TECHNICAL NOTES

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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THE EFFECT OF VALVE COOLING UPON MAXIMUM PERMISSIBLE ENGINE OUTPUT AS LIMITED BY KNOCK

By Maurice Munger, H. D. Wilsted,
and B. A. Mulcahy

Langley Memorial Aeronautical Laboratory
Langley Field, Va.

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SUMMARY

A Wright GR-1820-G200 cylinder was tested over a wide range of fuel-air ratios at maximum permissible power output as limited by knock with three different degrees of valve cooling. The valves used were stock valves (solid inlet valve and hollow sodium-cooled exhaust valve), hollow valves with no coolant, and hollow valves with flowing water as a coolant. Curves showing the variation in maximum permissible values of inlet-air pressure, indicated mean effective pressure, cylinder charge, and indicated specific fuel consumption with change in fuel-air ratio and valve cooling are shown. The use of valves cooled by a stream of water passing through their hollow interiors permitted indicated mean effective pressures 10 percent higher than the mean effective pressures permissible with stock valves when the engine was operated with fuel-air ratios from 0.055 to 0.065. Operation of the engine with lean mixtures with uncooled hollow valves resulted in power output below the output obtained with the stock valves. The data show an increase in maximum permissible indicated mean effective pressure due to cooling the valves, which averages only 2.1 percent with fuel-air ratios from 0.075 to 0.105.

INTRODUCTION

The temperatures of the surfaces of the combustion chamber of an internal combustion engine influence the condition of the compressed charge and therefore affect the combustion process and the engine performance. Rothrock and Biermann (reference 1, p. 2) state: "Pre-ignition is generally surface ignition at some hot spot in the combustion chamber. . . . The early start of combustion may also be accompanied by knock although this is not necessarily the case." Valve temperatures have been found by Colwell (reference 2) to be as high as 1300° F for the exhaust and 815° F for the intake. It is indicated that increased power output should be possible as a result of reductions in the operating temperatures of poppet valves.

An investigation to determine the benefits to be derived from increased cooling of the interior surfaces of engine combustion chambers is being carried out at the Langley Memorial Aeronautical Laboratory of the NACA.

As a part of this program, tests to determine the maximum permissible engine output as limited by knock were conducted with different degrees of cooling of the valves in an air-cooled engine cylinder. In one series of tests, the usual stock valves, a sodium-cooled exhaust valve,and a solid metal intake valve, were employed. In a second series of tests, hollow valves were cooled by means of a stream of water circulated through them. This method was adopted for research purposes and not with the idea that it could conveniently be used in service. In a third series of tests, the same hollow valves were operated without cooling.

This work was done from July to October 1941.

APPARATUS

In order to measure quantitatively the effect upon maximum engine performance of different degrees of cooling of both exhaust and intake valves, a single-cylinder test unit was set up. Figure 1 shows the arrangement with a Wright GR-1820-G200 cylinder. The bore of this cylinder is $6\frac{1}{8}$ inches. A stroke of 7 inches was used instead of a stroke of $6\frac{7}{8}$ inches as in the multicylinder engine because the crankshaft available had a $3\frac{1}{2}$-inch crank radius. With this stroke, the piston displacement was 206 cubic inches.

A series of tests was run with stock valves in both the intake and the exhaust positions. The exhaust valve was made with hollow head and stem and was sodium-cooled. The intake valve was of the tulip type, made of solid metal. The compression ratio with these stock valves was 6.75.

In a second series of tests, both intake and exhaust valves were internally cooled by a stream of water flowing through them. These water-cooled valves were made by removing the sodium from two stock exhaust valves through a hole in the stem tip and silver-soldering a stainless-steel tube into the open end. A water-supply tube extended (with valve closed) to within 1/8 inch of the inner surface of the valve head. This water-supply tube was stationary and was supported by the rocker-box cover. Suitable enclosures and packings served to direct the flow of the cooling water. Figure 2 shows the apparatus mounted upon the rocker-box covers. Substitution of a valve of exhaust form for the stock intake valve caused a change in the compression ratio to 6.96.

Operation of the valves with a central tube projecting from the stem tip was accomplished by the arrangement shown in figures 3 and 4. The yoke tappet, which was substituted for the usual roller, made it possible for the water tubes to pass through the fork of the rocker-arm and outside the rocker-box cover.

The water-supply lines to the exhaust valve and to the intake valve were each provided with a pressure gage and a control valve. The temperature of the water to and from the valves was determined by liquid-in-glass thermometers. The water discharged was collected and weighed on a small platform scale.

In a third series of tests, the water-supply system was removed from the valves used in the second series of tests and the engine was operated with no cooling of the valves. Removal of the water-supply tubes made it possible to see directly into the interior of the hollow valves and to observe the heat color of the valve head.

The air supply for the engine was taken from a laboratory main. The pressure of the air ahead of a measuring orifice was controlled by an adjustable reducing valve. From the measuring orifice, the air passed into a surge tank 24 inches in diameter and 70 inches long, having a volume of 18.3 cubic feet. The pressure drop across the measuring orifice was determined by a water manometer. The static pressures at the orifice and in the surge tank were determined by mercury manometers. Air temperatures were determined by liquid-in-glass thermometers.

The venturi passage and the fuel jets of a Bendix Stromberg model NA-R9 carburetor were used. The carburetor was connected to the surge tank by $25\frac{1}{2}$ inches of pipe with an inside diameter of $4\frac{1}{2}$ inches. A pipe 20 inches long with an inside diameter of $2\frac{1}{2}$ inches extended from the carburetor outlet to the engine port. The differential pressure between the fuel and the air was controlled by adjustment of a bypass valve in the fuel line so that the fuel pressure was kept at a value ranging from 3 to 12 pounds per square inch above the air pressure, depending upon the quantity of fuel required. The quantity of fuel supplied to the engine was controlled by a needle valve at the carburetor. For convenience in adjusting the rate of fuel flow for any particular fuel-air ratio, a rotameter was placed in the fuel line. Final determination of fuel quantities was made by means of a weighing device that measured the time required for the engine to use one-half pound of fuel.

Cooling air for the engine was supplied by a single-stage centrifugal blower. A metal cowling surrounding the engine cylinder directed the flow of cooling air. Temperatures of the cooling air and of the cylinder head and barrel at a number of locations were determined by means of iron-constantan thermocouples.

The ignition system consisted of a single pair of breaker points that interrupted the current in the primary windings of two standard 6-volt ignition coils connected in parallel. Two BG 298-GS spark plugs were thus supplied with simultaneous sparks. The cam for operating the breaker points was driven directly from the crankshaft to insure uniform rotation. The coils were energized every engine revolution; but, as the extra spark occurred 22° B.T.C. on the exhaust stroke, it had no effect upon engine operation.

The temperature of the exhaust gas was measured by means of an unshielded chromel-alumel thermocouple located $2\frac{1}{2}$ inches from the engine port in the center of the exhaust pipe.

METHOD

It was considered that the best basis on which to judge the effect upon engine output of the several valve-cooling conditions was to operate the engine with the maximum permissible inlet pressure, as limited by knock, and to vary the fuel-air ratio throughout the range in which engine operation was possible.

The engine was operated at a constant speed of 2000 rpm. The speed of the blower, which provided air for cooling the cylinder, was varied to maintain the cylinder-barrel material in the middle fin space at the rear at a constant temperature of 360° F. This temperature was measured by a thermocouple imbedded in the metal of the barrel. The oil-out temperature from the crankcase was maintained at 140° F to 150° F.

