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CYLINDER-HEAD TEMPERATURES AND COOLANT HEAT REJECTION OF A MULTICYLINDER LIQUID-COOLED ENGINE OF 1650-CUBIC-INCH DISPLACEMENT

By John H. Povolny, Louis J. Bogdan
and Louis J. Chelko

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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CYLINDER-HEAD TEMPERATURES AND COOLANT HEAT REJECTION OF A MULTICYLINDER LIQUID-COOLED ENGINE OF 1650-CUBIC-INCH DISPLACEMENT

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SUMMARY

An extensive investigation of the cooling characteristics of a multicylinder liquid-cooled engine of 1650-cubic-inch displacement was conducted at the NACA Cleveland laboratory. The results of this investigation showing the variation of the cylinder-head temperature and the coolant heat rejection with the pertinent engine and coolant variables are presented. These data include engine power outputs up to 2000 brake horsepower and wide ranges of engine speed, manifold pressure, fuel-air ratio, inlet-air temperature, ignition timing, exhaust pressure, and coolant flow, composition, temperature, and pressure. Runs were made with coolant flows as low as 49 gallons per minute in order to investigate the cooling characteristics of this engine under boiling conditions.

All the cylinder temperature data presented and all of the heat-rejection data, except those for which boiling of the coolant was most severe, were satisfactorily correlated with the primary engine and coolant variables by means of the NACA correlation method, which is based on the theory of heat transfer by forced convection. An example of the use of the correlation method for the prediction of head temperatures and the coolant heat rejection is presented.

INTRODUCTION

A knowledge of the cooling characteristics of reciprocating aircraft engines is essential for the prediction of engine

performance at extreme conditions of operation. Consequently, a research program was conducted at the NACA Cleveland laboratory in 1943 to investigate the cooling characteristics of liquid-cooled aircraft engines. The initial phase of this research program consisted of an investigation conducted on two single-cylinder engines to provide data for a fundamental study of the heat-transfer processes involved. These data, which isolate the effects of the various engine and coolant variables on the cylinder-head temperatures, are presented in reference 1. A semiempirical method based on the theory of heat transfer by nonboiling forced convection and used for correlating the cylinder-temperature data of reference 1 with the engine and coolant variables is presented in reference 2.

Concurrent with the investigation on the single-cylinder engines, the cooling characteristics of a multicylinder engine of 1710-cubic-inch displacement were investigated over wide ranges of engine and coolant conditions. The results of this investigation are reported in reference 3, which presents the variation of both the cylinder temperatures and coolant heat rejection with the primary engine and coolant variables. A correlation of the cylinder-head-temperature and coolant-heat-rejection data of reference 3 with the pertinent engine and coolant variables is presented in reference 4. The method of correlation is based on the theory of nonboiling forced-convection heat transfer and is similar to that developed in reference 2.

In order to determine if the trends presented in reference 3 are generally applicable to liquid-cooled multicylinder aircraft engines, a similar investigation of the cooling characteristics of an engine of 1650-cubic-inch displacement was conducted during 1946 and is reported herein. The cylinder-head temperatures and the coolant heat rejection were determined for power outputs up to 2000 brake horsepower over wide ranges of engine speed, manifold pressure, fuel-air ratio, inlet-air temperature, ignition timing, exhaust pressure, and for various conditions of coolant flow, composition, temperature, and pressure. Runs were made for coolant flows as low as 49 gallons per minute in order to investigate the cooling characteristics of this engine under boiling coolant conditions. The variation of the cylinder-head temperatures and the coolant heat rejection with the engine and coolant variables is presented and a comparison is made of the results of this investigation with those of the investigation of the engine of 1710-cubic-inch displacement (reference 3). These cylinder-head-temperature and coolant-heat-rejection data were correlated by means of the NACA correlation method, which is fully illustrated in reference 4, and the final results of the correlation are presented to aid in the application of the data. An example of the use of the correlation method for the prediction of cylinder-head temperatures and the coolant heat rejection is also included.

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APPARATUS

Engines

An investigation was conducted on two standard production-model Packard V-1650-7 engines, which shall hereinafter be designated engines A and B. The V-1650-7 engine is a 12-cylinder, liquid-cooled engine with a bore of 5.4 inches, a stroke of 6.0 inches, and a displacement of 1650 cubic inches. The compression ratio is 6.0 and the engine is fitted with a two-stage supercharger having impeller diameters of 12.0 and 10.1 inches. The two impellers are mounted on the same shaft and can be operated at speeds of either 5.802 or 7.349 times the engine speed. A liquid-type aftercooler is interposed between the supercharger outlet and the intake manifold. The valve overlap extends over a period of time equivalent to 43° rotation of the crankshaft. The ignition timing is controlled by the throttle position and varies from 29° B.T.C. for full-closed throttle to approximately 45° B.T.C. for from half- to full-open throttle. Both the intake and exhaust spark plugs are timed to fire simultaneously. For the variable-ignition-timing runs, the spark-control linkage was disconnected from the throttle and operated by an independent control.

General Engine Setup

A photograph of one of the engines mounted for the cooling investigation is shown in figure 1.

Power measurement. - The engines were mounted on a dynamometer stand equipped with a 3000-horsepower, water-gap, eddy-current dynamometer. The engine speed was electronically controlled and measured by a chronometric tachometer. The torque transmitted to the dynamometer was measured with a calibrated air-balanced diaphragm.

Combustion-air system. - Combustion air was supplied to the engine by the laboratory central-supply system and was metered with an adjustable orifice installed in the supply duct. The temperature of the air was regulated by passing it through either a heater or a refrigerator unit in the supply line; the air was cleaned by means of a filter unit installed in the line downstream of the orifice. Thermocouples and pressure taps were installed at the orifice and at the carburetor inlet to measure the temperature and the pressure of the air at these locations.

Exhaust system. - The engine exhaust gases were removed by means of the laboratory central exhaust system, which also provided the desired exhaust pressures. Water-jacketed exhaust stacks that had stack openings equal in area to the exhaust port openings were used for the investigation. The stacks were connected to a 10-inch-diameter header in which wall taps were installed for the measurement of the exhaust-gas back pressure. 931

Engine coolant system. - A diagrammatic sketch of the engine coolant system is shown in figure 2. An auxiliary pump installed in series with the engine pump permitted the coolant flow to be varied independently of the engine speed. The coolant flow was measured with a venturi. A throttle valve installed downstream of the venturi was used to increase the venturi-throat pressure sufficiently to prevent cavitation during operation at high coolant flows and low coolant pressures. Centrifugal-type vapor separators were installed in the engine coolant-outlet lines to remove air or any vapor that resulted from boiling of the coolant. Vent lines were run from both the vapor separators and the block outlets to an expansion tank. Sight glasses were installed in both sets of vent lines to permit visual observation of the coolant condition. A compressed-air and bleed-line combination on the coolant expansion tank permitted regulation of the expansion-tank pressure.

The coolant temperature-control unit consisted of two aircraft-type coolers and an air-operated, thermostatically controlled three-way mixing valve installed at the junction of a main line from and a bypass line around the coolers. Water was used to cool the engine coolant solutions and the flow was measured with calibrated rotameters. Thermocouples and pressure taps were installed at the locations indicated in figure 2.

The coolant flow path through the cylinder bank is schematically shown in figures 3 and 4. The coolant is distributed to the six barrels of each cylinder bank by means of an external coolant branch tube. This coolant branch tube, which is connected to the discharge of the engine coolant pump, has three outlets, each supplying coolant to two adjacent cylinder barrels. After entering the barrels, the coolant flows around each cylinder barrel and up into the cylinder heads through the 14 connector tubes. The coolant then passes over the cylinder heads and is discharged through a single outlet at the forward end of the cylinder bank. It may be seen from figure 4 that the flow over each cylinder head is equal to the total of all the flows through the connector tubes upstream of the cylinder head in question.

