inlet-air pressures above 50 inches of mercury absolute were available for the computations. Tests were therefore conducted on two small high-speed compression-ignition engines to determine the operating characteristics at inlet-air pressures up to 90 inches of mercury absolute. Particular emphasis was placed upon operation on the constant-pressure combustion cycle in an effort to produce high indicated mean effective pressures without excessive maximum cylinder pressure. Indicator cards were analyzed to obtain exhaust temperatures and pressures. These tests were run at the NACA Langley Field laboratory during the summer of 1941. No attempt is made herein to estimate the performance of the engine-turbine combination.

APPARATUS

Two four-stroke-cycle compression-ignition engines were used in these tests. One, which had a compression ratio of 13.1, consisted of a single water-cooled cylinder mounted on an NACA universal test engine crankcase. High turbulence in the combustion chamber was produced by the use of a displacer-type piston shown in figure 1 and more fully described in reference 3. The bore was 5 inches and the stroke $5\frac{1}{2}$ inches. The other engine, which had a compression ratio of 15, used an air-cooled cylinder with the same type of combustion chamber as the water-cooled cylinder and had a bore of $5\frac{3}{4}$ inches and a stroke of $5\frac{3}{4}$ inches.

The simultaneous discharge from both cylinders of a two-cylinder high pressure injection pump with plungers 11 millimeters in diameter metered fuel to a differential-area injection valve, which was spring-loaded to have an opening pressure of 2500 pounds per square inch. A multiple-orifice nozzle, designated K-4 in table I of reference 4, was used because it was found that this nozzle gave a good mixing of the fuel with the air.

The usual laboratory equipment was employed to measure power and fuel consumption. Combustion air was supplied at high pressure from a central supply system. Inlet-air pressure was measured at the surge tank and the air was unheated. Maximum cylinder pressures and indicator cards were obtained with a modified Farnboro engine indicator described in reference 5.

PROCEDURE

Four series of tests were run, three at a compression ratio of 13.1 and one at a compression ratio of 15. In the first series of tests inlet-air pressure, fuel flow, and injection advance angle for maximum power at several maximum cylinder pressures were determined with the engine having a compression ratio of 13.1. A maximum-power test at a compression ratio of 15 and a maximum cylinder pressure of 1225 pounds per square inch absolute was run in the second series of tests to show the effect of compression ratio on maximum power. In the third series the effect of injection advance angle, fuel-air ratio, and inlet-air pressure on fuel consumption at a maximum cylinder pressure of 1400 pounds per square inch absolute was determined. Indicator cards to show the effects of inlet-air pressure and injection advance angle on combustion were obtained during the fourth series of tests. The last two series of tests were run on the engine having a compression ratio of 13.1. All tests were run at an engine speed of 2200 rpm and atmospheric exhaust.

Maximum-power tests at low compression ratio. - At each of a number of fixed inlet-air pressures in the range from 45 to 80 inches of mercury absolute and with the fuel flow adjusted to give maximum power, the injection timing was advanced until an arbitrarily chosen maximum cylinder pressure was obtained. The inlet-air pressure, brake mean effective pressure, injection advance angle, and maximum cylinder pressure were recorded. The tests were repeated at successively higher inlet-air pressures; the injection was retarded in each case until the inlet-air pressure could not be further increased without increasing the maximum cylinder pressure, regardless of how much the injection was retarded. Similar tests were run at maximum cylinder pressures of 770, 955, 1135, 1225, 1315, and 1410 pounds per square inch absolute. The engine having a compression ratio of 13.1 was used for these tests.

The indicated mean effective pressure was calculated from the indicated power obtained by adding the measured brake power and the motoring power required to drive the engine at test speed and at test inlet-air pressure when no fuel was injected. Friction tests were made with a cold engine because ignition of the lubricating oil interfered with motoring measurements when the engine was warm.

Maximum-power tests at high compression ratio. - The maximum-power tests were conducted at a compression ratio of 15 in the same manner as the maximum-power tests at a compression ratio of 13.1, except that the only value of maximum cylinder pressure tested was 1225 pounds per square inch absolute.

Fuel-consumption test. - Fuel-consumption characteristics were obtained at inlet-air pressures of 68.2, 73, 76.5, 81.4, and 84.4 inches of mercury absolute. At each inlet-air pressure, sufficient fuel was injected to produce maximum power and the injection timing was adjusted to give a maximum cylinder pressure of 1400 pounds per square inch absolute. Observations were made of fuel consumption at various loads at the inlet-air pressure and the injection timing thus obtained.

Indicator-card tests. - Two series of indicator cards were obtained. All the cards in one series were taken on cycles having peak pressures of 1415 pounds per square inch absolute. At inlet-air pressures of 45, 60, 70, and 80 inches of mercury absolute, the injection was advanced until a maximum cylinder pressure of 1415 pounds per square inch absolute was obtained and indicator cards were taken. The fuel flow was that for maximum power at each inlet-air pressure.

The other series of indicator cards was obtained with the injection-advance angle so adjusted that combustion was as near constant pressure as possible. These indicator cards were obtained at inlet-air pressures of 48, 58, 70, 73.6, and 80 inches of mercury absolute and maximum-power fuel flow.

POWER AND FUEL-CONSUMPTION CHARACTERISTICS

Maximum-power tests. - Figure 2 shows that at any given maximum cylinder pressure between 770 and 1410 pounds per square inch absolute, the maximum power was attained when the compression pressure was approximately equal to the maximum cylinder pressure. For example, at a maximum cylinder pressure of 1225 pounds per square inch absolute, the maximum power was obtained when the inlet-air pressure was 75 inches of mercury absolute and the corresponding compression pressure without fuel was 1170 pounds per square inch absolute, which indicates that the engine was operating on approximately a constant-pressure combustion cycle. Because the compression pressure in the cylinder while the engine is under power is slightly higher than the compression pressure while the engine is being motored, the real agreement between the maximum cylinder pressure and the compression pressure is closer than is shown by figure 2.

The limit of supercharging for a given maximum cylinder pressure is reached when the compression pressure equals the maximum cylinder pressure. In all these tests the highest power was reached at the limit of supercharging as shown in figure 2.

Evaluation of the power characteristics of the compression-ignition engine by comparison with a spark-ignition engine may be obtained from figure 3. For the same power the spark-ignition engine requires an inlet-air pressure about 8 percent lower than the compression-ignition engine when the injection is advanced to give a constant-volume combustion cycle. When the injection is retarded to simulate a constant-pressure combustion cycle, however, the inlet-air pressure required for the same power by the compression-ingition engine is 25 percent greater than that required by the spark-ignition engine. When a maximum cylinder pressure is achieved in the compression-ignition engine as low as in the spark-ignition engine, this loss must be incurred. As can be seen from figure 3, the compression-ignition engine with a compression ratio of 13.1 has about the same ratio of maximum cylinder pressure to indicated mean effective pressure as a spark-ignition engine with a compression ratio of 6.4. The maximum-power fuel-air ratio is used in each case.

Effect of compression ratio on pressure-limited power. - The effect of compression ratio on the maximum power obtainable with a given maximum cylinder pressure is illustrated in figure 4. The maximum indicated mean effective pressure attainable with a maximum cylinder pressure of 1225 pounds per square inch absolute is 264 pounds per square inch at a compression ratio of 15 and 300 pounds per square inch at a compression ratio of 13.1. The corresponding inlet-air pressures were 62 and 75 inches of mercury absolute, respectively.

These data illustrate that, because a higher inlet-air pressure can be used, more power can be obtained by using a low compression ratio for a highly supercharged compression-ignition engine delivering exhaust gas to a turbine than by using a high compression ratio. The minimum compression ratio of a compression-ignition engine is limited by the ignitibility of the cylinder charge. Starting becomes difficult in warm climates when the compression ratio is less than 12 and in cold climates when the compression ratio is below 14 unless an air preheater is used. For increased power output for the compression-ignition engine-turbine combination, lower compression ratios seem desirable and may be practical if a hot plug is used to ignite the fuel.

Fuel consumption. - The fuel-air ratios were computed, assuming a volumetric efficiency of 0.90, and were plotted with fuel consumption and injection-advance angle in figure 5. The reciprocal of the indicated specific fuel consumption was used because the reciprocal is proportional to the indicated efficiency. This figure clearly shows that, when the injection is retarded, the indicated efficiency is reduced. Figure 6 shows the volumetric efficiency for a similar type engine.

Indicator-card tests. - An example of an original record obtained with the Farnboro indicator is shown in figure 7; records presented in such form are indistinct. The curves on the indicator cards obtained (figs. 8 and 9) were therefore drawn by hand to represent the mean of the points generated for each test condition by the Farnboro indicator. The pressure scales are not linear in the low ranges.