The method of engine operation employed was as follows: With constant fuel-air ratio, the inlet-air pressure, measured in the surge tank, was increased until steady audible knock became evident. A record was made of the rates of flow of fuel as shown by the rotameter and of air as shown by the differential and static pressures at the orifice. The pressure in the surge tank was also recorded. The severity of operation was then lessened by making a 7-percent reduction in both absolute inlet-air pressure and fuel flow as shown by the rotameter. Complete data were recorded under these conditions of operation, which were considered to be of maximum permissible severity. This method of engine operation is the same as that used at the Langley Memorial Aeronautical Laboratory for fuel-rating tests.

Preignition did not occur in any of the tests. This fact was determined by turning off the ignition switches at the completion of each test run and observing that firing ceased at once.

During the tests with the water-cooled valves, the water flowing from each valve in two minutes was collected separately and weighed. The temperature of the water to and from the valves was recorded. With an inlet temperature of 80° F to 90° F, approximately five pounds of water per minute to each valve was found to be sufficient to keep the temperature of the water flowing from the exhaust valve below 140° F.

Tests were made throughout the whole range of fuel-air ratio in which engine operation was possible. This range included the values from 0.051 in the lean region to 0.125 in the rich region. The spark timing was kept at 22° B.T.C. for all tests. This timing was chosen because it was found in preliminary tests to be the timing with which the engine developed maximum power. All the tests reported were made with 100-octane fuel, conforming to Army-Navy Fuel Specification No. AN-VV-F-781, from the same filling of a storage tank.

PRECISION

Determination of the inlet-air pressure that would barely produce distinct audible knock was not easy. With practice, however, it was possible to make check runs in which the 93-percent inlet pressure at any particular fuel-air ratio within the range of fuel-air ratios between 0.060 and 0.110, for which engine operation was stable, would usually agree within $\pm\frac{1}{2}$ inch of mercury.

The accuracy of the inlet-air measurement was checked by analyzing the exhaust gas with an Orsat apparatus. If the fuel-air ratio determined by the Orsat analysis is assumed to have been correct, the mean percentage difference in the quantity of air as measured by the orifice plate was found to be -1.2 percent and the greatest difference from this mean value was ±2.5 percent.

The weight of fuel delivered by the fuel-weighing device was found to be correct within 0.7 percent.

Thermocouple installations of the type used on the head and on the barrel of the cylinder are believed to make possible temperature determinations within ±5° F of the true value. The probable error in successive readings is less than ±2° F.

RESULTS AND DISCUSSION

Curves showing the relation between maximum permissible inlet-air pressure (93 percent of that producing distinct audible knock) and fuel-air ratio with the three valve-cooling conditions are shown in figure 5. For fuel-air ratios greater than 0.069, the difference between the maximum permissible inlet pressures for the stock valves and for the water-cooled valves is negligible. With leaner mixtures the permissible pressure for the cooled valves is higher than that for the stock valves. This difference is approximately 4 inches of mercury with a fuel-air ratio of 0.059. With the uncooled valves, a decrease in the permissible inlet pressure below the pressure found with the stock valves occurs at all fuel-air ratios except in the very lean region. Very little decrease occurs in the rich region, but at a fuel-air ratio of 0.066 the permissible inlet pressure with the uncooled valves has a minimum value of 34.8 inches of mercury. The minimum value with the stock valves, which occurs at the same fuel-air ratio, is 38.5 inches of mercury, and for the water-cooled valves the minimum is 39.9 inches of mercury at a fuel-air ratio of 0.070.

In order to install a water-cooled valve in the intake, a hollow exhaust valve, with a stem diameter of 0.6825 inch and a domed top, was substituted for the intake valve, which had a stem diameter of 0.560 inch and a tulip top. For this reason, the maximum permissible inlet pressure for these tests shows slightly higher values for equal cylinder charges than for the tests with stock valves. Any tendency toward an increase in volumetric efficiency during the tests with cooled valves because of reduction in valve temperature was more than overcome by the reduction in flow caused by the use of the exhaust valve in the intake position. This point is illustrated by figure 6, which shows the relation between cylinder charge and maximum permissible inlet-air pressure. The maximum permissible indicated mean effective pressure and the maximum permissible cylinder charge, however, should be independent of any effect due to the change in valve form.

A comparison of the cylinder charge when the engine was operating with cooled and with uncooled valves is also shown by figure 6. For these tests the only different condition was in the temperature of the valves. No attempt has been made to correlate the results with actual valve temperatures as sufficiently accurate information is lacking. Observation of the inlet valve during the tests with the uncooled valves showed a color indicating a probable maximum temperature of 1200° F at a fuel-air ratio of about 0.070. Figure 6 shows an increase in the cylinder charge of about 2 percent when the intake valve was cooled to a temperature close to 200° F as compared with the charge when the intake valve was uncooled.

Figure 7 shows the changes in indicated mean effective pressure for the three valve conditions. For fuel-air ratios between 0.075 and 0.105 the permissible indicated mean effective pressure with the cooled valves is higher than that with the stock valves by an average of 5 pounds per square inch, which is approximately a 2.1 percent increase.

In the lean region, cooling the valves makes it possible to maintain the indicated mean effective pressure with very little of the decrease observed with the stock valves. For fuel-air ratios between 0.055 and 0.065, the indicated mean effective pressure with the cooled valves is higher than that with the stock valves by an average of 21 pounds per square inch, or 10 percent.

With the uncooled valves there is a decrease in the permissible indicated mean effective pressure below the value obtained with the stock valves at all fuel-air ratios. For mixtures richer than 0.080 fuel-air ratio, the decrease is about 8 pounds per square inch. In the lean region, the drop is considerably greater and for fuel-air ratios from 0.060 to 0.075 it averages 26 pounds per square inch, which is 12 percent.

From data presented by Rothrock and Biermann (reference 3) it appears that an increase in the compression ratio of the magnitude caused by the change in valves would lower the permissible indicated mean effective pressure for the stock valves by approximately 4.6 percent and the permissible inlet-air pressure by about 3.8 percent for a fuel-air ratio of 0.076. For this reason, the total benefit due to cooling the valves would be slightly greater than that determined by results of the present tests. No difference would result in the reported comparison for the cooled and uncooled valves.

Figure 8 shows the maximum permissible cylinder charge with the three valve conditions. With the cooled valves, there is an average increase in the charge above the charge permissible with the stock valves for fuel-air ratios between 0.055 and 0.065 of 10 percent. This result agrees with the increase in indicated mean effective pressure for the same test conditions.

With the uncooled valves the permissible cylinder charge is reduced to a value below that obtained with the stock valves at all fuel-air ratios. For the fuel-air ratios ranging from 0.060 to 0.075, this reduction averages 15 percent.

Figure 9 shows the indicated specific fuel consumption for the several conditions of the tests. The minimum consumption is 0.36 pound at 0.060 fuel-air ratio.

The heat removed from the valves by the cooling water is shown by figure 10. With a fuel-air ratio of 0.065 at which, as shown by figure 11, the exhaust-gas temperature reaches a maximum, the heat removed from the valves was 214 Btu per minute for the exhaust valve and 75 Btu per minute for the intake valve.

As a check on the operating condition of the water-cooled exhaust valve, removable plugs were installed in the exhaust pipes of the test engine and of another engine with the same type of cylinder equipped with a sodium-cooled exhaust valve. When these plugs were removed, a direct view of the exhaust valves was possible. The test engine was run at a fuel-air ratio of 0.068 and an indicated mean effective pressure of 229.7 pounds per square inch. The water-cooled exhaust valve showed no color whatever due to heat.

The other engine was run at the same fuel-air ratio as the test engine. The indicated mean effective pressure, however, was only 190 pounds per square inch. Under these conditions of operation the sodium-cooled valve showed a distinctly red color. Comparison with a heat color chart published by the Bethlehem Steel Company (reference 4) would indicate that the valve temperature was between 1300° F and 1400° F. Because the water-cooled exhaust valve was not at a temperature higher than 200° F, the reduction in temperature due to the water cooling was in excess of 1000° F.

