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CORRELATION OF SINGLE-CYLINDER COOLING TESTS OF A PRATT & WHITNEY R-2800-21 ENGINE CYLINDER WITH WIND-TUNNEL TESTS OF A PRATT & WHITNEY R-2800-27 ENGINE

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SUMMARY

An equation with which it is possible to correlate the cooling characteristics of an engine has previously been developed. The validity of this equation is herein checked by the correlation of data over a wide range of engine and cooling conditions from single-cylinder laboratory tests of a cylinder of a Pratt & Whitney R-2800-21 engine. The data from the laboratory tests of this cylinder mounted on a single-cylinder test unit are compared with those from wind-tunnel tests of a Pratt & Whitney R-2800-27 multicylinder engine and a method is presented by which the pressure drop required for cooling the multicylinder engine was predicted, with little error, by means of the cooling equation for the single-cylinder engine. These predictions could be made, however, only after establishing the relationship between the hottest rear spark-plug temperature of the Pratt & Whitney R-2800-27 engine and the rear spark-plug temperature of the single-cylinder test unit. Data are also presented on the adequacy of the thermal bond between the aluminum muff and the steel barrel of the cylinder from the Pratt & Whitney R-2800-21 engine.

INTRODUCTION

An equation has been developed with which it is possible to predict the cooling characteristics of an engine (reference 1). By means of the cooling equation, it is also possible to compare the cooling of various engines. Tests have been made on many cylinders to check the validity of the equation (references 1 to 3), but the data in some cases were rather meager.

The use of a single-cylinder engine for cooling tests is advantageous in that changes can be made in a minimum of time and the cost involved in making changes is only a fraction of that for a multicylinder engine. Single-cylinder engines have, therefore, been used extensively for cooling tests. At times it is important to know the cooling performance of the multicylinder engine and much

time can be saved if this performance can be predicted from single-cylinder tests. A comparison has been made of the cooling performance of a single-cylinder engine with that of a multicylinder engine (reference 2).

The object of the present report is to present tests on the performance and cooling of an air-cooled cylinder, from a Pratt & Whitney R-2800-21 engine, mounted on a single-cylinder test base. The tests were run over a wide range of conditions to check further the correctness of the cooling equation. This cylinder had a machined aluminum-finned muff shrunk on the steel barrel, a type of construction similar to that tested with electrically heated coils. Another object of this report is to present results on the adequacy of the thermal bond between the muff and the steel barrel of this cylinder. The results of the single-cylinder-engine tests are compared with the cooling of a Pratt & Whitney R-2800-27 engine that was tested in a wind tunnel (reference 4). This comparison was possible because the cooling-surface areas of the cylinders of both engines were the same.

The cooling requirements of the multicylinder engine were obtained from the single-cylinder-engine cooling equation and from the correlation factor that was established between the single-cylinder and the multicylinder tests for several power ratings and fuel-air ratios for sea-level pressure and temperature.

This work was conducted at the Langley Memorial Aeronautical Laboratory, Langley Field, Va., during 1941.

ANALYSIS

An equation was developed in reference 1 by which the average temperature over the head and barrel of an engine cylinder could be calculated. The equation for the head, in the notation of the present paper is:

$$\frac{T_h - t_a}{T_g - T_h} = \frac{\bar{B}\, a_1\, I^{n'}}{K\, a_o \left(\Delta p/\rho_{60}\right)^m} \qquad (1)$$

where

T_h average temperature over cylinder-head surface when heat equilibrium is attained, °F

t_a temperature of cooling air at face of engine, °F

T_g mean effective gas temperature, $^{\circ}F$

$\bar{B}$ constant

a_1 internal-surface area of head of cylinder, square inches

I indicated horsepower for each cylinder

n' constant

K constant

a_o outside-wall area of cylinder head, square inches, (does not include rocker-box surfaces)

Δp pressure drop across cylinder, inches of water, (includes loss out of exit of baffle)

ρ average density of cooling air entering and leaving fins, (pound)(second)^2/foot^4

ρ_{60} density of air at 29.92 inches of mercury and 60° F, (pound)(second)^2/foot^4, 0.00245

m constant

It was also shown in reference 1 that

$$W = K_h \left(\Delta p \rho / \rho_{60} \right)^z \tag{2}$$

where

W weight of cooling air flowing over head of cylinder, pounds/second

K_h, z constants

A complete set of symbols for all equations used in this report is given in the appendix.

Equations similar to equations (1) and (2) can be written for the cylinder barrel.

As the inlet density is decreased or the quantity of heat is increased, the pressure drop across a resistance will increase for a given weight of air flowing across the resistance. This increase

is due to increased acceleration and friction losses caused by greater changes in velocity of the air through the resistance. An attempt has been made in equation (1) to compensate for such changes in density or heat output by using an average density based on the temperatures and pressures of the cooling air in front of and behind the cylinders. Tests for limited ranges of heat output and density have shown that, by using an average density in equation (1), correct values of pressure drop will be obtained from the equation. The quantitative effect of decreasing the inlet density or of appreciably increasing the heat output on Δp across an engine cylinder is not known and tests are needed to establish these relations. Attempts have been made (references 5 and 6) to estimate the pressure-drop requirements of air-cooled engines at high altitudes by simple theories involving the acceleration and friction losses. The velocity between fins may be large enough at high altitudes to cause compressibility shock to develop for which no correction to sea-level pressure drops has been made to date.

Because of the limitation involved in estimating temperatures of engines at high altitude from an equation of the form of equation (1) obtained at sea level, the data of the present report are presented in the form of an equation that is a combination of equations (1) and (2). Thus,

$$\frac{T_h - t_a}{T_g - T_h} = \frac{\overline{B}\, a_1\, I^{n'}}{\overline{K}\, a_o\, W^{m'}} \tag{3}$$

Except for a small error due to the change of viscosity of the air with altitude, equation (3) is applicable at all altitudes.

For a given cylinder, tests can be made in the laboratory by methods given in reference 1 to obtain the values of $\overline{K}$, m', $\overline{B}$, T_g, and n'. From the cylinder itself, a_o and a_1 can be obtained. The methods of obtaining a_o and a_1 for conventional cylinders are given in reference 1. When an aluminum muff is used on a cylinder barrel, a_o is the outside-surface area of the muff covered by the fins. The value of t_a to be used in equation (1) and of ρ_i, inlet density at the face of the cylinder, to determine ρ in equation (2) for the single-cylinder engine will be based on measured values of static pressure and temperature at the front of the cylinder.

The data of the cooling tests on the Pratt & Whitney R-2800-27 engine given in reference 4 were presented in an equation of the following form:

$$\frac{T_p - t_a}{T_g - T_p} = C \left[\frac{W_c^{\,n/m_1}}{\Delta p \rho_i/\rho_{60}} \right]^{m_1} \tag{4}$$

where

T_p average temperature at rear spark plugs of cylinders, $^{\circ}F$, (obtained with thermocouples embedded in spark-plug boss)

t_a temperature of air at face of engine, $^{\circ}F$

C constant

W_c weight of combustion air to engine, pounds/second

n constant

m_1 constant

Δp pressure drop across engine, inches of water, (includes loss out of exit of baffles)

ρ_i density of air at front of engine, (pound)(second)^2/foot^4

ρ_{60} density of air at 29.92 inches of mercury and 60° F, (pound)(second)^2/foot^4

T_g mean effective gas temperature, $^{\circ}F$

The data in reference 4 were also presented in the form

$$\frac{T_p - t_a}{T_g - T_p} = C_2 \left[\frac{W_t^{\,m_1'/n}}{W_c} \right]^{-n} \tag{5}$$

where

C_2 constant

m_1' constant

W_t weight of cooling air passing over engine, pounds/second

The quantity T_g is a function of fuel-air ratio, inlet manifold temperature, and spark timing. In the conventional multicylinder engine the true inlet manifold temperature is indefinite and difficult to measure because an unknown amount of fuel is vaporized in the carburetor and blower. In reference 4, the expedient was adopted of using, instead of true manifold temperature, a dry effective manifold temperature t_m defined as the sum of the carburetor-air temperature and the rise in air temperature in passing through the primary supercharger evaluated on the assumption that no vaporization of the fuel occurs. The following approximation for t_m was used

$$t_m = t_c + \frac{U_t^2}{gc_pJ} \tag{6}$$

where

t_c carburetor inlet-air temperature, °F

U_t tip speed of primary blower, feet/second

g acceleration of gravity, feet/(second)^2, 32.2

c_p specific heat of air at constant pressure, Btu/(pound)(°F), 0.24

J mechanical equivalent of heat, foot-pounds/Btu, 778

The blower tip speed U_t can be given in terms of the engine speed, the impeller diameter, and the impeller-gear ratio. The following formula was used for the effect of variation of t_m on T_g:

$$\Delta T_g = 0.8\ \Delta t_m \tag{7}$$

This equation was obtained from results given in reference 2. The effect of fuel-air ratio on the gas temperature was obtained from test data. The effect of spark advance was not required because all the tests were run at the same setting. When such methods were used to determine T_g, the data of reference 4 conformed very well to the values predicted by either equation (4) or (5).

The data of the present report are also presented in the forms of equations (4) and (5) for direct comparison between multicylinder-engine and single-cylinder-engine cooling. In the single-cylinder-engine equation T_p is the rear spark-plug temperature obtained

with a gasket-type thermocouple. In the single-cylinder-engine equation W_c was obtained by multiplying the quantity of charge air used by 18, and W_t was obtained by multiplying the sum of weights of cooling air passing over the head and the barrel of the cylinder by 18. The Pratt & Whitney R-2800-27 engine has 18 cylinders and such calculations would put the single-cylinder and multicylinder results on the same basis.

Estimates of cooling-pressure-drop requirements for the Pratt & Whitney R-2800-27 engine are made from the single-cylinder-engine equation of the form of equation (4) and from the relation between the single-cylinder-engine and the multicylinder-engine cooling. Because of the fact that ρ_i was used in equation (4), the equation is applicable only to a limited altitude range. All estimates are therefore restricted to sea-level requirements, where Army air temperature of 100° F and standard sea-level pressure of 29.92 inches of mercury are used for the atmospheric conditions. The first estimates are made by substituting these values of temperature and pressure in the cooling equation. The values of ρ_i and t_a in the equation should be for conditions of pressure and temperature that exist at the engine face as given in the definition of these symbols. At low airplane speeds ρ_i and t_a will be about the same as in the free air stream. At high speeds, however, the temperature and pressure will be greater at the engine face than in the free stream because the air slows down as it passes into the cowling. This slowing down will cause a compression of the air with little loss of pressure if the cowling entrance is well made. It is usually assumed that no loss of total pressure from the free stream to the engine face occurs and that the air has practically no velocity in front of the engine. The pressure at the engine face is then equal to the static pressure in the free stream plus the velocity pressure in the free stream. The compression is assumed to be adiabatic and the temperature at the engine face is calculated by use of the adiabatic compression exponent. The pressure and the temperature at the engine face are then

$$p_i = p_o + q_o = p_o + \frac{1}{2} \rho_o V_o^2 \qquad (8)$$

and

$$T_i = T_o \left(\frac{p_i}{p_o}\right)^{0.283} \qquad (9)$$

where

q_o dynamic pressure of air in free air stream

p_i pressure at face of engine

p_o static pressure in free air stream

T_i temperature at face of engine, $^{\circ}F$ absolute

T_o temperature of air in free air stream, $^{\circ}F$ absolute

V_o velocity of airplane

ρ_o density of air in free air stream

In the calculation of the pressure at the engine face, two factors - the factor of compressibility and the inlet-duct pressure loss - are omitted for the sake of simplicity. Inasmuch as the factor of compressibility is a positive correction and the inlet-duct-pressure loss is a negative one, the two corrections approximately cancel each other for the values given in this report.

The estimates of cooling-pressure-drop requirements for atmospheric pressure and temperature can be corrected for pressure and temperature at the engine face by means of the equation

$$\Delta p_c = \Delta p \frac{p_o}{p_i} \frac{T_i}{T_o} \left(\frac{T_p - t_o}{T_p - t_i} \right)^{\frac{1}{m_1}} \tag{10}$$

where t_o, T_p, and t_i are in $^{\circ}F$ and T_i and T_o are in $^{\circ}F$ absolute. The cooling-pressure-drop requirements based on 100° F and 29.92 inches of mercury are corrected by means of equation (10) for assumed values of airplane speed. The values of T_g to be used in the equation when estimates of cooling-pressure drop are made are obtained in the same manner as in reference 4 and as previously explained in this paper.

With fast-climbing airplanes, the temperatures of the cylinders do not reach equilibrium until some time after take-off. The cooling-pressure drop required is therefore lower than if equilibrium temperatures are reached. In order to determine this pressure drop, the cooling equations of the forms of equations (1) and (4) would have to be revised by methods given in reference 3 and would be very complex. The cooling-pressure drops are therefore estimated in this report for climb conditions from equations established for heat-equilibrium conditions. The estimated cooling-pressure drops are therefore somewhat greater than if the more complex equations for conditions of no heat equilibrium were used.

APPARATUS

The cylinder from the Pratt & Whitney R-2800-21 engine was mounted for testing on a single-cylinder test unit, as shown in figure 1. The bore and the stroke were $5\frac{3}{4}$ and 6 inches, respectively, and are the same as the bore and the stroke of the multicylinder engine. The compression ratio used in the tests was 6.7, which is also the same as in the multicylinder engine. The cams used in the tests gave a valve timing that was approximately the same as in the Pratt & Whitney R-2800-21 engine, namely,

Intake opens, degrees B.T.C.	20
Intake closes, degrees A.B.C.	76
Exhaust opens, degrees B.B.C.	76
Exhaust closes, degrees A.T.C.	20

The cylinder was enclosed in a sheet-metal jacket connected to a centrifugal blower that furnished the cooling air. A diagram of the jacket is shown in figure 2. The jacket had a wide entrance section to provide a low air velocity in front of the cylinder; the rear half of the jacket fitted closely against the fins in order that the air might be effectively used. The area of the exit of the jacket was 1.6 times the free-flow area between the fins. A partition was located in the exit duct of the jacket in order to separate the air that flowed over the head from the air that flowed over the barrel.

In some tests baffle plates were placed in front of the cylinder, as shown in figure 2, in order to change the direction of the air stream. The use of such devices has, in the past, resulted in making the cooling of the test cylinder more like that in flight. Thus, the cooling in the laboratory was a closer approach to the cooling encountered in flight when this turbulence device was used.

