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EFFECT OF INCREASING THE SIZE OF THE VALVE-GUIDE BOSS ON THE EXHAUST-VALVE TEMPERATURE AND THE VOLUMETRIC EFFICIENCY OF AN AIRCRAFT CYLINDER

By Max D. Peters

Aircraft Engine Research Laboratory
Cleveland, Ohio

NACA

WASHINGTON

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ADVANCE RESTRICTED REPORT

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SUMMARY

An exhaust-port design was developed that provides a heat-flow path of increased area from the valve stem to the outside surface of a Wright C9GC cylinder. The effects of the new port design on exhaust-valve temperature and volumetric efficiency were determined from single-cylinder engine tests. A reduction of 70° F in the exhaust-valve temperature was obtained with no decrease in volumetric efficiency or power output.

INTRODUCTION

The tests herein described are part of an investigation being conducted by the NACA laboratory in Cleveland, Ohio to determine and eliminate the causes of exhaust-valve failures in aircraft engines. Previous tests by the NACA (reference 1) indicated that a larger exhaust-valve-guide boss would provide a heat-flow path of greater area from the valve stem to the outside surface of the cylinder and would aid in preventing valve failures caused by overheating. It was realized, however, that an increase in the size of the boss would reduce the cross-sectional area of the port and retard gas flow through the port. Taylor, however, showed in reference 2 that the size of the exhaust port could be made much smaller than usual without a serious decrease in the flow coefficient. Taylor also found, as did Heron (reference 3) and Doman (reference 4), that the exhaust valves can be made smaller than the intake valves without causing a loss in engine power.

The first part of this report discusses the development of an exhaust-port design that has satisfactory gas-flow characteristics and a heat-flow path of materially increased area from the valve stem to the outside surface of the cylinder. The design was incorporated in an aircraft cylinder and was based on the results of a series of steady-flow tests conducted to determine the maximum size boss that could be contained in the exhaust port without a detrimental effect on engine air capacity. The second part of this report describes single-cylinder engine tests conducted by the NACA of an aircraft cylinder incorporating the altered port; the tests were run to determine the effect of the port on power output and exhaust-valve temperature.

DEVELOPMENT OF THE PORT DESIGN

Although a reduction in the operating temperature of the exhaust valve was the primary object of this investigation, it was first necessary to determine the effect of the port shape on gas flow through the port and the volumetric efficiency of the cylinder. The contours and cross-sectional areas of a standard Wright C9GC port and of three experimental ports were compared and their flow characteristics under steady-flow conditions were determined. The effect of a reduction in flow area on volumetric efficiency was determined from single-cylinder engine tests of a standard Wright C9GC cylinder equipped with an exhaust valve providing 85 percent of the flow area of the stock valve.

Apparatus

A diagrammatic layout of the apparatus used in the steady-flow tests of the experimental ports is shown in figure 1. The cylinder was mounted upon the tank outlet by means of an adapter plate. Valve lift was varied by a micrometer screw mechanism. The air flowing through the system was measured by a thin-plate orifice installed in accordance with A.S.M.E. standards. The air pressure in the tank was regulated by a hand-operated valve located ahead of the orifice. Air pressures were measured with mercury manometers and temperatures were determined with iron-constantan thermocouples.

The engine tests to determine the volumetric efficiencies of the standard cylinder and of the cylinder with the exhaust valve providing 85 percent of the flow area of the stock valve were run on an NACA universal test-engine crankcase. The intake pipe was 15 inches long and the exhaust pipe was 25 inches long. A schematic drawing of the standard test setup used is shown in figure 2.

Procedure

The effect of port shape on volumetric efficiency was determined by altering the shape of the port with modeling clay and measuring the air flow through the port at various valve lifts at constant tank pressure. The tank pressure was measured by manometer C shown in figure 1 and was considered to be the pressure causing flow through the port. The valve lift was varied from 0.050 inch to 0.600 inch.

As each port was flow-tested, it was necessary to determine the relation between the flow characteristics and the shape of the port. Accordingly, the cross-sectional areas of the ports were obtained by making a flexible mold of the port interior. The mold was withdrawn from the port, filled with plaster of Paris, and sawed into sections on planes, the traces of which are shown by the lines A, B, C and 1 to 7 in figure 3. The outlines of these sections were then drawn on paper and the areas were measured with a planimeter. Flow areas measured from several molds taken from two standard cylinder heads agreed within ±2 percent, indicating that the mold method is sufficiently accurate. The numbered half-sections of the standard port are shown in figure 4.

The cross-sectional areas of the ports plotted against distance measured along the mean flow paths of figure 3 are shown in figure 5. The areas of sections A, B, C, and 1 are the same for the standard port and ports 1 and 2. All of the sections except A, B, C, and 1 have equal areas for the port equipped with the 85-percent flow-area valve and the standard port. The locations of the numbered sections were the same as those shown on Wright Aeronautical Corporation drawing 112096 except for the locations in port 1. The areas at the lettered sections between the fully opened valve and the seat were calculated as the lateral area of a frustum of a cone. Corrections for the cross-sectional area of the valve stem were made for sections 1 and 2.

The engine tests to compare the volumetric efficiencies of the cylinder using the 85-percent flow-area exhaust valve and of the cylinder with the stock exhaust valve were run at a constant inlet-air temperature of 150° F and a manifold pressure of 40 inches of mercury absolute; standard valve timing was used in both tests. The exhaust back pressure was controlled to 28.2 inches of mercury absolute and the fuel-air ratio was 0.097.

Comparison of Test Results

Flow tests of the ports were made over a range of pressure drops across the valve up to 45 inches of mercury, but the test equipment limited the higher pressure-drop tests to the lower valve lifts. It was found, however, that the performance of the experimental ports that had a greater or lesser air flow than that of the standard port at any lift was consistent over the range of pressure drops used. For this reason, only the results of tests at a pressure drop of 10 inches of mercury are presented. Figure 6 compares the volume of air flowing through the ports plotted with valve lift. The flow reaches maximum values for the three modified ports at the higher lifts and the flow for the standard port is still increasing but at a slower rate. Flow tests made at pressure drops of 5 and 15 inches of mercury showed the same trends.

The following table compares the flow through the modified ports with the flow through the standard port at two valve lifts.

Port	Valve lift (in.)	
	0.320	0.562
	Flow (percent of standard)	
Standard Wright C9GC port	100	100
Port 1	100	80
Port 2	105	97
Valve with 85-percent flow area in standard port	82	77

The results of the flow tests of port 1 correspond exactly with those of the standard port at the lower lifts, but a maximum flow is reached by port 1 at a lower lift than the standard port. Port 1 contained an extremely heavy boss that reduced the flow area considerably in sections 2 to 6. (See fig. 5.) The flow through port 2 was greater than that through the standard port at all but the highest lifts. This port was designed with a slightly smaller boss in an effort to reduce the restriction to flow that port 1 possessed at the higher valve lifts. The half-sections of this port are shown in figure 7. The flow through the standard port equipped with the 85-percent flow-area valve was less than the flow through the other ports.

The volumetric efficiencies of the cylinder equipped with the 85-percent flow-area valve and of a standard cylinder are plotted in figure 8. No difference in the volumetric efficiencies was found over a range of engine speeds from 1200 to 2500 rpm (rated take-off speed) and one curve was faired through the two sets of test points.

Any port that has an air flow greater than that of the port with the 85-percent flow-area valve would therefore not be expected to cause a loss in volumetric efficiency. Port 2 would probably improve the exhaust process because it restricted flow the least at all but very high lifts and it is generally thought that most of the exhaust gas escapes at the lower lifts.

Figure 5 shows that sections 3 and 4 of port 1 are smaller than sections B and C of the standard port equipped with the 85-percent flow-area valve. Even though a smaller section existed in port 1, its restriction to air flow was not so great as that of the port equipped with the 85-percent flow-area valve (fig. 6). This condition might indicate that the flow area through the valve and seat is the most critical section of an exhaust port and that the rest of the port can be designed with less emphasis on gas flow and more consideration given to cooling of the exhaust valve.

ENGINE TESTING OF FINAL PORT DESIGN

Port 2 was chosen as the design to be engine-tested because it had satisfactory gas-flow characteristics and a heat-flow path of greater area. Cross sections of the standard Wright C9GC cylinder are presented in figure 9 to compare the sizes of the exhaust-valve-guide boss in the standard port and in port 2. The heat-flow paths at section A-A (fig. 9(a)) of port 2 and of the standard port are compared in figure 10.