As a further check upon the contrast between the temperature of the cooled and of the stock valves, observations of the color of the sodium-cooled exhaust valve were made with the engine operating at 0.106 fuel-air ratio and 229.9 pounds per square inch indicated mean effective pressure. Under these operating conditions, with a higher power output than when the first observations were made but with a probable reduction in exhaust-gas temperature due to the richer mixture, the valve showed no visible heat color. The absence of heat color would indicate a temperature below 1000° F.

The power with the stock valves showed a decrease with lean mixtures when the exhaust-gas temperature reached maximum values and the sodium-cooled exhaust valve showed a bright temperature color. With richer mixtures, lower exhaust-gas temperatures, and the valve showing no heat color, the power with cooled and with stock valves was substantially the same.

It is possible that the increase in power in the lean region with the water-cooled valves would occur even without cooling the valves to the same extent as was done with the water-cooling. Some means of producing controlled intermediate degrees of cooling would have to be devised to answer this question.

Engine temperatures recorded during the tests are shown on figures 12 and 13. Valve-seat temperatures were taken by thermocouples imbedded in the head material close to the seat inserts. The guide-bushing thermocouples were in the metal of the guides themselves.

The maximum head temperature with the stock valves shows a few points in the rich-mixture region that lie below the curve for the water-cooled valves. With these exceptions, the temperatures with the water-cooled valves are lower than those with the stock valves. The greatest reduction is in the temperature of the exhaust-valve guide bushing. This reduction averages 230° F.

The effectiveness of the sodium in carrying heat from the head of the valve to the guide is shown in figure 12 by the reduction in temperature of the exhaust-valve guide bushing during the tests with the uncooled valves. This temperature reduction averaged 95° F over the range of fuel-air ratio.

Figure 11 shows the exhaust-gas temperatures for the three valve-cooling conditions. The curves for the cooled valves and for the stock valves on this figure are almost identical. With the uncooled valves, however, there is shown a decrease in exhaust-gas temperature at all fuel-air ratios. At 0.065 fuel-air ratio this decrease is from 1585° F to 1445° F.

CONCLUSIONS

1. Cooling the valves in an air-cooled engine cylinder by circulating water through their hollow interiors

made possible an average increase of only 2.1 percent in the maximum permissible indicated mean effective pressure with fuel-air ratios from 0.075 to 0.105; with fuel-air ratios from 0.055 to 0.065 this increase averaged 10 percent.

2. The data show an increase of approximately 4 inches of mercury in the highest permissible inlet-air pressure with a fuel-air ratio of 0.059 when the valves were cooled.

3. Cooling the valves did not affect the minimum specific fuel consumption.

4. Engine operation with fuel-air ratios from 0.060 to 0.075 with the valve-cooling water shut off resulted in decreases of 12 and 15 percent, respectively, in the average permissible indicated mean effective pressure and cylinder charge as compared with the stock-valve results.

5. Comparison of the cylinder charge, at various inlet pressures, for the cooled and for the uncooled valves, shows an increase in the charge with the cooled valves of about 2 percent.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va.

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2. Colwell, A. T.: Modern Aircraft Valves. SAE Jour., vol. 46, no. 4, April 1940, pp. 147-165.

3. Rothrock, A. M., and Biermann, Arnold E.: Correlation of Knocking Characteristics of Fuels in an Engine Having a Hemispherical Combustion Chamber. T.N. No. 768, NACA, 1940.

4. Anon.: Bethlehem Alloy and Special Steels, Catalog-107. Bethlehem Steel Co., 1936.

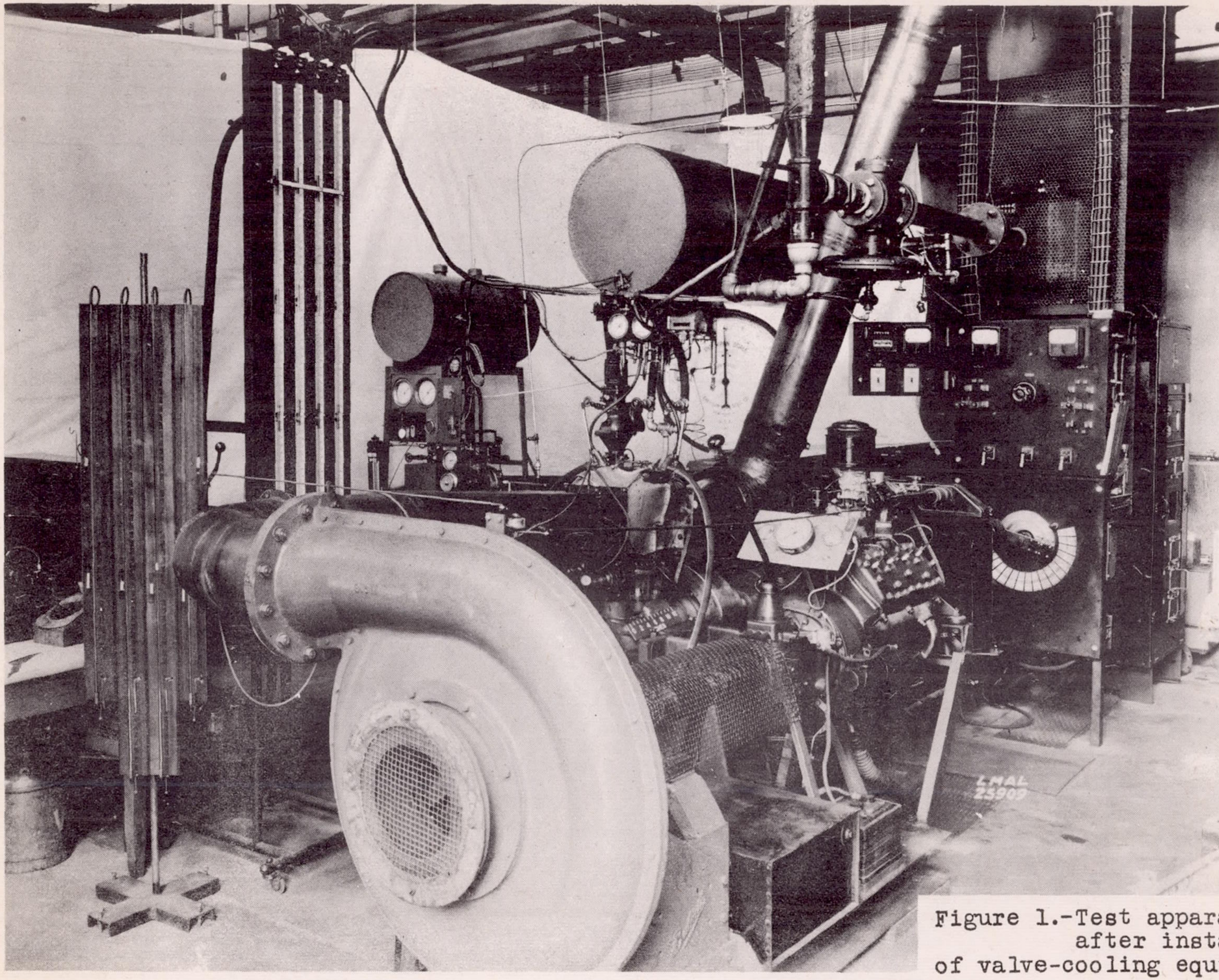


Figure 1.-Test apparatus after installation of valve-cooling equipment.