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For convenient identification, the manufacturer's designation of the banks and cylinders is used in this report. Thus, when facing the rear of the engine the right-hand bank is called bank A and the left-hand bank is called bank B. The cylinders of each bank are numbered from 1 to 6 starting at the front of the engine.

Lubricating-oil system. - A diagrammatic sketch of the oil system is shown in figure 5. A remote indicating oil-level and oil-flow device, which is described in reference 5, was incorporated in the oil reservoir tank. The oil temperature-control unit was similar to that used in the engine coolant system. Thermocouples and pressure taps were installed at the locations indicated on the diagram.

Aftercooler coolant system. - Figure 6 is a schematic diagram of the aftercooler coolant system. In addition to passing through the aftercooler unit, the aftercooler coolant also passed through a cooling jacket that surrounds the two-stage supercharger. An expansion tank was incorporated in the housing of the aftercooler unit and a relief valve set for a pressure of 20 pounds per square inch gage was mounted on the tank. The aftercooler coolant flow was regulated by means of a throttle valve and metered with a calibrated venturi. The aftercooler-coolant temperature-control unit was similar to that used in the coolant and oil systems. Thermocouples and pressure taps were located at the positions indicated on the diagram.

Coolants, fuel, and oil. - Water and several aqueous solutions of ethylene glycol were used as the coolants in the engine coolant system. A mixture of 30-percent ethylene glycol and 70-percent water was used as the coolant in the aftercooler coolant system. In order to inhibit corrosion, 0.2 percent by volume of sodium mercaptobenzothiazole (NaMBT) was added to all coolants.

The fuel AN-F-28, Amendment-2, was metered by calibrated rotameters. For knock-free engine operation at high power, 3 percent by volume of xylidines was added to the fuel; the tetraethyl-lead concentration was increased to 6 milliliters per gallon. The lubricating oil used throughout the investigation was Navy 1120.

Thermocouple Installation

Thermocouples for engine-temperature measurement. - The cylinder-head thermocouple installation is shown in figure 7. Thermocouples were installed on each cylinder in the cylinder head between the exhaust valves, between the intake valves, and in the center of the head. The thermocouple holes in the cylinder heads were drilled with the aid of jigs to insure uniformity and accuracy of location. The cylinder-head thermocouples were silver-soldered into brass plugs, which were peened into the bottom of the drilled holes. The leads inside the engine were packed in porcelain cement and encased in stainless-steel tubing. The leads outside the engine were insulated with flexible glass sleeves and protected with ignition-type shielding. The temperatures were read on a self-balancing direct-reading potentiometer and recorded on a self-balancing recording potentiometer. 931

The inlet-manifold temperature was measured with a single unshielded iron-constantan thermocouple located approximately 10 inches downstream of the aftercooler outlet.

Thermocouples for liquid-temperature measurement. - Two types of thermocouple, iron-constantan and copper-constantan, were installed in the engine-coolant, aftercooler-coolant, lubricating-oil, and cooling-water lines at the locations shown in figures 2, 5, and 6. The iron-constantan thermocouples were connected to both a self-balancing direct-reading potentiometer and a self-balancing recording potentiometer. The copper-constantan thermocouples, which were used for an accurate determination of the temperature differences across the engine and coolers, were connected to a portable precision-type potentiometer and balance was indicated on a light-beam galvanometer.

PROCEDURE

Conditions

Cylinder temperatures and coolant heat rejections were determined for the following ranges of conditions:

Engine power, bhp	560-2000
Engine speed, rpm	1800-3200
Manifold pressure, in. Hg absolute	31-78
Charge flow (air plus fuel), lb/sec	1.43-4.30
Fuel-air ratio	0.059-0.115
Carburetor-air temperature, °F	-57 - 198
Ignition timing, deg B.T.C.	20-52
Exhaust pressure, in. Hg absolute	8-61
Coolant flow, gal/min	49-253
Average coolant temperature, °F	168-278
Coolant-outlet pressure, lb/sq in. gage	20-45

Aqueous ethylene-glycol solutions containing 0, 30, and 70 percent by volume of ethylene glycol were used as coolants. In order to isolate the effect of the various engine and coolant variables on the cylinder temperatures and coolant heat rejection, one of these parameters was varied while, in general, the rest were held constant. No standard set of reference conditions however, was maintained. A complete summary of the experimental conditions is presented in table I.

The charge flow (air plus fuel) rather than the brake horsepower was held constant for all the tests in which one of the other engine parameters or one of the coolant parameters was varied. For the variable-engine-speed and variable-exhaust-pressure runs, the charge flow was maintained constant by varying the manifold pressure. For the variable-charge-flow runs, the engine speed was held constant while the manifold pressure was varied. The inlet-manifold temperature was maintained at the desired values indicated in table I by varying the aftercooler-coolant temperature.

For the entire investigation, the carburetor-inlet pressure was maintained at 1 atmosphere, the engine supercharger was operated in the lower of the two gear ratios, and the oil-inlet temperature was 172° ±4° F. For all of the runs except those in which ignition timing was the primary variable, the ignition timing was approximately constant at 45° to 46° B.T.C. For all the tests in which the engine parameters were varied, the coolant composition was 30-percent ethylene glycol and 70-percent water, the average coolant temperature was maintained at 240° F, the coolant-outlet pressure at 35 pounds per square inch gage, and the coolant flow at 201 to 205 gallons per minute with the exception of the variable-ignition-timing runs for which

the coolant flow was 167 gallons per minute. (For current flight installations, the coolant flow at an engine speed of 3000 rpm would probably be not less than 170 or more than 210 gal/min.)

The coolant flow was varied at several engine powers for several combinations of coolant temperature, pressure, and composition. The highest coolant flow attainable in each case was limited either by the cavitation characteristics of the pumps at low coolant pressure or by the capacities of the pumps at high coolant pressure. The lowest coolant flow was limited by severe boiling of the coolant in the engine. Depending upon the type of test conducted, either the average or the outlet temperature of the coolant was held constant. The desired coolant-outlet pressures were maintained by regulating the coolant expansion-tank pressure.

The ethylene-glycol concentration of the coolant was determined from the specific gravity of samples taken at intervals throughout the investigation.

Calculations

Temperature averages. - The average temperature for each thermocouple location in the cylinders was determined by averaging the temperatures measured at that location in all 12 cylinders. The average coolant temperature was taken as the arithmetic mean of the measured inlet and outlet temperatures.

Heat rejected to coolant. - The heat rejected to the coolant was determined by two methods: (1) from the measured temperature rise and coolant flow through the engine; and (2) from the measured temperature rise and flow of the coolant cooling water. The heat rejected to the engine coolant is presented on the basis of method (2) because at low flows when large amounts of vapor were formed method (1) would not include the heat of vaporization, and at high flows difficulty was experienced in accurately measuring the small temperature rise of the coolant incurred in passing through the engine. The maximum difference between the heat rejections as determined by each of the two methods, however, was not more than 10 percent. The external heat loss from the coolant piping and expansion tank was estimated to be less than 2 percent of the coolant heat rejection. The data were not corrected for this loss.

RESULTS AND DISCUSSION

The variation of the cylinder-head temperature and the coolant heat rejection with the basic engine and coolant parameters was, in some cases, obtained for only a single set of operating conditions. It is believed, however, that the variations shown are general inasmuch as the more comprehensive investigation of reference 3 indicated that the trends of cylinder temperatures or heat rejection with primary engine and coolant variables were the same for several operating conditions.

