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A COOLING-CORRELATION EQUATION FOR A DOUBLE-ROW RADIAL ENGINE BASED ON THE TEMPERATURE OF THE EXHAUST-VALVE SEAT

By James M. Jagger and Fred O. Black, Jr.

Aircraft Engine Research Laboratory
Cleveland, Ohio

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MEMORANDUM REPORT

for the

Army Air Forces, Air Technical Service Command

A COOLING-CORRELATION EQUATION FOR A DOUBLE-ROW RADIAL ENGINE BASED ON THE TEMPERATURE OF THE EXHAUST-VALVE SEAT

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SUMMARY

A cooling-correlation equation was obtained based on the average temperature of the exhaust-valve seats of a large double-row radial engine. The exhaust-valve seat was chosen as the basis for the correlation because endurance tests have shown it to be the critical region of this engine cylinder. From this correlation equation predictions may be made of the cooling-air pressure drop required to cool the critical region of the cylinders.

The accuracy of predicting engine cooling-air pressure-drop requirements, based on the cooling correlation, was substantially increased when a modified carburetor was used.

INTRODUCTION

At the request of the Army Air Forces, Air Technical Service Command, tests have been conducted at the NACA Cleveland Laboratory to improve the cooling of a large double-row radial aircraft engine. In connection with the program, a study has been made of the cooling characteristics of the engine by the NACA cooling correlation method. (See reference 1.) Cooling correlations have previously been based on the temperatures of the rear spark-plug boss and the rear spark-plug gasket. (See reference 2.)

The exhaust-valve seat has been shown to be a critical region of the rear-row cylinders. The initial failure in endurance tests made at excessive cylinder temperatures was that of the exhaust-valve seat. Operation at this condition resulted in burning of the exhaust-valve seat and warping of the exhaust valve and the exhaust-valve guide.

A cooling-correlation equation based on this critical cylinder temperature was successfully developed. This correlation is compared with one based on the temperature of the rear spark-plug boss. Tests were made with a modified carburetor and a standard carburetor; the modified carburetor produced a more consistent relation between the average temperature of all engine cylinders and the temperature of the hottest cylinder.

ANALYSIS

The NACA cooling correlation method (reference 1) is used to present the cooling characteristics of the engine tested.

The correlation equation is of the form

$$\frac{T_h - T_a}{T_g - T_h} = C\,\frac{{W_e}^x}{(\sigma\Delta p)^y} = C\left(\frac{{W_e}^{x/y}}{\sigma\Delta p}\right)^y \tag{1}$$

where

T_h	reference cylinder-head temperature, °F
T_a	cooling-air stagnation temperature in front of engine, °F
T_g	mean effective gas temperature, °F
C	constant
W_e	weight flow of engine charge air, pounds per second
σ	ratio of density of cooling air in front of engine to density of standard sea-level air
Δp	cooling-air pressure drop, inches of water

The mean effective gas temperature T_g represents the mean gas temperature effective in the transfer of heat from the combustion gases to the cylinder head. With constant spark advance and exhaust back pressure, T_g is principally dependent upon fuel-air ratio and the temperature of the charge air entering the cylinder and is represented by the equation

$$T_g = T_{g80} + \Delta T_g \tag{2}$$

where T_{g80} is the mean effective gas temperature for a charge-air temperature of 80° F without supercharging and ΔT_g represents the variation of T_g with manifold-air temperature T_m. For a fuel-air ratio of 0.08, a value of 1150° F is assumed for the reference mean effective gas temperature of the cylinder head. (See reference 1.) The relation has been empirically determined for the cylinder heads to be approximately

$$\Delta T_g = 0.8(T_m - 80) \tag{3}$$

The manifold-air temperature T_m is calculated from the carburetor inlet-air temperature T_c and the theoretical supercharger temperature rise assuming no fuel vaporization. For the engine tested, the relation is

$$T_m = T_c + 19.8 \left(\frac{N}{1000}\right)^2 \tag{4}$$

where N is the engine speed, rpm.

In order to distinguish between the correlations presented, the subscript 1 refers to the correlation based on the average temperature of the 18 exhaust-valve seats and the subscript 2 refers to the correlation based on the average temperature of the 18 rear spark-plug bosses.

ENGINE TEST EQUIPMENT

The 3350-cubic-inch-displacement engine tested was fitted with a cowling from a four-engine airplane and installed in an engine test stand. (See fig. 1.) The cooling-air flow was provided by an exhauster, which reduced the pressure at the rear of the engine. The combustion air was supplied by a blower to the ducting forward of the intercooler.

The cylinder-head temperatures were measured by thermocouples. The positions of the thermocouples used to measure the temperatures of the exhaust-valve seat and rear spark-plug boss are shown in figure 2. The engine cooling-air pressure drop was obtained by taking the difference between the average of five total-pressure measurements on each front-row cylinder and the average of three static-pressure measurements taken at the rear of each rear-row cylinder. The position of these pressure tubes on the engine cylinders is shown in figure 3.

TEST PROCEDURE

Two series of tests were made to establish the correlation at a fuel-air ratio of 0.08 with the modified carburetor:

1. Cooling-air flow was varied while fuel-air ratio, carburetor-air temperature, engine speed, and charge-air flow were held constant to determine the variation of engine temperature with cooling-air flow.

2. Charge-air flow was varied while fuel-air ratio, carburetor-air temperature, engine speed, and cooling-air flow were held constant to determine the variation of engine temperature with charge-air flow.

In order to determine the effect of fuel-air ratio on mean effective gas temperature, tests were made with the modified and the standard carburetors at various fuel-air ratios, powers, and engine speeds at a constant carburetor-air temperature.

RESULTS AND DISCUSSION

Figure 4 establishes the relation between the cooling temperature differential $(T_h - T_a)/(T_g - T_h)$ and the cooling-air pressure drop $\sigma\Delta p$. The slope of the curve in figure 4(a) based on the temperature of the exhaust-valve seat determines the exponent y_1 in the equation

$$\frac{T_{h1} - T_a}{T_{g1} - T_{h1}} = C_1 \frac{W_c^{x_1}}{(\sigma\Delta p)^{y_1}} \tag{5}$$

A similar curve is shown in figure 4(b) based on the temperature of the rear spark-plug boss. The slope of this curve determines the exponent y_2 in the equation

$$\frac{T_{h2} - T_a}{T_{g2} - T_{h2}} = C_2 \frac{W_e^{x_2}}{(\sigma\Delta p)^{y_2}} \tag{6}$$

A comparison of figures 4(a) and 4(b) shows that the temperature of the critical region of the cylinder is less affected by engine cooling-air flow than is the temperature of the rear spark-plug boss.

Figure 5 establishes the relation between the cooling temperature differential $(T_h - T_a)/(T_g - T_h)$ and the weight flow of engine charge air W_e. The slopes of the curves in figure 5(a) and figure 5(b) determine the exponents x_1 and x_2 in equations (5) and (6), respectively. The slopes of the curves in figure 5 are essentially the same, which indicates that the temperatures of the critical region of the cylinder and the rear spark-plug boss are equally affected by the weight flow of engine charge air.