Figure 8 shows that, when the inlet-air pressure is increased and the injection retarded, the cycle changes from an approximation of the constant-volume combustion cycle to a constant-pressure combustion cycle. It is not always possible, however, to obtain a constant-pressure combustion cycle with a retarded injection, as shown in figure 9. At low maximum cylinder pressure, a large delay in ignition of the fuel injected resulted in most of the fuel igniting at one time and causing a sharp rise in pressure. The injection consequently was considerably retarded after top center to prevent the maximum cylinder pressure from exceeding the compression pressure. A test at a maximum cylinder pressure lower than 770 pounds per square inch absolute was attempted but it was found necessary to retard the injection to such an extent that ignition was irregular as a result of the low compression temperature during injection.

The data from the Farnboro indicator for a constant-pressure combustion cycle were transferred to a pressure-volume basis and are presented in figure 10. A true Diesel cycle was not attained because the expansion after cut-off was isothermal (the expansion exponent being approximately 1.0) resulting from the continuation of combustion during the expansion stroke. The isothermal expansion is undesirable because the economy and power at a given maximum cylinder pressure are less than those obtained with a true Diesel cycle having adiabatic expansion. The close approach to the Diesel cycle shown by the indicator diagram in figure 10, however, is considered remarkable for a small compression-ignition engine operating at a high engine speed of 2200 rpm.

Energy available in the exhaust gas. - The energy available in the exhaust gas was obtained by analysis of an indicator card. Observations of exhaust-gas temperatures as measured by a thermocouple could not be used for this purpose because temperature measured in this manner is much lower than the true value, as is explained in reference 6.

A description of the method of analyzing the indicator cards is given in appendix A and the results of this analysis are shown in figures 11 and 12. From figure 11 the release temperature and pressure may be obtained for computation of exhaust temperature and energy available at various operating conditions. Such computations were made for the special case of a fuel-air ratio of 0.065 and sea-level ambient pressure with intake-manifold and exhaust-manifold pressures equal. The method of making this computation is described in appendix B and the results are shown in figure 12 together with the energy required for supercharging. The energy available in the exhaust at high inlet-air pressures exceeds the required supercharger power. The combined efficiency of a representative supercharger and turbine is 50 percent. The required energy to the turbine, computed by dividing the energy required for supercharging by 0.5, is shown to be less than the available energy, thereby indicating the possibility that the turbine can develop excess energy, which could be used to augment the power of the engine, particularly at high inlet-air pressure.

SUMMARY OF RESULTS

From the tests made on small high-speed compression-ignition engines to determine the suitability of this type of engine for use in an engine-gas turbine combination, the following results were obtained:

1. The maximum power attainable in a cylinder with maximum pressure limited by structural strength was obtained by supercharging until the compression pressure equaled the maximum cylinder pressure. The lowest ratio of maximum cylinder pressure to indicated mean effective pressure was 4 when the compression ratio was 13.1.

2. Engine operation with approximately constant-pressure combustion was found possible with the displacer-type chamber at an engine speed of 2200 rpm.

3. The exhaust energy available with rich mixtures at sea level with the exhaust back pressure equal to the inlet-air pressure exceeded the energy required to drive a turbosupercharger.

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APPENDIX A

INDICATOR-CARD ANALYSIS

In order to find the temperatures and pressures before release, an analysis of indicator cards was made. The release pressure cannot be directly read from the indicator cards obtained with a modified Farnboro engine indicator because the pressure scale is inaccurate below 300 pounds per square inch.

An analysis was made of the cycle shown in figure 10 using the following symbols:

n_c exponent for compression stroke in equation $PV^n = k$

n_e exponent for expansion stroke

v_{co} volume above piston at cut-off

v_c clearance volume

v_d displacement volume

r_c cut-off ratio, $\frac{v_{co}}{v_c}$

r compression ratio, $\frac{v_c + v_d}{v_c}$

T_4 temperature at release, $^{\circ}R$

P_4 pressure at release, lb/sq ft absolute

T_1 inlet-air temperature, $^{\circ}R$

P_1 inlet-air pressure, lb/sq ft absolute

Experimental values of n_e, n_c, and r_c for rich mixtures were obtained from indicator cards taken in the tests reported herein. The cards used for lean mixtures were taken from figure 4 of reference 7. These cards were taken with an optical indicator with the engine under power for only one cycle. The compression ratio was 13.9. The exponent n_c was found to be approximately 1.35, as shown in figure 13. Figure 13 also shows n_e for a range of fuel-air ratios from 0.0581 to 0.0027. The value of the cut-off ratio was available only for rich mixtures from the indicator cards obtained in the present test. It was assumed that the cut-off ratio is constant for rich mixtures and is proportional to the fuel-air ratio up to the theoretically correct mixture. The variation of n_e, n_c, and r_c with fuel-air ratios are shown in figure 14.

Release temperatures and pressures for most operating conditions at a compression ratio of 13.1 are plotted in figure 11. These values were obtained by substituting the experimental values of the constants from figure 14, in the following equations:

$$T_4 = T_1 r^{n_c - n_e} r_c^{n_e} \tag{1}$$

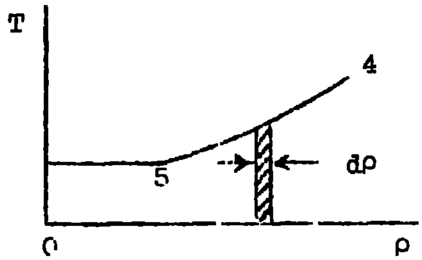
$$P_4 = P_1 \frac{T_4}{T_1} \tag{2}$$

With this information, the exhaust energy can be calculated for almost any condition.

APPENDIX B

CALCULATION OF EXHAUST-GAS TEMPERATURE AND AVAILABLE ENERGY

The temperature of the exhaust gas can be calculated from the conditions before release. During the release process two actions occur. Each small volume of gas escaping from the cylinder is assumed to undergo a free expansion with practically no change of temperature. The gases remaining in the cylinder undergo an approximately adiabatic expansion. The relation between the temperature and the density will then be approximately as shown in the following figure:



Both the temperature T and the density ρ decrease until the pressure becomes equal to the exhaust pressure. The temperature remains constant while the remainder of the gases are exhausted from the cylinder. The average temperature of the exhaust gas can therefore be obtained by integrating the area under the curve and dividing by the abscissa.

The equation for the average temperature is derived as follows:

$$T_{av} = \int_{\rho_4}^{\rho_5} \frac{T d\rho + T_5 \rho_5}{\rho_4} \qquad (1)$$

From the gas equations:

$$T = T_4 \left(\frac{\rho}{\rho_4} \right)^{\gamma - 1}$$

$$T_5 = T_4 \left(\frac{P_5}{P_4} \right)^{\frac{\gamma - 1}{\gamma}}$$

$$\rho_5 = \rho_4 \left(\frac{P_5}{P_4} \right)^{\frac{1}{\gamma}}$$

Substituting the expression for T, T_5, and ρ_5 in equation (1) and integrating

$$T_{av} = \frac{T_4}{\gamma}\left[1 + (\gamma-1)\,\frac{p_5}{p_4}\right] \qquad (2)$$

When the release conditions and the exhaust-back pressure are known, the exhaust-gas temperature can be calculated from equation (2). The values of γ were obtained from figures 2 and 4 of reference 8.

After the exhaust-gas temperature was calculated from equation (2), the energy available for a turbine cycle was obtained from figure 9 of reference 8. The results are shown plotted in figure 12.

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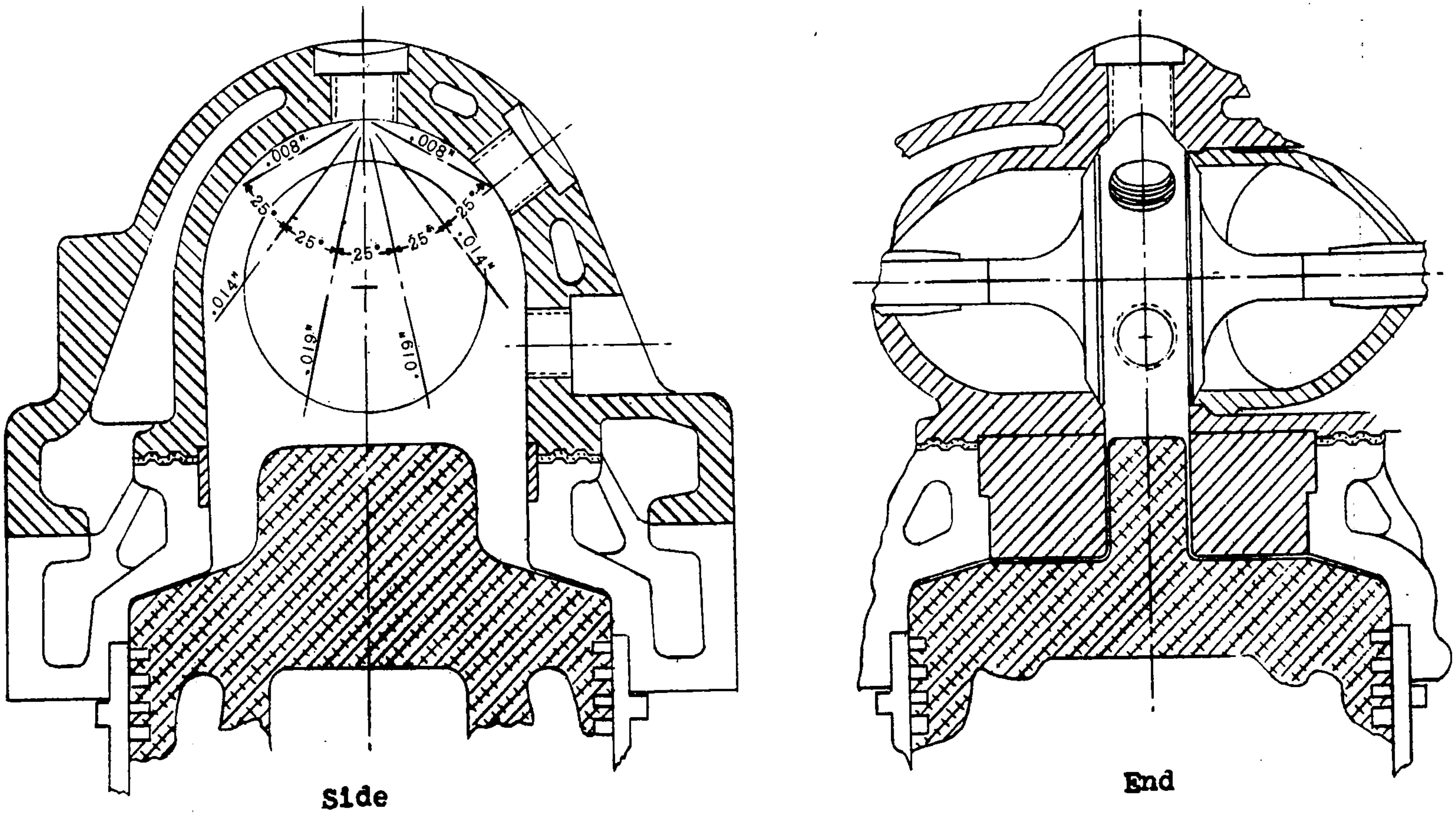