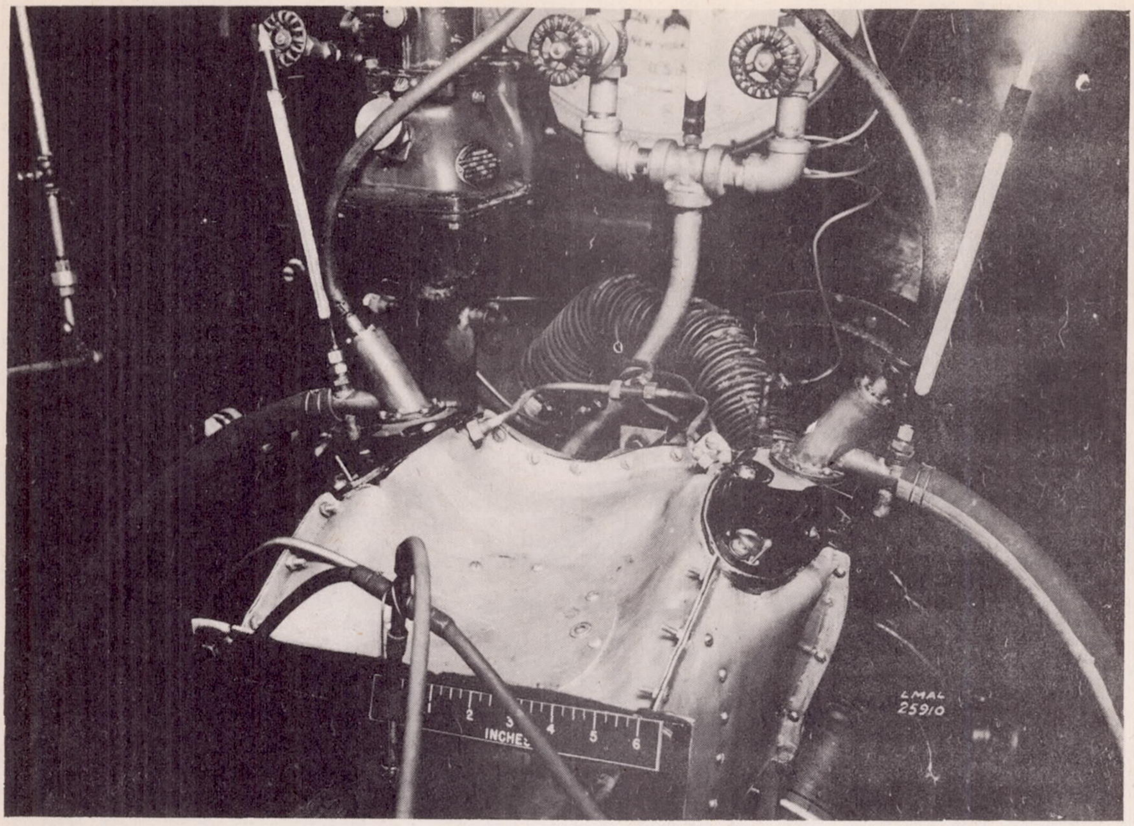


Figure 2.-Water-supply system for valve-cooling.

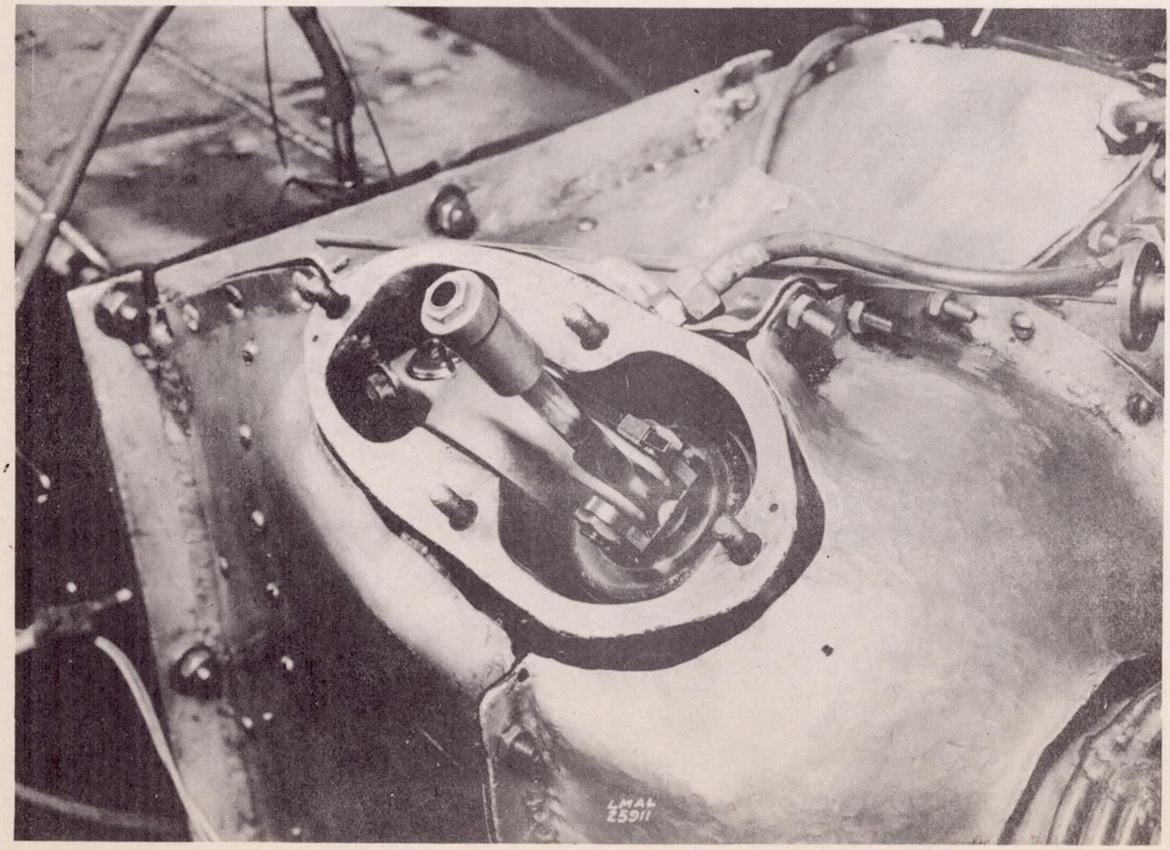


Figure 3.-Alterations to valve and valve-operating mechanism to provide cooling-water supply.