When these exponents are substituted in equation (1) and the constants evaluated, the cooling-correlation equations (7) and (8) are determined based on the average temperature of the exhaust-valve seat and the average temperature of the rear spark-plug boss, respectively.

$$\frac{T_{h1} - T_a}{T_{g1} - T_{h1}} = 0.576 \left(\frac{W_e^{2.50}}{\sigma \Delta p}\right)^{0.20} \tag{7}$$

$$\frac{T_{h2} - T_a}{T_{g2} - T_{h2}} = 0.500 \left(\frac{W_e^{1.79}}{\sigma \Delta p}\right)^{0.29} \tag{8}$$

A plot of equations (7) and (8) is presented in figure 6.

The variation of the reference mean effective gas temperature T_{g80} with fuel-air ratio based on tests with the modified carburetor is shown in figure 7. The reference mean effective gas temperature was calculated by solving equations (7) and (8) for T_g and then calculating the reference mean effective gas temperature T_{g80} using equations (2), (3), and (4). Similar curves showing the variation of the reference mean effective gas temperature with fuel-air ratio, based on tests using the standard carburetor are shown in figure 8.

Figures 7 and 8 show that the scatter of test results based on the temperature of the exhaust-valve seat is less than the scatter when based on the temperature of the rear spark-plug boss. Figure 7, based on results obtained with the modified carburetor, shows the effect of improved fuel-air-ratio distribution in that it closely approximates unpublished single-cylinder test data. The curves have a sharp peak at a fuel-air ratio of approximately 0.068 and the range of stable lean engine operation has been extended to a fuel-air ratio of 0.052.

APPLICATION OF RESULTS

Inasmuch as the correlation is based on the average temperature of all engine cylinders, whereas the most important temperature is that of the hottest cylinder, it is necessary that accurate information be available relating the temperature of the hottest cylinder to the average temperature of all cylinders.

Figure 9 shows the difference between the temperature of the hottest exhaust-valve seat and the average temperature of the exhaust-valve seats for all cylinders plotted against fuel-air ratio for each power condition investigated. A similar plot of the difference between the temperature of the hottest rear spark-plug boss and the average temperature of the rear spark-plug bosses for all cylinders is given in figure 10. In addition to the data obtained with the standard and modified carburetors in the present tests (engine A), unpublished data are included from various engines of the same model (B, C, and D) equipped with the standard carburetor. These data indicate no consistent relation between maximum and average cylinder temperatures for engines equipped with the standard carburetor. The engine equipped with the modified carburetor, however, produced a consistent pattern in repeated tests and an accurate determination of the average temperature for all cylinders may be obtained from a given temperature of the hottest cylinder.

Another important parameter required for cooling-correlation predictions is the weight flow of engine charge air. The variation of the brake specific air consumption with fuel-air ratio for various engine operating conditions is shown in figure 11.

An accurate determination of these parameters substantially increases the accuracy of predicting engine cooling-air pressure-drop requirements based on the cooling correlation.

SUMMARY OF RESULTS

From test-stand results to determine the cooling correlation of large double-row radial engine, with a 3350-cubic-inch displacement, the following results may be summarized:

1. A satisfactory cooling correlation equation based on the temperature of the exhaust-valve seat was obtained.

2. Improved fuel-air-ratio distribution to the engine cylinders produced by the modified carburetor substantially increased the accuracy of predicting engine cooling-air pressure-drop requirements based on the cooling correlation.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, April 30, 1945.

REFERENCES

1. Pinkel, Benjamin: Heat-Transfer Processes in Air-Cooled Engine Cylinders. NACA Rep. No. 612, 1938.

2. Koenig, Robert J., and Engelman, Helmuth W.: A Cooling Correlation of the Wright R-2600-8 Engine Showing the Effect of Water as an Internal Coolant. NACA ARR No. E5A29, 1945.