Figure 1.- Combustion chamber in the water-cooled cylinder.

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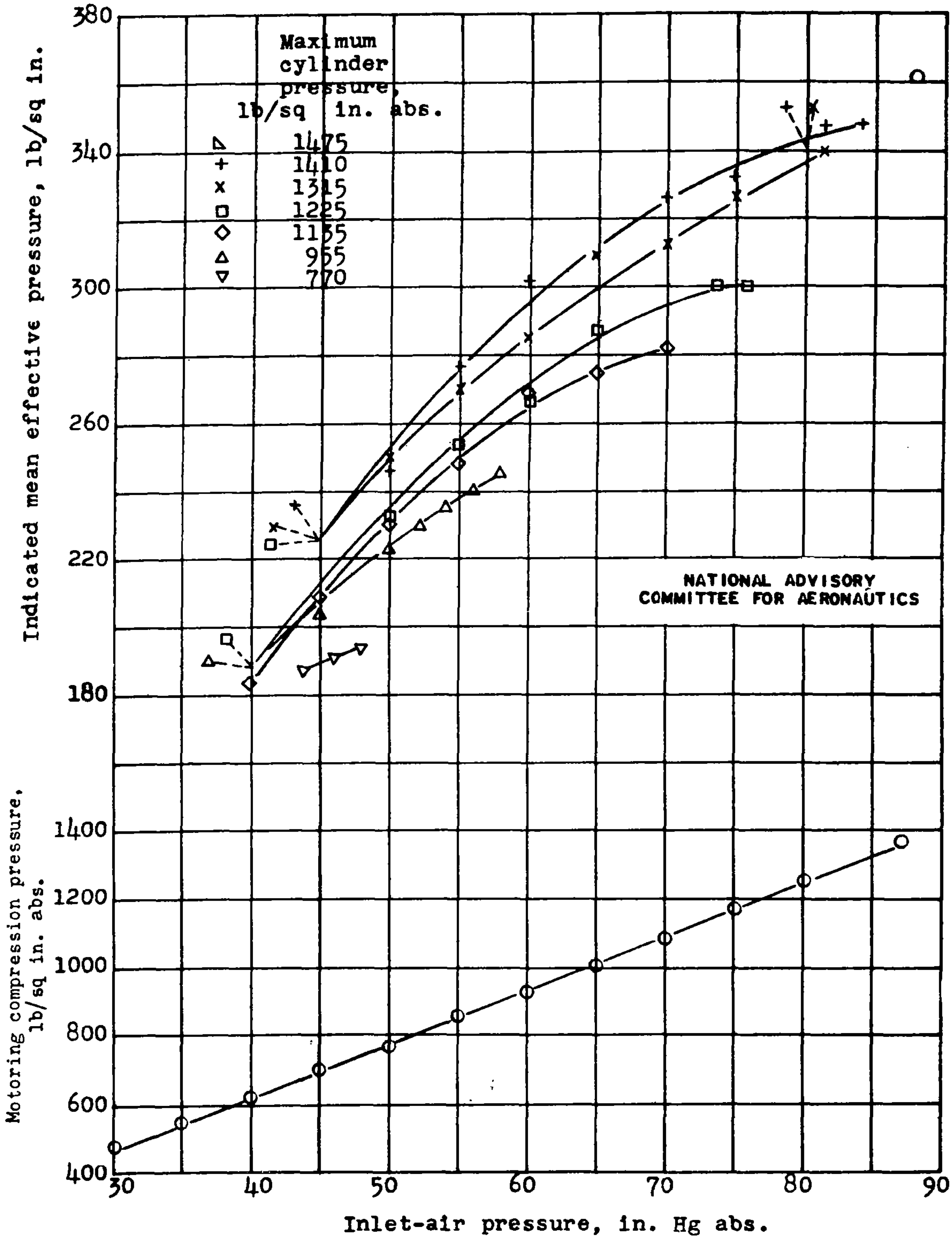

Figure 2. - Effect of inlet-air pressure on maximum power and compression pressure in a pressure-limited cycle. Compression ratio, 13.1; engine speed, 2200 rpm; inlet-air temperature 95° F.

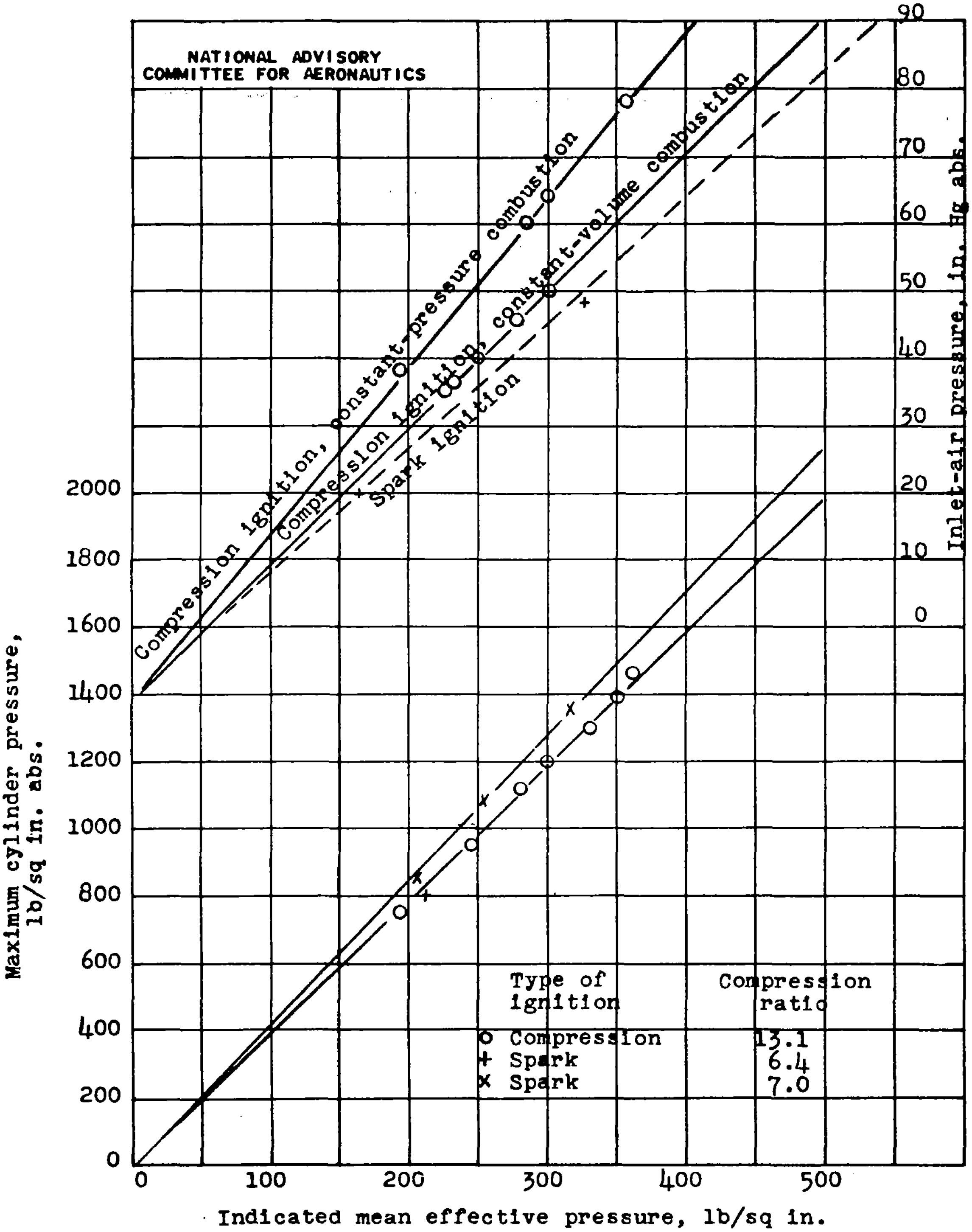

Figure 3. - Comparison of inlet-air pressures and maximum cylinder pressures in compression-ignition and spark-ignition engines.