Cooling air was furnished by a centrifugal blower connected to the cylinder jacket by an air duct. The air quantity was measured with thin-plate orifices in the ends of an orifice tank that was connected to the inlet of the blower.

A Nash blower was used to provide inlet manifold pressures above atmospheric pressure. A surge tank was installed in the inlet system above the engine to reduce pulsations. The engine power was absorbed by an interconnected water brake and an electric dynamometer, and the torque was measured by dial scales. Standard test-engine equipment was used in measuring engine speed and fuel consumption. (See fig. 1.)

Iron-constantan thermocouples and a potentiometer were used for measuring the cylinder temperatures. The thermocouples were made of wire of 0.016-inch diameter, enameled and silk-covered. The temperatures were measured at 22 points on the head, 2 points on the flange, and 20 points on the barrel, as shown in figure 3. The 22 thermocouples on the head and 10 of the thermocouples on the barrel were peened into the aluminum as shown in figure 4. The other 10 thermocouples on the barrel were placed on the steel through holes drilled through the aluminum muff. Hard-rubber plugs were placed in the holes as shown in figure 4 and the metal at the top of the hole was peened around the plug to hold it in place. Whenever a thermocouple was placed on the steel, another was placed on the aluminum muff as close as possible to the first thermocouple. The thermocouples on the muff are shown by the designation (Al) in figure 3.

The temperatures of the cooling air passing over the cylinder barrel and the cylinder head were measured separately. The cooling-air temperatures were measured at the inlet of the jacket near the cylinder by two multiple thermocouples consisting of two thermocouples connected in series and at the outlet of the jacket by two multiple thermocouples consisting of four thermocouples connected in series. Cold junctions of all thermocouples were located in an insulated box. The cold-junction temperatures, the air temperatures at the thin-plate orifices, and the air temperatures in the surge tank above the carburetor were measured with liquid-in-glass thermometers.

The pressure drop across the cylinder was measured by a static ring located around the air duct ahead of the engine cylinder, where the velocity pressure was negligible. The pressure drop across the cylinder with the baffle plates placed in the jacket entrance (fig. 2) was obtained with total-head tubes placed between the fins of the barrel and the head near the front of the cylinder. The locations of the static ring and the total-head tubes are shown in figure 2. The static ring and the total-head tubes were connected to water manometers. An alcohol-filled micromanometer was used to measure the pressure in the orifice tank, and a mercury manometer was used to measure the engine-inlet pressure in the surge tank above the carburetor.

The quantity of engine combustion air was measured with a thin-plate orifice in conjunction with a multiple manometer that indicated both the total pressure upstream of the orifice in inches of mercury and the pressure drop across the orifice in inches of water. Fuel-air ratios were calculated from the measurements of air and fuel consumption.

METHODS AND TESTS

Engine Cooling

Tests were conducted to determine the cooling and the power characteristics of the cylinder to be used in predicting performance as limited by cooling. Calibration tests were made to determine the weight of air flowing over the head and barrel of the cylinder per unit time as a function of the pressure drop across the cylinder. The constants in equations for the barrel and head of the form of equation (2) are obtained from these tests. The calibration curves are shown in figure 5. These curves are applicable for the jacket both with and without the baffle plates. The conductance of the baffles calculated in the conventional manner is 0.0903, based on the frontal area of the 2800 engine.

Tests of the cylinder were conducted to determine the constants in equation (1) and in a similar equation for the cylinder barrel by methods fully described in reference 1. These tests were made both with the jacket of wide entrance and with the jacket having baffle plates in the entrance. From these tests, the cooling characteristics of the cylinder with turbulence at the jacket entrance can be compared with the cooling characteristics without turbulence at the jacket entrance. Likewise from these tests, the cooling characteristics of the cylinder can be compared with those of other cylinders.

The value of T_g to be used in equation (1) depends on the fuel-air ratio, the spark setting, the intake manifold temperature, the compression ratio, and the exhaust pressure being used. The effective gas temperature has been found to vary with all of these factors except the compression ratio, for which no tests have been made to date (references 1 and 2). Tests were made on the present cylinder to determine the values of T_g for various fuel-air ratios only by the methods given in references 1 and 2, and these values were used in the estimation of cooling-pressure drops. The variation of T_g with spark setting and exhaust pressure would be small for the range of conditions covered by these performance-prediction calculations.

Tests were also made over a wide range of engine and cooling conditions to determine the temperature distribution over the cylinder and to establish the validity of the cooling equations. These conditions will be given in detail later in the paper.

Engine Performance

Tests were conducted at wide-open throttle with a maximum-power mixture over a range of speeds from approximately 1600 to 2600 rpm.

In addition, tests were made at wide-open throttle and at 2100 rpm to determine the variation of power and specific fuel consumption for a range of fuel-air ratios. From these speed and fuel-air-ratio tests, the values of fuel-air ratio for maximum-power, maximum-economy, and full-rich mixtures were obtained.

A series of tests was made at wide-open throttle, at maximum-power mixture, and at 2100 rpm for a range of manifold pressures from approximately 29 to 49 inches of mercury absolute. The object of these tests was to determine the cooling of the cylinder near its rated power output.

A spark setting of 20° B.T.C. and gasoline conforming to Army Specification No. 2-92, grade 100 (100-octane number, Army method) were used for all the tests. The engine horsepowers given in this paper are all observed values. The friction horsepowers were determined by motoring the engine at the inlet pressures and speeds used in the power runs. The methods of computing the results are fully described in references 1, 2, and 7.

RESULTS AND DISCUSSION

Engine Cooling

Cooling equations for cylinder of Pratt & Whitney R-2800-21 engine. - From the test data, constants were obtained to insert in equations for the cylinder head and the barrel with and without baffle plates in the jacket entrance. For the case of the jacket without baffle plates, the equations are

$$\text{(Head)} \qquad \frac{T_h - t_a}{T_g - T_h} = \frac{0.0499\, a_1\, I^{0.61}}{1.48\, a_o\, W^{0.69}} \qquad (11)$$

$$\text{(Barrel)} \qquad \frac{T_b - t_a}{T_g - T_b} = \frac{0.0152\, a_1\, I^{0.81}}{2.87\, a_o\, W^{0.64}} \qquad (12)$$

The equations with baffle plates in the jacket entrance are

$$\text{(Head)} \qquad \frac{T_h - t_a}{T_g - T_h} = \frac{0.0499\, a_1\, I^{0.61}}{1.86\, a_o\, W^{0.69}} \qquad (13)$$

$$\text{(Barrel)} \qquad \frac{T_b - t_a}{T_g - T_b} = \frac{0.0152\, a_1\, I^{0.81}}{3.44\, a_o\, W^{0.64}} \qquad (14)$$

The equations for the barrel are based on the average temperature of the 10 thermocouples on the steel liner.

The areas of the cylinder are as follows:

	Head	Barrel
Effective internal-wall area a_1, square inches	78.4	85.5
Effective external-wall area a_o, square inches	145.0	72.2

The outside-wall heat-transfer coefficient U may be obtained from the equation in reference 1:

$$U = \overline{K}W^{m'} \qquad (15)$$

Also

$$U = \frac{H}{a_o\ (T_h - t_a)} \qquad (16)$$

where H is the heat transferred from cylinder wall to cooling air, Btu/hour. The method of calculating H is given in references 1 and 2. If U is obtained from equation (16) and plotted against W, the values of $\overline{K}$ and m' can be obtained.

The values of $\overline{K}$ and m' in equations (13) and (14) were determined in this manner and the values of $\overline{K}$ and m' in equations (11) and (12) were checked by this method. The curves of U plotted against W are shown in figure 6. From these curves the values of $\overline{K}$ and m' for the case of no baffle plates are

	Head	Barrel
$\overline{K}$	1.49	2.87
m'	.69	.64

which are close to the values in equations (11) and (12). The baffle plates increased the values of U about 20 percent as compared with the cooling without baffles.

Equations (11), (12), (13), and (14) can be converted to the form of equation (1) by substituting in these equations the relation between W and $\Delta p\rho/\rho_{60}$ given in figure 5.

Variation of T_g with fuel-air ratio. - The results of the tests to determine the variation of the effective gas temperature with fuel-air ratio are shown in figure 7. It has become general

practice to reduce T_g to a dry intake manifold temperature of 80° F (reference 4) by making use of equations (6) and (7). No supercharger was used in the present tests between the carburetor and the engine, and the temperature of the carburetor air was 88° F (fig. 7), which is equivalent to the dry manifold temperature. Therefore, T_g at 80° F will be equal to the results of figure 7 minus 0.8 (88 - 80). This correction should result in the head curve of figure 7 passing through 1150° F and the barrel curve through 600° F at a fuel-air ratio of 0.08. The temperatures of 1150° and 600° F were used to establish the cooling equations of the engine from tests at fuel-air ratios of approximately 0.08. Applying the temperature corrections to figure 7 will not result in the curves passing through the temperature expected. The results of figure 7 were obtained from only one series of tests and, although the curves did not pass through the values expected when corrected to 80° F dry manifold temperature, the results were close enough for practical purposes and no further tests were made. The values of T_g of figure 7 were used in determining correlation curves to be presented without correction to 80° F dry manifold temperature. The tests made to determine the correlation curve were at dry manifold temperatures in the range of the tests made to establish figure 7.

When only one series of tests is made to establish the variation of T_g with fuel-air ratio and the values of T_g are a little different than expected at 0.08 fuel-air ratio and 80° F dry manifold temperature, the curve should be presented in the following manner for greater accuracy of results: plot the ratio of values of T_g obtained from the tests at any fuel-air ratio to the T_g obtained at 0.08 fuel-air ratio against fuel-air ratio. Then use the curve to find T_g at 80° F by multiplying these ratios by the T_g used to establish the cooling equations (in the present case 1150° F for the head and 600° F for the barrel).

General correlation curve. - The cooling of a cylinder or engine can be determined by means of the indices $\dfrac{T_h - t_a}{T_g - T_h}$ and $\dfrac{T_b - T_a}{T_g - T_b}$ (reference 2). The lower the value of the indices for given engine and cooling conditions the better the cooling of the cylinder. From equation (3) it can be seen that, if such indices are plotted against $\dfrac{(I/v)^{n'}}{W^{m'}}$ in which I is the power output and v is the displacement of the cylinder or engine, on paper having logarithmic ordinate and abscissa scales, straight lines with a slope of unity should be obtained. All data should fall on the same line regardless of engine and cooling conditions if the correct T_g is used in the cooling indices. Such lines are proof of the validity of the cooling equation.

Tests were made for a large range of engine and cooling conditions, as shown in the table of figure 8, and calculations of the foregoing parameters were made. The results are plotted in figure 8. From equation (3) it can be shown that the value of the cooling index when $\frac{(I/v)^{n'}}{W^{m'}} = 1$ should be $\frac{\overline{B}\, a_1\, v^{n'}}{\overline{K}\, a_o}$. From the present cylinder $v = 155.9$ cubic inches. When the values of $\overline{B}$ and $\overline{K}$ for equations (11) and (12), a_1, a_o, n', and v are substituted in this formula, the value of the head index is 0.396 and the barrel index 0.377. A 45° line drawn through the points for the head and another through the points for the barrel intersect the cooling indices at 0.396 and 0.389, respectively. The line through the head points does not quite fit some of the points, but the difference in head temperature between the points and the line is not very great. The intercept at $\frac{(I/v)^{n'}}{W^{m'}} = 1$ for the barrel is a little higher than the calculated intercept but the difference is very small. In general, the points for the barrel fall very near the line except for a few at the upper end of the curve. From these data the application of the cooling equation for a wide range of engine and cooling conditions is verified.

Temperature distribution over cylinder. - In reference 2 equations were developed in which temperatures at individual points on the cylinder were set up as functions of the engine and cooling conditions. To determine temperatures at individual points by such equations is rather tedious and, if some simple relations between the average cylinder temperatures and pertinent individual temperatures could be obtained, only the equations for the average temperatures and these simple relations would be needed to determine the individual temperatures.

Various individual cylinder temperatures have been plotted in figure 9 against either the average head or the average barrel temperature, depending upon the location of the thermocouple whose temperature is being plotted. The various kinds of symbols used to plot the test results indicate a wide range of cooling and engine conditions, as shown in the table in figure 9. No trend of individual temperature with variation of cooling or engine conditions for a given average head or barrel temperature could be noted, and faired curves have been drawn through all the test points for each location. These curves show that the temperatures at individual points on the cylinder barrel remained approximately constant for a given average barrel temperature; the same condition was true of individual temperatures on the cylinder head for a given average head temperature, regardless of engine and cooling conditions. An analysis of the

cooling equations for the head and the barrel of the cylinder and the curve of average head temperature against rear spark-plug temperature will show that, for a given rear spark-plug temperature, although T_h is constant for any engine and cooling condition, T_b will not remain constant as engine and cooling conditions are varied. The only case in which T_b will remain constant, when the rear spark-plug temperature is constant but the cooling and engine conditions are varying, occurs when the values of m' and n' are the same in both the head and the barrel equations. From curves such as are shown in figure 9, the limiting value of T_h to be used in equation (11) for determining the cooling performance can be obtained if the limiting value of the rear spark-plug temperature is known.

Most of the data in figure 9 are for the tests without baffle plates. One curve is shown, however, for the tests with baffle plates, namely, T_h plotted against rear spark-plug temperature. For the same value of T_h the rear spark-plug temperature is much hotter with baffle plates than without baffle plates. From the results of figures 6 and 9 for the same weight of cooling air and horsepower, the decrease of rear spark-plug temperature is small when baffle plates are used as compared with the temperature when baffle plates are not used. The baffle plates, however, appreciably decreased the T_h for given cooling and engine conditions. (See fig. 6.)

Plotted in figure 9 are the average barrel temperatures based on the temperatures of the aluminum muff against the average barrel temperatures T_b based on the steel temperatures. A curve with a slope of unity fits the data closely, which shows that the temperature drop from the steel to the aluminum muff is negligible or that the thermal bond between the muff and the steel is very good for the cylinder tested.