Apparatus

The Wright C9GC cylinder was altered by building up the boss with arc-welded aluminum in order to incorporate the design of port 2. Molds of the port were taken during the alteration process to insure that the desired design was being followed.

The temperature of the exhaust valve was measured with a steel-constantan thermocouple installed in the valve itself, as shown in figure 11. This thermocouple was calibrated in an electric furnace by comparing it with a standard thermocouple that was spot-welded to the valve crown. Although an aluminum-valve-steel couple was produced as a parallel circuit to the valve-thermocouple circuit when the valve was seated in the head, tests showed that the effect of this additional couple is negligible. The calibration of the valve thermocouple was found to be unchanged after the valve had been used in the engine.

The crankcase and the test equipment that were used in the volumetric-efficiency tests were used in the engine tests to determine the effect of port 2 on exhaust-valve temperature and power output.

Procedure

The following engine conditions were held constant during the comparative tests:

Engine speed, rpm	2200
Fuel-air ratio	0.099
Combustion-air temperature, °F	150
Cooling-air pressure, inches of water	16
Spark timing, degrees B.T.C.	22.5
Exhaust back pressure, inches of mercury absolute	28.6

The same thermocouple-equipped valve was used in both cylinders; valve temperatures were recorded as a function of indicated mean effective pressure. The temperatures were recorded at each test condition after readings had been stabilized for approximately 5 minutes. The indicated mean effective pressures were obtained by adding the brake mean effective pressures to the friction mean effective pressures determined by motoring the engine.

Comparison of Engine-Test Results

The operating temperatures of the exhaust valve in the standard cylinder and in the cylinder with port 2 are plotted in figure 12. Exhaust-valve temperatures were reduced approximately 70° F in the cylinder with the enlarged boss. The rear spark-plug-bushing temperature was very nearly the same for both tests. A greater valve-temperature reduction might be accomplished if more cooling-fin surface were provided around the exhaust-valve-guide boss. Figure 9(b) indicates the possible increase of fin area that might be incorporated in the port 2 design.

The relation between intake manifold pressure and indicated mean effective pressure is plotted in figure 13 for both tests. Changing the exhaust-valve-guide boss had no effect on the power output of the cylinder.

SUMMARY OF RESULTS

Comparative tests of a standard cylinder and a modified cylinder with an enlarged exhaust-valve-guide boss gave the following results:

1. Increasing the size of the exhaust-valve-guide boss reduced the exhaust-valve temperature 70° F.

2. As predicted by the steady-flow tests of the altered port, the increased size of the exhaust-valve-guide boss did not result in a decrease in power output of the cylinder.

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

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3. Heron, S. D.: Exhaust-Valves and Guides for Aircraft Engines. SAE Jour., vol. XV, no. 2, Aug. 1924, pp. 122-132.

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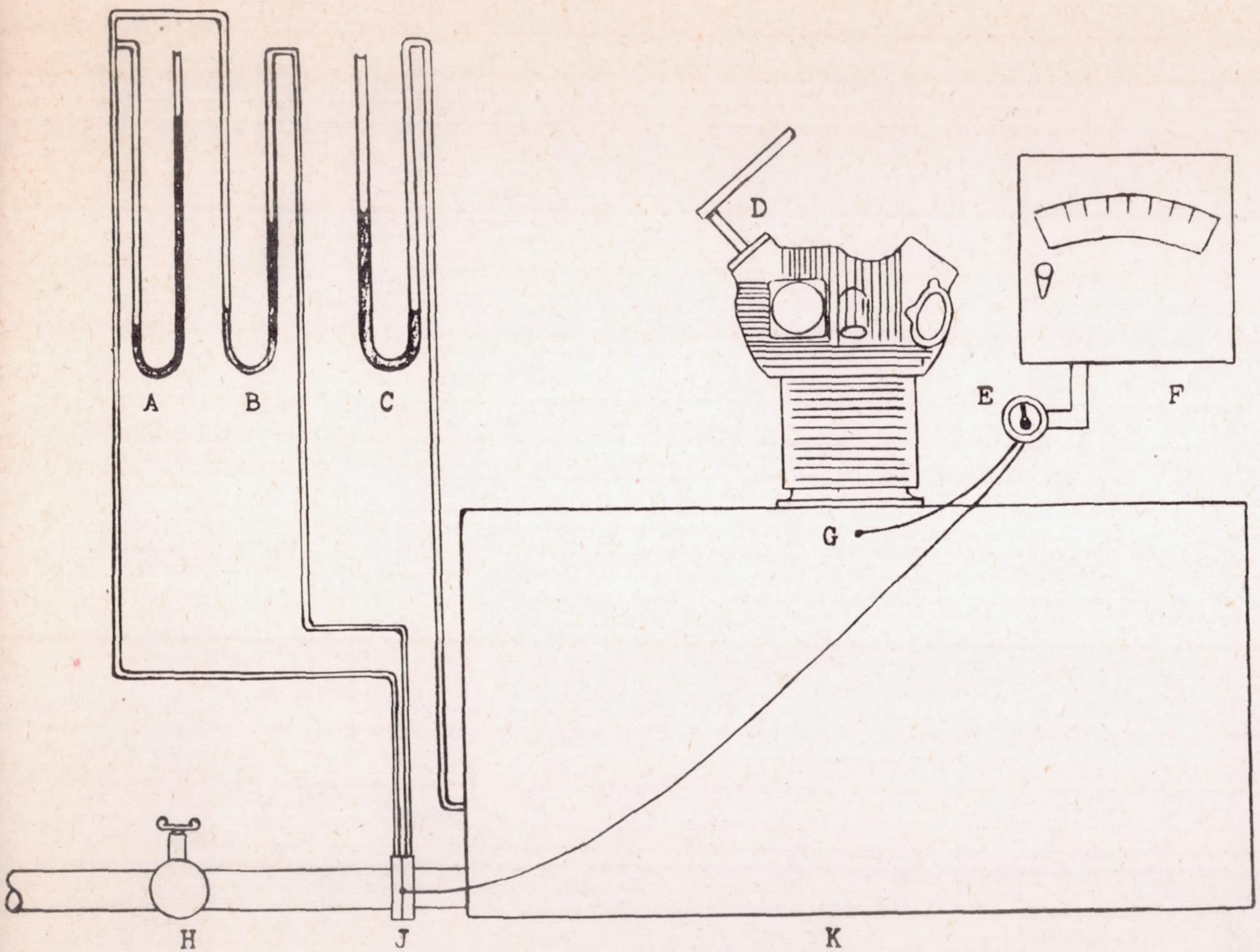

Manometers to measure:
A Inlet pressure
B Orifice differential
C Tank pressure
D Valve-lift mechanism
E Selector switch
F Potentiometer
G Tank thermocouple
H Valve in air line
J Orifice and orifice thermocouple
K Tank