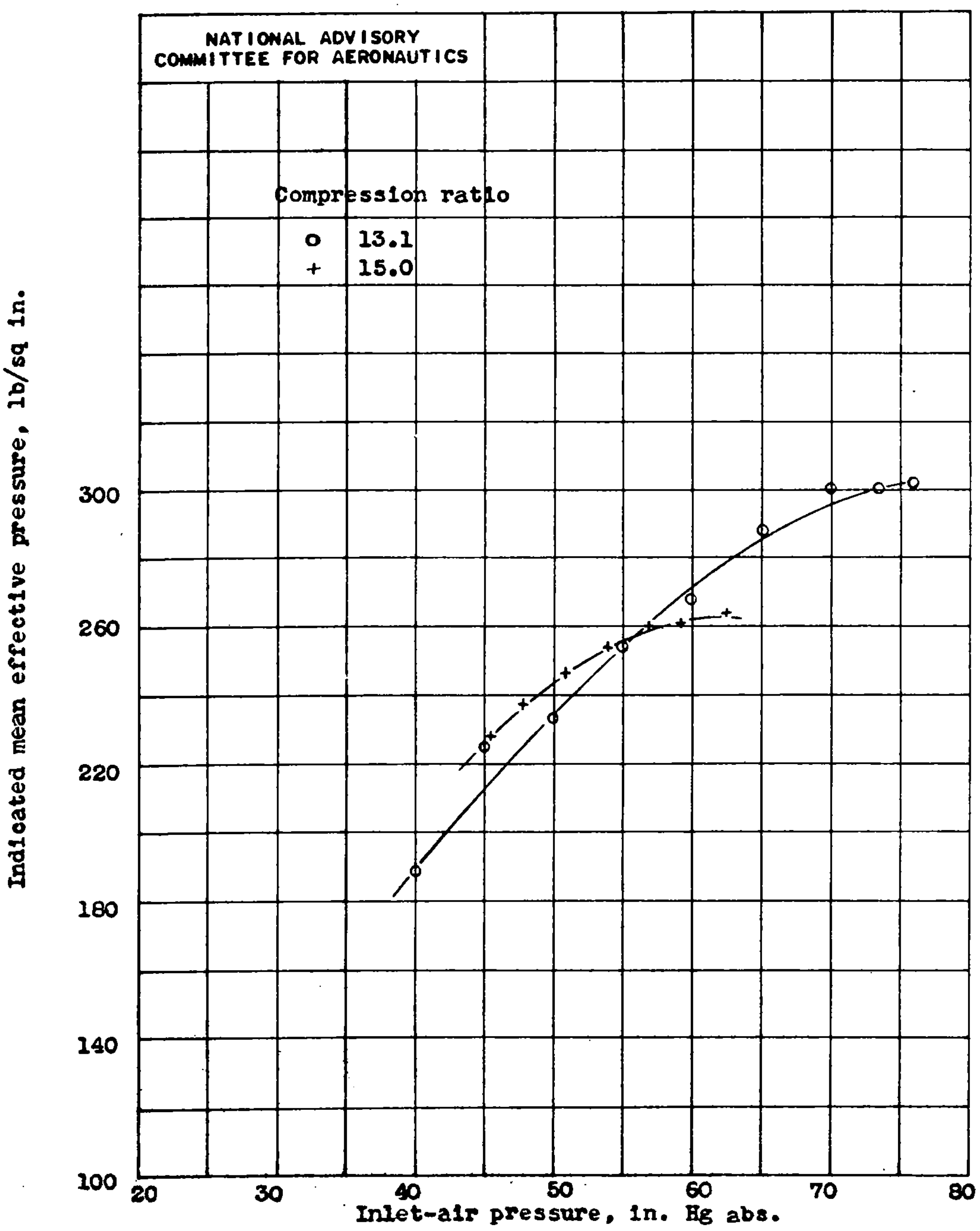

Figure 4. - Effect of compression ratio on the maximum power obtainable from pressure-limited cycles in a compression-ignition engine. Maximum cylinder pressure, 1225 pounds per square inch; engine speed, 2200 rpm; inlet-air temperature, 95° F.

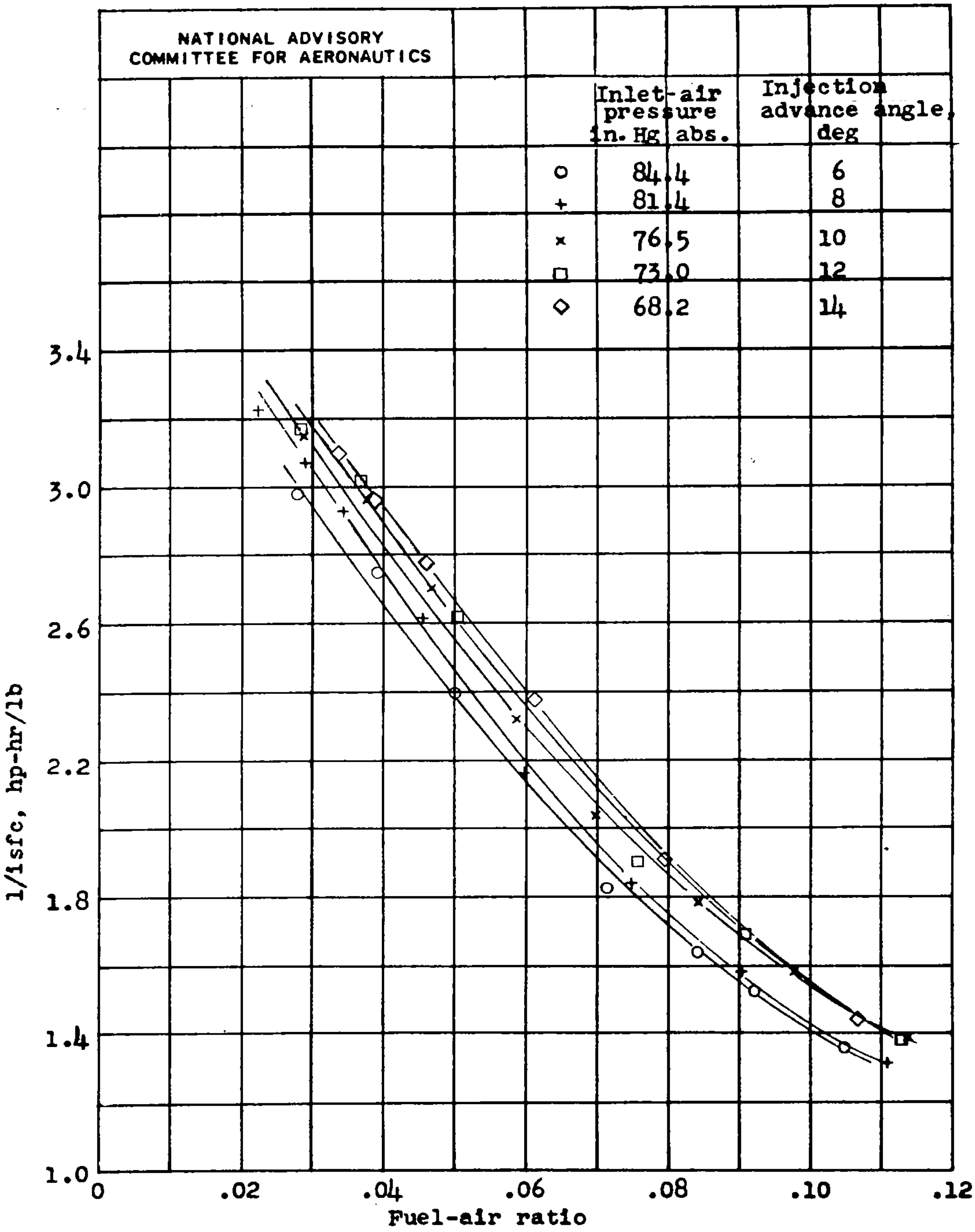

Figure 5.- Effect of injection advance angle and fuel-air ratio on fuel consumption of compression-ignition engine when maximum cylinder pressure is maintained at 1400 pounds per square inch. Compression ratio, 13.1; engine speed, 2200 rpm.