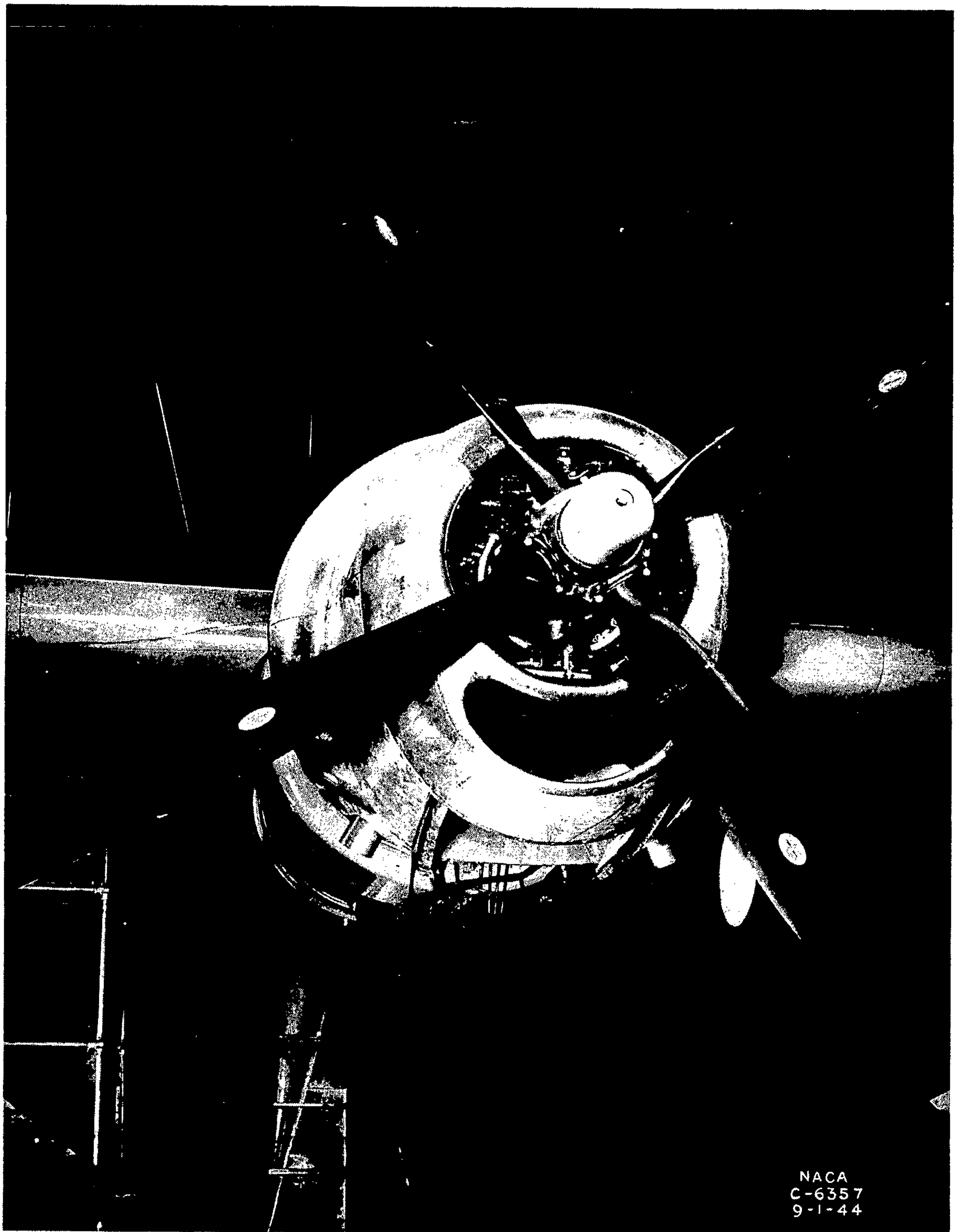
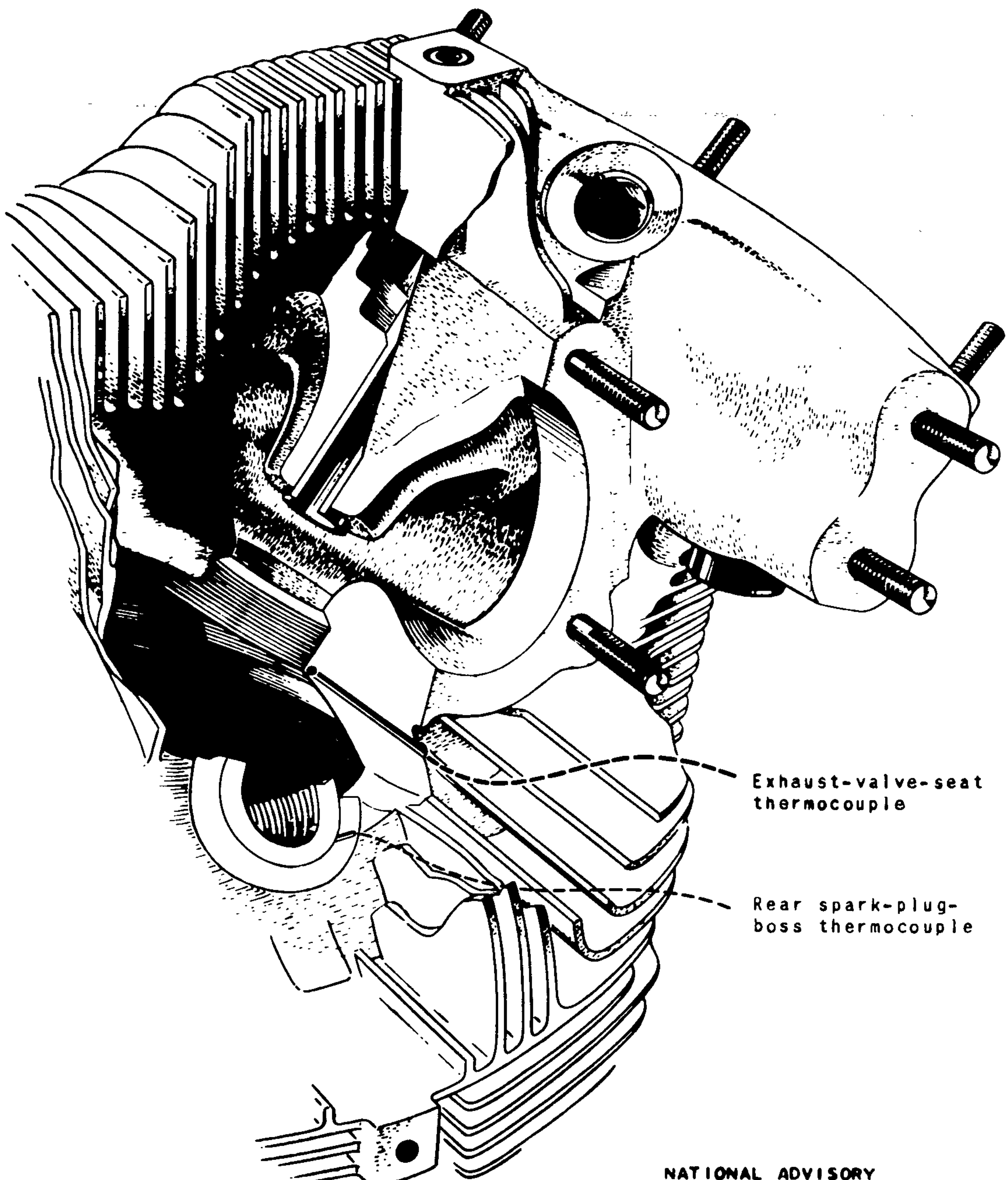
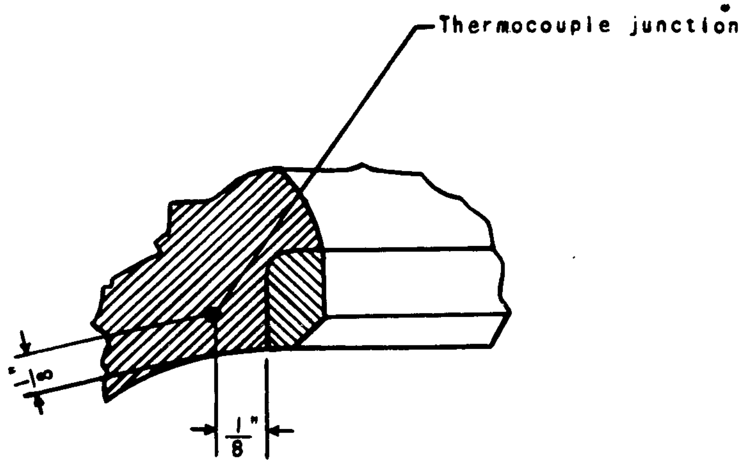


Figure 1. - Test-stand installation of 3350-cubic-inch-displacement engine fitted with cowling from four-engine airplane.

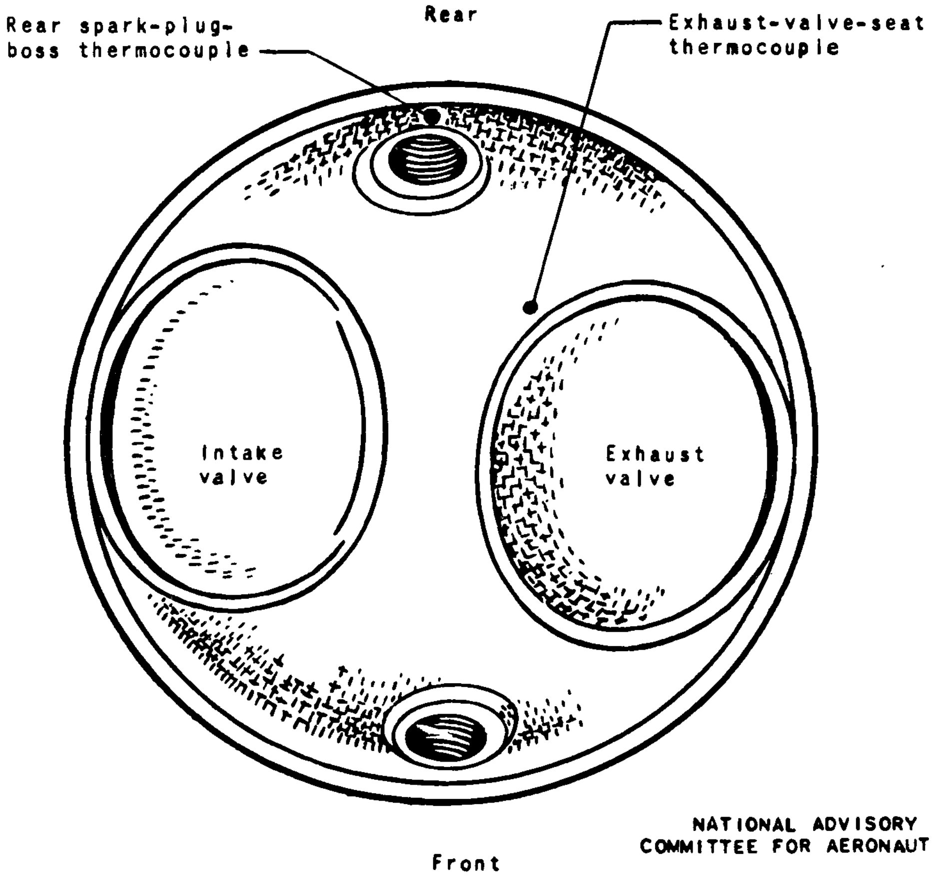


(a) Cutaway of cylinder.