Comparison of multicylinder and single-cylinder engine cooling. - The results of the tests on the Pratt & Whitney R-2800-27 engine of reference 4 and the present tests plotted in the forms of equations (4) and (5) are shown in figure 10. For the multicylinder tests n/m_1 was equal to 1.76 and m_1 was 0.32; m_1'/n was 1.14 and n was 0.565. Plotting the single-cylinder results in like manner gives the same exponents, as shown in figure 10. Points are shown on the curves for the single-cylinder engine plotted in the form of equation (4) but no points are shown for the curves of this engine plotted in the form of equation (5). The curves in the form of equation (5) were obtained from the equations of the curves in the form of equation (4) and from plots of W against $\Delta p\rho_i/\rho_{60}$. The value of T_p in the cooling index for the multicylinder-engine curves is the average of the rear spark-plug temperatures of all cylinders, and the value of T_p for the single-cylinder-engine curves is the temperature of the rear spark plug.

The results show that the cooling of the engines is in better agreement on a pressure-drop basis than on a weight-of-air basis. The results of the tests with baffle plates on the single-cylinder-engine setup were in better agreement with the multicylinder results than the results of the tests without baffle plates. For a value of $\frac{W_c^{1.76}}{\Delta p \rho_i/\rho_{60}}$ of 0.3, T_g of 1150° F, and t_a of 100° F, the difference between T_p for the multicylinder engine and T_p for the single-cylinder engine with baffle plates is 18° F. This difference is not very great but, owing to the form of the cooling equation, a small change in T_p makes a large change in Δp. It is therefore essential to know the difference between multicylinder and single-cylinder cooling in order that predictions of Δp for multicylinder engines from single-cylinder results will be in the range of cooling-pressure drop required by multicylinder engines.

As brought out in Apparatus, standard baffles, such as were used in reference 4, were not used in the single-cylinder tests. A special jacket was used. This special jacket gave slightly better cooling than for the multicylinder condition as indicated by the curves in figure 10. Later single-cylinder tests with standard baffles gave better cooling than either the multicylinder tests or the single-cylinder tests with the special jacket. From this result it follows that single-cylinder cooling with standard baffles will be better than multicylinder-engine cooling. Once a relation has been established for one set of conditions between multicylinder-engine cooling and single-cylinder-engine cooling, however, further conditions can be tested on the single-cylinder engine and engine cooling can be predicted from the results.

In multicylinder-engine cooling, sufficient air must be passed across the engine to cool the hottest cylinder even at the expense of overcooling the other cylinders. In order to determine the pressure drop required for cooling the engine, not only the relations shown in figure 10 but also the relation between the hottest rear spark-plug temperature and the average rear spark-plug temperature must be known. This relation for some conditions for the Pratt & Whitney R-2800-27 engine is shown in the curve of figure 9(b), which was obtained from reference 4. When the power of the engine was increased, this curve changed somewhat, but it is an approximation of what the hottest plug temperature will be for all conditions. In general, the hottest plug temperature was about 50° F higher than the average of the rear spark-plug temperatures.

The mean effective gas temperature T_g was obtained in reference 4 for a dry manifold temperature of 80° F for various fuel-air ratios. The method of obtaining T_g referred to a t_m of 80° F

has been given in the Analysis. The values are plotted in figure 11. The values of T_g at 80° F for the single-cylinder engine are also given in figure 11. The values for the single-cylinder engine were obtained using an equation of the form of equation (4) and the same factor to correct to 80° F as was used in the case of the multicylinder tests. The agreement of the results is excellent above a fuel-air ratio of 0.07 but below 0.07 the single-cylinder curve decreases more rapidly than the multicylinder-engine curve. It would be expected that the curves would agree above 0.07. If T_g was known for each cylinder of the multicylinder engine but only the fuel-air ratio of the entire engine was known, a series of curves of the shape of the single-cylinder curve of figure 11 would be obtained and would be displaced along the fuel-air ratio axis. It can be shown that an average of these displaced curves would be a curve different below a fuel-air ratio of 0.07 than the single-cylinder curve. The single-cylinder curve represents the true curve for each cylinder of the multicylinder engine if all factors of the cooling equations for these cylinders were known.

Engine Performance

Effect of engine speed. - The results of the tests with wide-open throttle for varying engine speed are given in figure 12. All values of mean effective pressure and horsepower are observed readings. The manifold pressure was less than atmospheric, owing to the drop through the air-measuring system. For a given manifold pressure and temperature, the brake and the indicated mean effective pressures increased and then decreased with an increase of speed within the range tested. The tests were made with a maximum-power mixture, and the results show that the indicated fuel consumption and the fuel-air ratio remained almost constant over the range of speed tested.

Effect of fuel-air ratio. - The variations of power, mean effective pressure, and specific fuel consumption with fuel-air ratio at wide-open throttle and at an engine speed of 2100 rpm are shown in figure 13. The power remains fairly constant for fuel-air ratios from 0.075 to 0.09 but decreases below a fuel-air ratio of 0.07. A fuel-air ratio of 0.07 will be taken as corresponding to the maximum-economy mixture even though the minimum specific fuel consumption is obtained with a fuel-air ratio of 0.065. The ratio used will be 0.07 instead of 0.065 because there is little difference in the specific fuel consumption with either ratio and the power is greater with the higher ratio. A fuel-air ratio of 0.08 will be used to denote the maximum-power mixture.

Effect of manifold pressure. - The variation of performance with manifold pressure is shown in figure 14. The indicated horsepower varies linearly with manifold pressure and, for the greater part of the range of manifold pressure, the indicated specific fuel consumption remains approximately constant. These data provided values near the rated power output of the R-2800-21 engine that could be used to extend the curves of figure 8.

Estimated Cooling-Pressure-Drop Requirements

Determination of equivalent rear spark-plug temperature used in cooling equation for single-cylinder engine. - Before the cooling-pressure-drop determinations for various cooling and engine conditions are given, the relation between the hottest rear spark-plug temperature of the multicylinder engine and the rear spark-plug temperature of the single-cylinder engine must be determined. The object of determining this relation is to find what rear spark-plug temperature must be assumed in the cooling equation for the single-cylinder engine in order that the cooling-pressure drop determined will be that needed to cool the multicylinder engine to some stated hottest-plug temperature.

From figure 10, the cooling equation of the single-cylinder engine with baffle plates is

$$\frac{T_p - t_a}{T_g - T_p} = 0.485 \left[\frac{{W_c}^{1.76}}{\Delta p \rho_i/\rho_o}\right]^{0.32} \tag{17}$$

and for the multicylinder engine, is

$$\frac{T_p - t_a}{T_g - T_p} = 0.542 \left[\frac{{W_c}^{1.76}}{\Delta p \rho_i/\rho_o}\right]^{0.32} \tag{18}$$

The relation between the maximum rear spark-plug temperature and the average rear spark-plug temperature for the multicylinder engine shown in figure 9(b) can be represented by the equation

$$T_p = \frac{T_{hp} - 48}{1.02} \tag{19}$$

where T_{hp} is the maximum rear spark-plug temperature, $^{\circ}F$. For the same values of W_c and $\Delta p\rho_i/\rho_o$ in equations (17) and (18), the relation between T_{hp} and T_p for the single-cylinder engine becomes

$$T_p = \frac{0.485\ T_g\left(T_{hp}-48-1.02t_a\right)+0.542t_a\left(1.02\ T_g-T_{hp}+48\right)}{0.485\left(T_{hp}-48-1.02t_a\right)+0.542\left(1.02\ T_g-T_{hp}+48\right)} \tag{20}$$

Calculations of T_p for various values of T_{hp} for several fuel-air ratios and cooling-air temperatures were made and the results are shown in figure 15. The calculations were made on the assumption that the dry manifold temperature was equal to the cooling-air temperature, and T_g was obtained from figure 11 for each fuel-air ratio. Equation (20) applies only for a range of fuel-air ratios from 0.07 to 0.11 where the single-cylinder- and multicylinder-engine T_g values agree. Below a fuel-air ratio of 0.07, a relation corresponding to equation (20) would have to be derived involving T_g for both the single-cylinder and the multicylinder engines.

Figure 15 shows that, for the range of fuel-air ratios and cooling-air temperatures chosen, there is little difference in T_p for a given T_{hp}. The air temperatures chosen are the Navy, Army, and Civil Aeronautics Authority standard sea-level temperatures. The main change in the values of T_p of figure 15 will occur with a change of inlet manifold temperature, as when a supercharger is used. There is approximately 75° F difference between the maximum rear spark-plug temperature of the multicylinder engine and the rear spark-plug temperature of the single-cylinder engine in figure 15 for the same cooling and engine conditions.

<u>Cooling-pressure-drop requirements</u>. - Cooling-pressure-drop requirements have been calculated from equation (17) for the following conditions of the Pratt & Whitney R-2800-27 engine:

Condition	Atmospheric pressure, p_a (in.Hg abs.)	Atmospheric temperature, t_a (°F)	Blower-rim temperature, t_m (°F)	Maximum rear spark-plug temperature, T_{hp} (°F)	F/A	Power	Altitude
1	29.92	100	[1]L.B.	500	0.10	Take-off	Sea level
2	29.92	100	L.B.	500	.08	---do---	Do.
3	29.92	100	L.B.	500	.11	---do---	Do.
4	29.92	100	L.B.	550	.10	---do---	Do.
5	29.92	100	100	500	.10	---do---	Do.
6	29.92	100	L.B.	450	.08	Normal rated	Do.

[1]L.B. refers to temperature at blower rim using low-blower gear ratio.

The engine would probably fail for condition 2 with present-day fuels because of knock, but this condition was assumed to illustrate the effect of fuel-air ratio on cooling-pressure drop.

The take-off power of the Pratt & Whitney R-2800-27 engine is 2000 brake horsepower at 2700 rpm and the normal rated power is 1600 brake horsepower at 2400 rpm. The friction horsepower of this engine, taken from unpublished data, in low blower at 2700 rpm is 398 and at 2400 rpm is 294. Then

Take-off power = 2398 indicated horsepower
Normal rated power = 1894 indicated horsepower

The blower-rim temperatures were calculated from equation (6) on the assumption that t_c equals t_a. The blower-rim temperatures are thus represented by the dry manifold temperatures. The tip speeds of the impeller were determined from the engine speed, from the impeller diameter, which is 11 inches for this engine, and from the impeller-gear ratio, which is 7.60:1 in low blower.

The values of T_g were found from the fuel-air ratios given in the table and the values of t_m were calculated from figure 7 and equation (7). The values of the rear spark-plug temperature to use in the single-cylinder cooling equation (17) in order to get equivalent multicylinder cooling-pressure drops were calculated by means of equation (20) with the values of T_{hp} and t_a given in the table of conditions and the calculated values of T_g.

The weights of charge air W_c needed to develop take-off and normal rated powers were obtained from the curve of figure 16, which was determined from data of this report. This curve checks closely with a similar curve obtained from unpublished test data on the multicylinder-engine air consumption. For the first calculations of Δp, the cooling-air temperature in both equations (17) and (20) was assumed to be atmospheric temperature and ρ_i was calculated from the atmospheric pressure and temperature. The results of the calculations are given in the following table:

Condition	T_p (°F)	t_a (°F)	t_m (°F)	T_g (°F)	W_c (lb/sec)	ρ_i $[(lb)(sec)^2/ft^4]$	Δp (in. water)
1	418	100	262	1104	3.85	0.0708	13.34
2	417	100	262	1296	3.85	.0708	29.23
3	420	100	262	1004	3.85	.0708	7.91
4	466	100	262	1104	3.85	.0708	6.85
5	420	100	100	974	3.85	.0708	6.70
6	370	100	228	1268	3.05	.0708	34.26

The pressure drop required for cooling for condition 2 was checked by using the multicylinder-engine cooling equation (18) and the relation between T_p and T_{hp} for the multicylinder engine given by equation (19). The value of T_g was found from the fuel-air ratio given and the previously calculated values of t_m by use of the multicylinder-engine curve on figure 11 and equation (7). The calculated Δp was 29.42 inches of water, which is a good check of the 29.23 inches of water from the single-cylinder-engine cooling equation. The foregoing results show that multicylinder cooling-pressure drops can be predicted with little error from single-cylinder-engine cooling equations if the relation between T_p of the single-cylinder engine and T_{hp} of the multicylinder engine is known or can be estimated with fair accuracy.

Effect of adiabatic compression of the air at the engine face on cooling-pressure drops. - Calculations of cooling-pressure drops for conditions 1 to 6 given in the preceding table have been made by using the pressure and temperature at the engine face, an assumed airspeed of 200 miles per hour for conditions 1 to 5, and an assumed airspeed of 350 miles per hour for condition 6. The pressure and the temperature at the engine face were calculated from equations (8) and (9) and the pressure drop, from an equation of the form of equation (10). The exponent m_1 of equation (18) and the relation between T_p and T_{hp} given in equation (19) were used in equation (10). The values of Δp used in equation (10) were the values given in the preceding table. The values of Δp_c and Δp are given in the following table for comparison of the adiabatic compression effect:

Condition	Δp	Δp_c
1	13.34	13.79
2	29.23	30.23
3	7.91	8.18
4	6.85	7.02
5	6.70	6.93
6	34.26	39.42

The results show that, for 200 miles per hour, using the atmospheric pressure and temperature in the cooling equation instead of the pressure and temperature at the engine face resulted in only a small difference in the estimated cooling-pressure drop. For condition 6 at 350 miles per hour, however, the difference in the cooling-pressure drops was appreciable: 34.26 inches of water by using p_o and t_o and 39.42 inches of water by using p_i and t_i. Calculations for 40,000 feet show that the adiabatic compression effect on Δp is much smaller than the effect at sea level.

Comparison of Δp_c for conditions 1, 2, and 3 shows the large effect on cooling-pressure drop of "leaning out" the mixture. For a fuel-air ratio of 0.11, a pressure drop of 8.18 inches of water was required to cool take-off power at sea level; for a fuel-air ratio of 0.10 and the other conditions the same, 13.79 inches of water; and for a fuel-air ratio of 0.08, 30.23 inches of water. The effect of increasing the allowable maximum rear spark-plug temperature is shown by comparing Δp_c for conditions 1 and 4. Improving engines to such a point that they could be safely operated at rear spark-plug temperatures of 550° F would, for the conditions assumed, decrease the cooling-pressure drop required from 13.79 inches of water to 7.02 inches of water or would approximately halve the original pressure drop. The effect of decreasing the intake manifold temperature by using intercoolers, and therefore decreasing T_g, is illustrated by conditions 1 and 5. The cooling-pressure drop required is again approximately halved - from 13.79 to 6.93 inches of water.