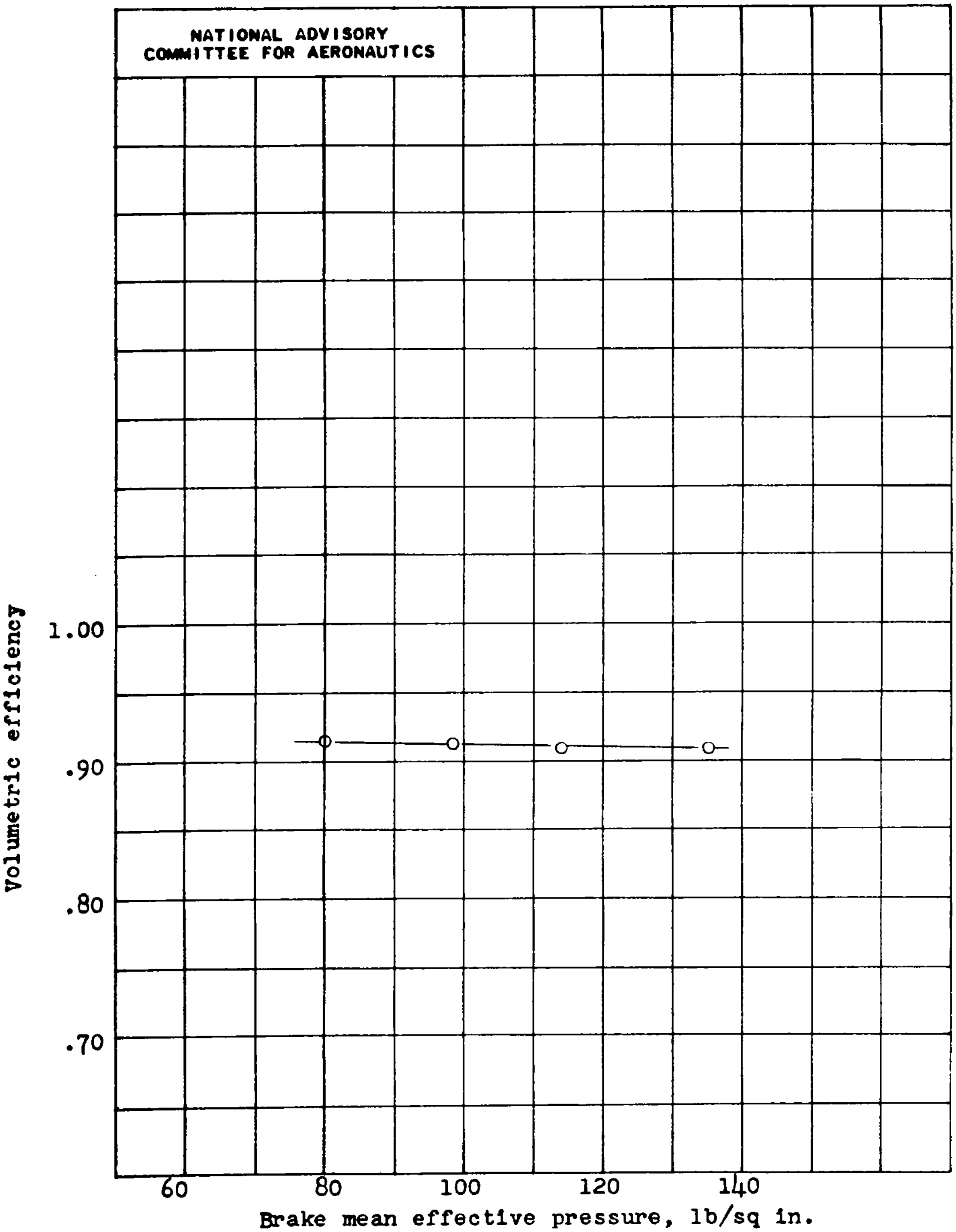

Figure 6.- Volumetric efficiency of compression-ignition engine with displacer type piston. Engine speed, 2250 rpm; inlet-air pressure, 40 inches mercury absolute.

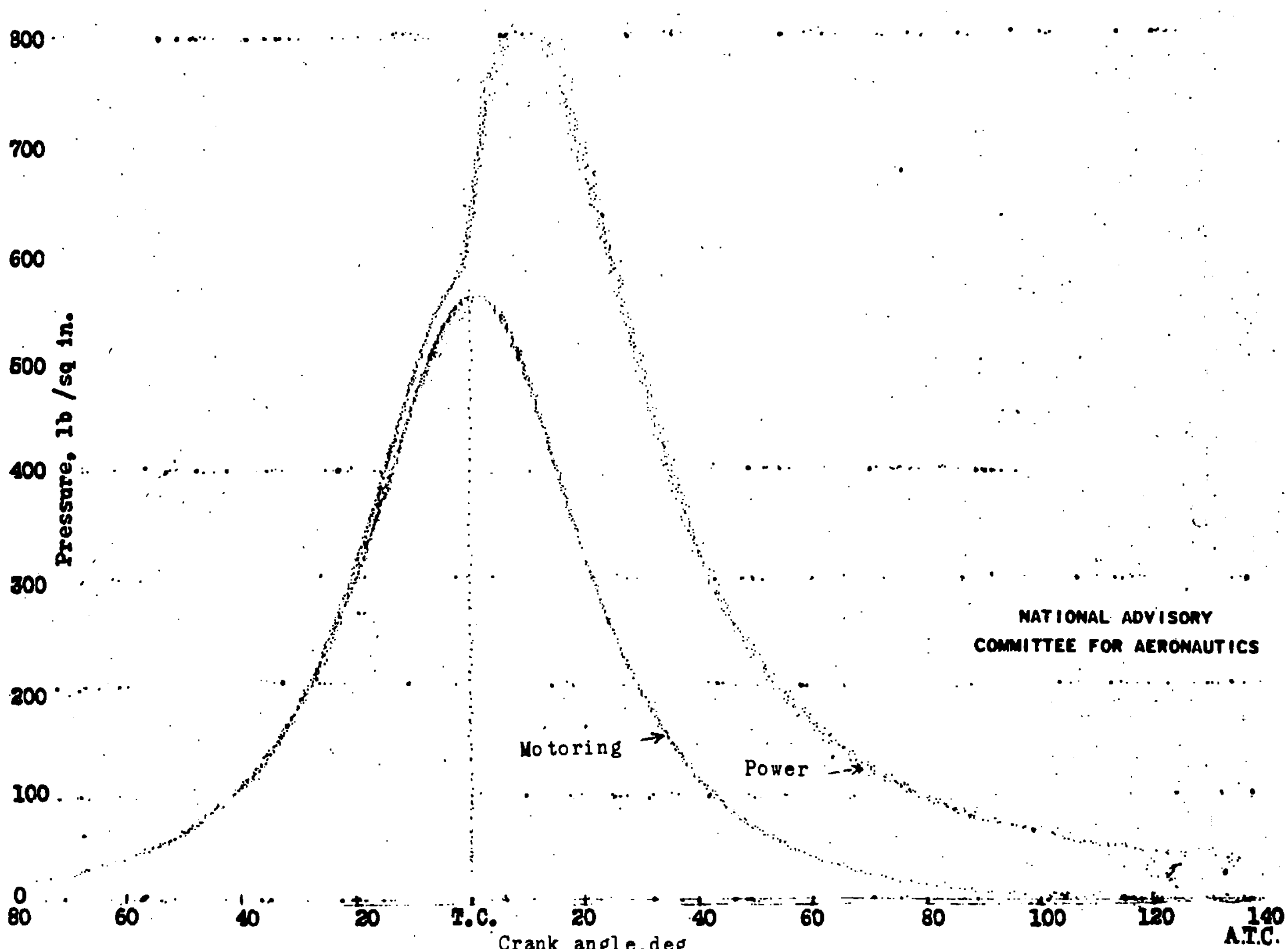

Figure 7. - Typical record obtained with a Farnboro indicator.