Figure 2. - Location of thermocouples installed in cylinder.



(b) Section through valve seat.



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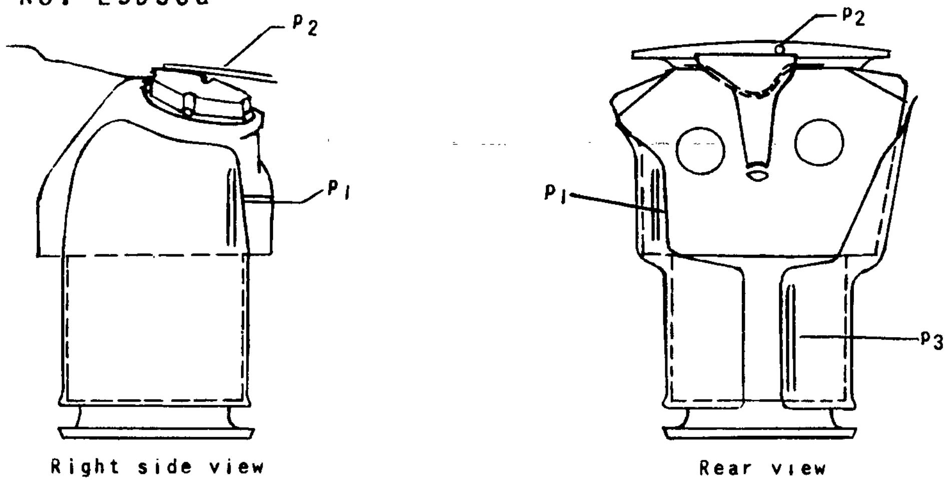
(c) Looking into barrel.

Figure 2. - Concluded. Location of thermocouples installed in cylinder.



(a) Static-pressure tubes on rear-row cylinder.

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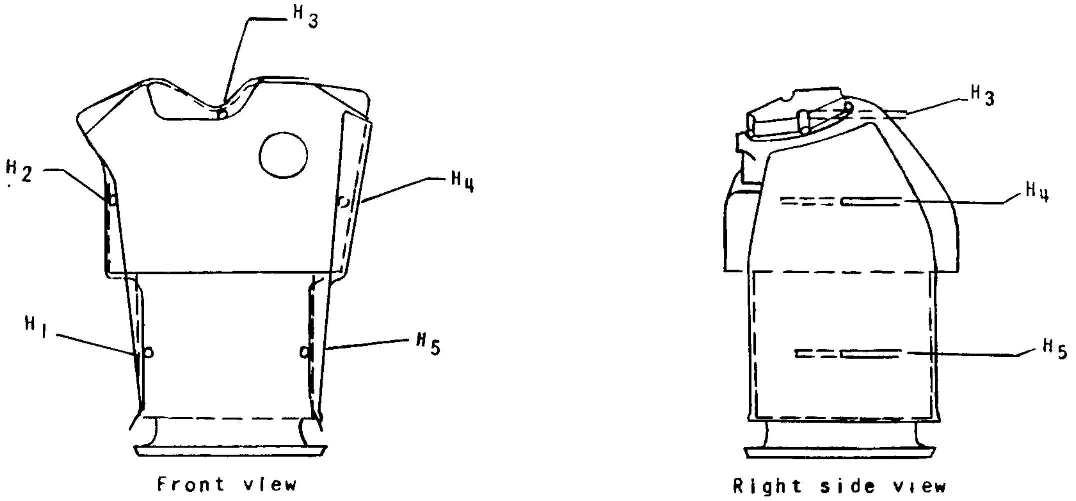
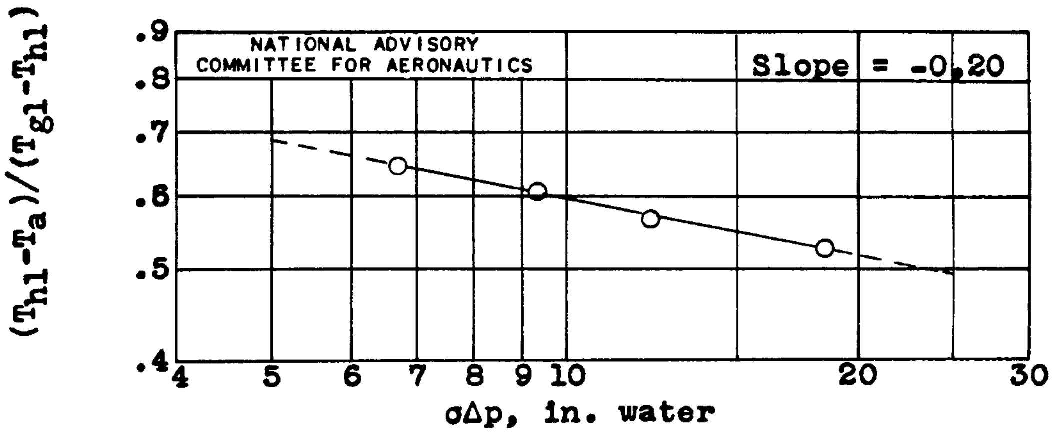
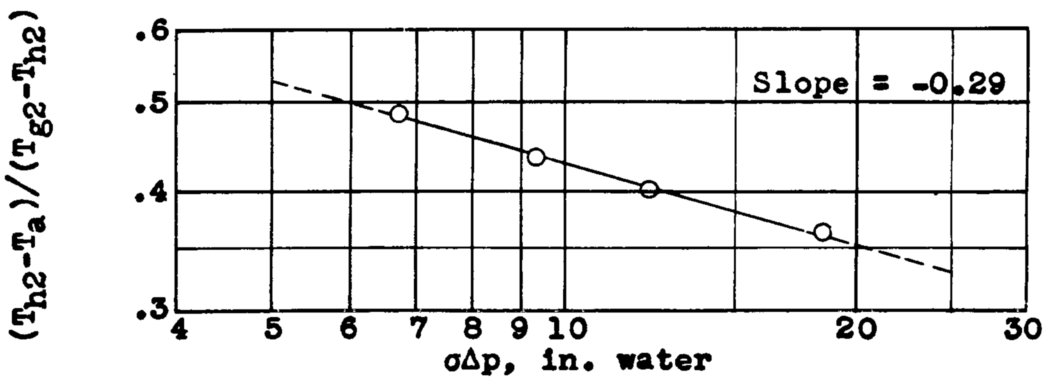


(b) Total-pressure tubes on front-row cylinder.