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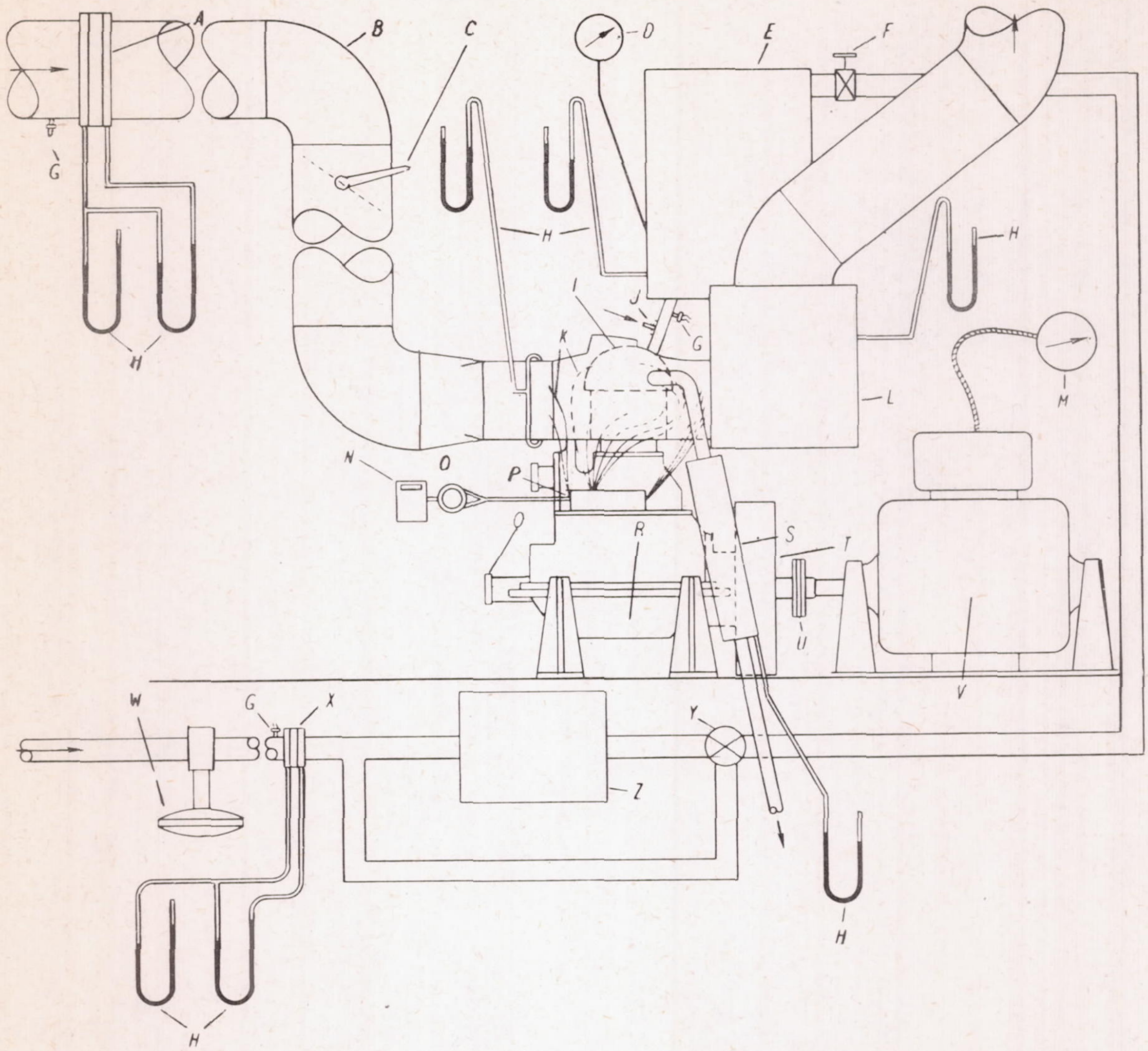
Figure 1.- Diagrammatic layout of apparatus used in steady-flow tests of the experimental ports.



A Cooling-air orifice
B Cooling-air duct
C Cooling-air pressure regulator
D Remote-reading thermometer
E Surge tank
F Throttle
G Thermocouple
H Manometer
I Test cylinder
J Manifold fuel injector
K Thermocouple leads
L Downstream cooling-air expansion tank
M Remote-reading dynamometer scale
N Self-balancing potentiometer
O Selector switch
P Thermocouple cold-junction box
Q Distributor
R Crankcase
S Exhaust expansion tank
T Flywheel
U Flexible coupling
V Dynamometer
W Combustion-air pressure regulator
X Combustion-air orifice
Y Motor-controlled mixing valve
Z Air heater

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Figure 2. - Test engine, auxiliary equipment, and instruments.

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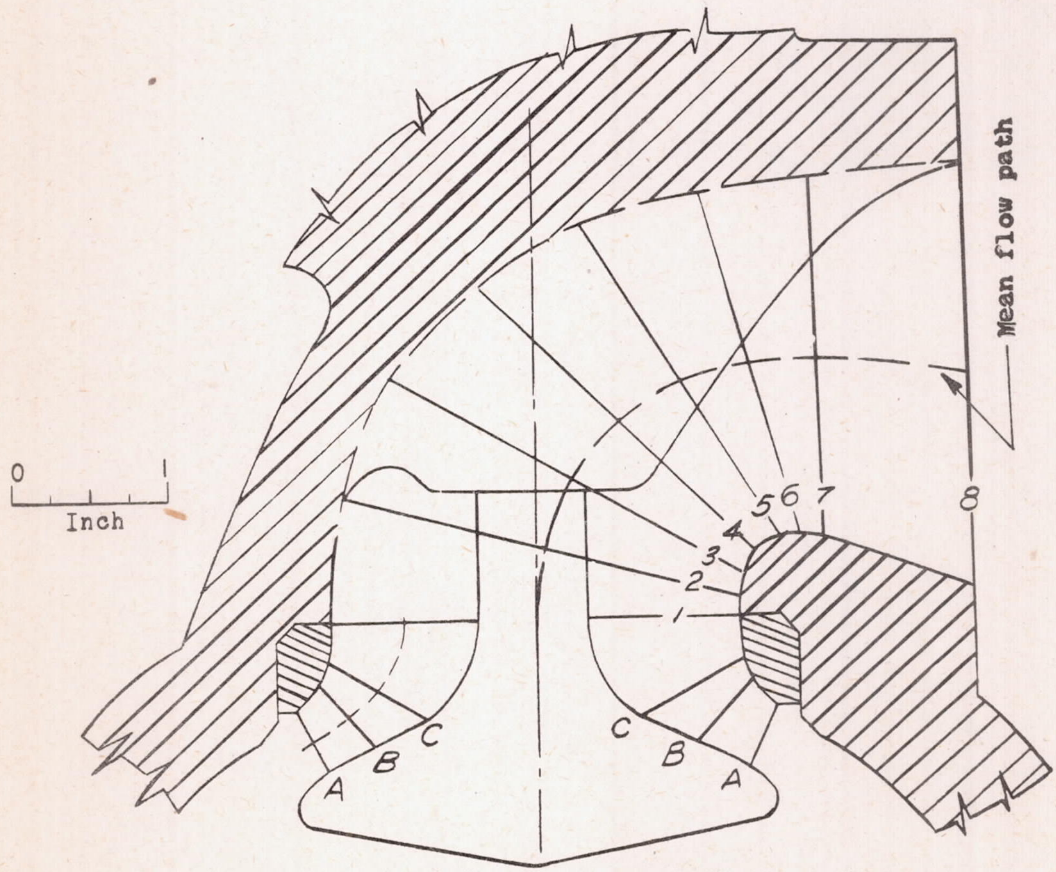

Valve in full-open position

Figure 3.- Locations of exhaust-port sections in standard Wright C9GC cylinder, with valve in full-open position.