The large pressure drops required for cooling at even normal rated power at sea level with economical fuel-air ratios are illustrated by the 39.42 inches of water required for condition 6. The pressure drops calculated are only those required across the engine and do not include cowling-duct losses. It can be shown that, for some conditions of flight, the pressure drop across a duct in a moving stream can be as much as twice as great as the pressure drop across the heater in the duct. The usual procedure at present for cooling engines under certain conditions in which cooling-pressure drops required are not available is to enrich the mixture. The

effect of this enrichment on Δp_c is shown in the calculations for conditions 1, 2, and 3. It is thought that, at high altitudes (approximately 40,000 ft), even the additional cooling provided by enriching the mixture will be insufficient to cool engines whose power outputs are greater than, but whose fins are the same as, those of present-day engines. The greater power outputs will be possible because of improved superchargers. Some research on the causes of the poor temperature distribution from cylinder to cylinder of present-day engines is needed. From figure 9(b) it was shown that the average temperature of the rear spark plugs of the Pratt & Whitney R-2800-27 engine was about 50° F lower than the maximum rear spark-plug temperature. In order to limit the hottest cylinder to 450° F on the rear spark plug, enough air must be furnished in order that the engine as a whole is operating at only about 400° F on the rear plugs. The improvement of the cooling of the one or two hot cylinders could possibly result in decreases of cooling-pressure drop required of the order of decrease of Δp_c for conditions 1 and 4.

CONCLUSIONS

For the range of conditions and for the engines tested in this investigation the following conclusions can be drawn:

1. An equation to determine the cooling characteristics of air-cooled engines was proved valid for a cylinder from a Pratt & Whitney R-2800-21 engine mounted on a single-cylinder engine for a large range of engine and cooling conditions.

2. The pressure drop required for cooling a Pratt & Whitney R-2800-27 engine was determined with little error by means of the cooling equation of the single-cylinder engine with a cylinder from a Pratt & Whitney R-2800-21 engine mounted on it. These predictions were made only after establishing the relationship between the hottest rear spark-plug temperature of the multicylinder engine and the rear spark-plug temperature of the single-cylinder engine.

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Cleveland, Ohio.

APPENDIX

SYMBOLS

a_o	outside-wall area of head (or barrel) of cylinder, sq in., (does not include rocker-box surface)
a_1	internal-surface area of head (or barrel) of cylinder, sq in.
$\overline{B}$	constant (See equation (1).)
C	constant (See equation (4).)
C_2	constant (See equation (5).)
c_p	specific heat of air at constant pressure, Btu/(lb)(°F), 0.24
F/A	fuel-air ratio
g	acceleration of gravity, ft/(sec)^2, 32.2
H	heat transferred from cylinder head (or barrel) to cooling air, Btu/hr
I	indicated horsepower per cylinder
J	mechanical equivalent of heat, ft-lb/Btu, 778
K	constant (See equation (1).)
$\overline{K}$	constant (See equation (3).)
K_h	constant (See equation (2).)
m	constant (exponent in equation (1))
m'	constant (exponent in equation (3))
m_1	constant (exponent in equation (4))
m_1'	constant (exponent in equation (5))
n	constant (exponent in equations (4) and (5))
n'	constant (exponent in equations (1) and (3))
p_i	pressure of cooling air at engine or cylinder face, in. Hg abs.
p_o	static pressure of air in free air stream, in. Hg abs.

q_o dynamic pressure of air in free air stream, in. Hg

t_a temperature of cooling air at face of cylinder or engine, $^{\circ}F$

T_b average temperature over outside cylinder-barrel surface when equilibrium is attained, $^{\circ}F$

t_c carburetor inlet-air temperature, $^{\circ}F$

T_g mean effective gas temperature, $^{\circ}F$

T_h average temperature over outside cylinder-head surface when equilibrium is attained, $^{\circ}F$

T_{hp} maximum rear spark-plug temperature, $^{\circ}F$

T_i temperature of cooling air corrected for adiabatic compression of air from free stream to engine face, $^{\circ}F$ abs.

$t_i = T_i - 460$, $^{\circ}F$

t_m dry effective inlet manifold temperature, $^{\circ}F$

T_o temperature of air in free stream, $^{\circ}F$ abs.

$t_o = T_o - 460$, $^{\circ}F$

T_p average of temperatures of rear spark plugs on multicylinder engine or rear spark-plug temperature of single-cylinder engine, $^{\circ}F$

U over-all heat-transfer coefficient from outside cylinder wall of engine to cooling air, based on difference between average cylinder temperature and inlet-cooling-air temperature, Btu/(hr)(sq in.)($^{\circ}F$)

U_t tip speed of primary blower, fps

v displacement volume of cylinder, cu in., 155.9

V_o speed of airplane, mph

W weight of cooling air passing over head (or barrel) of cylinder, lb/sec

W_c weight flow of charge air to engine, lb/sec, (for the single-cylinder engine of the present tests W_c = weight of charge to the cylinder × 18 to be comparable with the flow to the P. & W. R-2800-27 engine)

W_t weight of cooling air flowing across multicylinder engine, lb/sec, (for the single-cylinder engine W_t = weight of cooling air flowing across head and barrel of cylinder $\times$ 18 to be comparable with P. & W. R-2800-27 engine cooling-air flow)

z constant (exponent in equation (2))

ρ average density of cooling air, $(lb)(sec)^2/ft^4$ (based on temperatures and pressures of cooling air before and behind cylinder)

ρ_i density of cooling air at face of cylinder or engine, $(lb)(sec)^2/ft^4$

ρ_o density of air in free air stream, $(lb)(sec)^2/ft^4$

ρ_{60} density of air at 29.92 in. Hg and 60° F, $(lb)(sec)^2/ft^4$

Δp cooling-pressure drop across cylinder or engine, in. water, (includes loss caused by expansion of air from exit of baffle)

Δp_c cooling-pressure drop across engine corrected for adiabatic compression of air at face of engine, in. water

ΔT_g increment of mean effective gas temperature, °F

Δt_m increment of dry effective inlet manifold temperature, °F

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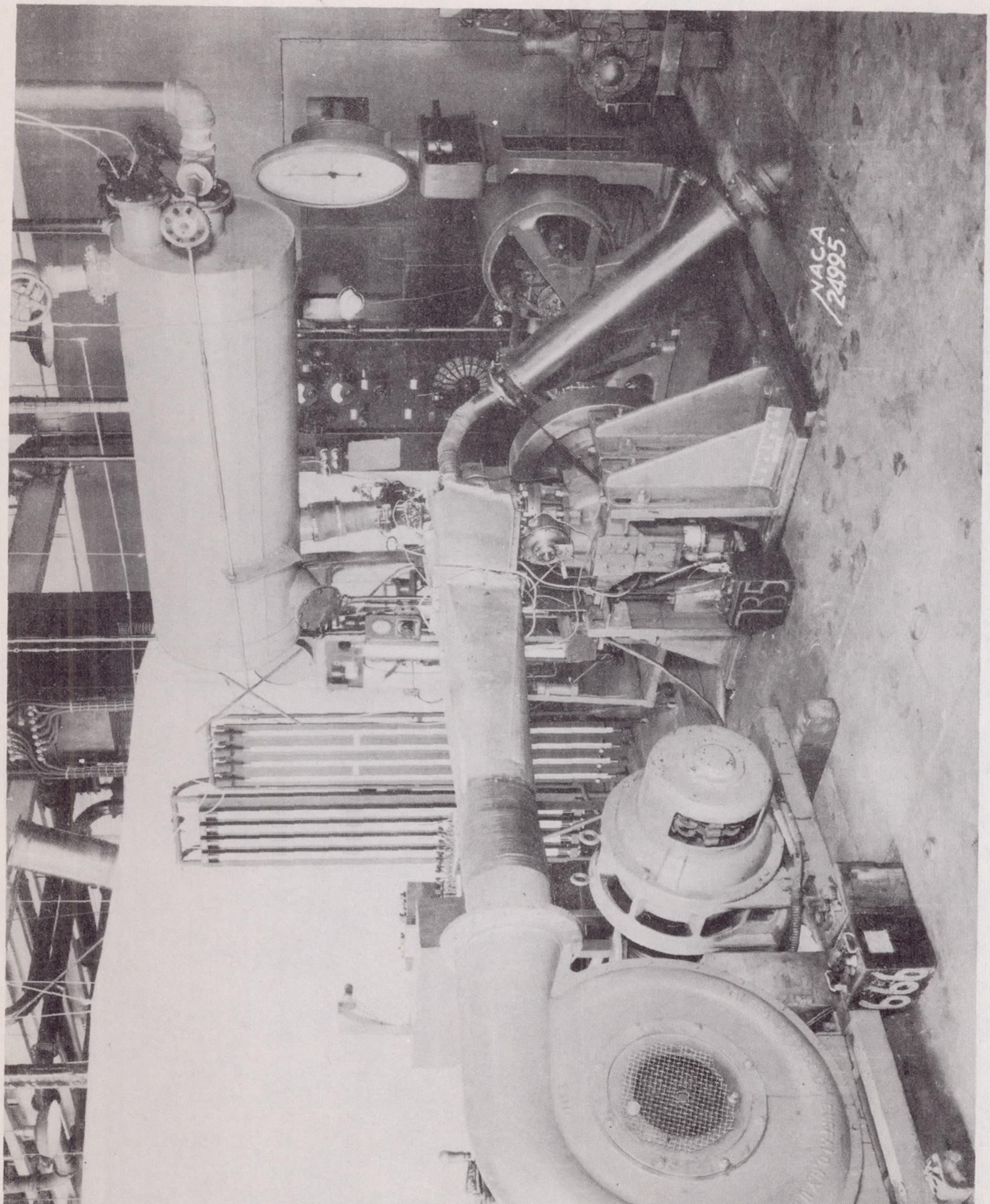

Figure 1.- Setup of single-cylinder air-cooled engine.

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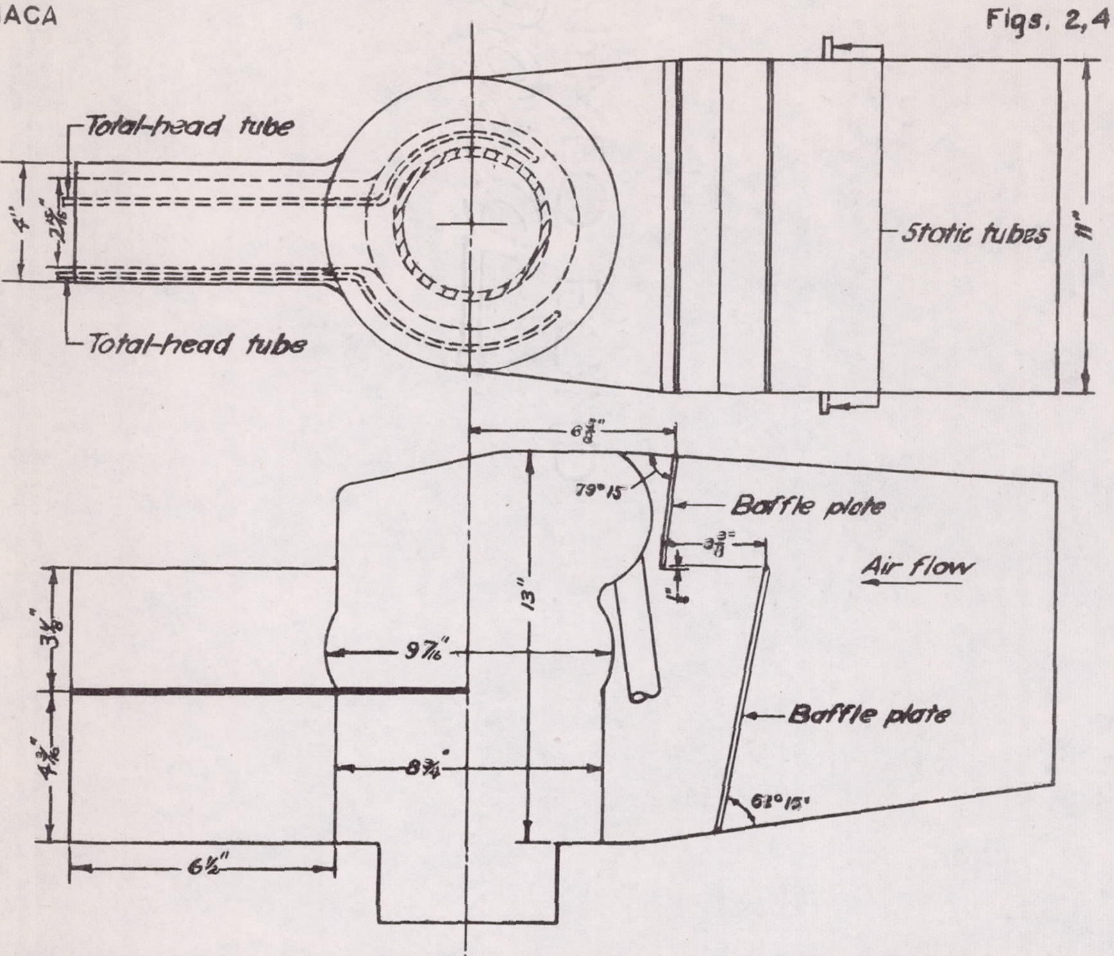

Figure 2. - Diagram of engine cylinder and jacket showing baffles.

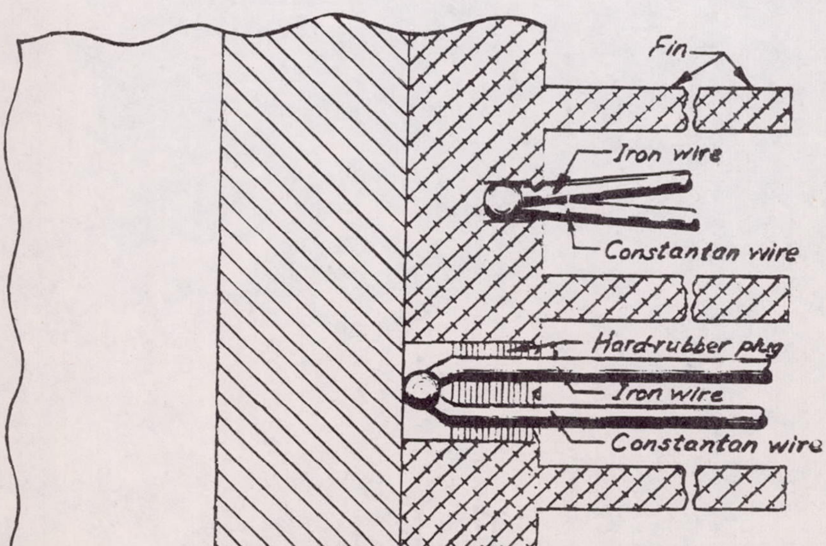

Figure 4 - Method of attaching thermocouples to steel base and aluminum muff.