Figure 3. - Location of cooling-air pressure tubes installed on cylinders.

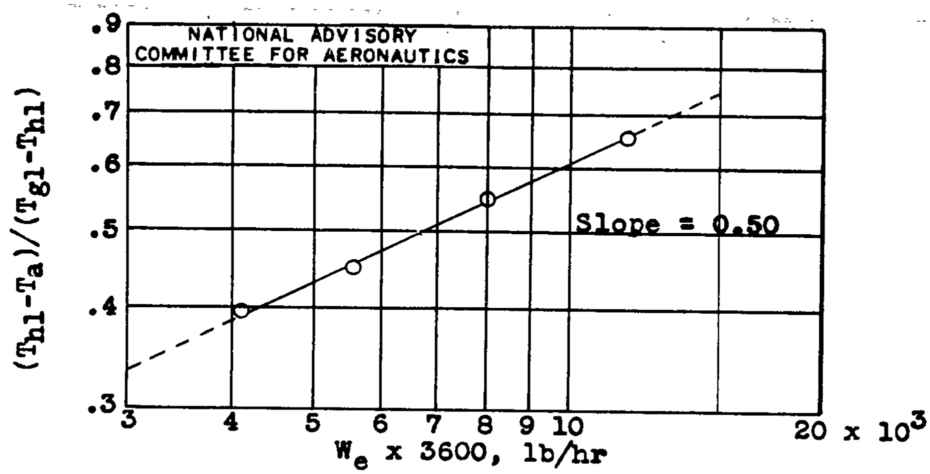


(a) Based on temperature of exhaust-valve seat T_{h1}.

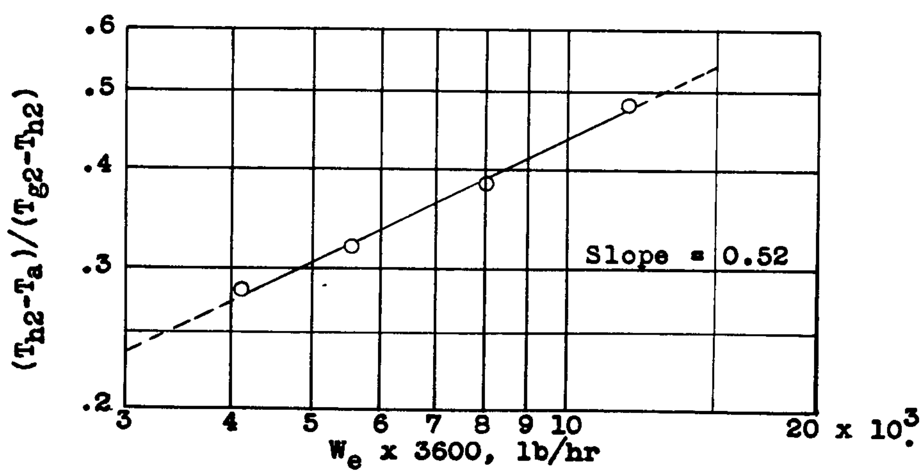


(b) Based on temperature of rear spark-plug boss T_{h2}.

Figure 4. - Variation of cooling temperature differential $(T_h-T_a)/(T_g-T_h)$ with sea-level cooling-air pressure drop $\sigma\Delta p$. Charge-air flow, 9800 pounds per hour; fuel-air ratio, 0.08.

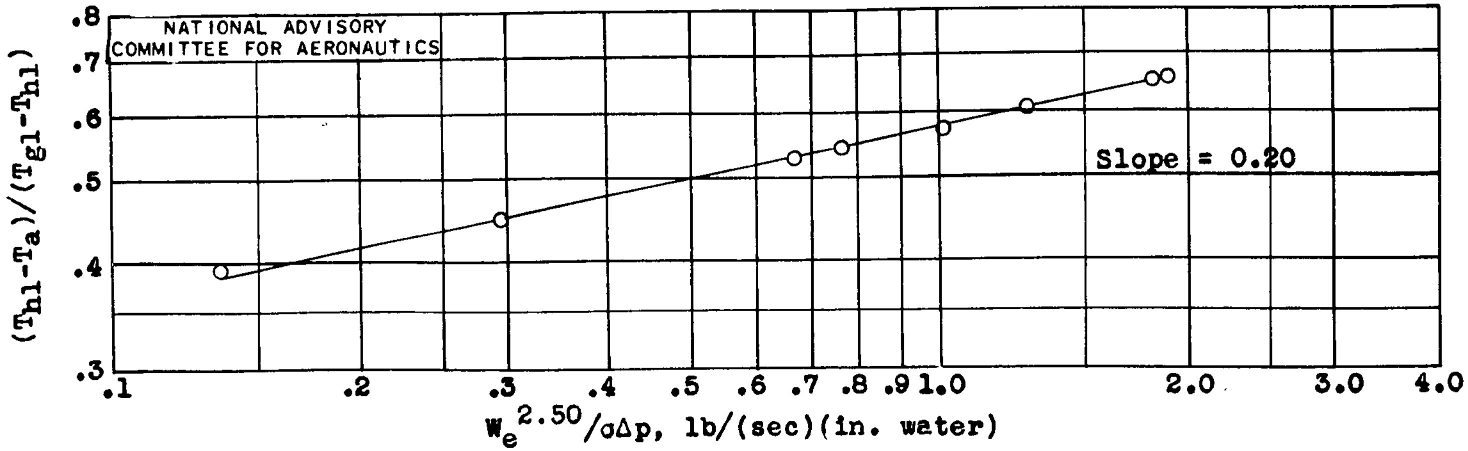


(a) Based on temperature of exhaust-valve seat T_{h1}.