Part of this investigation was conducted with engine A and part with engine B; therefore all the variables considered were not investigated on a single engine. A comparison of data from each engine at similar operating conditions showed, however, that the trends and the magnitudes of both the cylinder temperatures and heat rejection were the same for both engines.

Relations Among Cylinder Temperatures

The relation between the average temperature in the center of the cylinder head and the average temperature in the cylinder head between the exhaust valves is shown in figure 8 for all the data for both engines at the various operating conditions. A linear relation exists between these temperatures and the scatter of the data is within ±15° F. The average temperature in the center of the head was from 50° to 70° F lower than the average temperature in the cylinder head between the exhaust valves.

The variation of the average temperature in the cylinder head between the intake valves with the average temperature in the cylinder head between the exhaust valves is presented in figure 9 for engine A at three values of engine running time. A separate straight line fits the data for each running time and the slope of the lines is the same. The effect of engine running time on the cylinder temperatures is subsequently discussed in connection with another figure. The mean scatter of the data is about ±5° F and the average temperature in the cylinder head between the intake valves ranged from about 45° to 70° F lower than that between the exhaust valves at an engine running time of 15 hours, and about 100° to 110° F lower at an engine running time of 95 hours.

As a result of the linear relation existing between the temperatures in the various locations in the cylinder heads, the

variation of only one of them with the primary engine and coolant variables is presented. The average cylinder-head temperature between the exhaust valves has been chosen for this purpose because it was the highest average temperature measured and is therefore most indicative of critical cooling conditions.

The relation between the average temperature of the 12 cylinders in the head between the exhaust valves and the temperature of the hottest cylinder (maximum) measured for the same location is presented in figure 10 for all the data. For the conditions investigated, the maximum temperature generally occurred on cylinder A6 and occasionally on cylinders B6 and A5. A linear variation is noted for both engines for the entire range of temperatures measured and the mean scatter of the data is about ±15° F. The maximum temperature ranged from 30° to 60° F higher than the average temperature; the difference increased with the magnitude of the temperature. A linear variation was also obtained between the maximum and average temperatures for the other thermocouple locations.

The variation of the average cylinder-head temperatures with engine running time is presented in figure 11. The data presented were obtained at a reference operating condition from time to time during the course of the runs on engine A. The coolant used for this reference operating condition was composed of 30-percent ethylene glycol and 70-percent water. Coolants of other compositions were used, however, between the runs at the reference operating condition. For an increase in engine running time from 15 to 115 hours, both the average temperatures in the cylinder head between the exhaust valves and in the center of the head increased about 45° F, whereas the temperature between the intake valves remained approximately constant. A close inspection of the coolant passages in a scrapped cylinder head revealed scale deposits in the exhaust side and in the center of the head but none in the intake side. The increase of the temperatures in the exhaust side and in the center of the cylinder head are attributed to these scale deposits. The effect of engine running time noted in figure 9 is the result of this increase in the temperature of the cylinder head between the exhaust valves and the constancy of the temperature between the intake valves with engine running time.

The data presented in the succeeding figures are not corrected for the effect of engine running time so that temperatures obtained for the same operating conditions but at different running times will not always be the same. The variations presented in each

figure were not greatly influenced by engine running time, however, because, as indicated in table I, all runs of a series were completed within a relatively short interval of time.

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Effect of Engine Variables on Average Cylinder-Head Temperature Between Exhaust Valves

Charge flow and engine power. - The variation of the average cylinder-head temperature between the exhaust valves T_h with charge flow is presented in figure 12. The cylinder-head temperature increased linearly with charge flow throughout the range investigated and, for an increase in charge flow of 0.5 pound per second, an increase of approximately 25° F in T_h occurred.

In order to permit a determination of the variation of the cylinder-head temperature with engine power, a secondary scale of brake horsepower is included in figure 12. This scale of power is applicable only for the engine speed of 3000 rpm at which the runs were made.

Inlet-manifold temperature. - The effect of measured inlet-manifold temperature on T_h is presented in figure 13 for both variable carburetor-air temperature and variable engine speed (constant charge flow). The increase in inlet-manifold temperature with an increase in engine speed is the result of an increase in the temperature rise across the supercharger. For both sets of data, the increase in T_h with inlet-manifold temperature was the same and amounted to about 9° F for an increase of 100° F in inlet-manifold temperature.

Fuel-air ratio. - The effect of fuel-air ratio on T_h is presented in figure 14. A maximum value of T_h occurred at a fuel-air ratio of 0.067, which is approximately equal to that for the chemically correct mixture. For an increase in fuel-air ratio from 0.067 to 0.115, T_h decreased approximately 70° F.

Ignition timing. - The variation of T_h with ignition timing for three values of engine speed is shown in figure 15. For all three engine speeds investigated, an increase in the ignition timing from 20° to 36° B.T.C. resulted in negligible change in T_h; further increase in ignition timing to 52° B.T.C. resulted in an increase in the average cylinder-head temperature of about 10° F.

Exhaust pressure. - The variation of T_h with exhaust pressure for various fuel-air ratios and for engine speeds of 2400 and 3000 rpm is presented in figure 16. For comparable fuel-air ratios, the effect of exhaust pressure is slightly greater at 3000 than at 2400 rpm and for each speed it is slightly greater at the leaner fuel-air ratios. The increase in T_h for an increase in exhaust pressure from 8 to 60 inches of mercury absolute ranged from about 45° F at an engine speed of 2400 rpm and a fuel-air ratio of 0.095 to about 70° F at an engine speed of 3000 rpm and a fuel-air ratio of 0.067. 931

Effect of Coolant Variables on Average Cylinder-Head Temperature Between Exhaust Valves

Coolant composition. - The average cylinder-head temperature between the exhaust valves for aqueous ethylene glycol coolant mixtures containing 0-, 30-, and 70-percent ethylene glycol is presented in figure 17 for a range of coolant flows. These data were obtained for constant average coolant temperature and nonboiling coolant conditions. The condition of the coolant with respect to boiling and nonboiling was determined by observation of the coolant in the sight glasses. For all conditions, an increase in the ethylene-glycol concentration of the coolant resulted in an increase in T_h. A comparison of the values of T_h obtained when using 100-percent water as the coolant with those obtained when using the other coolants shows a difference of about 15° F at all coolant flows for the coolant containing 30-percent ethylene glycol and differences of about 33° F at a coolant flow of 250 gallons per minute and about 47° F at a coolant flow of 50 gallons per minute for the coolant containing 70-percent ethylene glycol. It is shown in reference 3 that boiling of the coolant did not appreciably alter the relative effect of coolant composition from that obtained for nonboiling coolant conditions.

Coolant flow. - The effects of coolant flow on T_h are illustrated in figure 17 for constant average coolant temperature and nonboiling coolant conditions, and in figure 18 for two constant coolant-outlet temperatures and boiling coolant conditions. In general, the effect of coolant flow was less for the data shown in figure 18 than for that shown in figure 17. This lesser effect of coolant flow (fig. 18) is probably the result of a decrease in the average coolant temperature and an increase in the intensity of

boiling as the coolant flow was reduced (reference 3). For the data of figure 18, the rate of increase of T_h with decreased coolant flow was greater at the low coolant pressures and low coolant flows. This greater rate of increase as the coolant flow is reduced is attributed to a transition from nuclear to film-type boiling wherein large amounts of insulating vapor are formed resulting in reduced heat transfer and increased cylinder-head temperatures. For constant average coolant temperature and nonboiling coolant conditions (fig. 17), the effect of coolant flow was slightly greater with the coolant containing the greatest percentage of ethylene glycol (the 70-percent-glycol coolant mixture) than with either the 30-percent-glycol mixture or 100-percent water. Thus, it is apparent from the foregoing considerations that the effect of coolant flow on T_h is dependent upon the operating conditions. For example, a decrease in the coolant flow from 200 to 50 gallons per minute resulted in an increase in T_h ranging from practically zero for the high-pressure, high-coolant-temperature data of figure 18 to about 25° F for the 70-percent-glycol coolant mixture data of figure 17. In reference 3 it was found that the effect of coolant flow for the conditions of constant average coolant temperature and boiling of the coolant, and constant coolant-outlet temperature and no boiling of the coolant was less than that for the coolant conditions of figure 17 and greater than that for the coolant conditions of figure 18.