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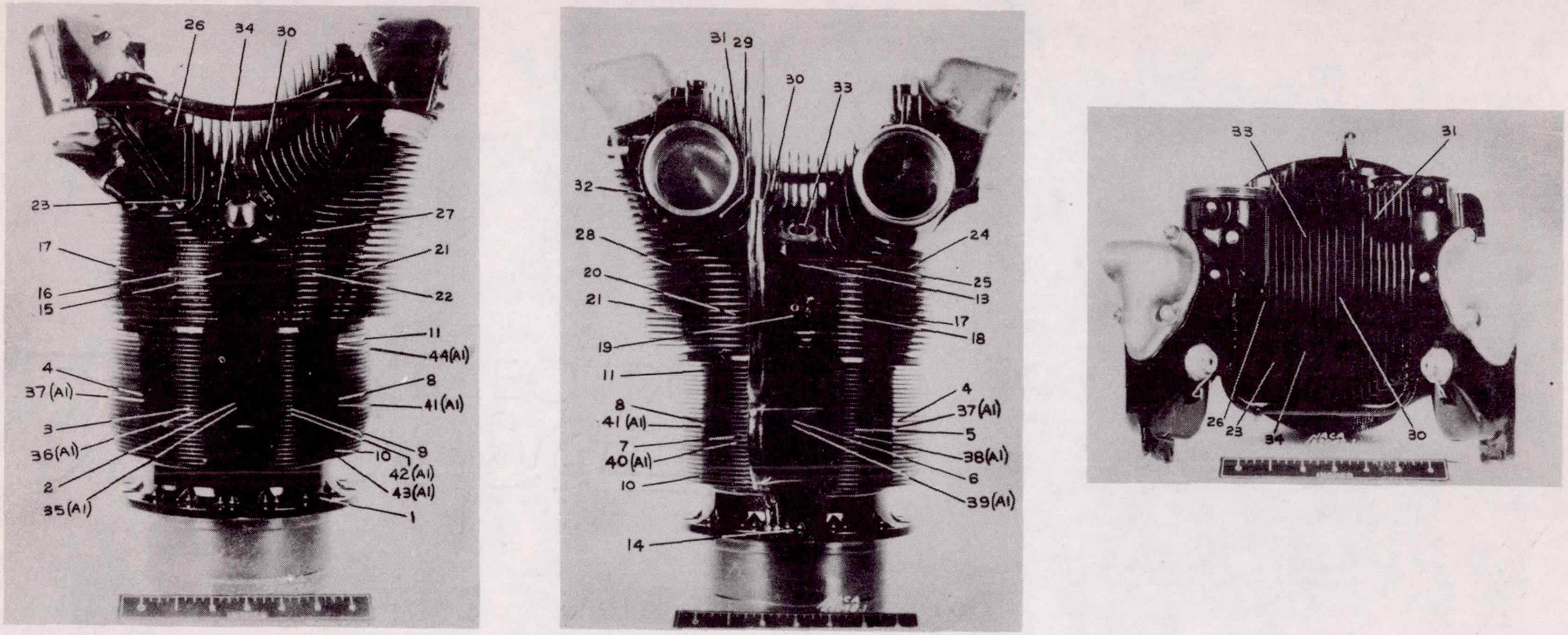

Figure 3.- Three views of Pratt & Whitney R-2800-21 cylinder showing location of thermocouples.

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Fig. 5

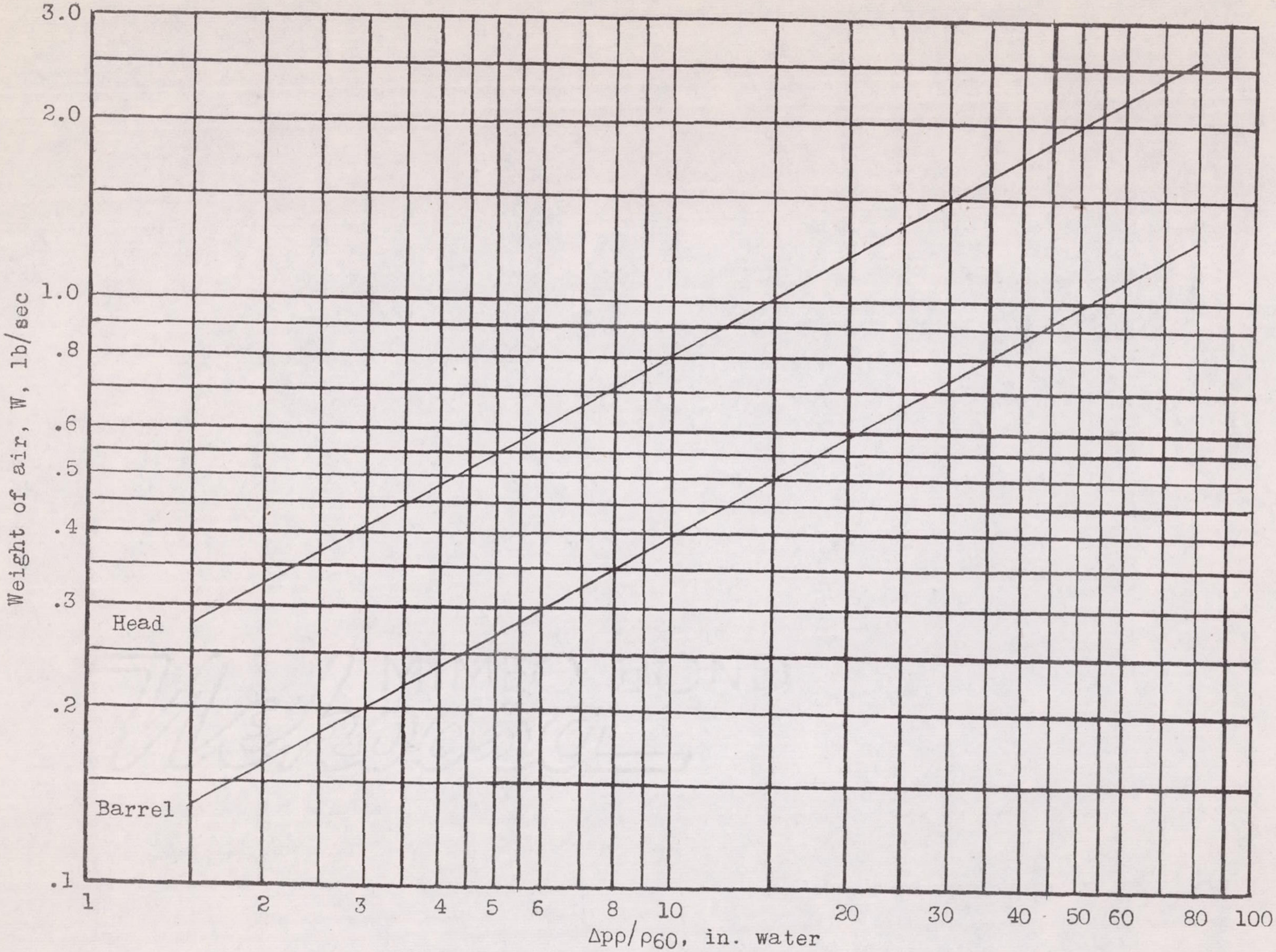

Figure 5.- Calibration of the jacket as an air duct.

	Series	7 ○	8-a △	8-b ◇	14 □	21 +
Engine speed, rpm		2109	2108	2119	2097	2106
Brake mean effective pressure, lb/sq in		232.8	137.9	220.5	135.9	137.2
Fuel-air ratio		0.081	0.082	0.082	0.083	0.083
Spark setting, deg B.T.		20	20	20	20	20
Carburetor-air temperature, °F		76	80	78	84	80
Cooling-air temperature, °F		87	94	75	89	88

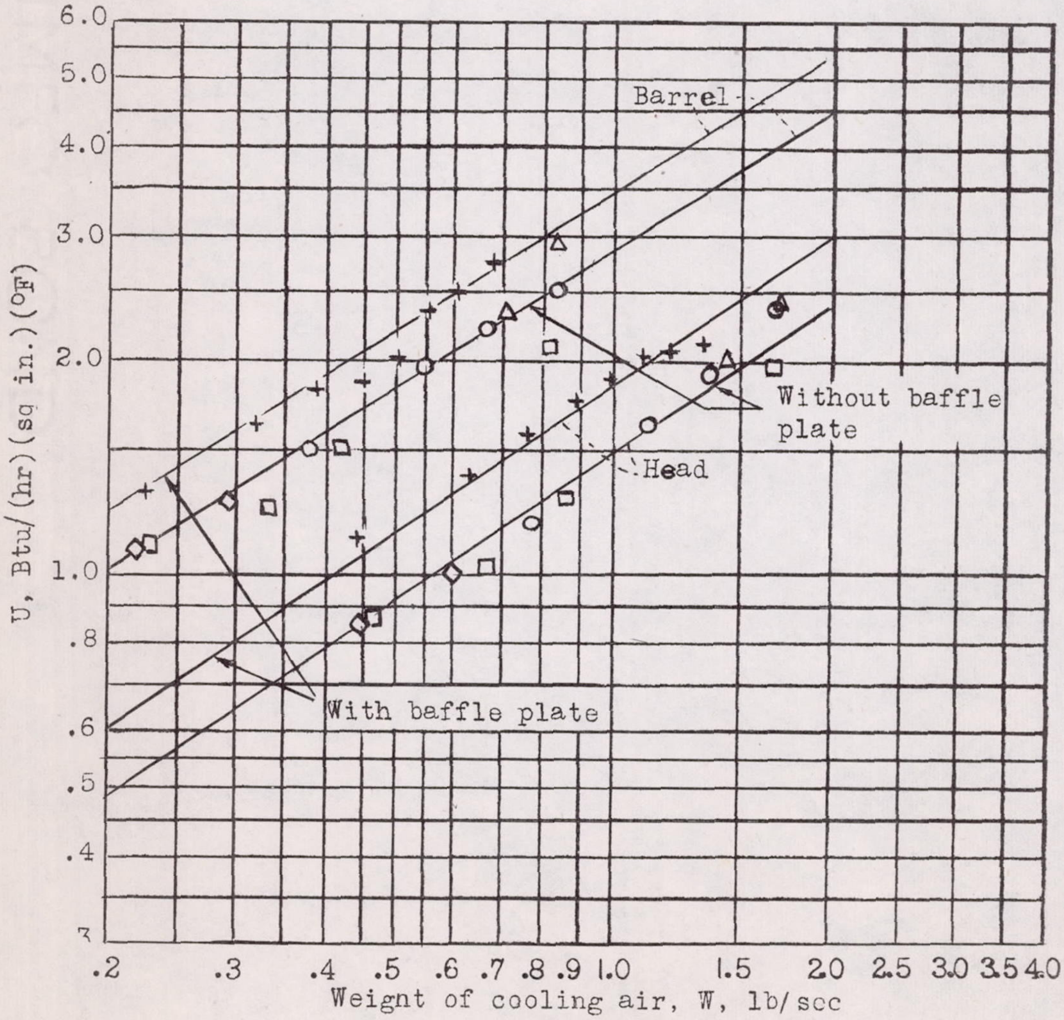

Figure 6.- Variation of outside-wall heat-transfer coefficient with weight of cooling air across cylinder.

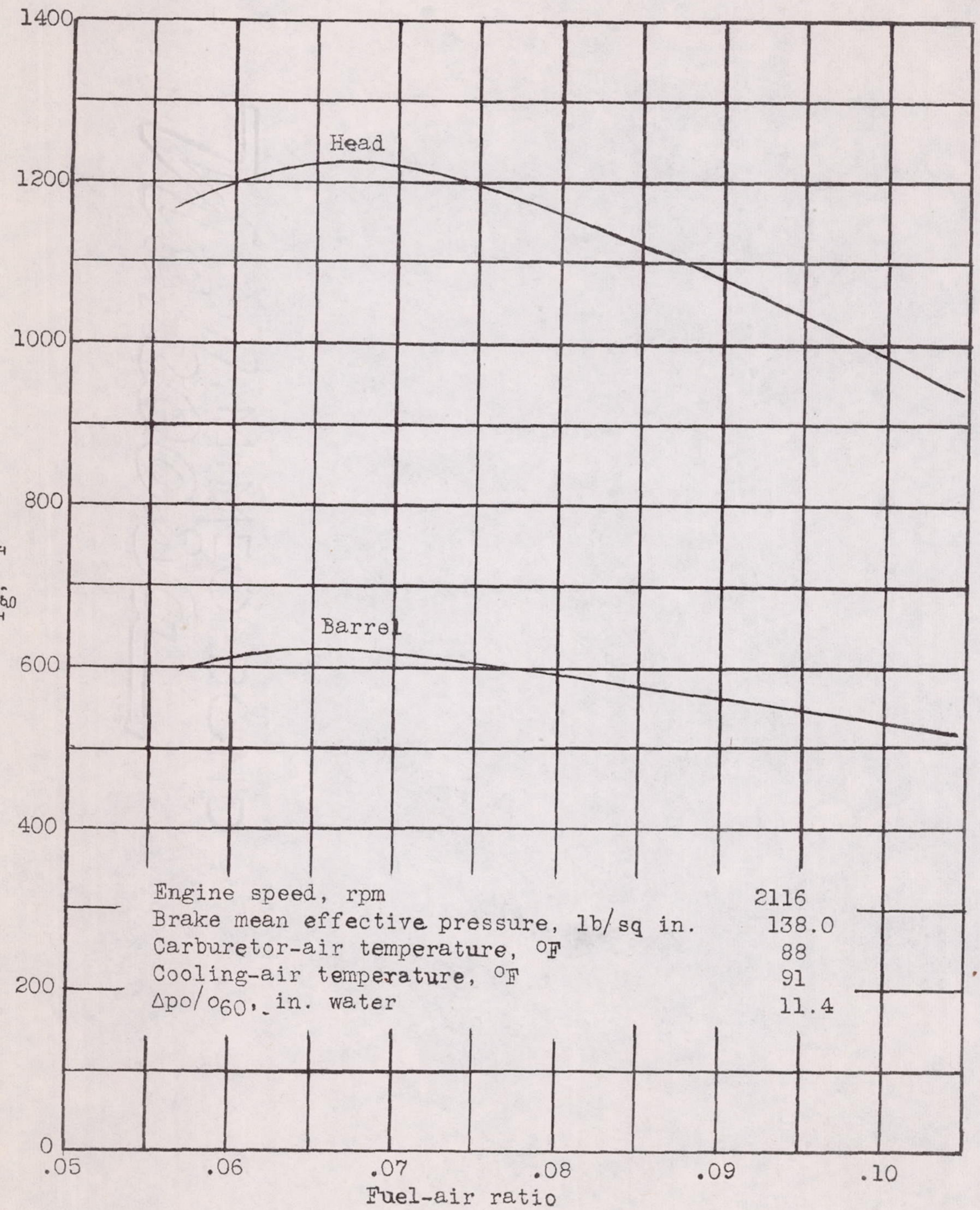

Figure 7.- Variation of the effective gas temperature with fuel-air ratio.