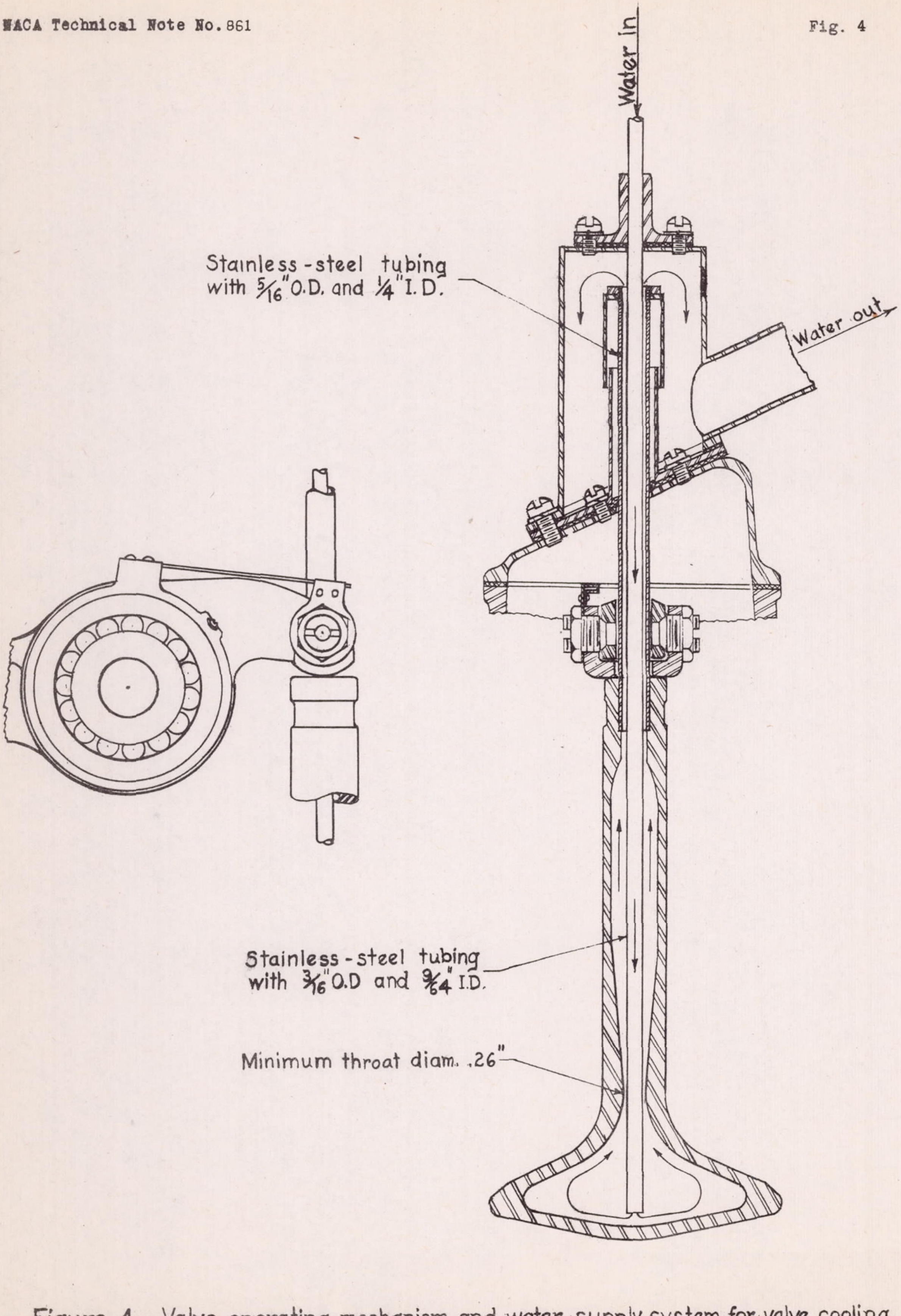


Figure 4.- Valve-operating mechanism and water-supply system for valve cooling.

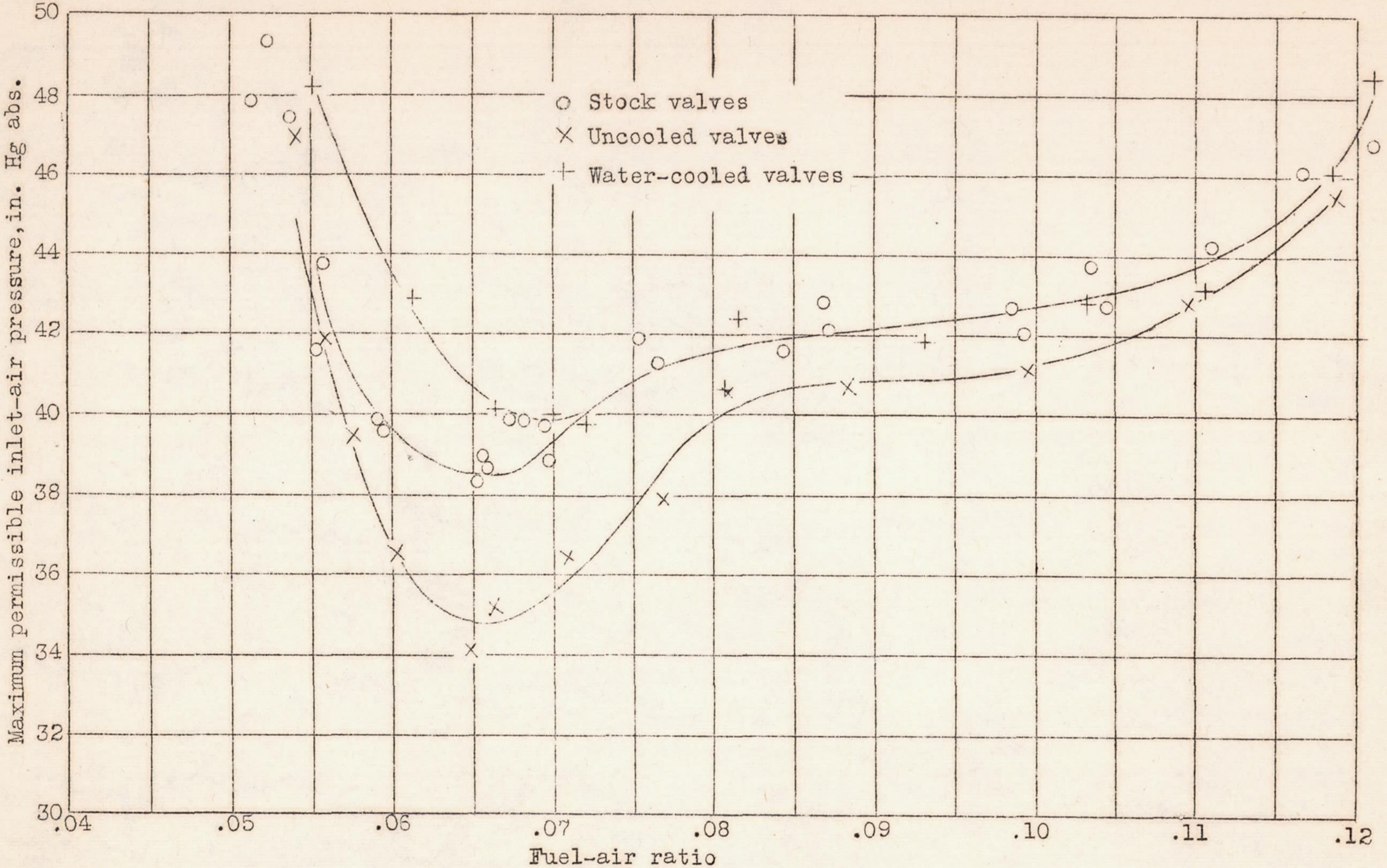


Figure 5.- Comparison of maximum permissible inlet-air pressure at various fuel-air ratios. Fuel, 100-octane.