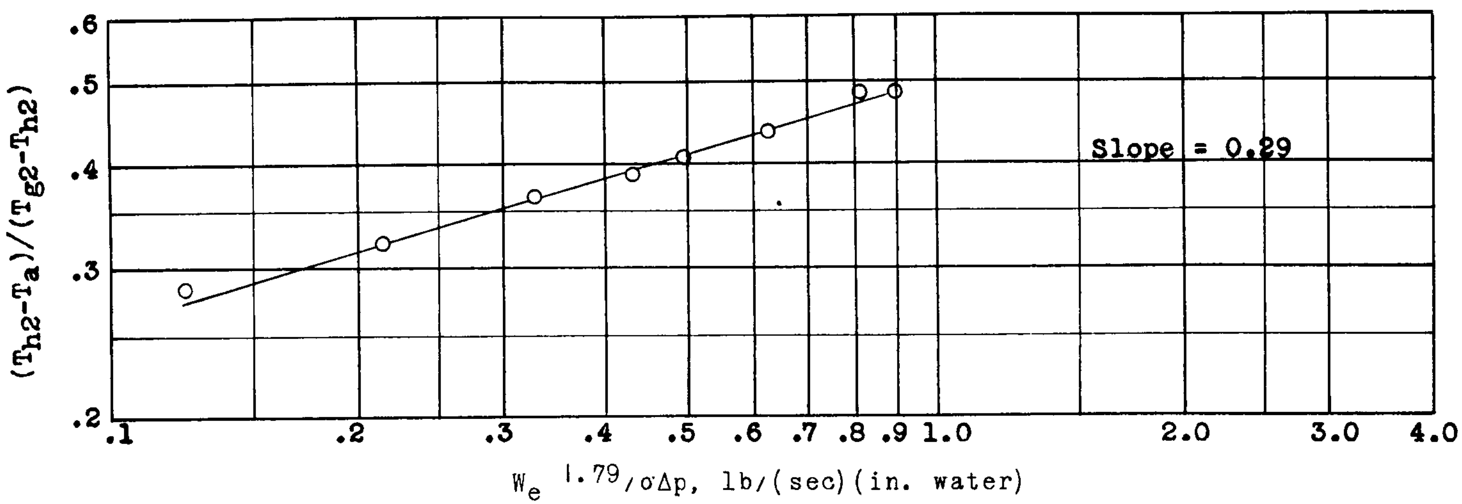


(b) Based on temperature of rear spark-plug boss T_{h2}.

Figure 5. - Variation of cooling temperature differential $(T_h-T_a)/(T_g-T_h)$ with charge-air flow W_e. Cooling-air pressure drop, 18 inches of water; fuel-air ratio, 0.08.



(a) Based on temperature of exhaust-valve seat T_{h1}.



(b) Based on temperature of rear spark-plug boss T_{h2}.

Figure 6. - Variation of cooling temperature differential $(T_h-T_a)/(T_g-T_h)$ with ratio of charge-air flow to cooling-air pressure drop. Fuel-air ratio, 0.08.

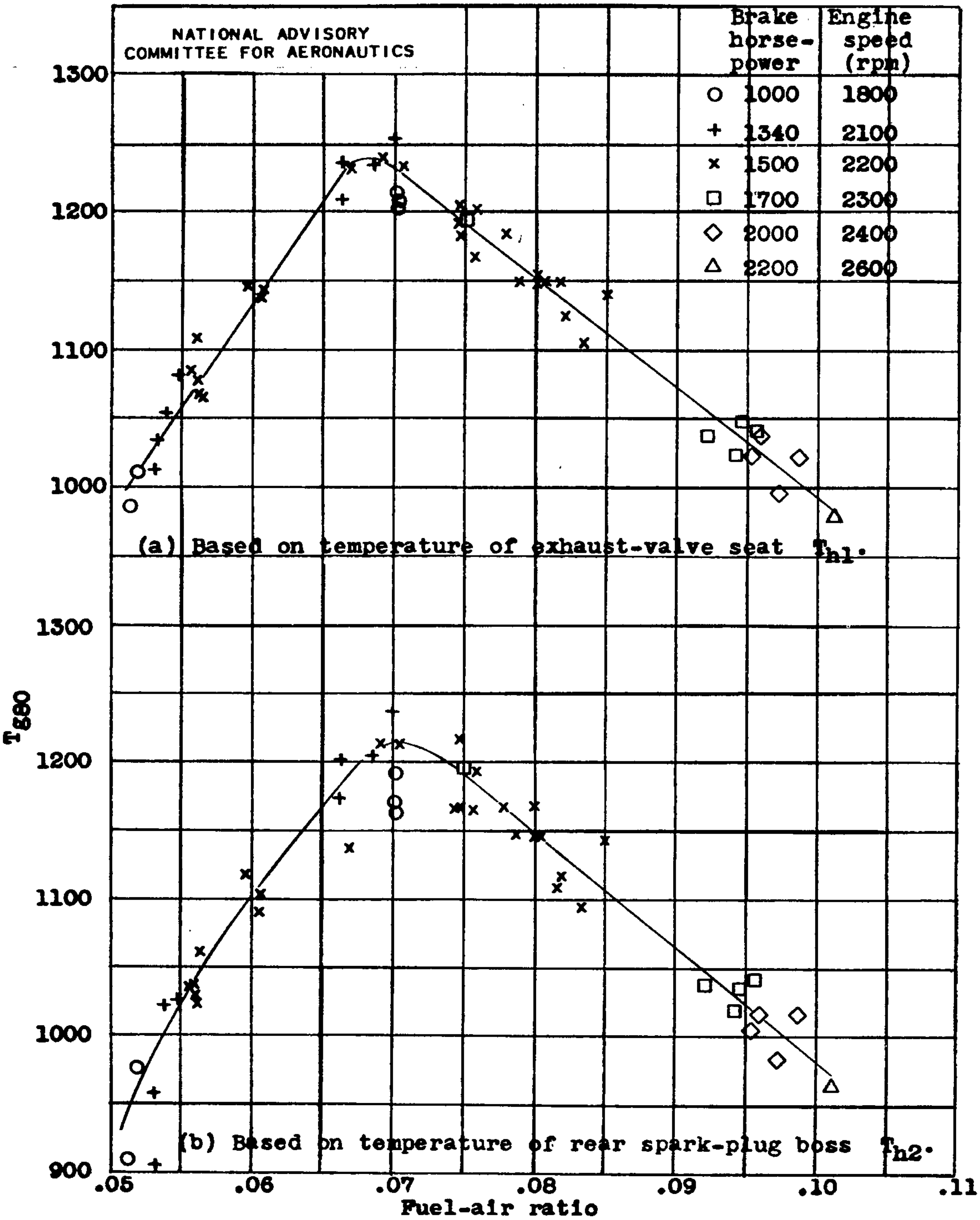


Figure 7. - Variation of reference mean effective gas temperature T_{g80} with fuel-air ratio for cylinder heads obtained with modified carburetor.