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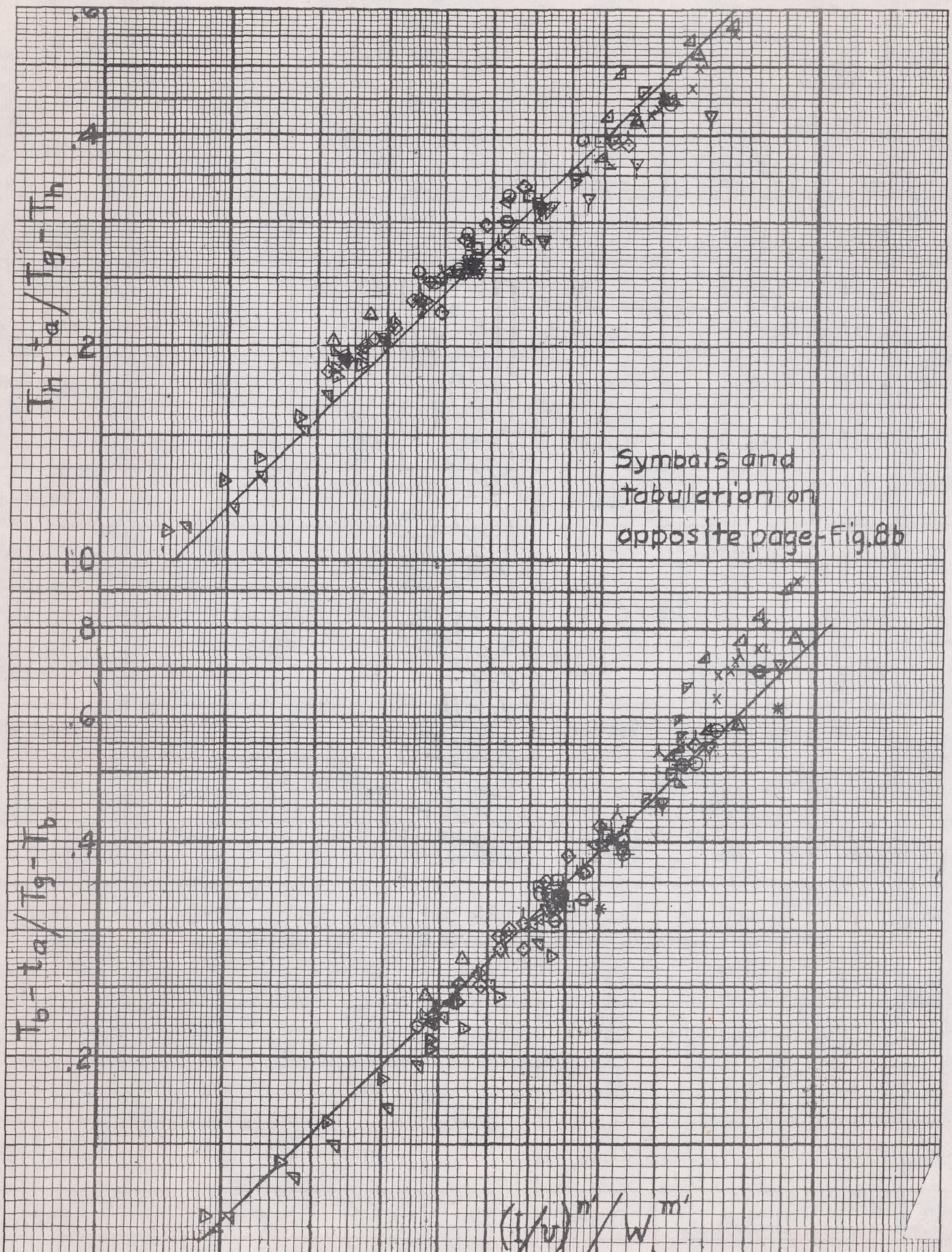

Figure 8.- Variation of $(T_h-t_a)/(T_g-T_h)$ and $(T_b-t_a)/(T_g-T_b)$ with $(I/v)^{n'}/W^{m'}$ for single-cylinder engine.

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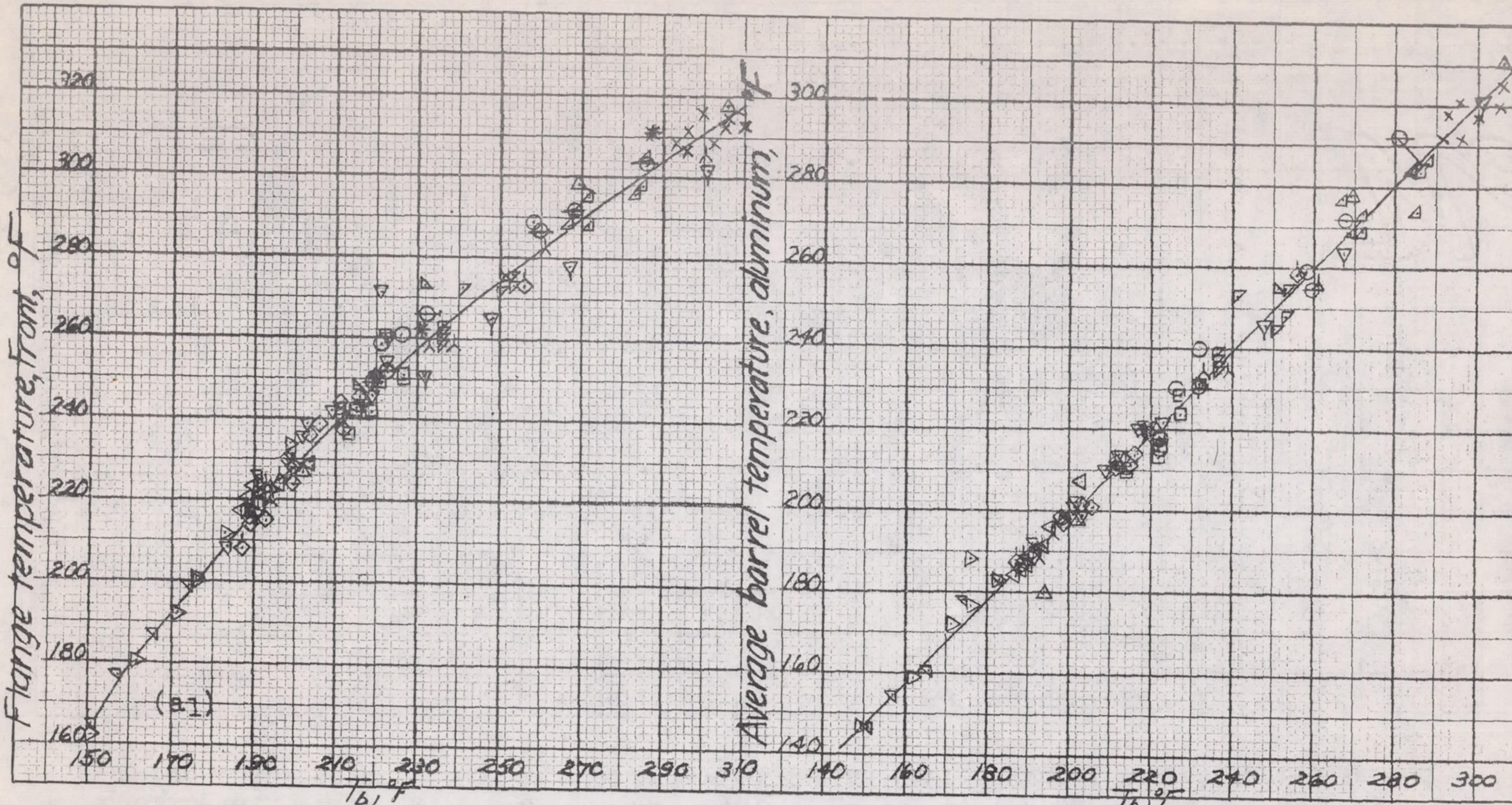

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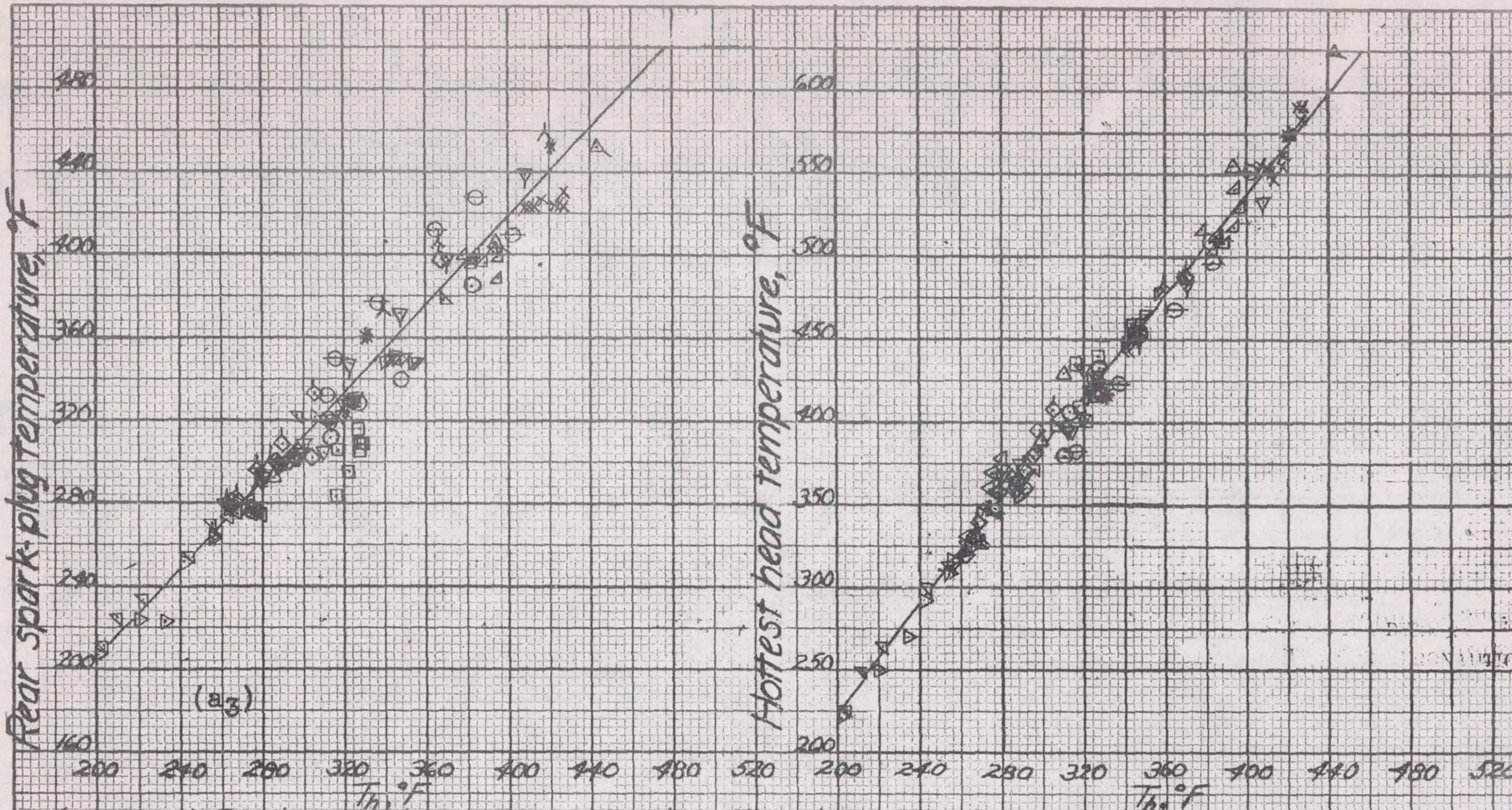
	○	△	△	□	▽	◇	▽	◁	△
Engine speed, rpm	2109	2108	2119	2110	2116	2123	2112	Varied	2097
Bmep, lb/sq in.	232.8	137.9	221.0	Varied	138.0	Varied	Varied	Varied	135.9
Fuel-air ratio	0.081	0.082	0.082	Varied	Varied	Varied	Varied	0.085	0.083
Spark setting, °B.T.C.	20	20	20	20	20	20	20	20	20
Carb.-air temp. °F	76	80	78	84	88	89	82	88	84
Cooling-air temp. °F	87	94	75	85	91	97	85	100	89
$\Delta p \rho / \rho_{60}$, in. water	Varied	Varied	Varied	Varied	11.1	Varied	Varied	36.9	Varied

Fig. 9a1

#53



(a_1, a_2, a_3) Single-cylinder tests without baffle plates ahead of cylinder.