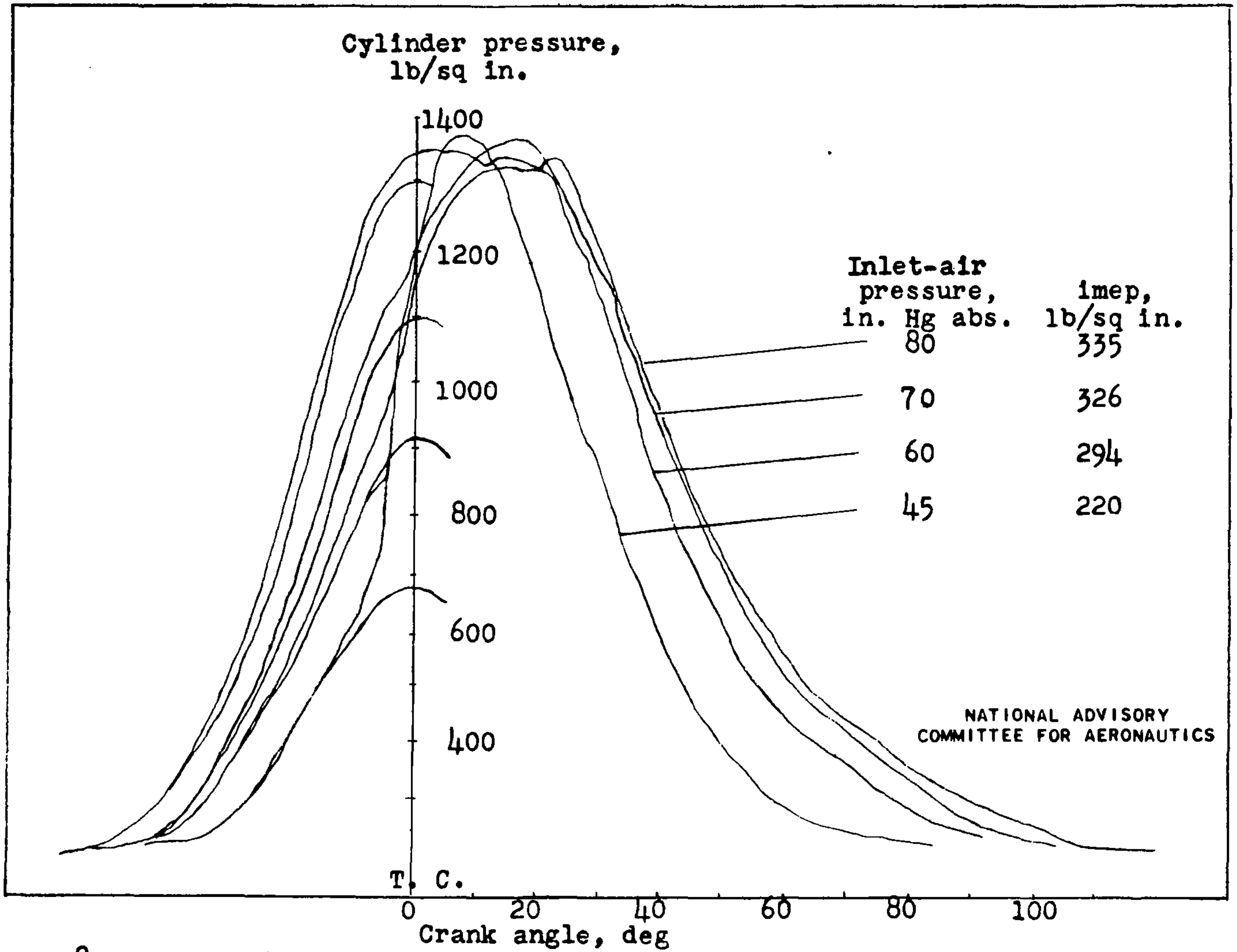

Figure 8.- Effect of inlet-air pressure on cyclic pressure when the maximum cylinder pressure is held at 1310 pounds per square inch. Compression ratio, 13.1; engine speed, 2200 rpm; inlet-air temperature, 95° F.

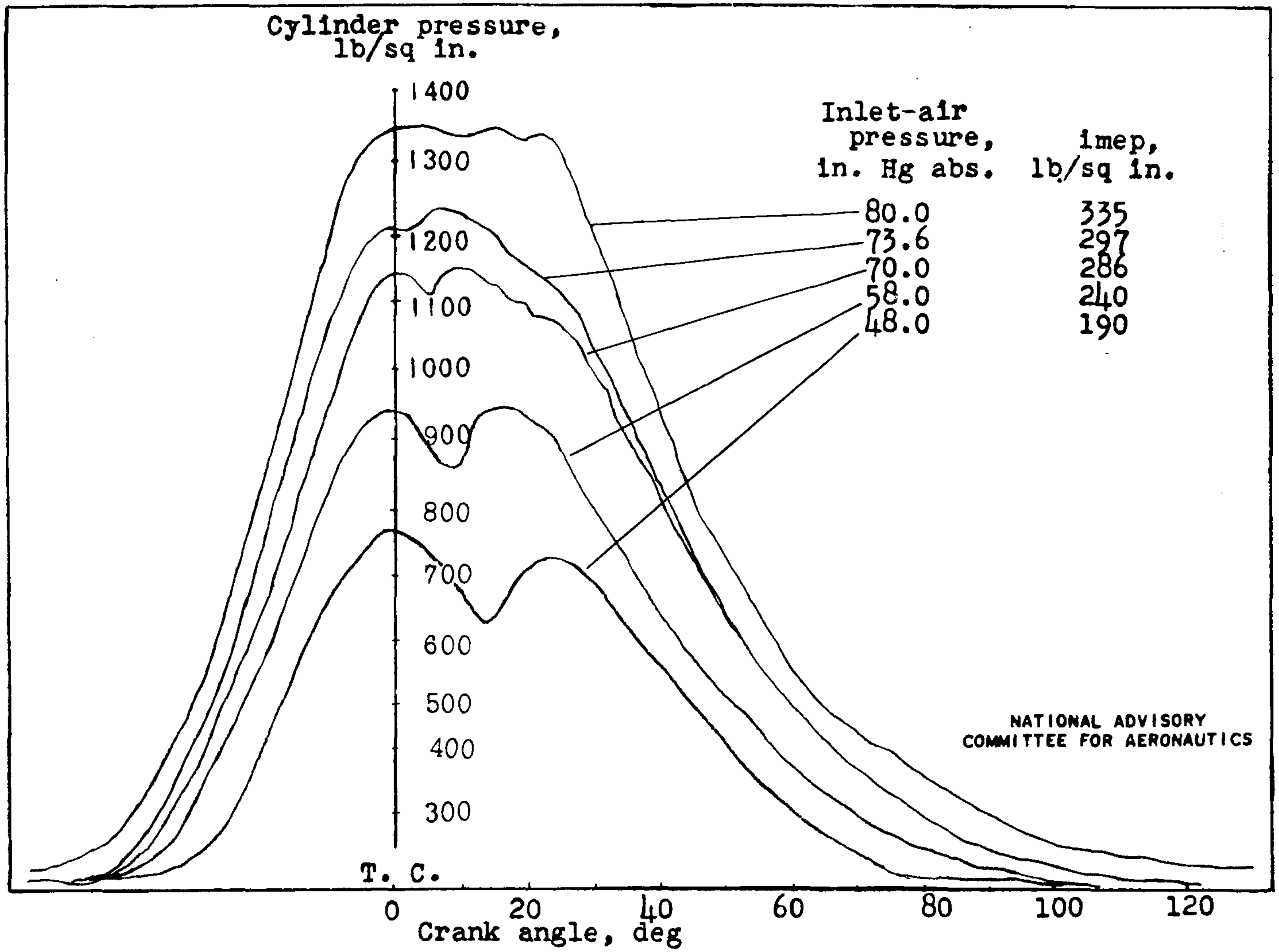

Figure 9.- Effect of maximum cylinder pressure on cyclic pressure when the maximum cylinder pressure equals the compression pressure. Compression ratio, 13.1; engine speed, 2200 rpm; inlet-air temperature, 95° F.