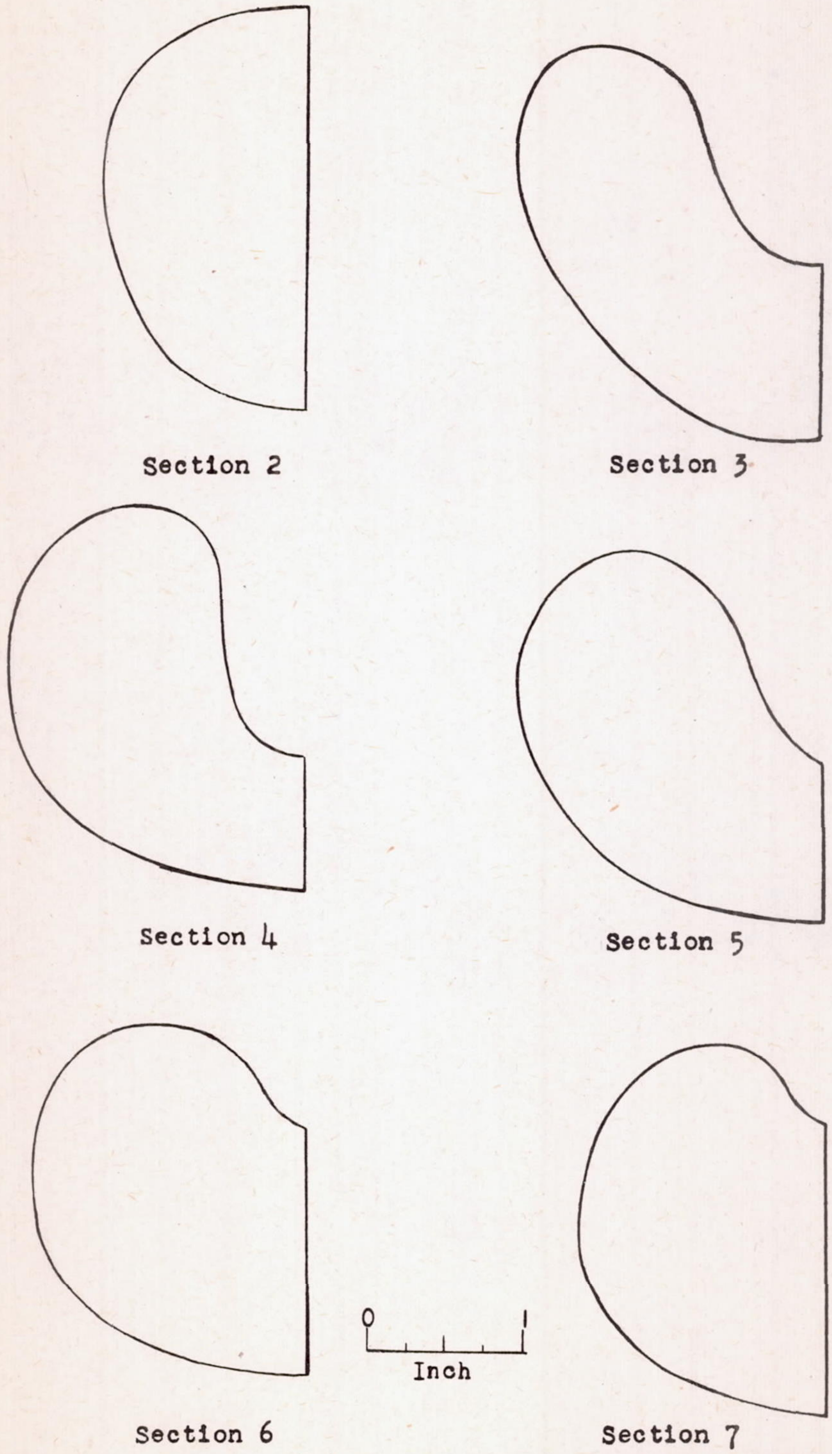

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Figure 4.- Half-sections of standard exhaust port in a Wright C9GC cylinder.

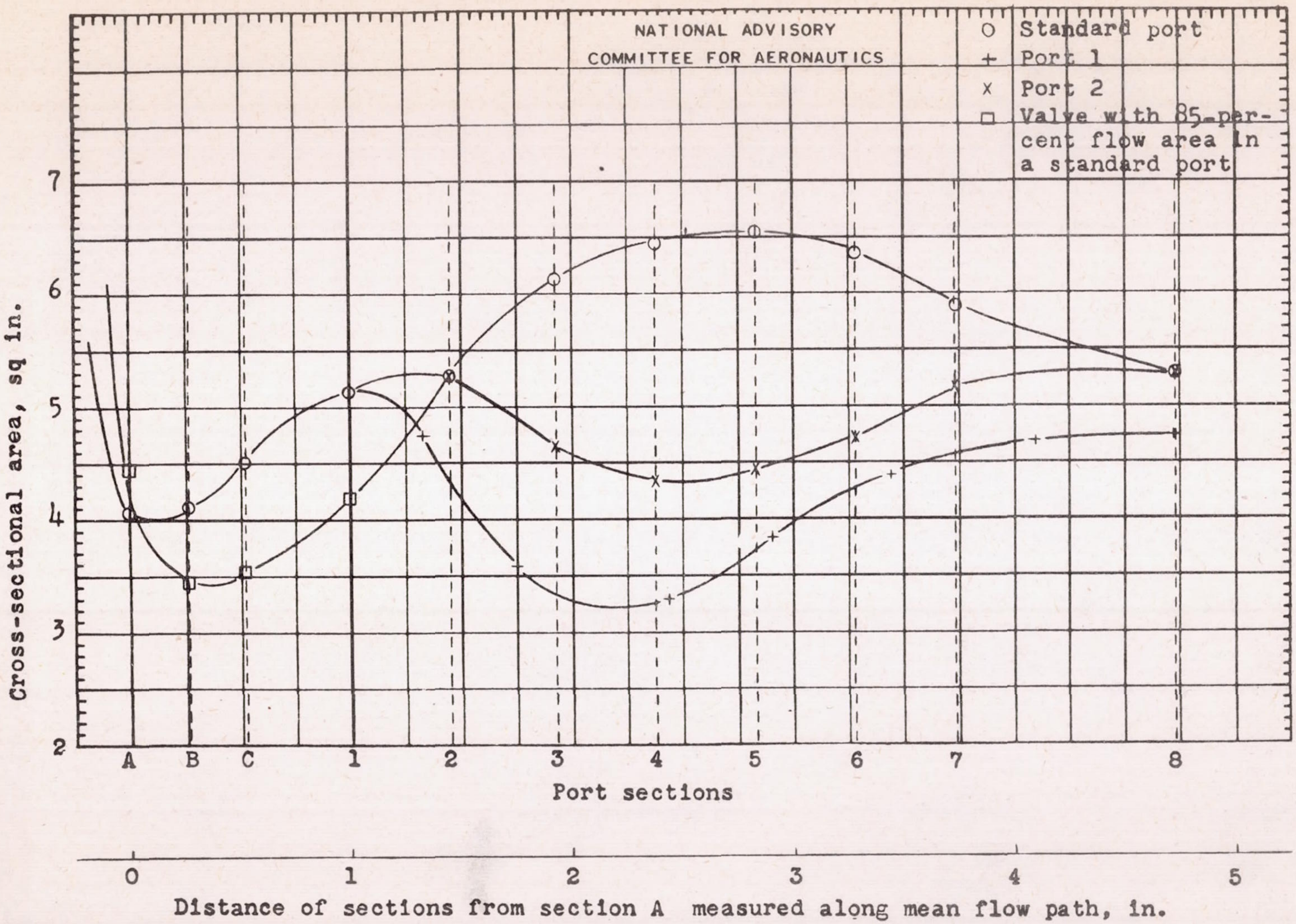


Figure 5.- Cross-sectional areas of the standard and of the modified exhaust ports in a Wright C9GC cylinder.