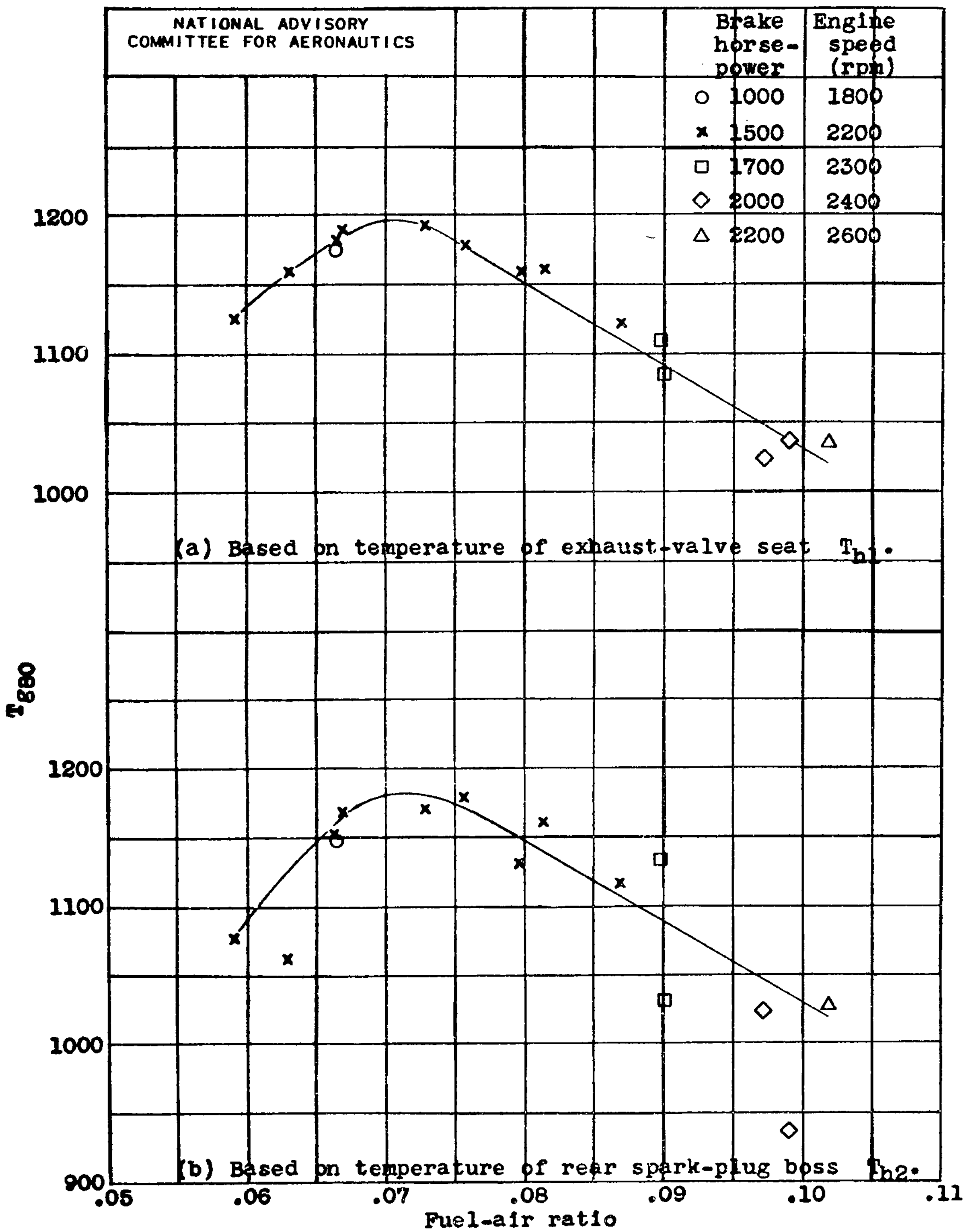


Figure 8. - Variation of reference mean effective gas temperature T_{g80} with fuel-air ratio for cylinder heads obtained with standard carburetor.

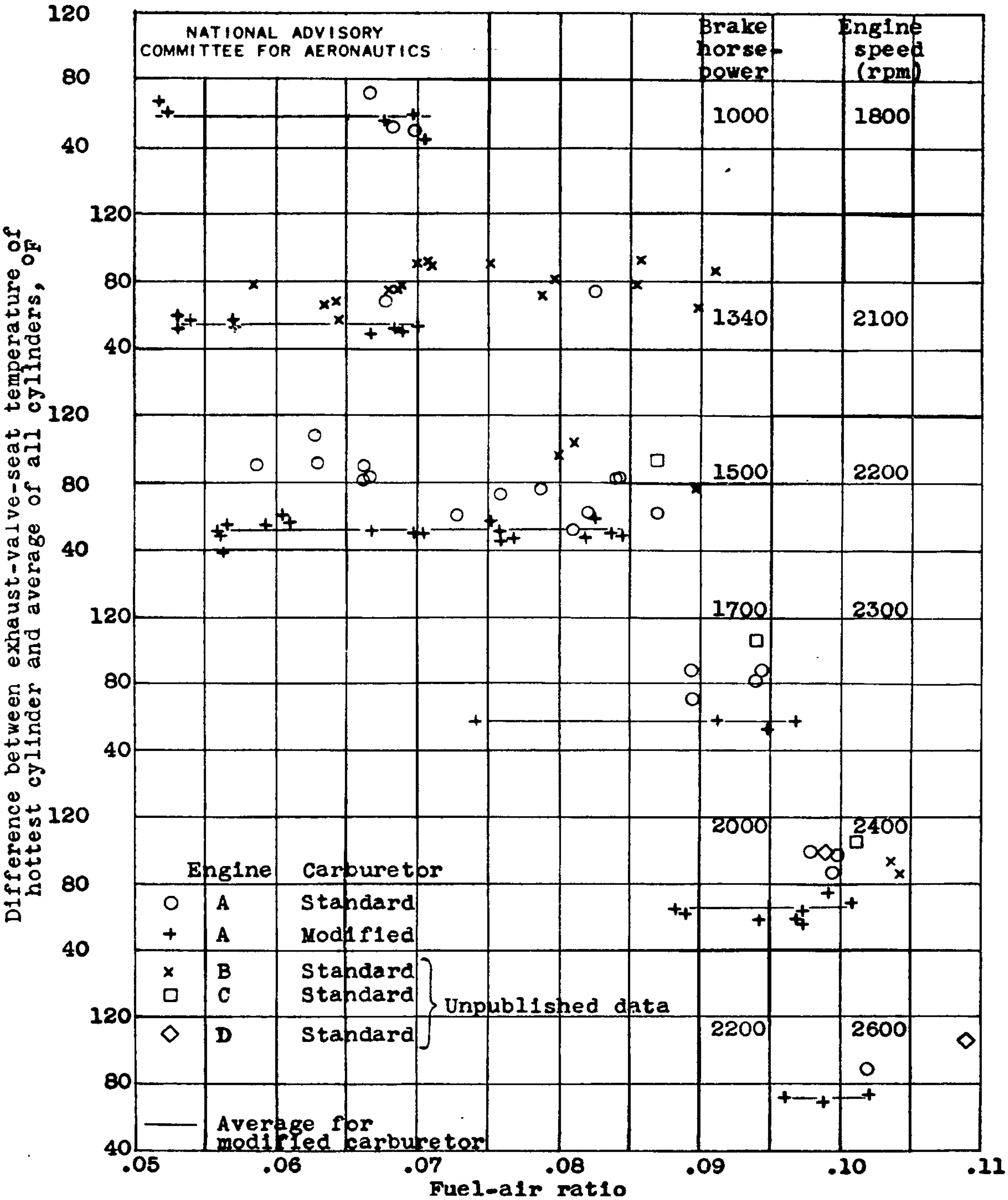


Figure 9. - Exhaust-valve-seat temperature spreads obtained with standard and modified carburetors.