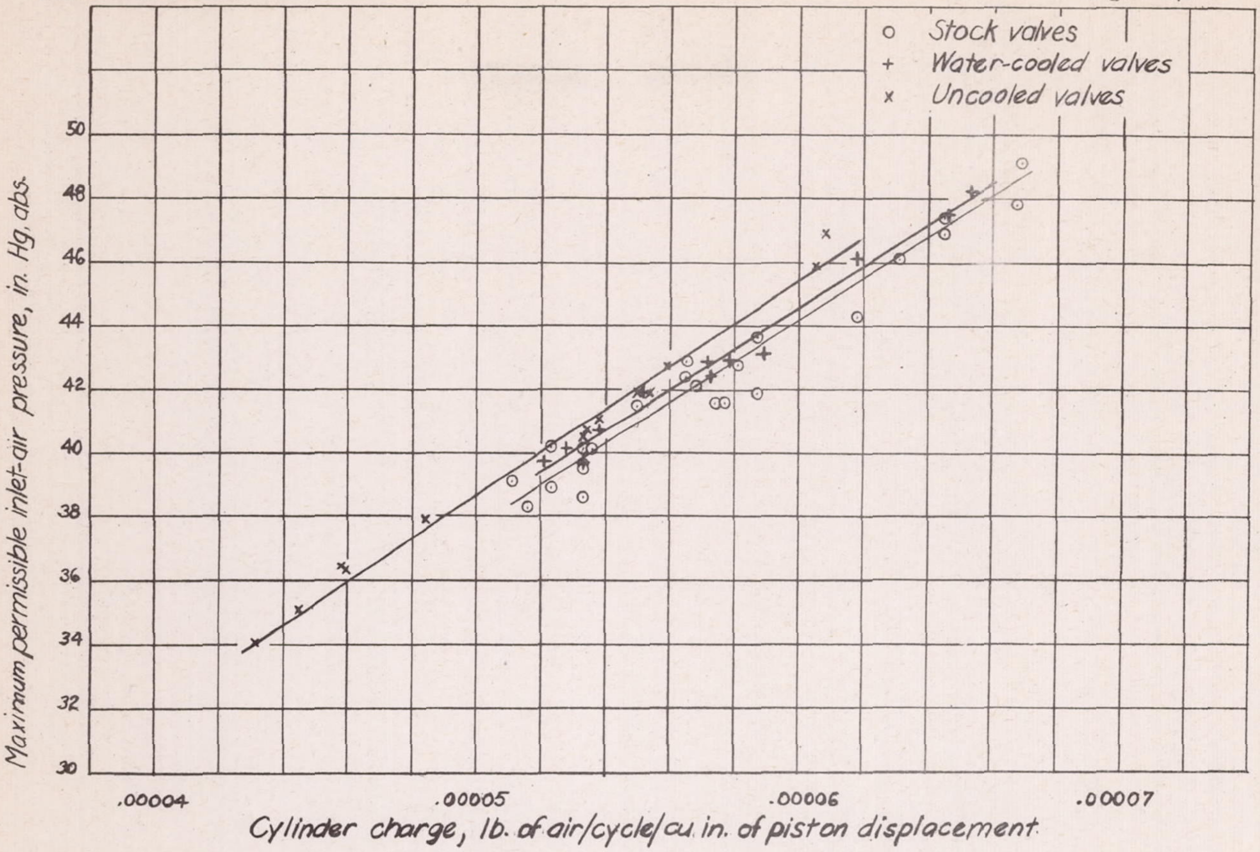


Figure 6.- Cylinder-charge comparison; stock, water-cooled and uncooled valves. Fuel, 100-octane.

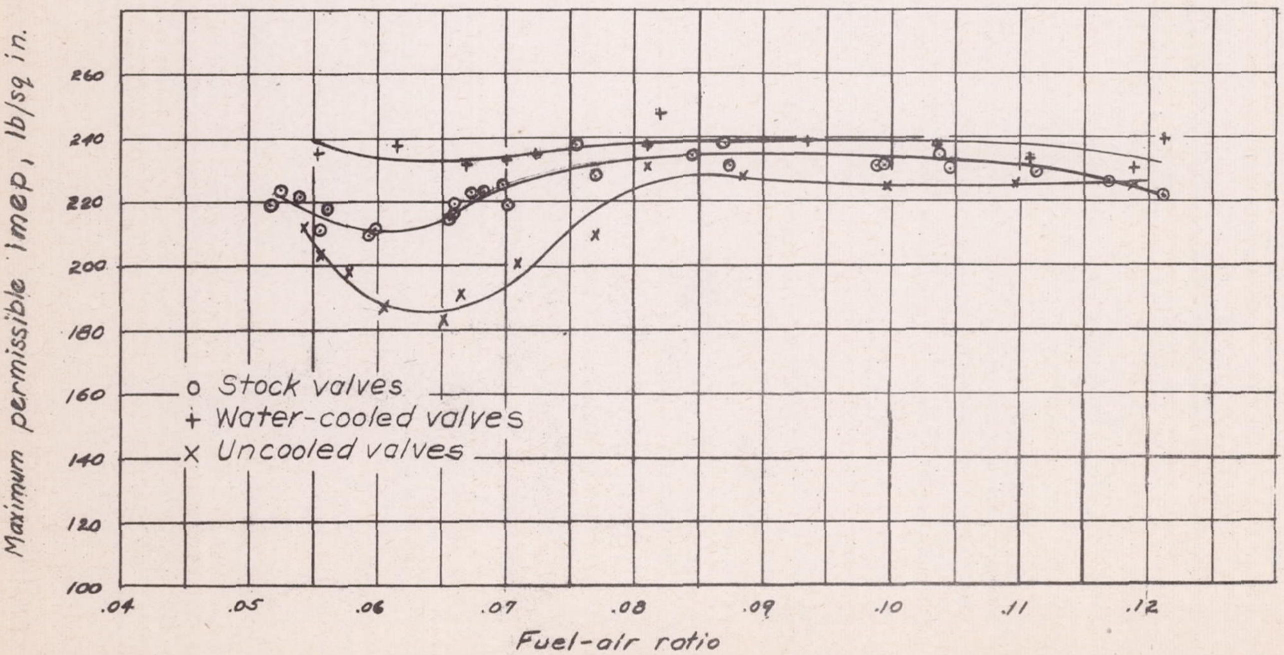


Figure 7 – Comparison of maximum permissible indicated mean effective pressure at various fuel-air ratios. Fuel, 100-octane.