Coolant-outlet pressure. - The decrease in T_h with decrease in coolant-outlet pressure, which is illustrated in figure 18, became slightly greater as the coolant flow was reduced from 200 to 100 gallons per minute. Further reduction to a coolant flow of 50 gallons per minute resulted in a decreasing effect of coolant pressure and in some instances a reversal of the effect previously noted. The maximum increase in T_h for an increase in coolant-outlet pressure of 20 pounds per square inch was about 15° F. For all conditions, the decrease in T_h with a decrease in coolant pressure is attributed to an increase in the localized nuclear boiling of the coolant, which resulted in an increase in the heat-transfer coefficient on the liquid side. The increase in T_h with a decrease in coolant pressure noted at the low coolant flows is probably the result of a transition from nuclear to film-type boiling wherein large amounts of insulating vapor are formed and result in a decrease in the heat-transfer coefficient on the liquid side.

Average coolant temperature. - The effect of average coolant temperature on T_h for nonboiling coolant mixtures containing

30- and 70-percent ethylene glycol is shown in figure 19. For both of these coolants, the variation was linear and amounted to an increase of about 60° F for an increase in average coolant temperature of 100° F.

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Effect of Engine Variables on Heat Rejection to Coolant

Charge flow and engine power. - The variation of the coolant heat rejection with charge flow is presented in figure 20, which includes a secondary scale of brake horsepower. As did the head temperature, the heat rejection increased linearly with charge flow. For an increase in charge flow of 0.5 pound per second, the coolant heat rejection increased approximately 40 Btu per second.

Inlet-manifold temperature. - The effect of measured inlet-manifold temperature on the coolant heat rejection is presented in figure 21 for both variable carburetor-air temperature and variable engine speed (constant charge flow). For both sets of data, the increase in the coolant heat rejection amounted to about 20 Btu per second for an increase of 100° F in inlet-manifold temperature.

Fuel-air ratio. - The effect of fuel-air ratio on the coolant heat rejection is presented in figure 22. As for the corresponding cylinder-head-temperature data, a maximum value occurred at a fuel-air ratio of 0.067. For an increase in fuel-air ratio from 0.067 to 0.115, the coolant heat rejection decreased about 80 Btu per second.

Ignition timing. - The variation of the coolant heat rejection with ignition timing for three values of engine speed is presented in figure 23. For all three engine speeds, an increase in the ignition timing from 20° to 36° B.T.C. resulted in a negligible change in the coolant heat rejection; further increase to 52° B.T.C. resulted in an increase in the coolant heat rejection of about 12 Btu per second.

Exhaust pressure. - The variation of the coolant heat rejection with exhaust pressure for various fuel-air ratios and for engine speeds of 2400 and 3000 rpm is presented in figure 24. As for the cylinder-head temperatures, the effect of exhaust pressure is slightly greater at 3000 than at 2400 rpm for comparable fuel-air ratios and for each speed it is greater at the leaner fuel-air ratios. The increase in the coolant heat rejection for an increase

in exhaust pressure from 8 to 60 inches of mercury absolute ranged from about 55 Btu per second at an engine speed of 2400 rpm and a fuel-air ratio of 0.095 to about 90 Btu per second at an engine speed of 3000 rpm and a fuel-air ratio of 0.067.

Engine running time. - No measurable change in the heat rejection to the coolant occurred with engine running time. Although scaling of the coolant passages probably would decrease the heat rejections, the magnitude of the change was less than could be detected within the normal accuracy of the data. It would also be expected that the heat rejections would be affected less than the head temperatures, because the scale formation occurred on only part of the coolant passages in the cylinder head.

Effect of Coolant Variables on Coolant Heat Rejection

Coolant composition. - A comparison of the coolant heat rejections obtained when using the aqueous ethylene-glycol coolant mixtures containing 0-, 30-, and 70-percent ethylene glycol is presented in figure 25 for a range of coolant flows. These data were obtained for constant average coolant temperature and nonboiling coolant conditions. For the entire flow range investigated, the coolant heat rejection was about 5 Btu per second lower for the 30-percent glycol mixture and about 11 Btu per second lower for the 70-percent glycol mixture than that for water.

Coolant flow. - The effects of coolant flow on the coolant heat rejection are illustrated in figure 25 for constant average coolant temperature and nonboiling coolant conditions and in figure 26 for two coolant-outlet temperatures and boiling coolant conditions. The effect of coolant flow was the same for the three coolants illustrated (fig. 25) when the average coolant temperature was held constant and when there was no boiling of the coolant; for a decrease in coolant flow from 200 to 50 gallons per minute, the heat rejection decreased about 10 Btu per second. When the coolant-outlet temperature was held constant and there was boiling of the coolant (fig. 26), coolant flow had no effect on the coolant heat rejection except at 2000 brake horsepower, for which there was a slight decrease of about 7 Btu per second for a reduction in coolant flow from 100 to 50 gallons per minute. The constancy of the coolant heat rejection with coolant flow for these conditions is a direct reflection of the constancy of the head temperature with coolant flow noted for figure 18. The decrease in the coolant heat rejection at 2000 brake horsepower for a decrease in coolant

flow from 100 to 50 gallons per minute is probably the result of a greater increase in film-type boiling at this power than at the other powers.

Coolant-outlet pressure. - Although the data of figure 26 were obtained for several coolant-outlet pressures, the scatter was such that no definite trends of coolant heat rejection with coolant-outlet pressure could be determined.

Average coolant temperature. - The variation of the coolant heat rejection with average coolant temperature for coolant mixtures containing 30- and 70-percent ethylene glycol is presented in figure 27. For both of these coolants, the variation was linear and amounted to a decrease of about 50 Btu per second for an increase in average coolant temperature of 100° F.

Comparison of Engines of 1650- and 1710-Cubic-Inch Displacement

In general, the engine variables had a slightly greater effect on both the average cylinder-head temperature between the exhaust valves and the heat rejected to the coolant for the 1650-cubic-inch-displacement engine than for the 1710-cubic-inch-displacement engine of reference 3. The effect of coolant variables on both the cylinder-head temperature and the coolant heat rejection was similar for both engines except that for an increase in average coolant temperature of 100° F the increase in cylinder-head temperature was about 60° F for the 1650 engine as compared with 85° F for the 1710 engine. For comparable engine and coolant conditions, the average cylinder-head temperature between the exhaust valves for the 1650 engine was from about 30° to 40° F higher than that for the 1710 engine and the coolant heat rejections were approximately the same.