	○	△	ч	□	▽	◇	◸	◁	◺
Engine speed, rpm	2109	2108	2119	2110	2116	2123	2112	Varied	2097
Bmep, lb/sq in.	232.8	137.9	221.0	Varied	138.0	Varied	Varied	Varied	135.9
Fuel-air ratio	0.081	0.082	0.082	Varied	Varied	Varied	Varied	0.085	0.083
Spark setting, °B.T.C.	20	20	20	20	20	20	20	20	20
Carb.-air temp. °F	76	80	78	84	88	89	82	88	84
Cooling-air temp. °F	87	94	75	85	91	97	85	100	89
$\Delta p \rho / \rho_{60}$, in. water	Varied	Varied	Varied	Varied	11.1	Varied	Varied	36.9	Varied

NACA

Fig. 9a3

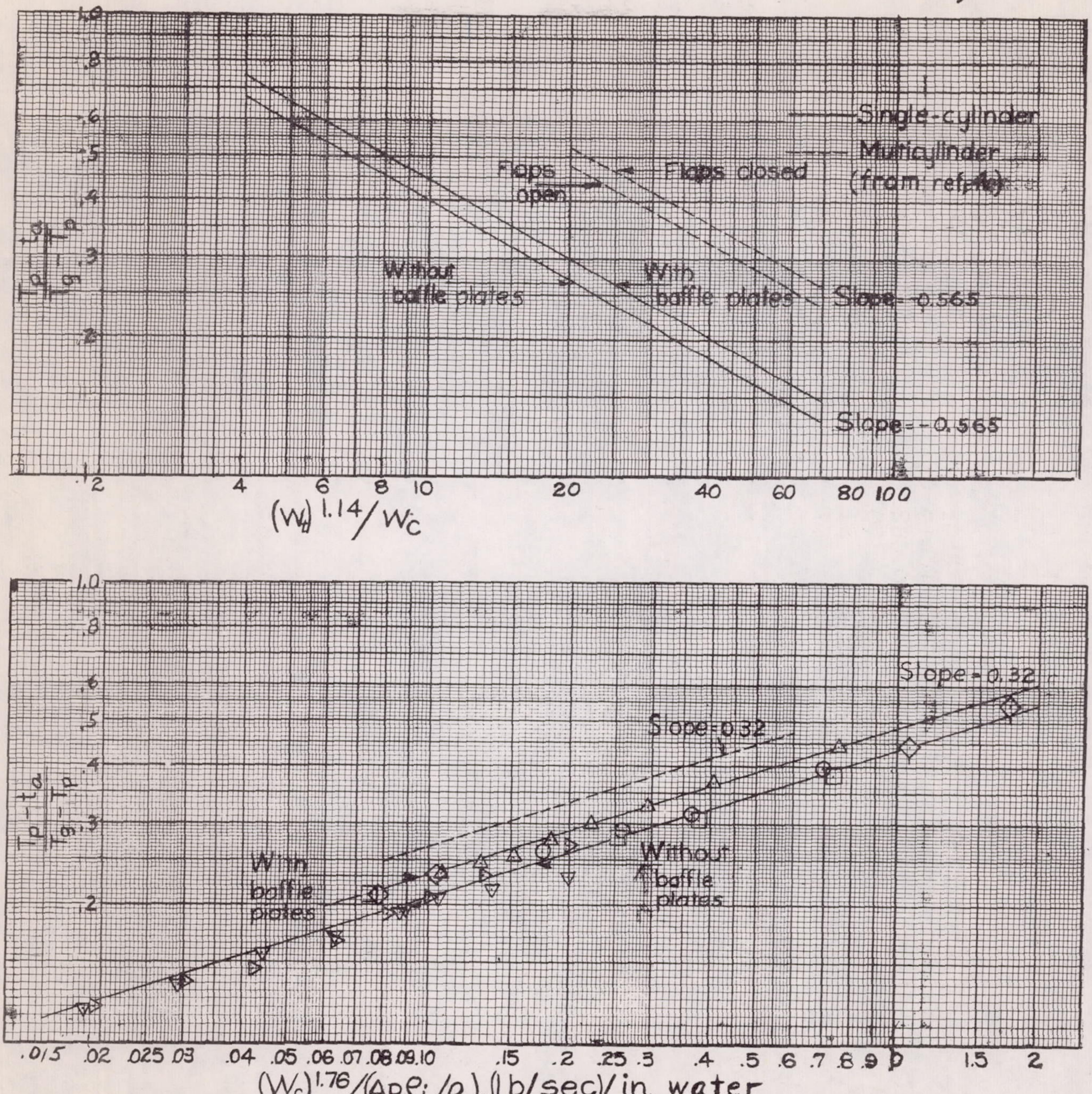


Fig. 10.- Variation of $(T_p - t_a)/(T_g - T_p)$ with $(W_c)^{1.76}/(\Delta p \rho_i/\rho_o)$ and $(W_t)^{1.14}/W_c$

	○	◇	▷	□	▽	△	◇
Engine speed, rpm	2109	2108	Varied	2097	Varied	2106	2119
Brake mean effective pressure, lb/sq in.	232.8	137.9	Varied	135.9	Varied	137.2	221.0
Fuel air ratio	0.081	0.082	0.085	0.083	0.081	0.083	0.082
Spark setting, deg B.T.C.	20	20	20	20	20	20	20
Carburetor-air temperature, °F	76	80	88	84	82	80	78
Cooling-air temperature, °F	87	94	100	89	95	88	75
$\Delta p \rho/\rho_{60}$ in. water	Varied	Varied	36.9	Varied	36.7	Varied	Varied

#53

#53

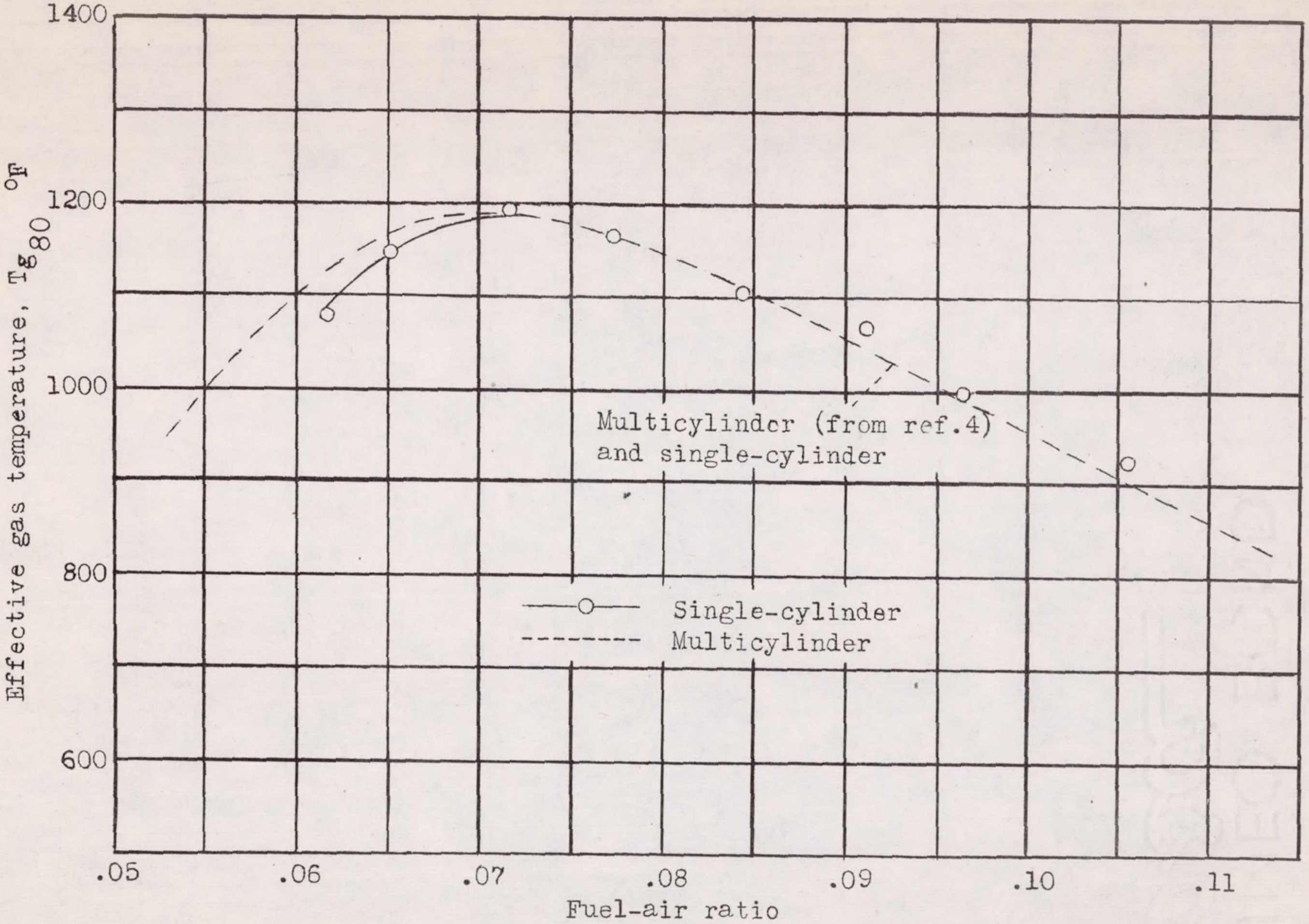


Figure 11.- Comparison of single-cylinder and multicylinder effective gas temperatures.

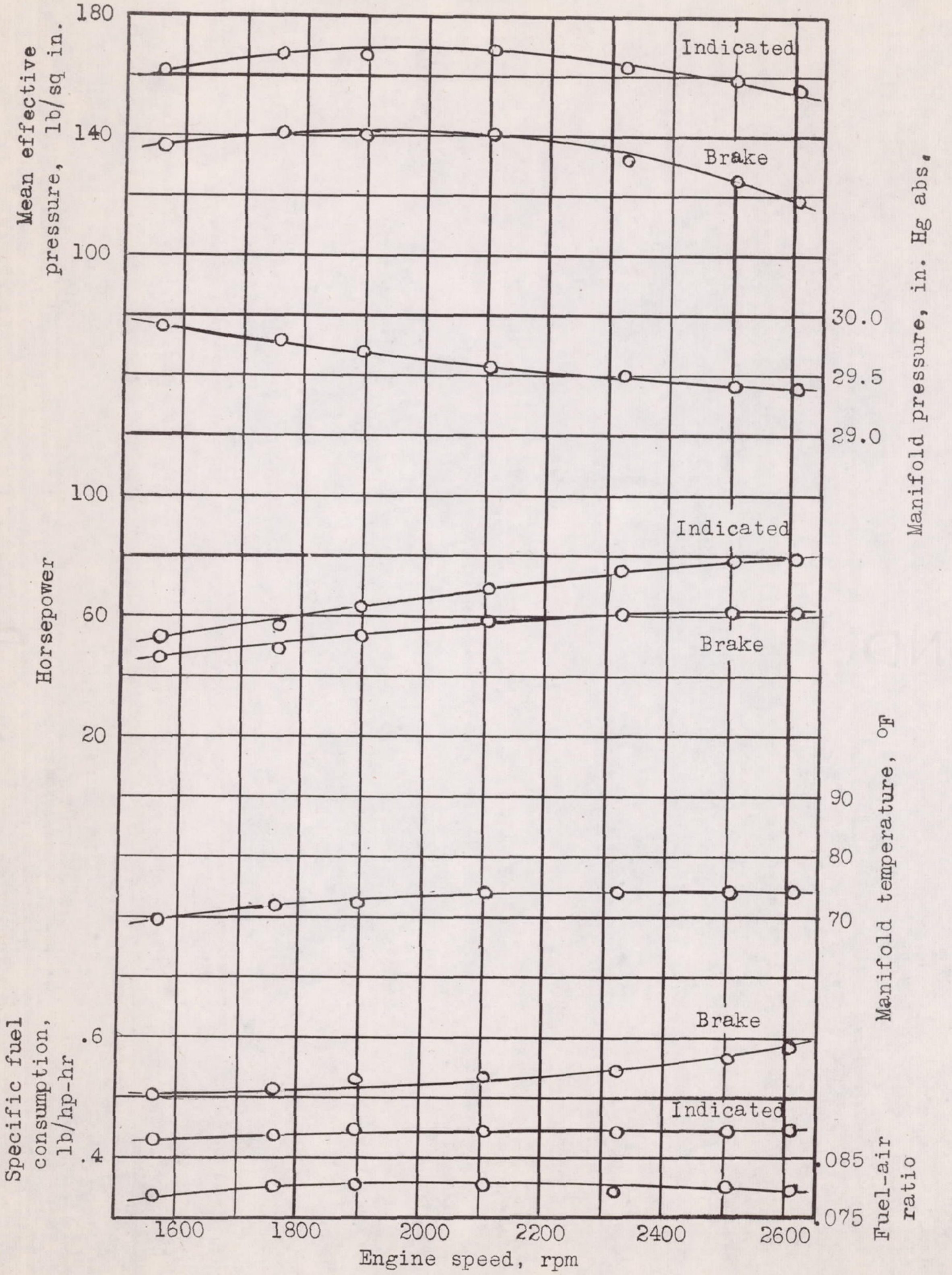


Figure 12.- Variation of performance with engine speed.