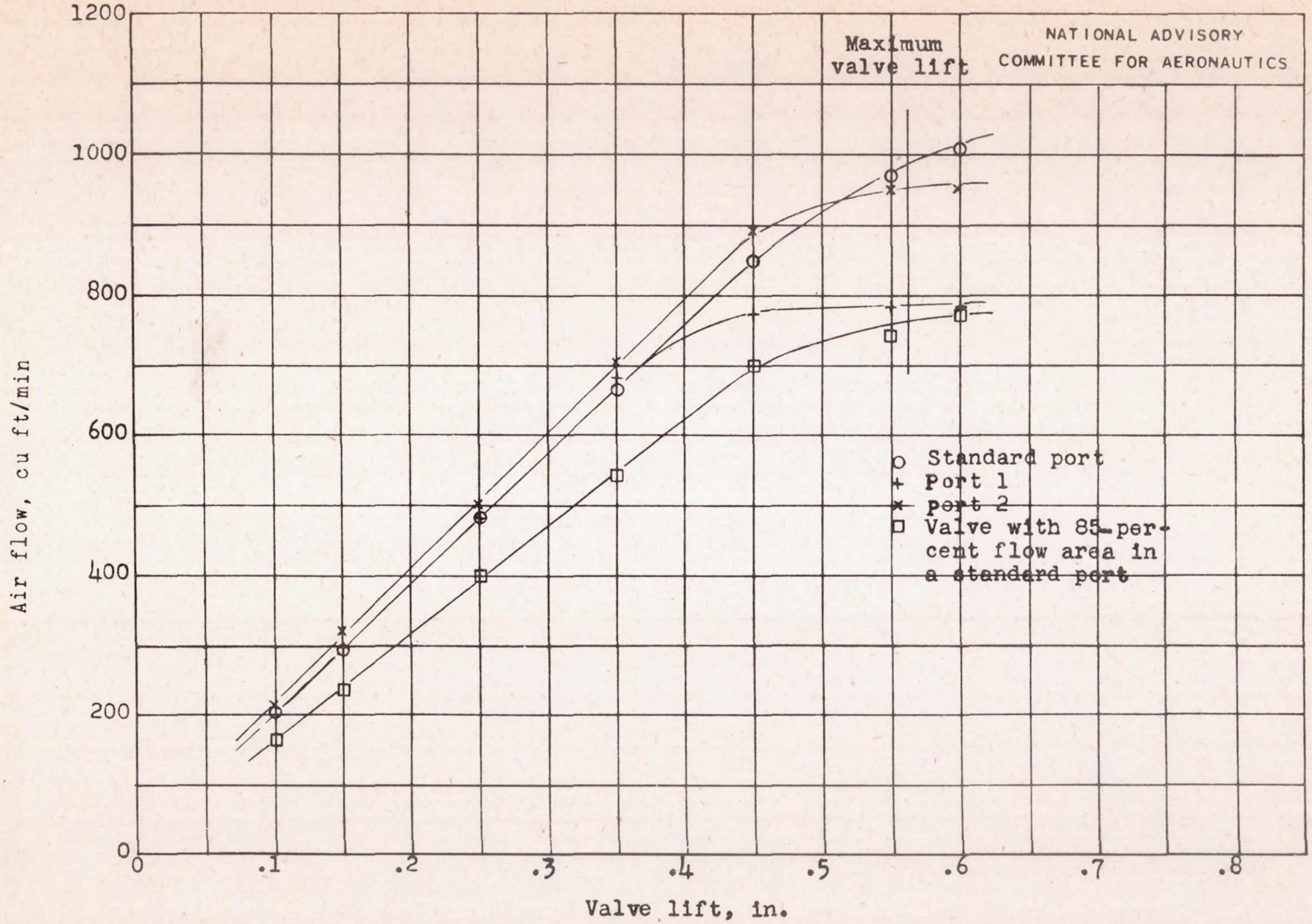


Figure 6. - Comparison of air flow through standard and modified exhaust ports in a Wright C9GC cylinder. Pressure drop through ports, 10 inches of mercury absolute. Air flow corrected to standard conditions (pressure, 29.92 inches of mercury absolute; temperature, 59°F).

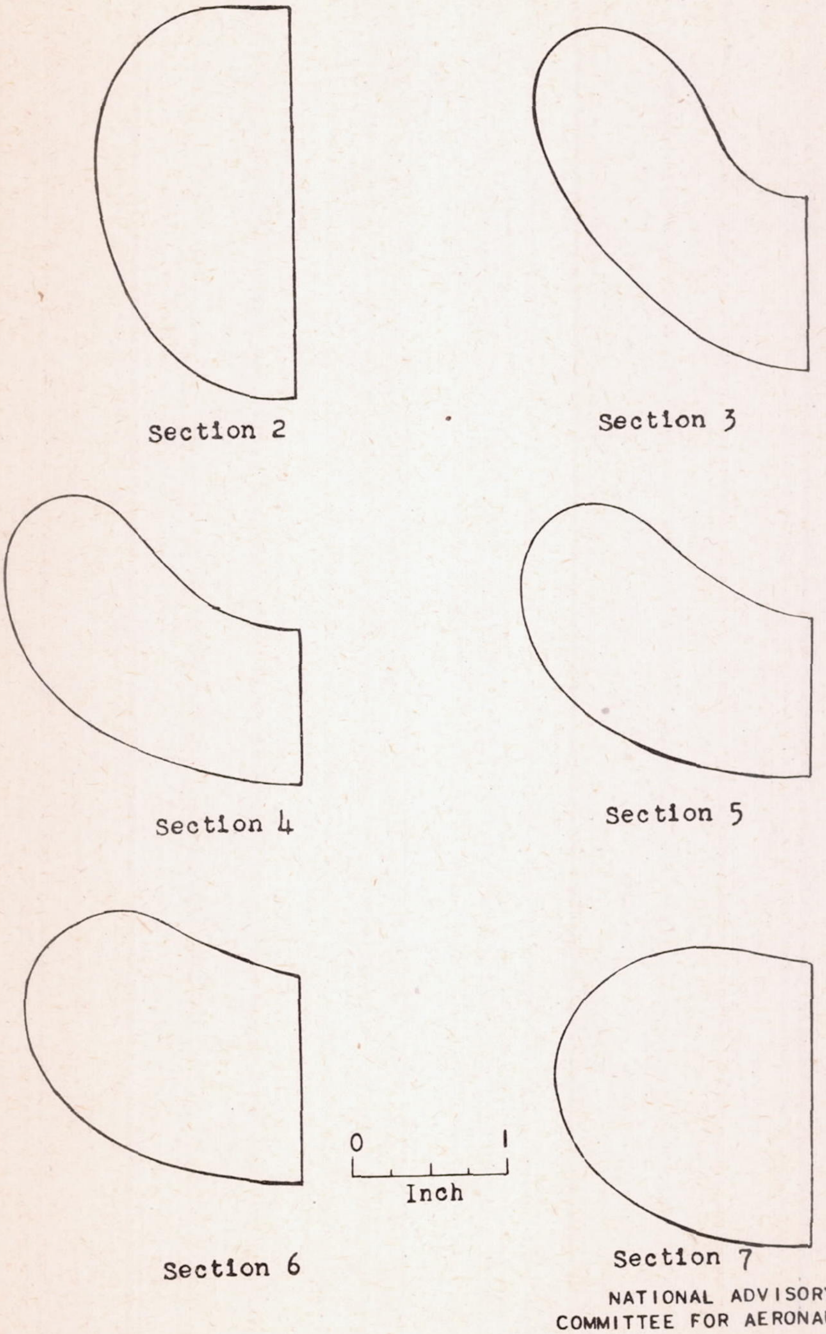


Figure 7. - Half-sections of port 2.