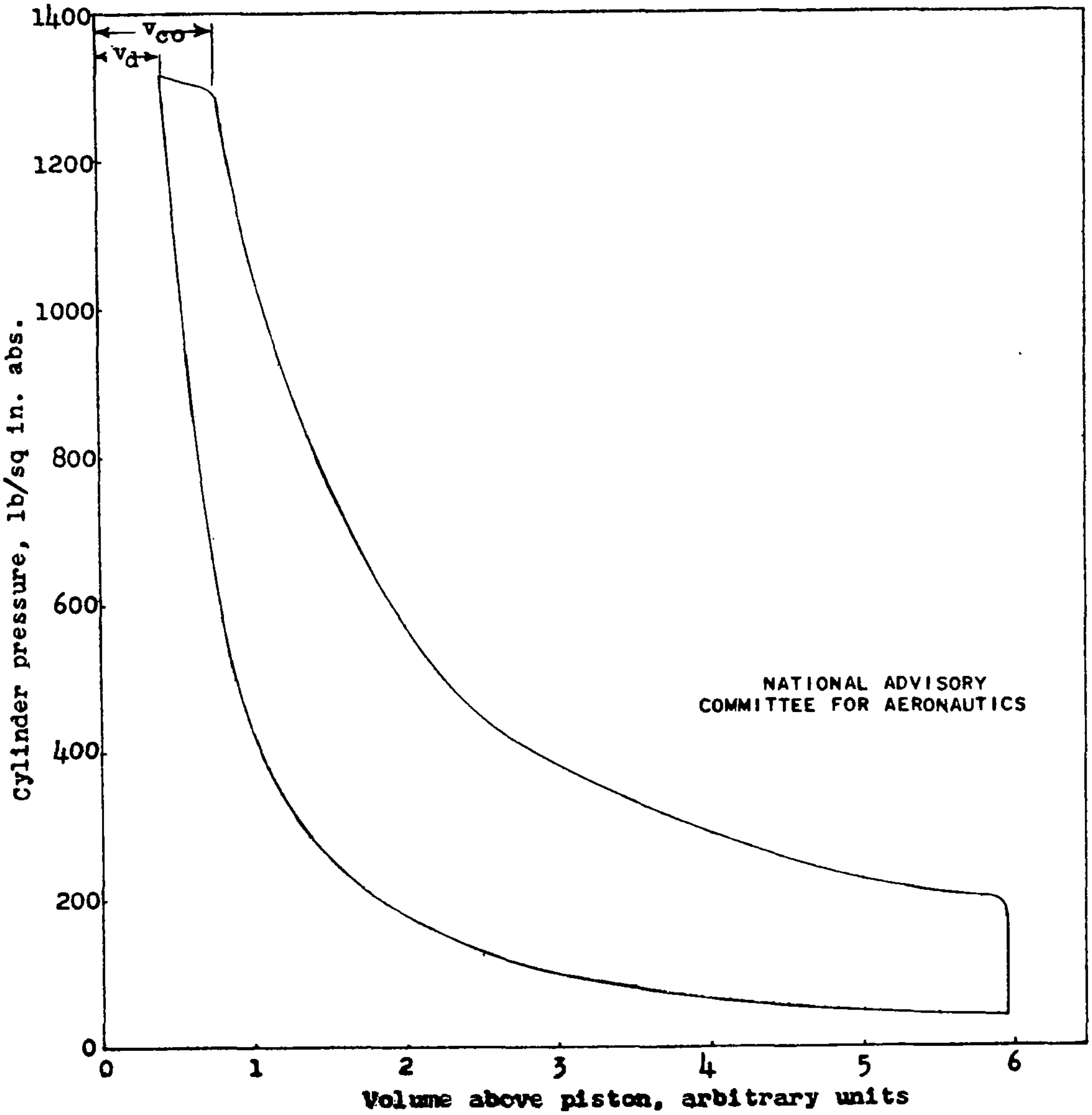

Figure 10. - High-pressure diagram obtained with Farnboro indicator and translated to pressure-volume card. Compression ratio, 13.1; engine speed, 2200 rpm; inlet-air pressure, 80 inches mercury absolute; inlet-air temperature, 95° F; measured indicated mean effective pressure, 335 pounds per square inch; area of card, 8.74 square inches; equivalent mean effective pressure, 318 pounds per square inch.

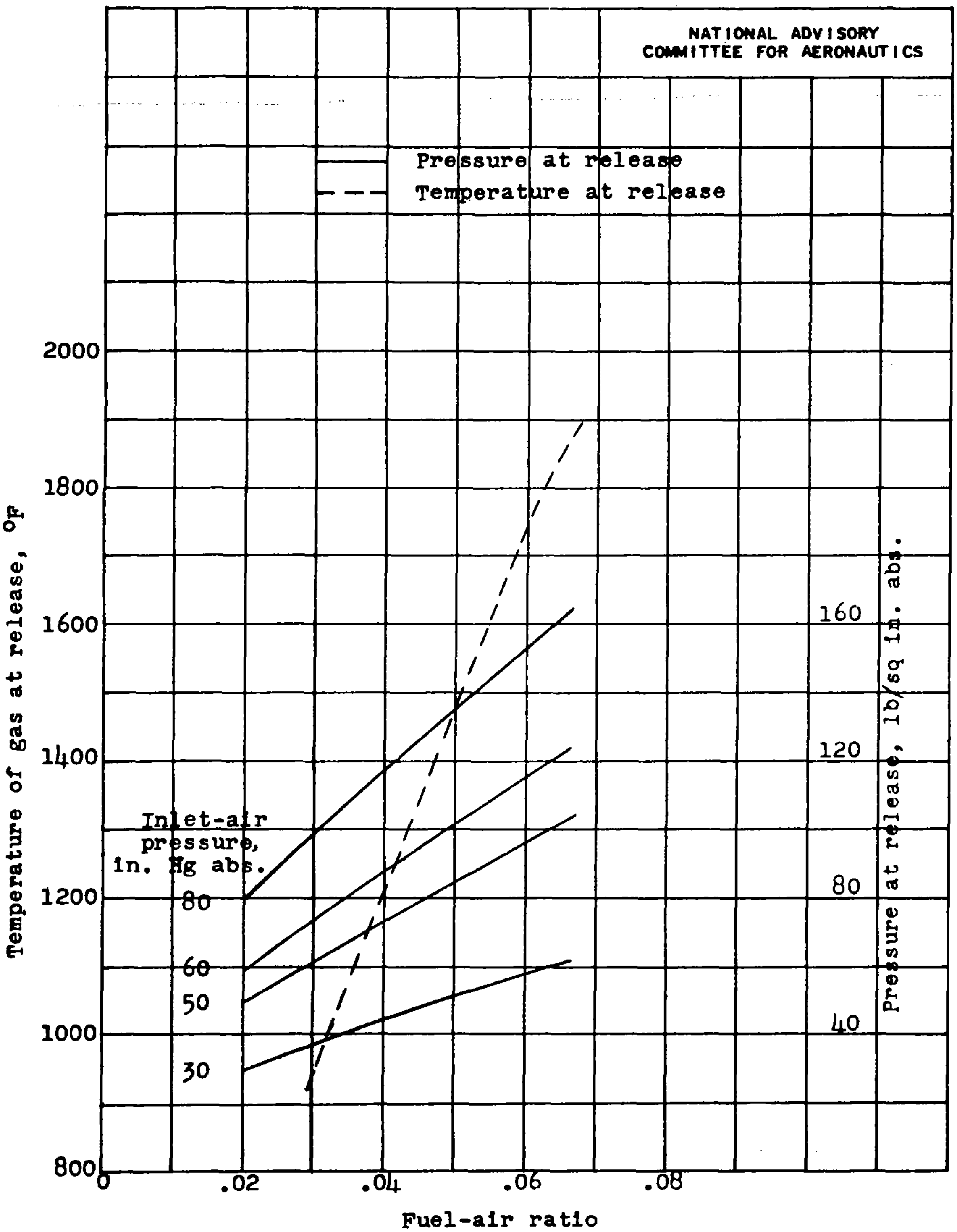

Figure 11. - Release temperatures and pressures estimated from an analysis of indicator cards obtained on a compression-ignition engine. Compression ratio, 13.1; inlet-air temperature, 95° F.

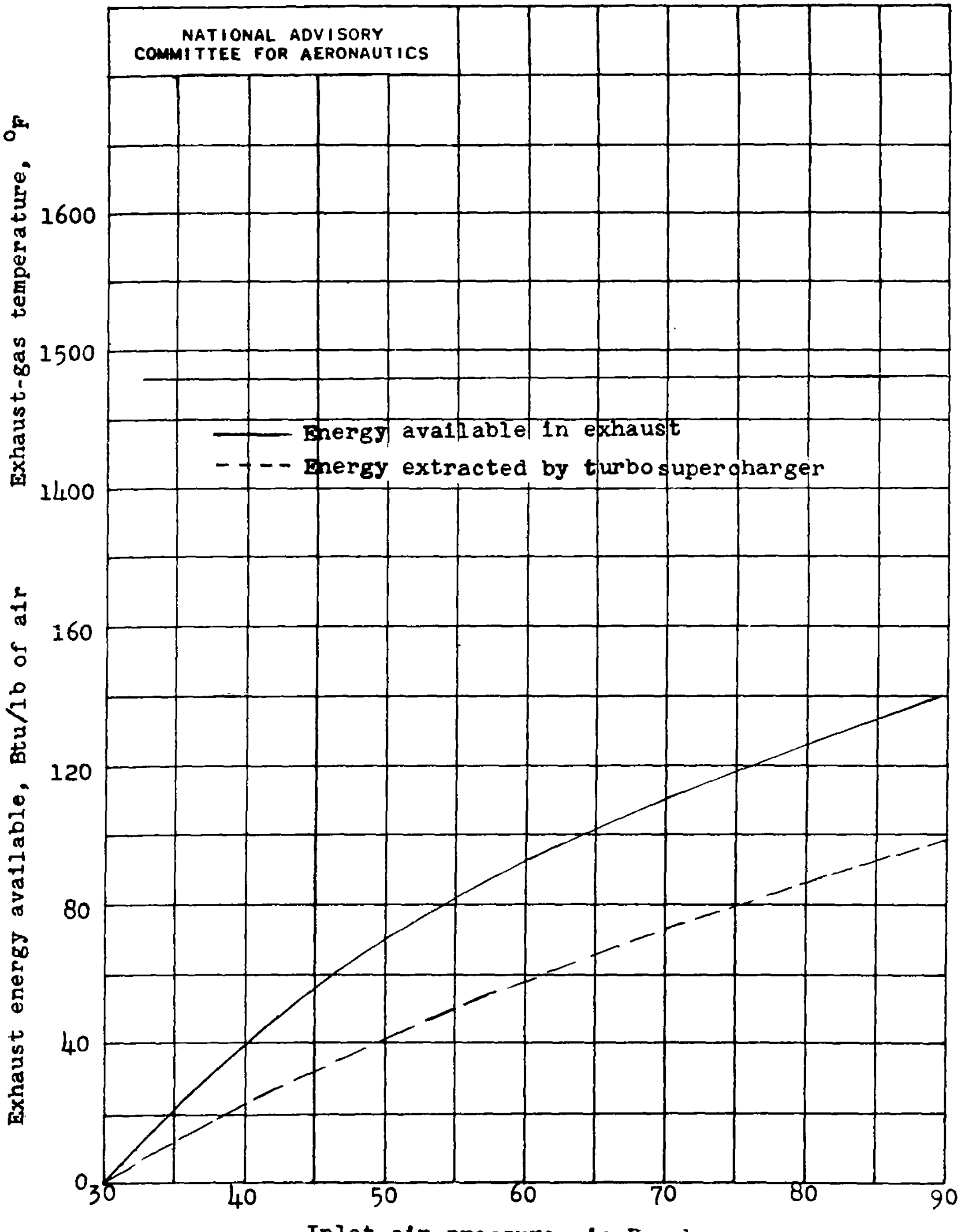

Figure 12.- Exhaust-gas temperatures and energy available to turbine estimated from analysis of indicator cards obtained on a compression-ignition engine. Compression ratio, 13.1; fuel-air ratio, 0.067; inlet-air temperature, 95° F; engine speed, 2200 rpm; exhaust back pressure is equal to the inlet-air pressure.