Correlation of Cylinder-Head Temperatures

All the cylinder-head-temperature data were satisfactorily correlated with the primary engine and coolant variables, and the final results are presented herein to aid in application of the data. The NACA correlation method, which is developed in reference 2 and is based on the theory of heat transfer by forced convection, was applied in a manner similar to that presented in

reference 4 for the multicylinder liquid-cooled engine of 1710-cubic-inch displacement. The resulting correlation plot is presented in figure 28 and all the data points fall within the dashed lines, which represent a variation of ±20° F. The equation of this straight-line curve, which is the correlation equation that relates all the pertinent variables, is

$$\left(\frac{T_g - T_h}{T_h - T_l}\right)\left[\left(\frac{0.000492}{W_l^{0.36}}\right)\left(\frac{\mu^{0.36}}{k\ Pr^{0.33}}\right) + Z\right] = W_c^{-0.60} \qquad (1)$$

where T_g is the cylinder-gas temperature effective in the transfer of heat from the cylinder gases to the inside of the cylinder wall, and is considered a function of fuel-air ratio, inlet-manifold temperature, exhaust pressure, and ignition timing. The constants and the exponents in the equation were evaluated by a data analysis similar to that presented in reference 4. The remaining symbols in equation (1) and others to be used subsequently are defined as follows:

H coolant heat rejection, (Btu)/(sec)

k thermal conductivity of coolant, (Btu)/(sec)(sq ft)(°F/ft)

N engine speed, rpm

Pr Prandtl number of coolant, $c\mu/k$

T_c charge-air temperature at carburetor inlet, (°F)

T_g effective cylinder-gas temperature, (°F)

T_h average cylinder-head temperature between exhaust valves, (°F)

T_l average coolant temperature, (°F)

T_m dry inlet-manifold temperature (calculated), (°F)

W_c engine charge flow (air plus fuel), (lb)/(sec)

W_l coolant flow, (lb)/(sec)

Z factor that accounts for temperature drop through cylinder head

μ absolute viscosity of coolant, (lb)/(ft)(sec)

ΔT_{aft} temperature drop through aftercooler, (°F)

ΔT_s temperature rise across supercharger, (°F)

The variation of T_g with fuel-air ratio for a range of exhaust pressures and for a dry inlet-manifold temperature of 80° F is presented in figure 29. For the range covered by the variable-ignition-timing device (29° to 45° B.T.C.) on the engines used, ignition timing was found to have a negligible effect on T_g and thus no correction for this parameter is required. Correction of T_g to a dry inlet-manifold temperature other than 80° F is made according to the relation

$$\Delta T_g = 0.35\ (T_m - 80) \tag{2}$$

where T_m is defined as the air temperature at the carburetor inlet plus the temperature rise of the air incurred in passing through the superchargers (assuming no fuel vaporization) minus the temperature drop of the charge mixture incurred in passing through the aftercooler (as may be determined from the heat rejected to the aftercooler coolant) and may be written

$$T_m = T_c + \Delta T_s - \Delta T_{aft} \tag{3}$$

For the two-speed two-stage supercharger on the engines used in this investigation, the temperature rise across the supercharger is defined by the following relations:

Low-blower operation,

$$\Delta T_s = 25.17 \left(\frac{N}{1000}\right)^2 \tag{4}$$

and high-blower operation,

$$\Delta T_s = 40.38 \left(\frac{N}{1000}\right)^2 \tag{5}$$

As for the multicylinder liquid-cooled engine of reference 4, the Z factor increased with initial engine running time. This increase, which was attributed to a progressive scale buildup in the coolant passages, is shown in figure 30. Extrapolation of

the curve beyond an engine running time of 120 hours was made using the shape of the corresponding curve of reference 4 as a guide. This extrapolation indicates that for engine running times greater than about 120 hours a constant value of Z equal to 0.22 may be used.

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In order to facilitate the computation of head temperatures by means of the correlation equation, values of the coolant-property parameter $\mu^{0.36}/k\ Pr^{0.33}$ are presented in figure 31 for various coolant mixtures of ethylene glycol and water over a range of coolant temperatures.

Correlation of Coolant Heat Rejection

All of the coolant-heat-rejection data except those for which boiling of the coolant was most severe (all runs at 2000 bhp) were satisfactorily correlated with the primary engine and coolant variables and the final results are presented to aid in application of these data. The NACA correlation method was applied, in a manner similar to that presented in reference 4 for the multicylinder liquid-cooled engine of 1710-cubic-inch displacement. The resulting correlation plot is presented in figure 32 and all the data points fall within ±5 percent of the coolant heat rejection determined by the straight-line curve through the data points. The equation of this straight-line curve, which relates all the pertinent variables, is

$$0.49\left(\frac{T_g - T_l}{H}\right) - \left(\frac{0.000154}{W_l^{0.21}}\right)\left(\frac{\mu^{0.21}}{k\ Pr^{0.38}}\right) - 0.32 = W_c^{-0.97} \qquad (6)$$

The constants and the exponents in this equation were evaluated by a data analysis similar to that presented in reference 4. For this correlation, the Z factor is a constant and is represented by the value 0.32 in equation (6).

The variation of effective exhaust-gas temperature T_g for the heat-rejection correlation with fuel-air ratio for a range of exhaust pressures and for a dry inlet-manifold temperature of 80° F is presented in figure 33. For the same operating conditions, the value of T_g is considerably lower than the value of T_g for the

head-temperature correlation (fig. 29). A lower value of T_g for the heat-rejection correlation would be expected because it represents the effective value for the heat-transfer processes in the entire cylinder, including the contribution of heat from the cylinder barrel, whereas for the head-temperature correlation only the processes in the hot region of the cylinder head are involved. 931

As in the head-temperature correlation, ignition timing was found to have a negligible effect on T_g over the range covered by the variable-ignition-timing device on the engines used, and thus no correction for this parameter is required. Correction of T_g to a dry inlet-manifold temperature other than 80° F is made with equations (2), (3), and (4) or (5).

In order to facilitate the computation of the coolant heat rejection by means of the correlation equation, values of the coolant-property parameter $\mu^{0.21}/k\ Pr^{0.38}$ are presented in figure 34 for various coolant mixtures of ethylene glycol and water over a range of coolant temperature.

Example of Correlation Method

In order to illustrate the use of the correlation equations, the following example is presented.

The maximum cylinder-head temperature between the exhaust valves and the coolant heat rejection are to be determined for the following conditions:

Charge (air plus fuel) flow, lb/sec	2.50
Engine speed, rpm	3000
Fuel-air ratio	0.085
Carburetor-air temperature, °F	80
Temperature drop through aftercooler, °F	60
Exhaust pressure, in. Hg absolute	30
Coolant flow, lb/sec	27.5
Average coolant temperature, °F	240
Coolant composition, ethylene glycol-water, percent by volume	30-70
Accumulated engine running time, hr	over 120
Supercharger gear ratio	low

The average cylinder-head temperature is first evaluated from equation (1) and the maximum cylinder-head temperature then determined from reference to figure 10 in the following manner:

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The dry inlet-manifold temperature is computed from the carburetor-air temperature, the engine speed, and the temperature drop through the aftercooler by means of equation (3)

$$T_m = T_c + \Delta T_s - \Delta T_{aft}$$

$$= T_c + 25.17 \left(\frac{N}{1000}\right)^2 - \Delta T_{aft}$$

$$= 80 + 25.17 \left(\frac{3000}{1000}\right)^2 - 60$$

$$= 246.5^\circ \text{ F}$$

In order to determine the effective cylinder-gas temperature T_g, a value of T_g for a fuel-air ratio of 0.085, an exhaust pressure of 30 inches of mercury absolute, and a dry inlet-manifold temperature of 80° F is first determined from figure 29 to be 1118° F. The correction to T_g for the dry inlet-manifold temperature of 246.5° F is determined from equation (2).

$$\Delta T_g = 0.35 \ (T_m - 80)$$

$$= 0.35 \ (246.5 - 80)$$

$$= 58.3^\circ \text{ F}$$

The value of T_g is then determined by adding the correction for manifold temperature to the value obtained from figure 29.

$$T_g = 1118 + 58.3 = 1176.3^\circ \text{ F}$$

Because a constant value of 0.22 for the Z factor for accumulated engine running time over 120 hours is indicated by figure 30, this constant value is used. The coolant-property parameter $\mu^{0.36}/k \ Pr^{0.33}$ is determined from figure 31 for the specified coolant and coolant temperature and is equal to 445.