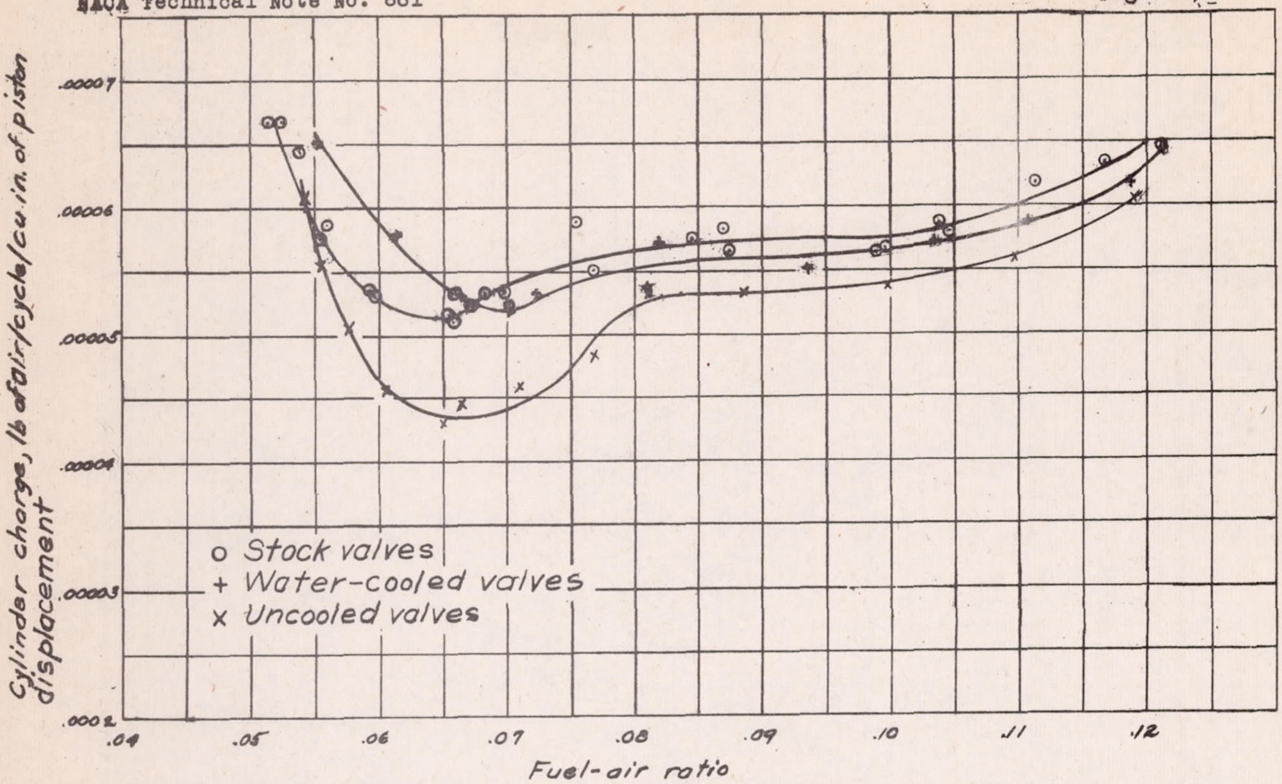
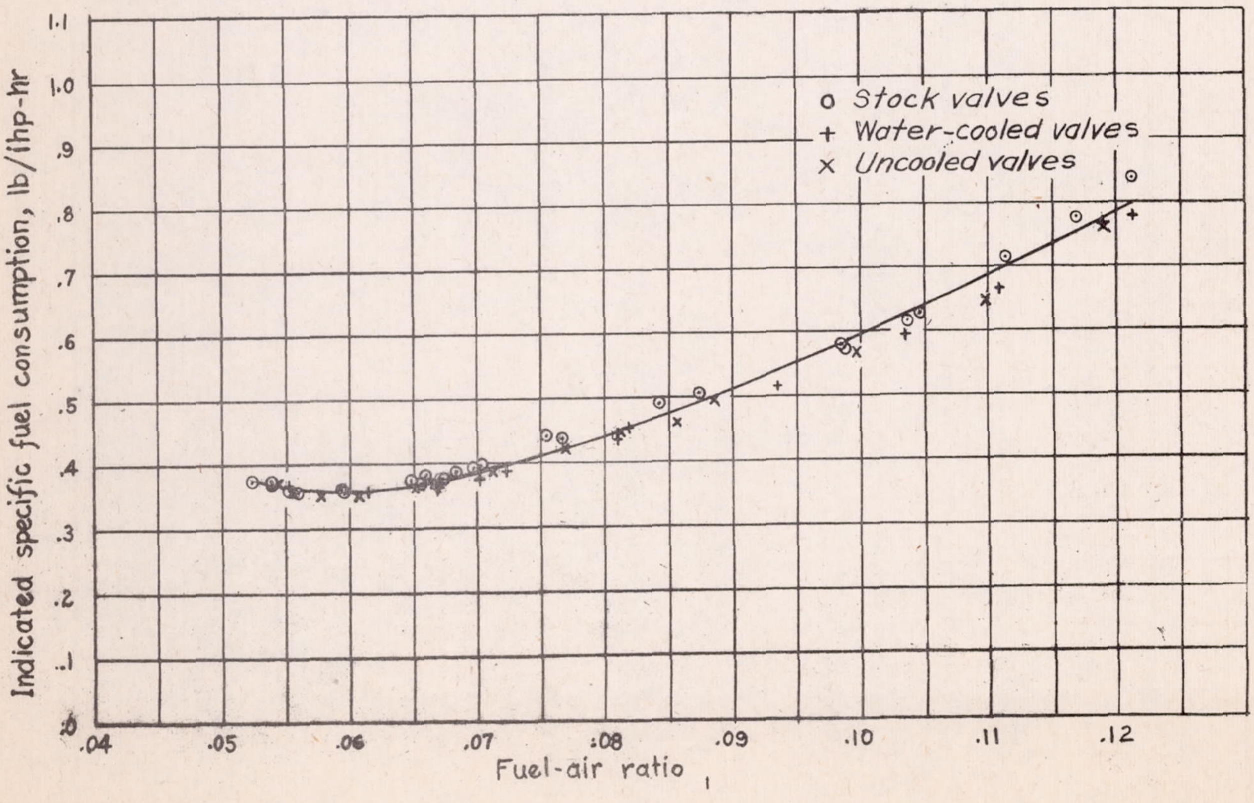


Figure 8.- Comparison of maximum permissible cylinder charge at various fuel-air ratios. Fuel, 100-octane.



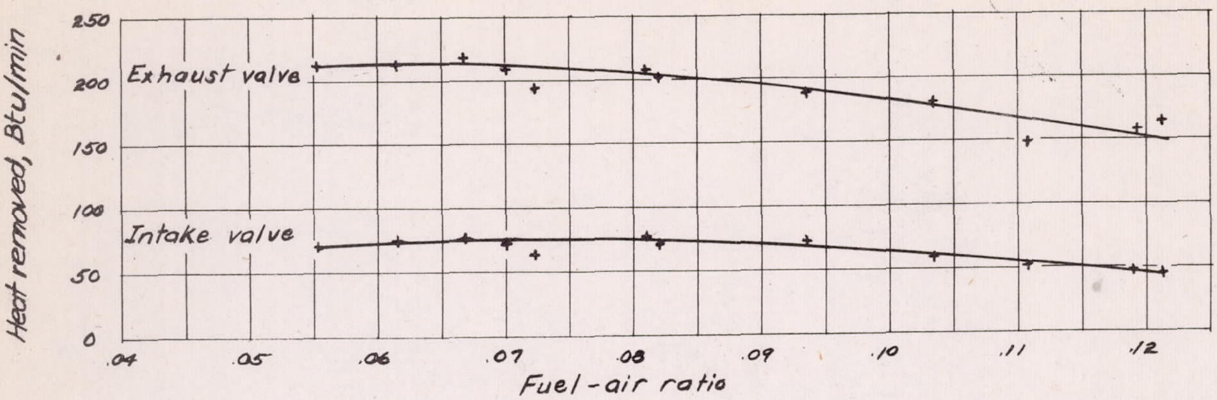


Figure 10.- Heat removed from valves by cooling water at maximum permissible engine output for various fuel-air ratios. Fuel, 100-octane.