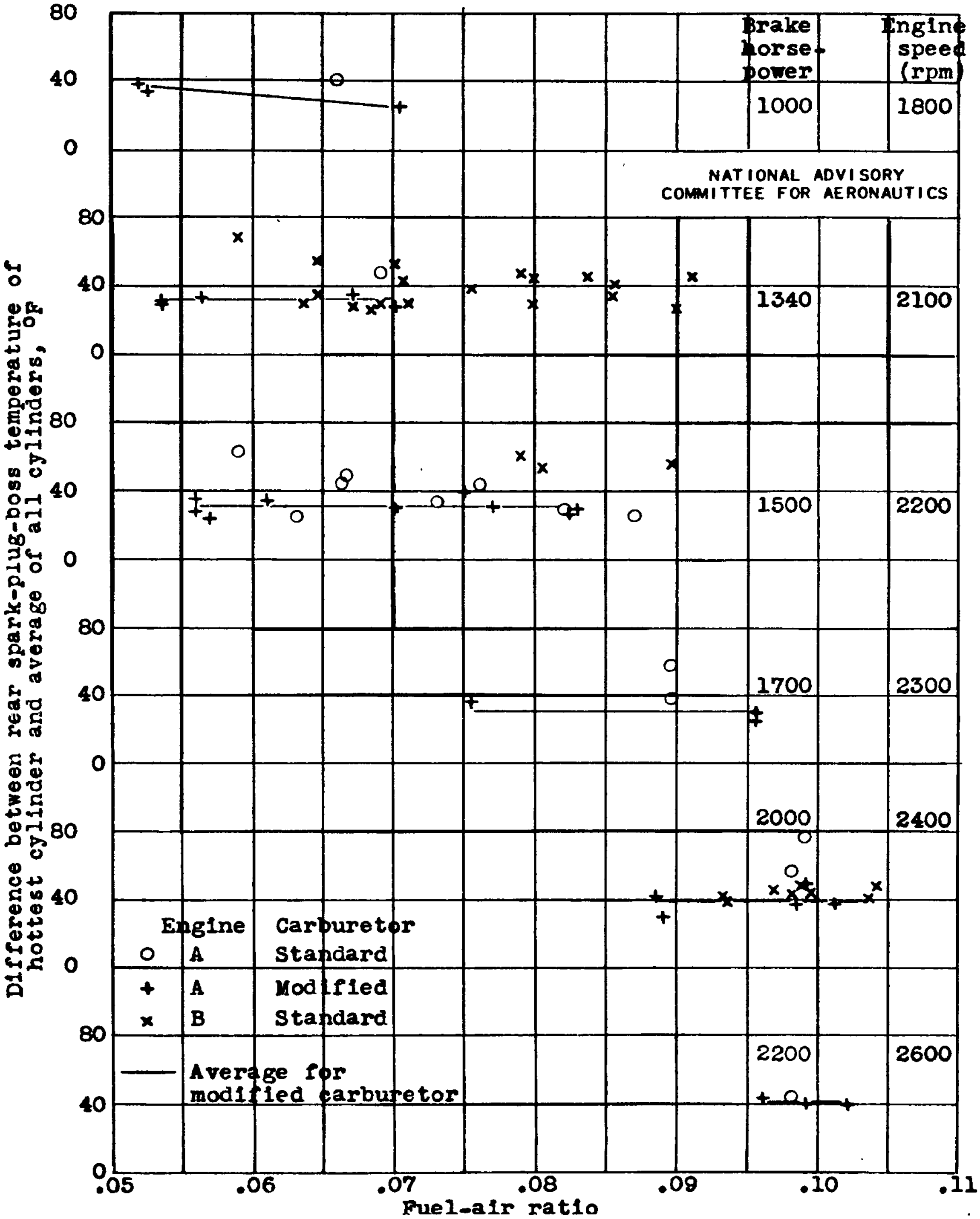


Figure 10. - Rear spark-plug-boss temperature spreads obtained with standard and modified carburetors.

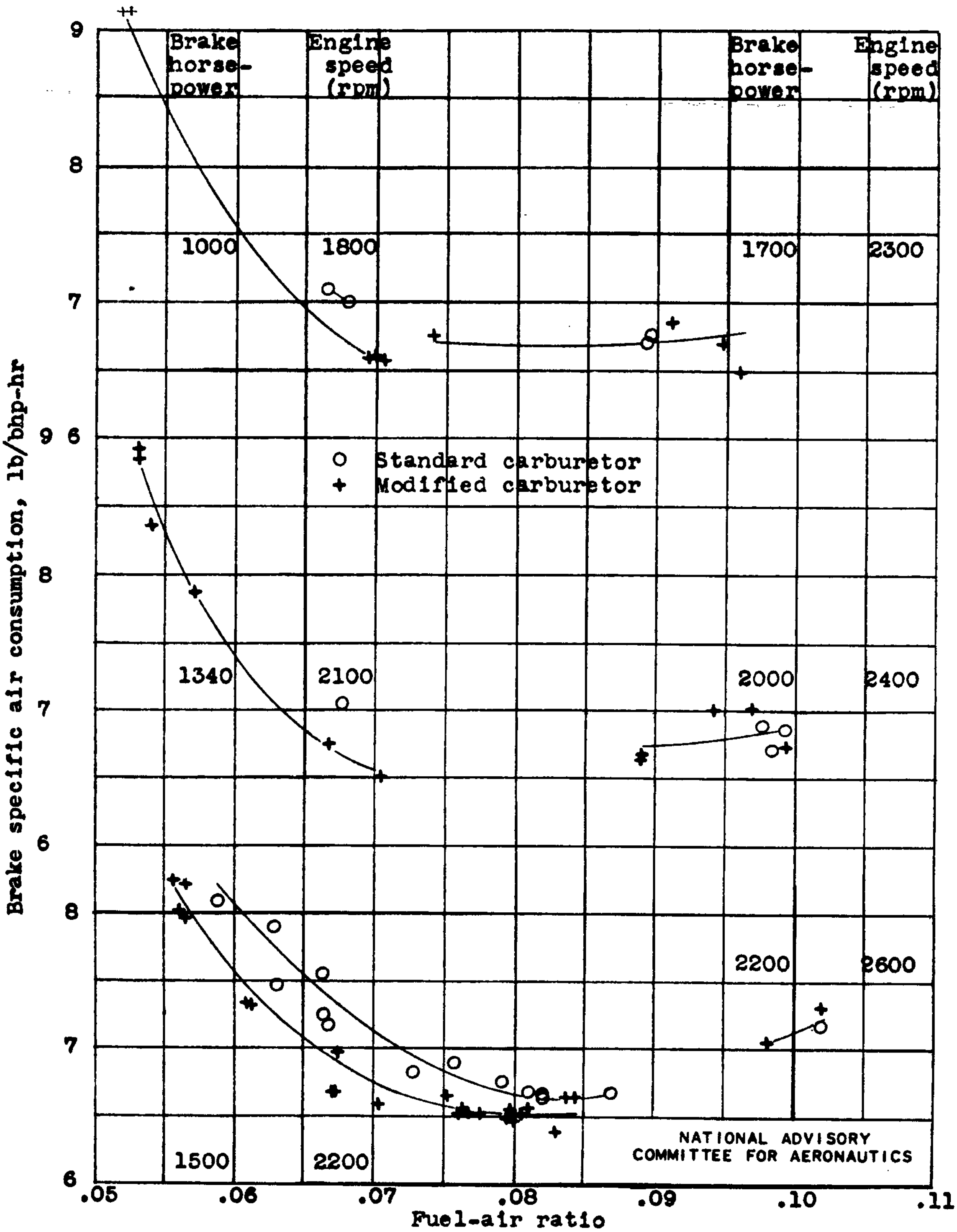


Figure 11.- Comparison of brake specific air consumption obtained with standard and modified carburetors.