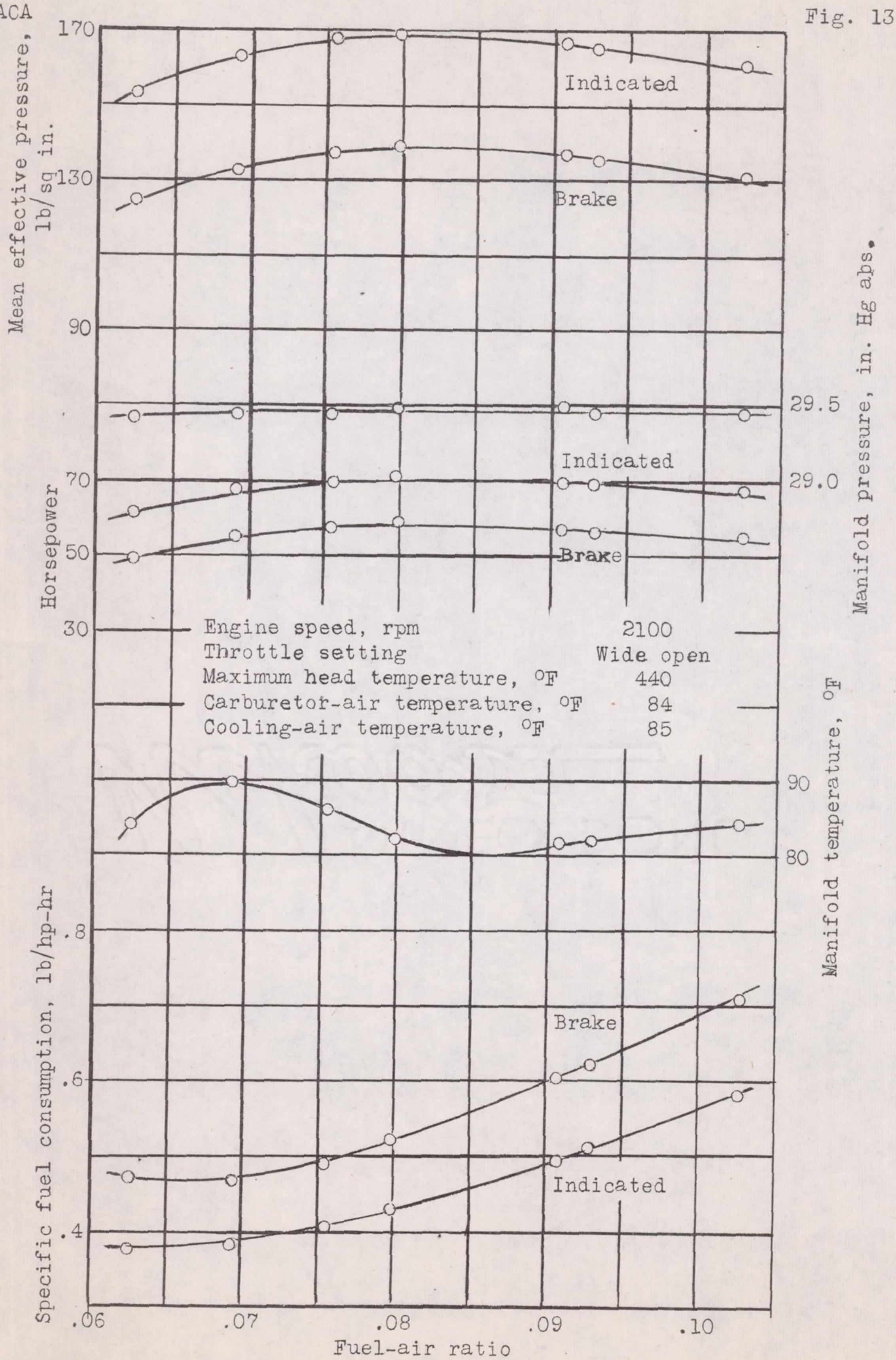


Figure 13.- Variation of performance with fuel-air ratio.

#53

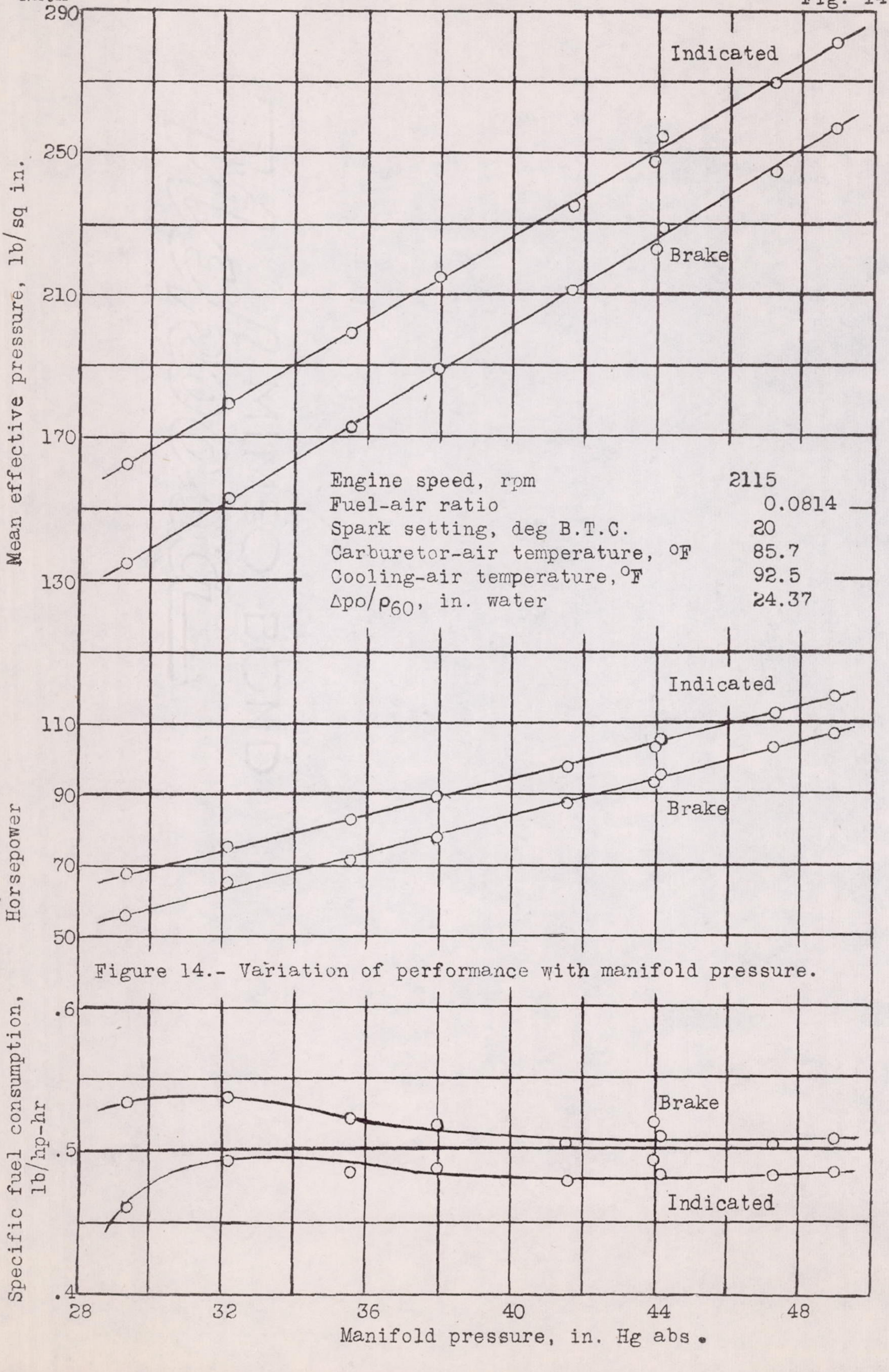


Figure 14.- Variation of performance with manifold pressure.

#53

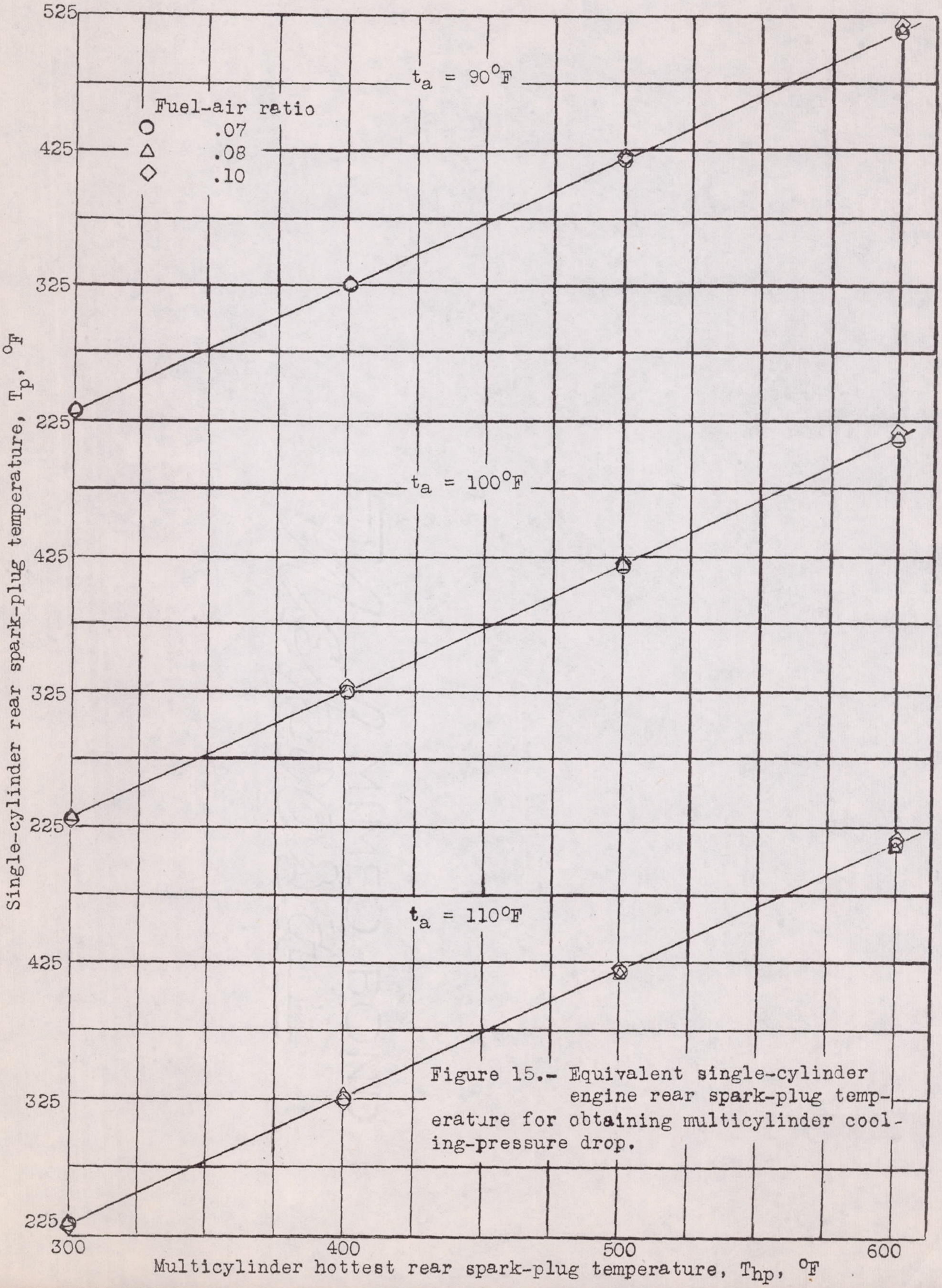


Figure 15.- Equivalent single-cylinder engine rear spark-plug temperature for obtaining multicylinder cooling-pressure drop.

#53

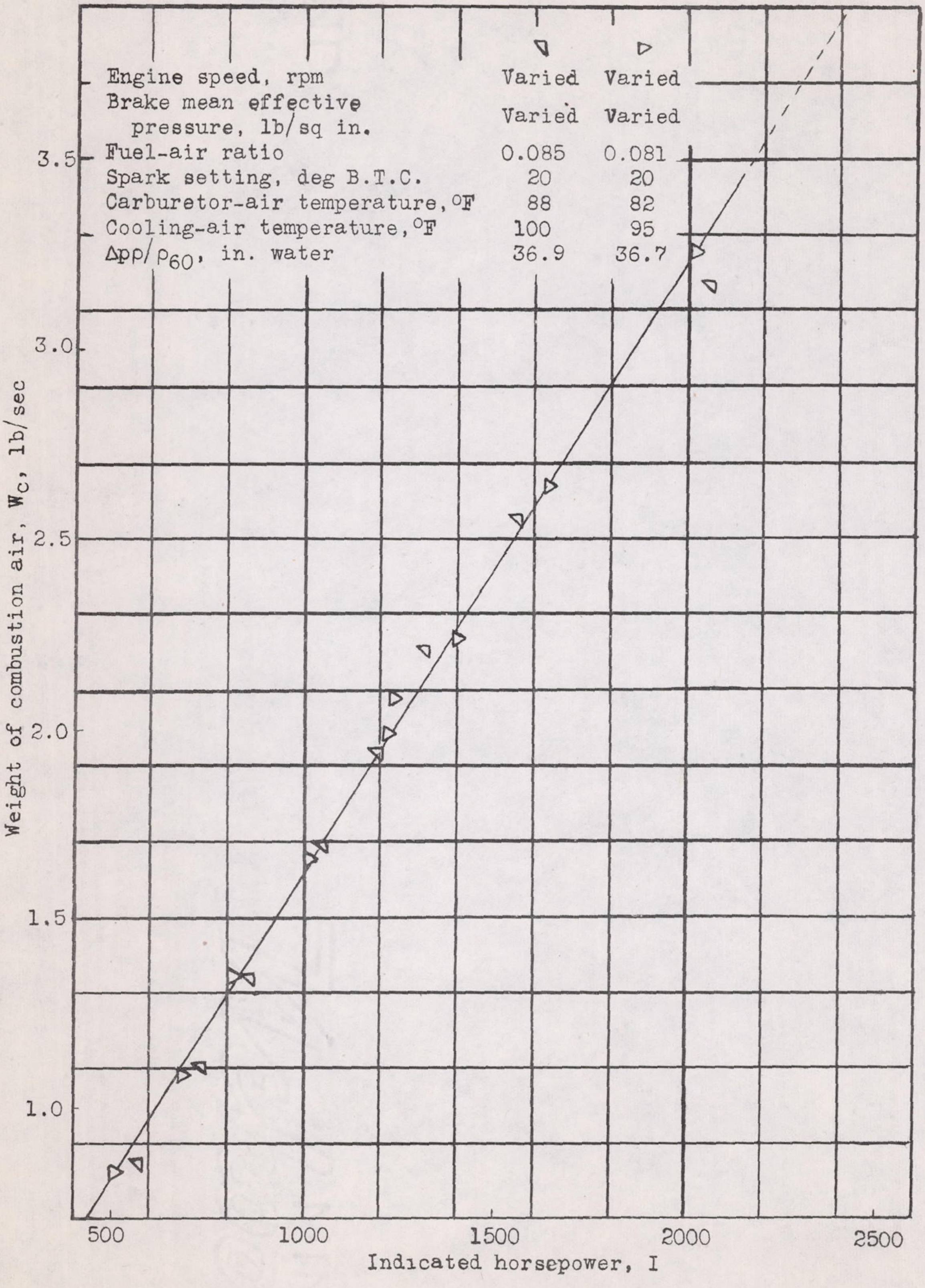


Figure 16.- Relationship between charge-air weight and indicated horsepower.

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