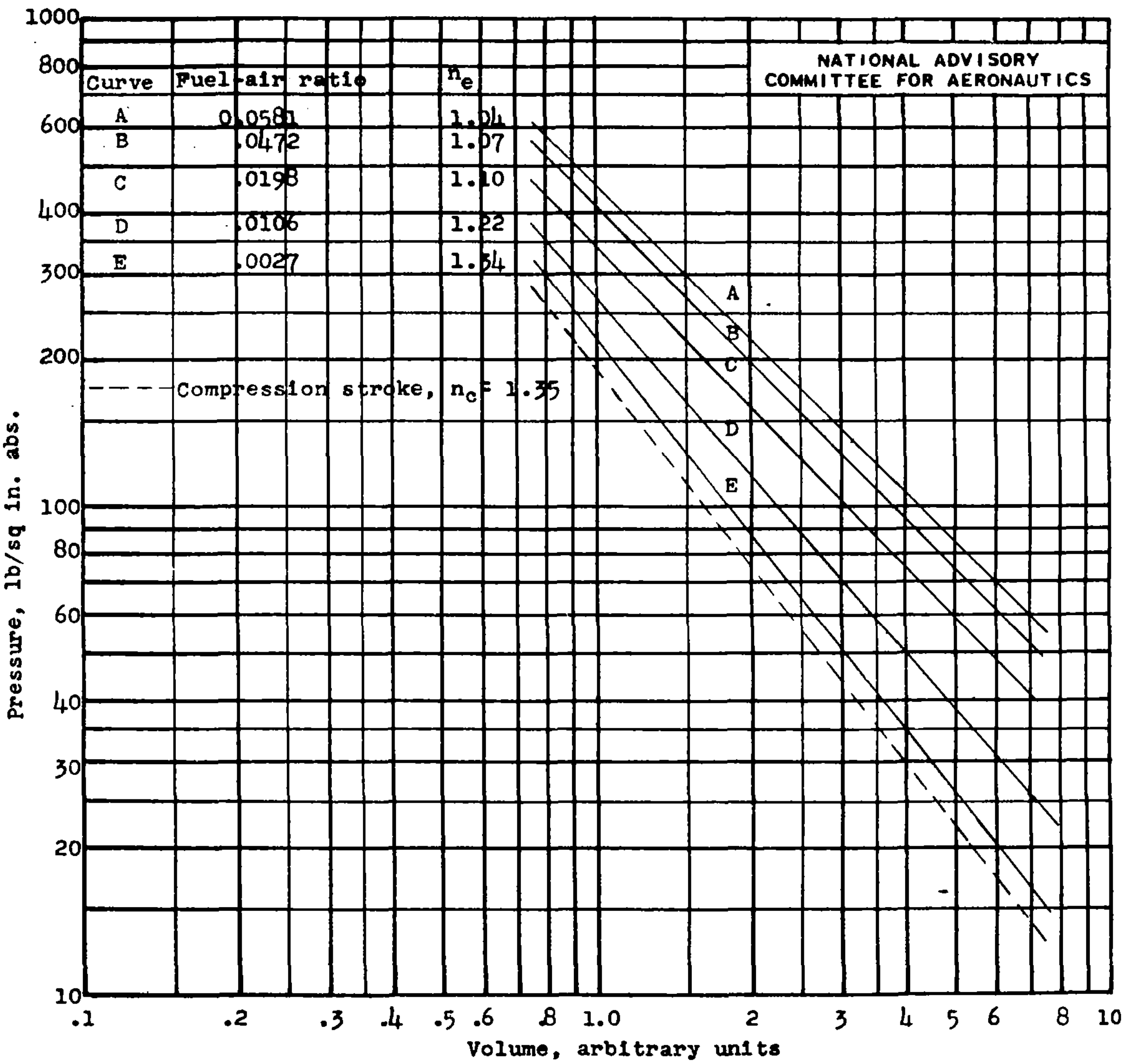

Figure 13. - Variation of expansion exponent n_e with fuel-air ratio. Compression ratio, 13.9. (Data from reference 7.)

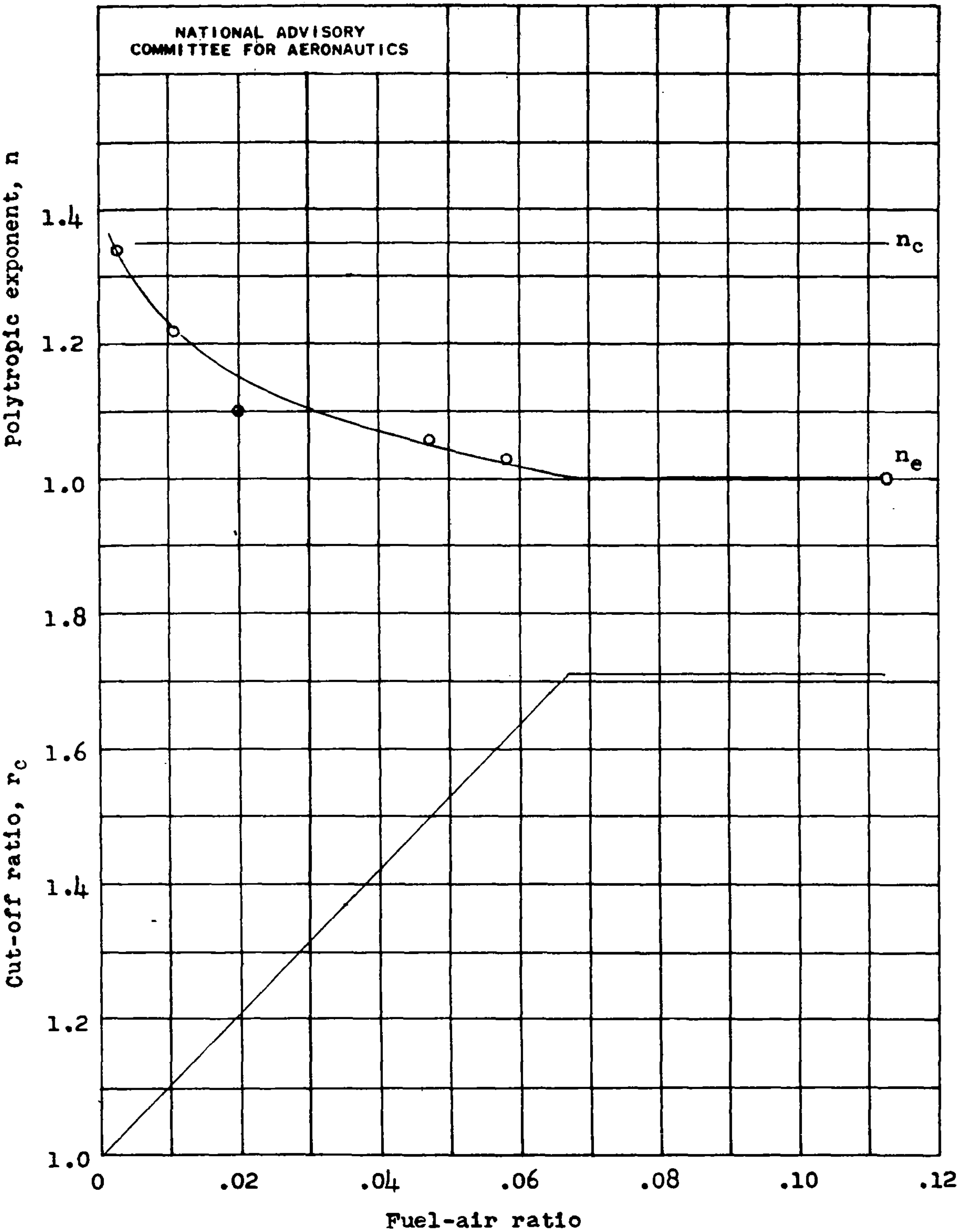

Figure 14. - Effect of fuel-air ratio on cycle exponents and cut-off ratio of compression-ignition engine.