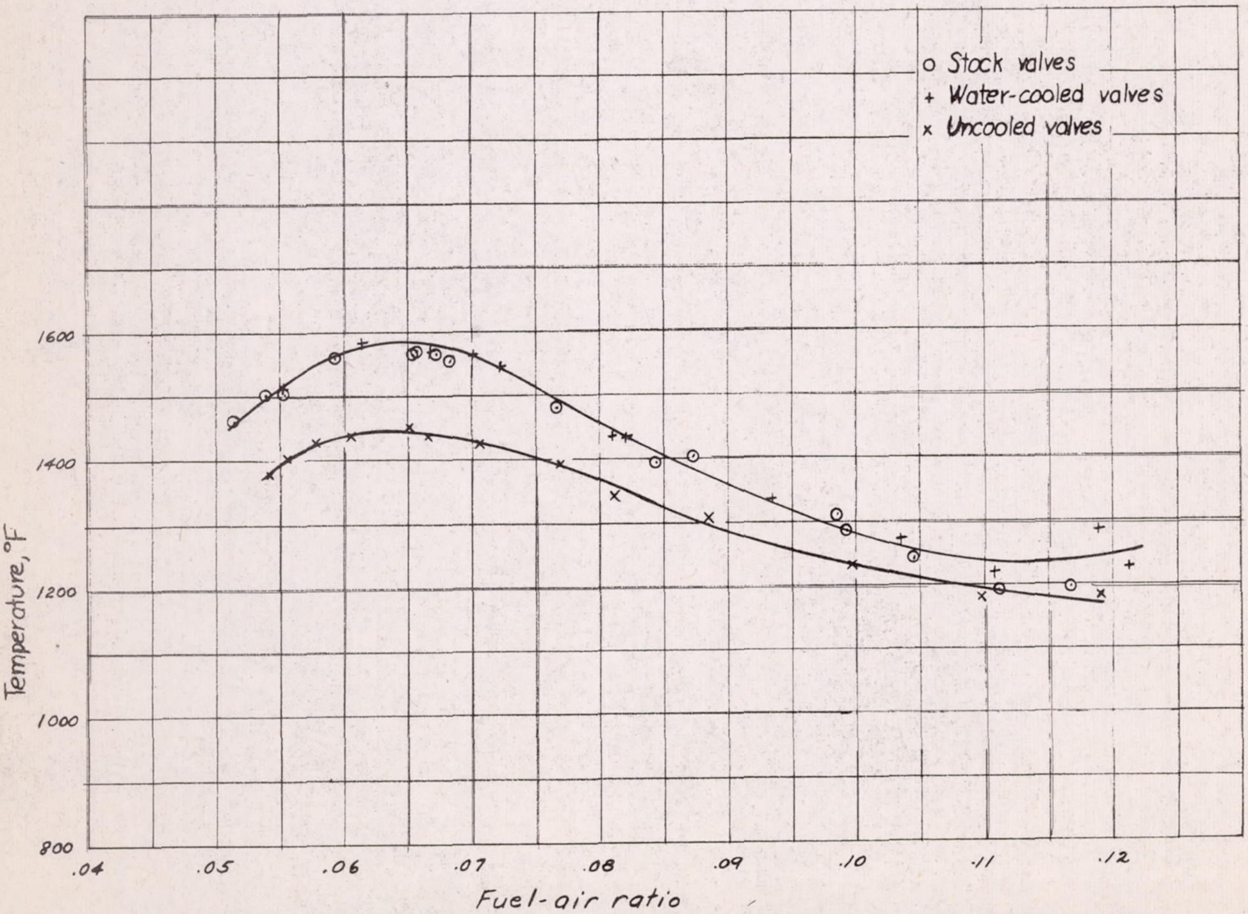


Figure 11. - Exhaust-gas temperature at maximum permissible output with various fuel-air ratios Fuel, 100-octane.

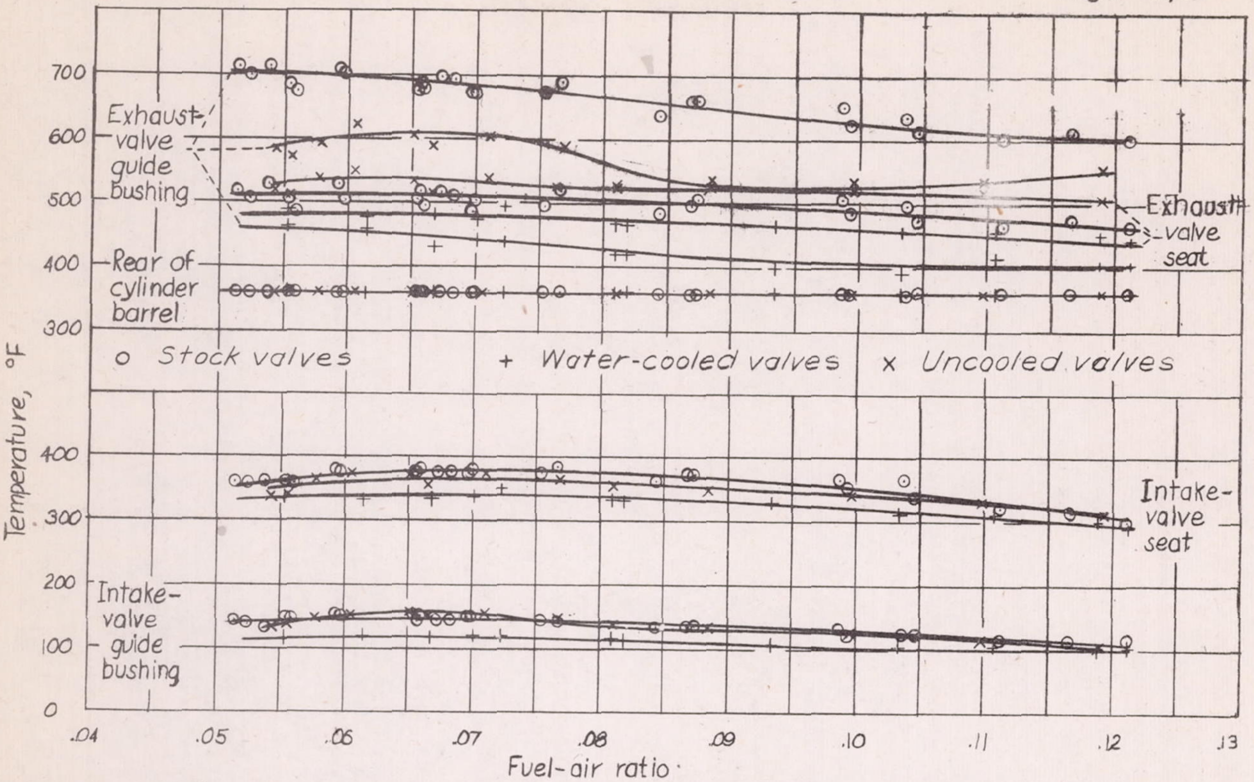


Figure 12.- Valve-seat and valve-guide temperatures at maximum permissible output with various fuel-air ratios. Fuel, 100-octane.

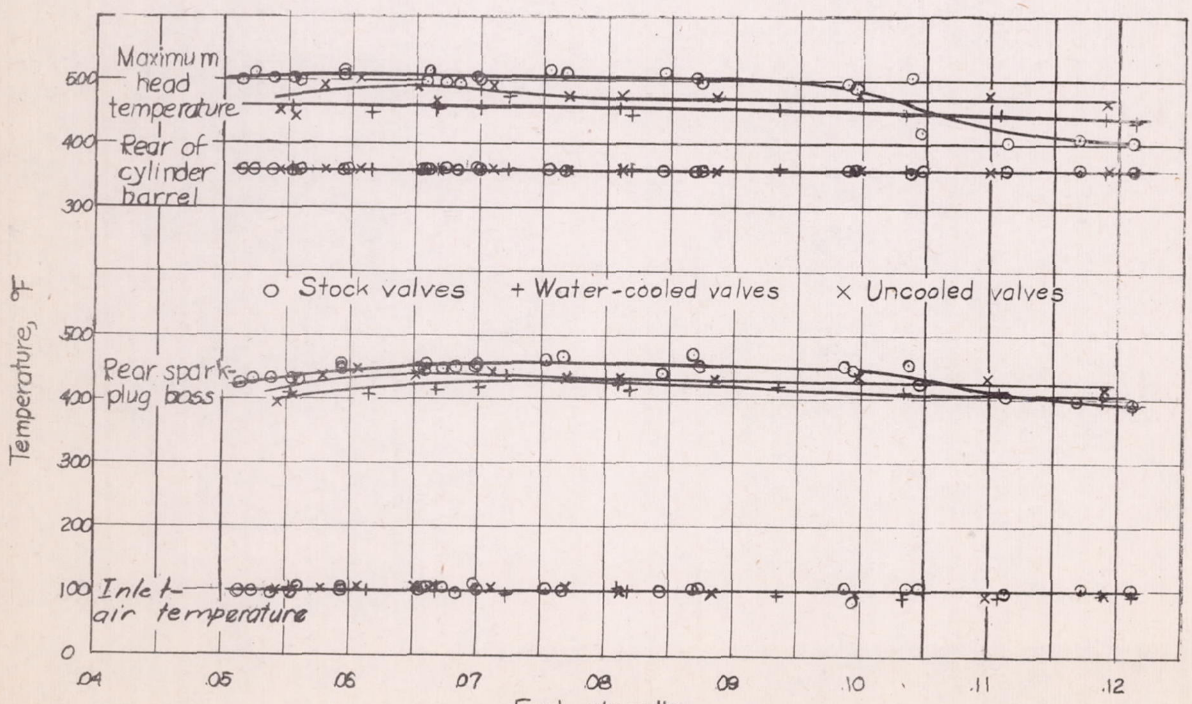


Figure 13.- Temperatures of inlet air and rear spark-plug boss and maximum head temperature at maximum permissible output with various fuel-air ratios. Fuel, 100-octane.