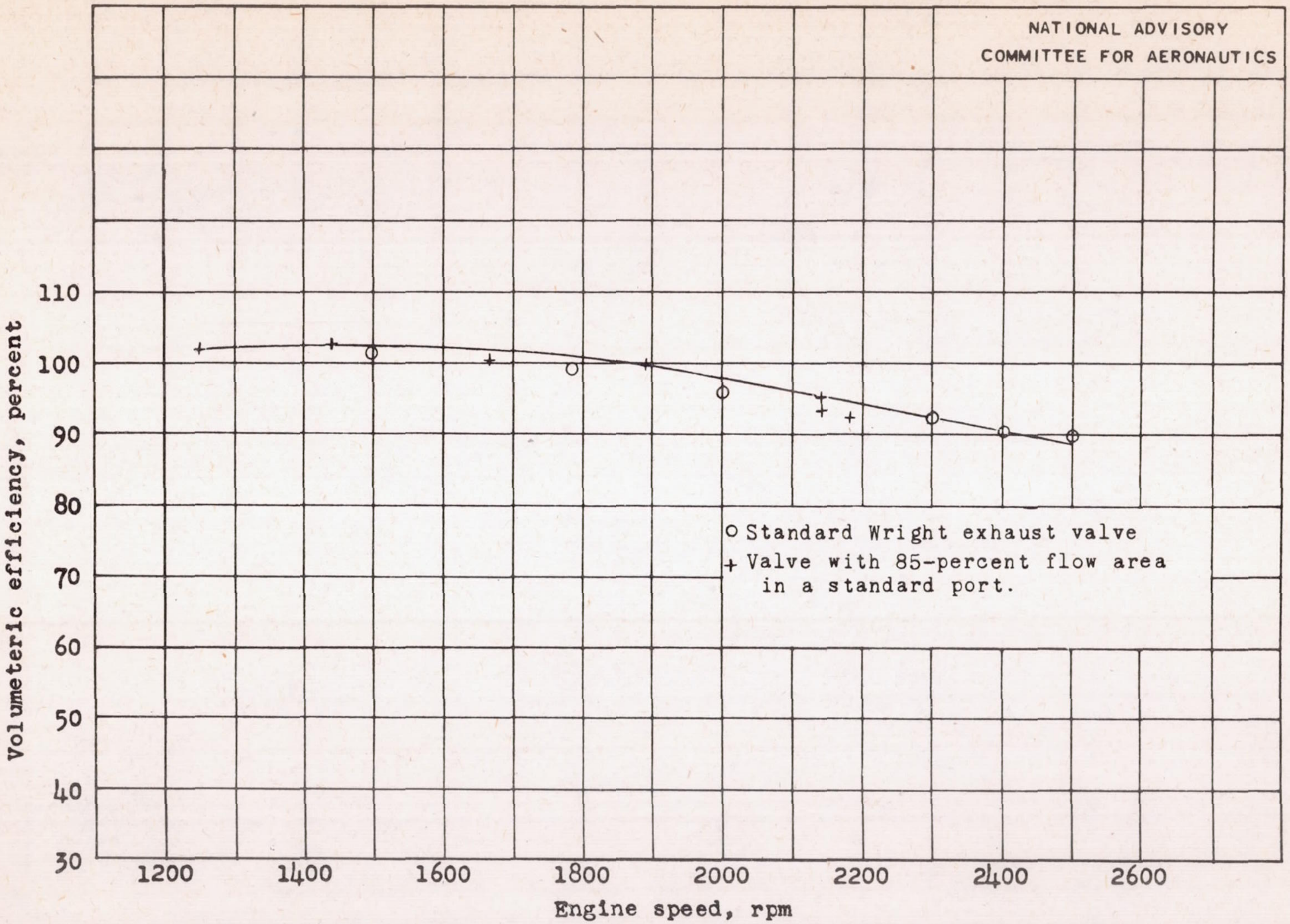
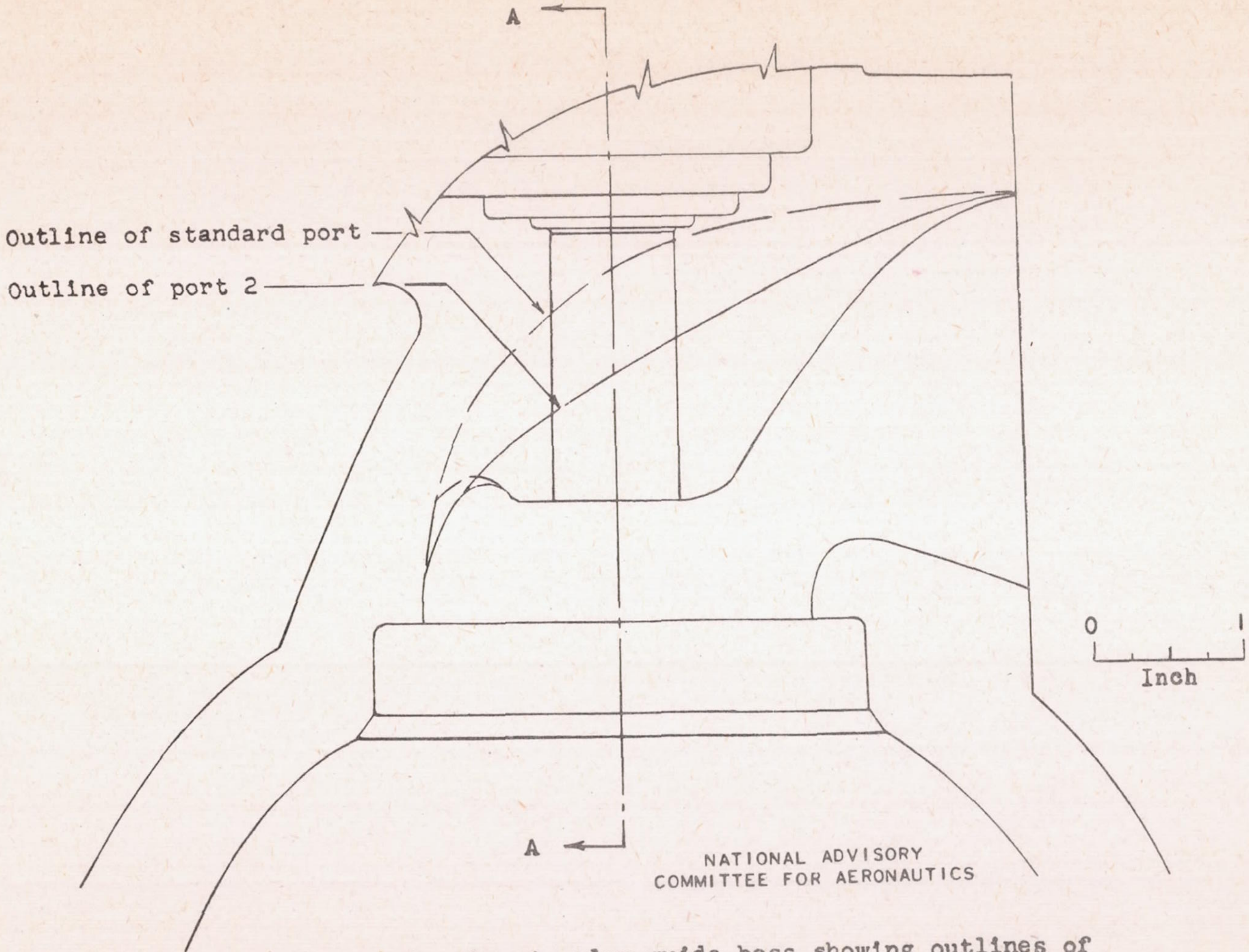


Figure 8. - Comparison of the volumetric efficiencies of a Wright C9GC cylinder with a standard and an 85-percent flow-area exhaust valve. Intake manifold pressure, 40 inches of mercury absolute; combustion-air temperature, 150° F; exhaust pressure, 28.2 inches of mercury absolute; fuel-air ratio, 0.097.



(a) Cross section through exhaust-valve-guide boss showing outlines of standard port and port 2.

Figure 9.- Comparison of size of exhaust-valve-guide boss in standard port and in port 2 of Wright C9GC cylinder.

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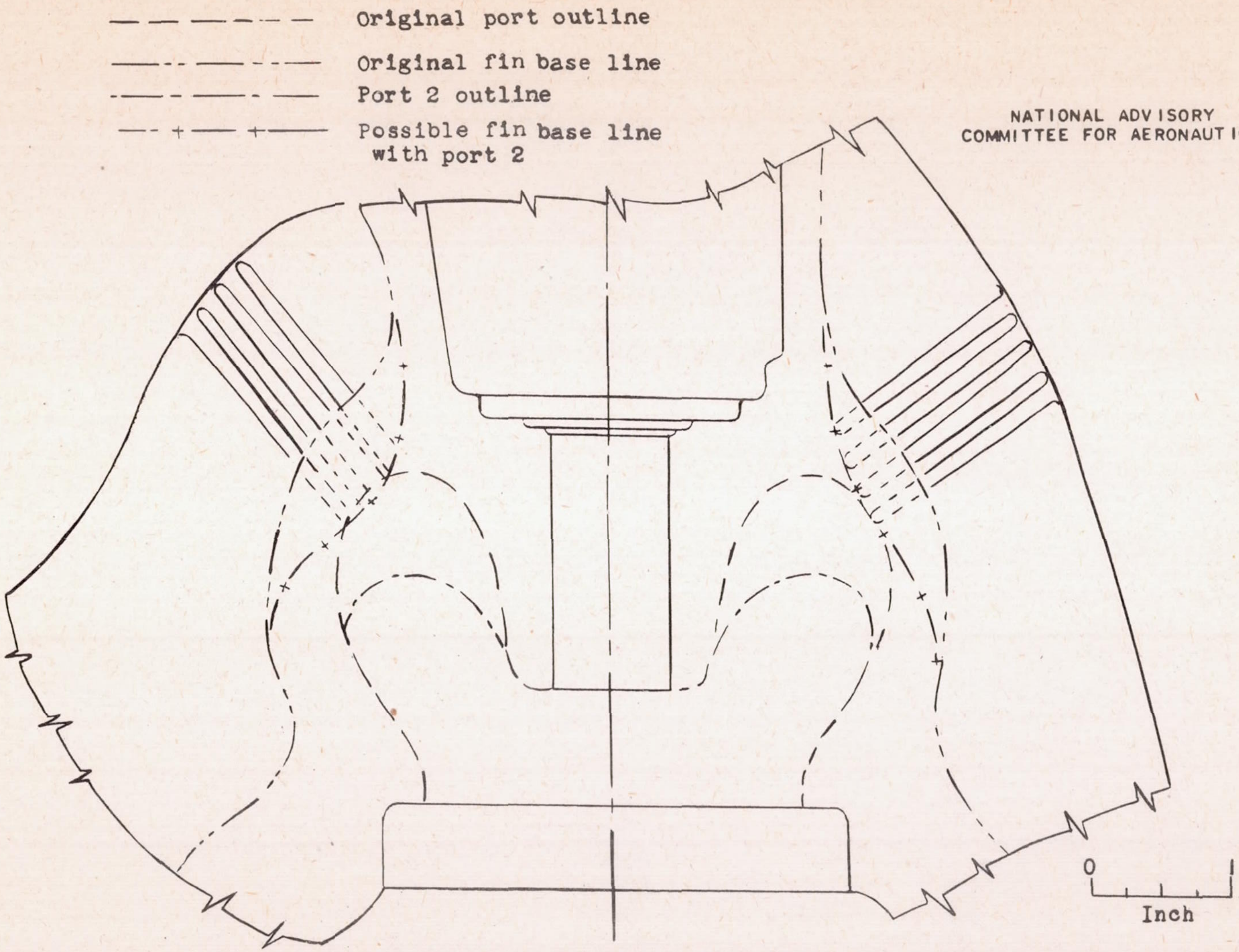

(b) Section A-A showing built-up exhaust-valve-guide boss in Wright C9GC cylinder.
Figure 9.- Concluded.