Substitution of the values of the various parameters into equation (1)

$$\left(\frac{T_g - T_h}{T_h - T_l}\right)\left[\left(\frac{0.000492}{W_l^{0.36}}\right)\left(\frac{\mu^{0.36}}{k\ Pr^{0.33}}\right) + Z\right] = W_c^{-0.60}$$

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gives the following

$$\left(\frac{1176.3 - T_h}{T_h - 240}\right)\left[\left(\frac{0.000492}{27.5^{0.36}}\right)(445) + 0.22\right] = 2.50^{-0.60}$$

$$T_h = 550.7^\circ\ F$$

For this value of the average cylinder-head temperature, the corresponding value of the maximum cylinder-head temperature between the exhaust valves is found to be 598.5° F (fig. 10).

A comparison of the calculated average cylinder-head temperature between the exhaust valves with that indicated by primary data may be made by referring to the variable-charge-flow data of figure 12 for which all of the conditions of the example except engine running time are duplicated. In this figure, an average cylinder-head temperature of 505° F is indicated for a charge flow of 2.50 pounds per second. The correction of this head temperature from the engine running time of 26 hours for the runs to the engine running time of the example is determined from figure 11 to be 37° F. The correction is added to the value from figure 12.

$$T_h = 505 + 37 = 542^\circ\ F$$

This value of T_h obtained from the primary data is within 9° F of the calculated value.

In determining the coolant heat rejection by means of equation (6), the manifold temperature is 246.5° F, as previously calculated. The initial value of T_g is determined from figure 33 to be 742° F for a fuel-air ratio of 0.085, an exhaust pressure of 30.0 inches of mercury absolute, and a dry inlet-manifold temperature of 80° F. The correction to T_g for the dry inlet-manifold temperature of 246.5° F is also the same as previously calculated, 58.3° F.

The value of T_g is then determined by adding the correction for manifold temperature to the value from figure 33.

$$T_g = 742 + 58.3 = 800.3^\circ \text{ F}$$

The coolant-property parameter $\mu^{0.21}/k\ Pr^{0.38}$ is determined from figure 34 for the specified coolant and coolant temperature as equal to 1450.

Substitution of the values of the various parameters into equation (6)

$$0.49\left(\frac{T_g - T_l}{H}\right) - \left(\frac{0.000154}{W_l^{0.21}}\right)\left(\frac{\mu^{0.21}}{k\ Pr^{0.38}}\right) - 0.32 = W_c^{-0.97}$$

gives the following

$$0.49\left(\frac{800.3 - 240}{H}\right) - \left(\frac{0.000154}{27.5^{0.21}}\right)(1450) - 0.32 = 2.50^{-0.97}$$

$$H = 326 \text{ Btu/sec}$$

A comparison of this calculated value of coolant heat rejection with that indicated by primary data may be made by referring to the variable-charge-flow data of figure 20 for which all conditions of the example are duplicated. In this figure, a value of 324 Btu per second for the coolant heat rejection is indicated at a charge flow of 2.50 pounds per second. This value of H obtained from the primary data is within 1 percent of the calculated value.

SUMMARY OF RESULTS

The following results were obtained from an investigation of the cooling characteristics of a multicylinder liquid-cooled engine of 1650-cubic-inch displacement:

1. For the range of conditions investigated, both the cylinder-head temperature between the exhaust valves and the coolant heat rejection increased:

(a) Considerably with charge flow - The increases amounted to about 25° F and 40 Btu per second, respectively, for an increase in charge flow of 0.5 pound per second.

(b) A small amount with measured inlet-manifold temperature (9° F and 20 Btu/sec, respectively, for an increase of 100° F in inlet manifold temperature).

(c) Rapidly as the fuel-air ratio was leaned to a value of about 0.067 and then decreased with further leaning - For a decrease in fuel-air ratio from 0.115 to 0.067, the increases amounted to about 70° F and 80 Btu per second, respectively.

(d) Slightly as the ignition timing was increased from about 36° to 52° B.T.C. (about 10° F and 12 Btu/sec, respectively) - Decreasing the ignition timing from 36° to 20° B.T.C. resulted in negligible change in both the cylinder-head temperature and coolant heat rejection.

(e) With exhaust pressure - The maximum increases occurred at an engine speed of 3000 rpm and a fuel-air ratio of 0.067 and amounted to about 70° F and 90 Btu per second, respectively, for an increase in exhaust pressure from 8 to 60 inches of mercury absolute.

2. The cylinder-head temperatures increased and the coolant heat rejection decreased:

(a) With an increase in the ethylene-glycol concentration of the coolant - For a change in the ethylene-glycol concentration of the coolant from 100-percent water to 70-percent ethylene glycol and 30-percent water, the cylinder-head temperature increased about 33° F at a coolant flow of 250 gallons per minute and about 47° F at a coolant flow of 50 gallons per minute, and the coolant heat rejection decreased about 11 Btu per second for all coolant flows.

(b) With a decrease in coolant flow for constant average coolant temperature and nonboiling coolant conditions - For a decrease in coolant flow from 200 to 50 gallons per minute, the maximum increase in the cylinder-head temperature occurred for the coolant mixture containing the greatest percentage of ethylene glycol used (70 percent) and amounted to about 25° F; the corresponding decrease in the coolant heat rejection was the same for all coolants tested and amounted to about 10 Btu per second. For constant coolant-outlet temperature and boiling coolant conditions, the change in cylinder-head temperature and coolant heat rejection with coolant flow was negligible.

(c) With an increase in the average coolant temperature - For an increase in average coolant temperature of 100° F, the cylinder-head temperature increased about 60° F and the coolant heat rejection decreased about 50 Btu per second.

3. A linear relation existed between both the average temperatures for all the thermocouple locations and the maximum and average temperature for each location.

4. In general, the cylinder-head temperature increased slightly with coolant-outlet pressure; the maximum increase amounted to about 15° F for an increase in coolant pressure of 20 pounds per square inch. No significant variation of the coolant heat rejection with coolant-outlet pressure could be detected.

5. The temperatures in the exhaust side and center of the cylinder heads increased about 45° F for an increase in engine running time from 15 to 115 hours, probably as a result of a progressive scale buildup on the coolant passages in these regions. No significant variation of the coolant heat rejection with engine running time could be detected.

6. All the cylinder-head-temperature data and all of the coolant-heat-rejection data, except those for which boiling of the coolant was most severe, were satisfactorily correlated by means of the NACA correlation method, which is based on the theory of heat transfer by forced convection. Application of this correlation permits the prediction of both the cylinder-head temperatures and the coolant heat rejection for a wide range of operating conditions.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio. July 18, 1949.

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2. Pinkel, Benjamin, Manganiello, Eugene J., and Bernardo, Everett: Cylinder-Temperature Correlation of a Single-Cylinder Liquid-Cooled Engine. NACA Rep. 853, 1946.

3. Povolny, John H., and Chelko, Louis J.: Cylinder-Head Temperatures and Coolant Heat Rejections of a Multicylinder, Liquid-Cooled Engine of 1710-Cubic-Inch Displacement. NACA TN 1606, 1948.