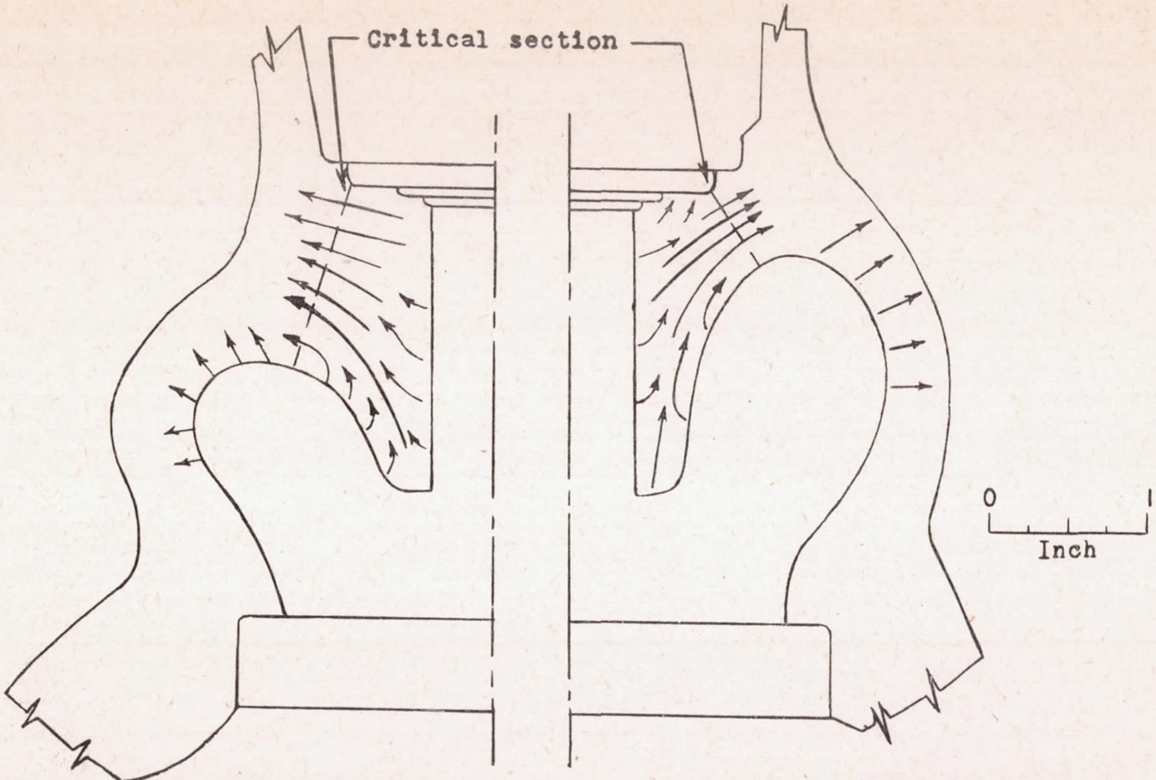

Figure 10.- Comparative heat-flow paths of port 2 and standard port at section A-A (See fig. 9(b).) Wright C9GC cylinder.

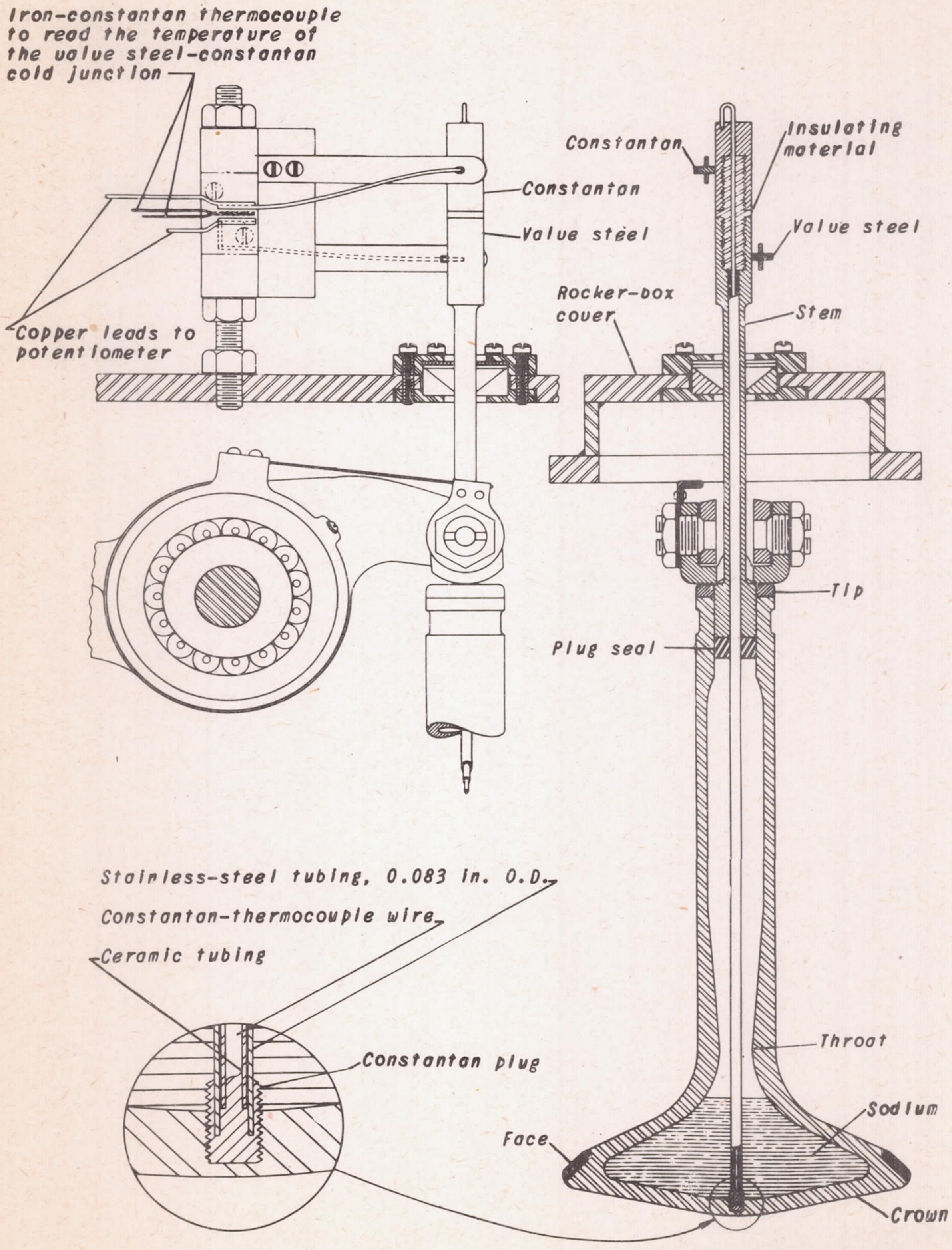

Figure 11. - Details of exhaust valve equipped with a thermocouple.