4. Lundin, Bruce T., Povolny, John H., and Chelko, Louis J.: Correlation of Cylinder-Head Temperatures and Coolant Heat Rejections of a Multicylinder, Liquid-Cooled Engine of 1710-Cubic-Inch Displacement. NACA Rep. 931, 1949.

5. Koffel, William K., and Biermann, Arnold E.: A Unit Laboratory Engine Oil System Providing for a Remote Indication of Oil Flow and Oil Consumption Together with Blow-by Measurement. NACA TN 952, 1944.

TABLE I - SUMMARY OF

Variable	Engine power (bhp)	Engine speed (rpm)	Manifold pressure (in. Hg absolute)	Charge flow (lb/sec)	Fuel-air ratio	Carburetor-air temperature (°F)
Charge flow	763-1378	3000	35-55	1.71-2.84	0.085	80
Engine speed	560-766	1800-3200	31-44	1.43	0.080	80
Fuel-air ratio	739-865	2400	38-40	1.68	0.059-0.115	80
Carburetor-air temperature	1115	2700	45-55	2.23	0.081	-57 - 198
Ignition timing	658-726 739-807 777-876	2400 2700 3000	36 36 37	1.46 1.68 1.90	0.080 .080 .080	83 82 82
Exhaust pressure	616-789 802-977	2400 3000	34-45 38-48	1.50 2.04	0.063-0.095 .063- .095	82 82
Coolant conditions	920 1085 1115 1530 1540 1700 1670 2000 780 745	2700 2700 2700 3000 3000 3000 3000 3200 2700 2700	40 46 46 61 61 68 67 78 36 34	1.87 2.16 2.19 3.22 3.22 3.55 3.46 4.30 1.57 1.57	0.080 .077 .077 .093 .090 .090 .090 .092 .080 .080	83 80 80 80 81 81 82 80 82 82
Engine running time	1100	2700	46	2.20	0.080	80

[a]Constant coolant-outlet temperature.

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EXPERIMENTAL CONDITIONS

Inlet-manifold temperature (°F)		Ignition timing (deg B.T.C.)	Exhaust pressure (in. Hg absolute)	Coolant gylcol-water (percent by volume)	Coolant flow (gal/min)	Average coolant temperature (°F)	Coolant-outlet pressure (lb/sq in. gage)	Engine	Engine running time (hr)
Measured	Calculated								
175	246	46	29	30-70	202	240	35	A	25.5-27.7
116-259	162-336	46	29	30-70	202	240	35	A	23.0-25.5
116-122	165-192	46	29	30-70	202	240	35	A	27.7-31.1
89-293	125-380	46	29	30-70	202	240	35	A	16.1-23.0
140	184	20-52	29	30-70	167	240	35	A	107.0 ---
142	190	20-52	29	30-70	167	240	35	A	---- ----
142	196	20-52	29	30-70	167	240	35	A	--- 119.0
158	200	45	8-61	30-70	205	240	35	B	12.1-23.8
156	212	45	8-61	30-70	201	240	35	B	24.2-31.2
140	190	46	29	0-100, 30-70, 70-30	51-253	240	35	A	92.5-100.5
157	212	46	29	30-70	50-200	[a]250	20,30,40	A	42.5-46.5
136	190	46	29	30-70	49-200	[a]275	28,35,45	A	57.6-61.2
170	243	46	29	30-70	49-200	[a]250	20,30,40	A	46.5-50.6
170	240	46	29	30-70	51-205	[a]275	28,35,45	A	66.1-70.0
176	246	46	29	30-70	50-205	[a]250	20,30,40	A	50.6-54.3
173	245	45	29	30-70	76-202	[a]275	28,35,45	A	77.8-81.1
176	260	46	29	30-70	52-201	[a]250	20,30,40	A	70.1-75.5
140	185	45	29	30-70	205	168-278	35	B	4.5-8.9
135	190	45	29	70-30	200	170-274	35	B	31.2-34.0
135	190	46	29	30-70	200	[a]245	35	A	14.0-113.2

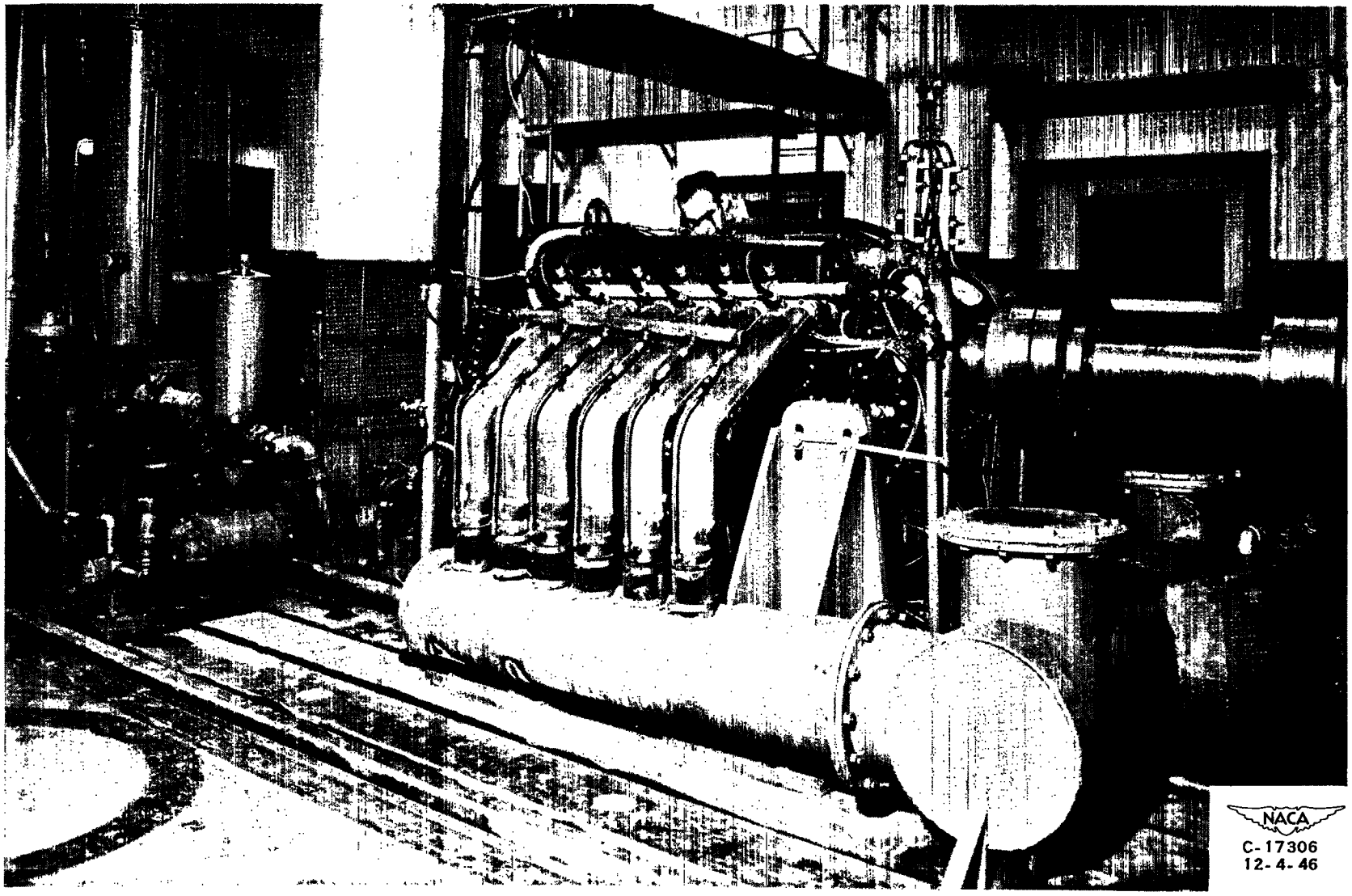

Figure 1. - Engine mounted for cooling investigation.