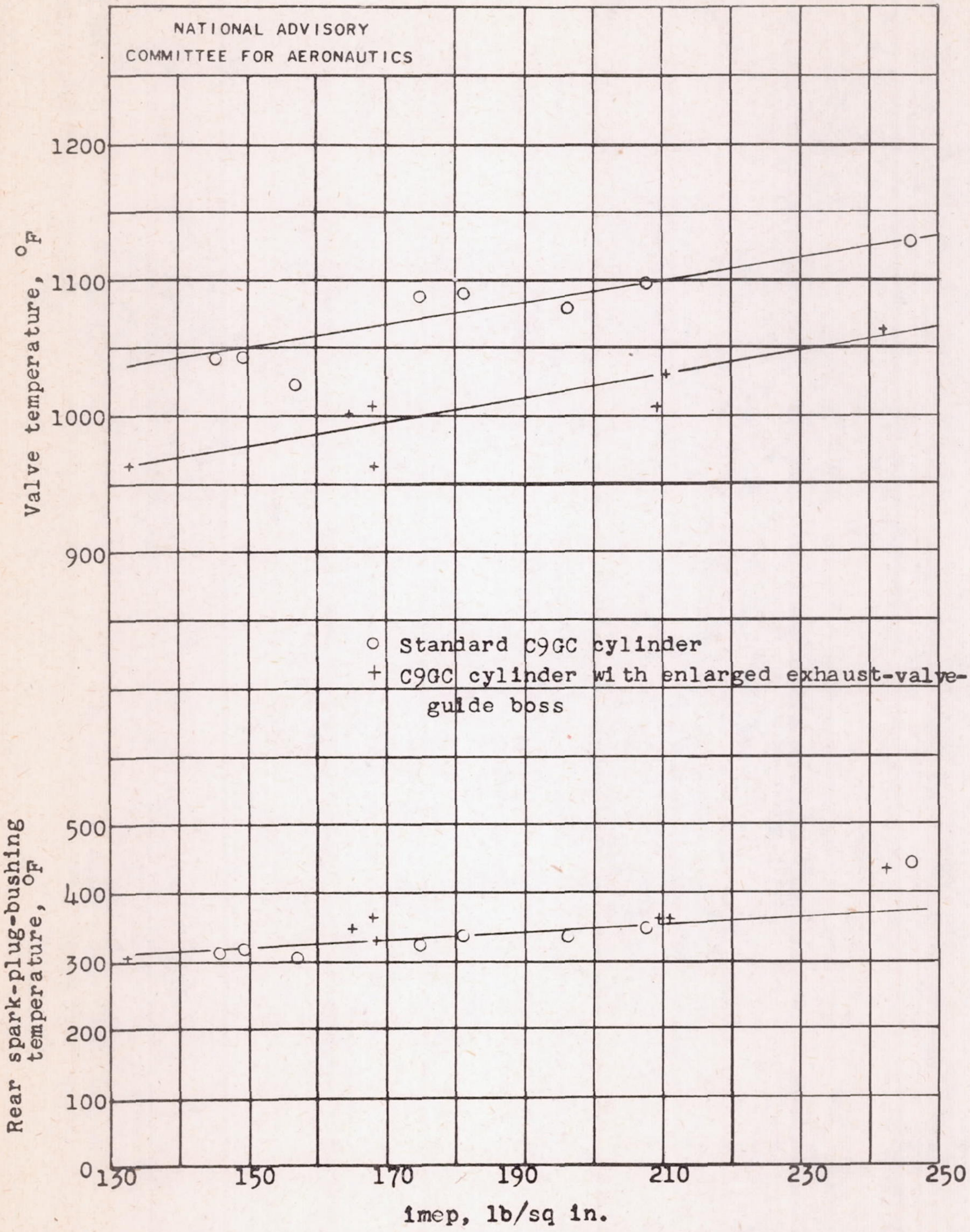

Figure 12. - Comparison of exhaust-valve temperature in a Wright C9GC cylinder and a C9GC cylinder with an enlarged exhaust-valve-guide boss. Engine speed, 2200 rpm; cooling-air pressure drop, 16 inches of water; spark advance, $22\frac{1}{2}^{\circ}$ B.T.C.; fuel-air ratio, 0.099; combustion-air temperature, 150° F; cooling-air inlet temperature, 72° F; exhaust back pressure, 28.6 inches of mercury absolute.

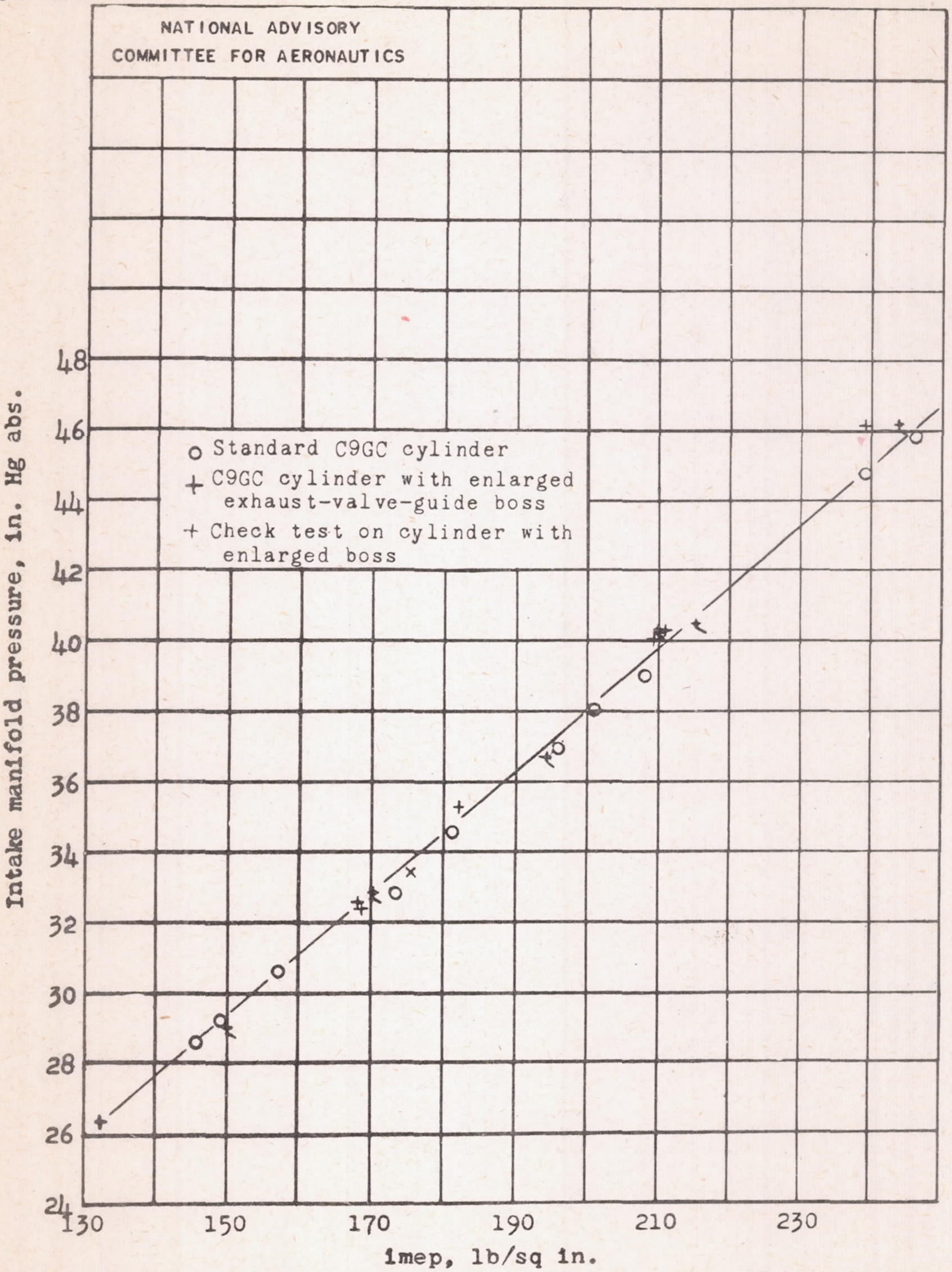

Figure 13.- Effect of intake-manifold pressure on indicated mean effective pressure for a standard C9GC cylinder and a C9GC cylinder with an enlarged exhaust-valve-guide boss. Engine speed, 2200 rpm; spark advance, $22\frac{1}{2}^{\circ}$ B.T.C.; fuel-air ratio, 0.099; cooling-air pressure drop, 16 inches of water; combustion-air temperature, 150° F. Cooling-air inlet temperature, 72° F; exhaust back pressure, 28.6 inches of mercury, absolute.