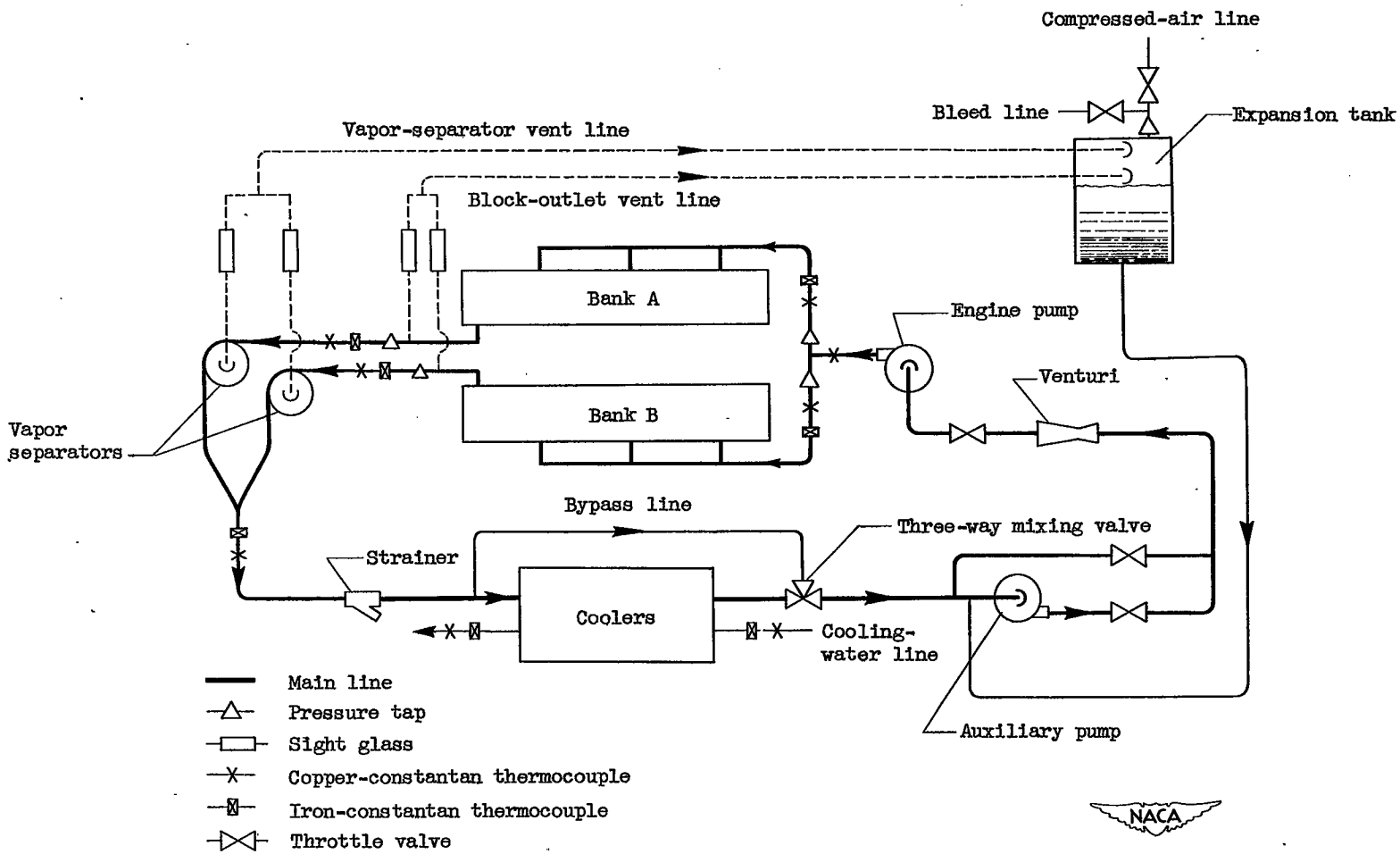

Figure 2. - Diagramatic sketch of engine coolant system.

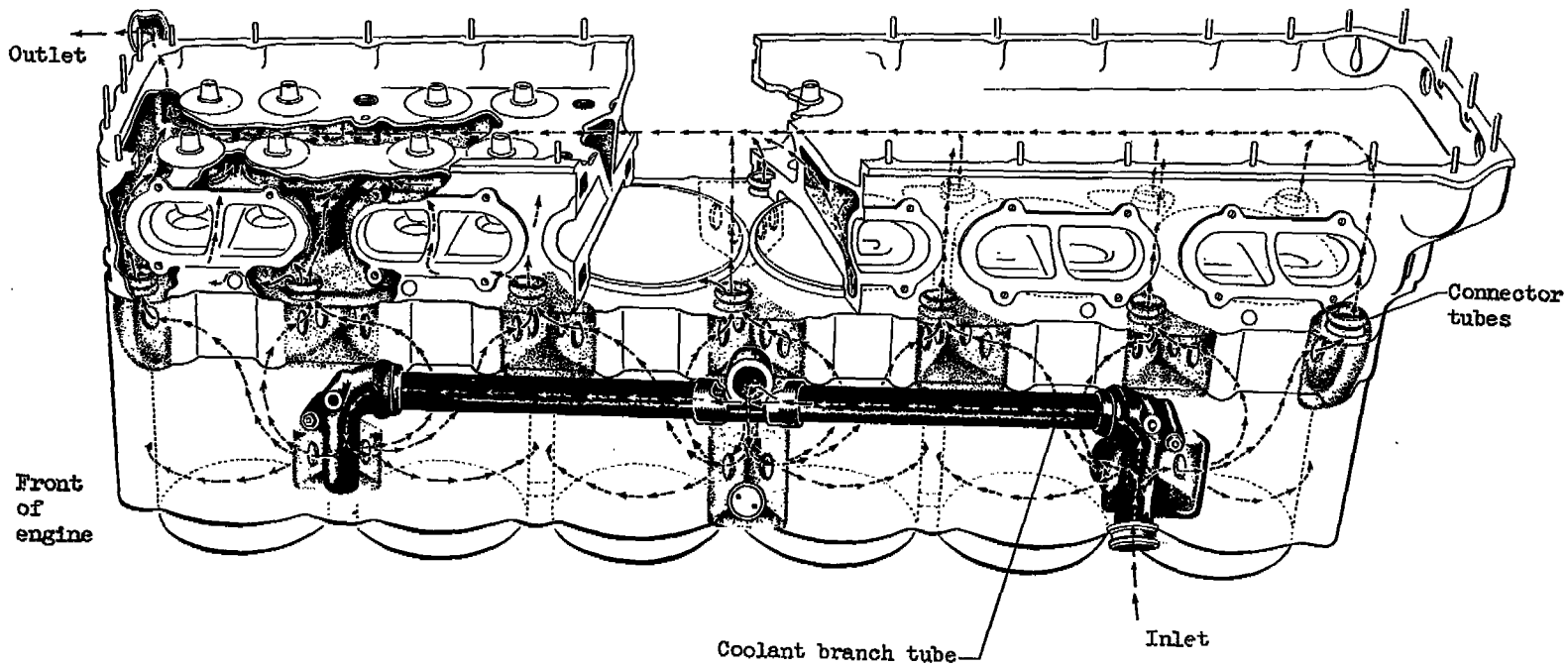

Figure 3. - Sketch of cylinder bank showing network of coolant passages.

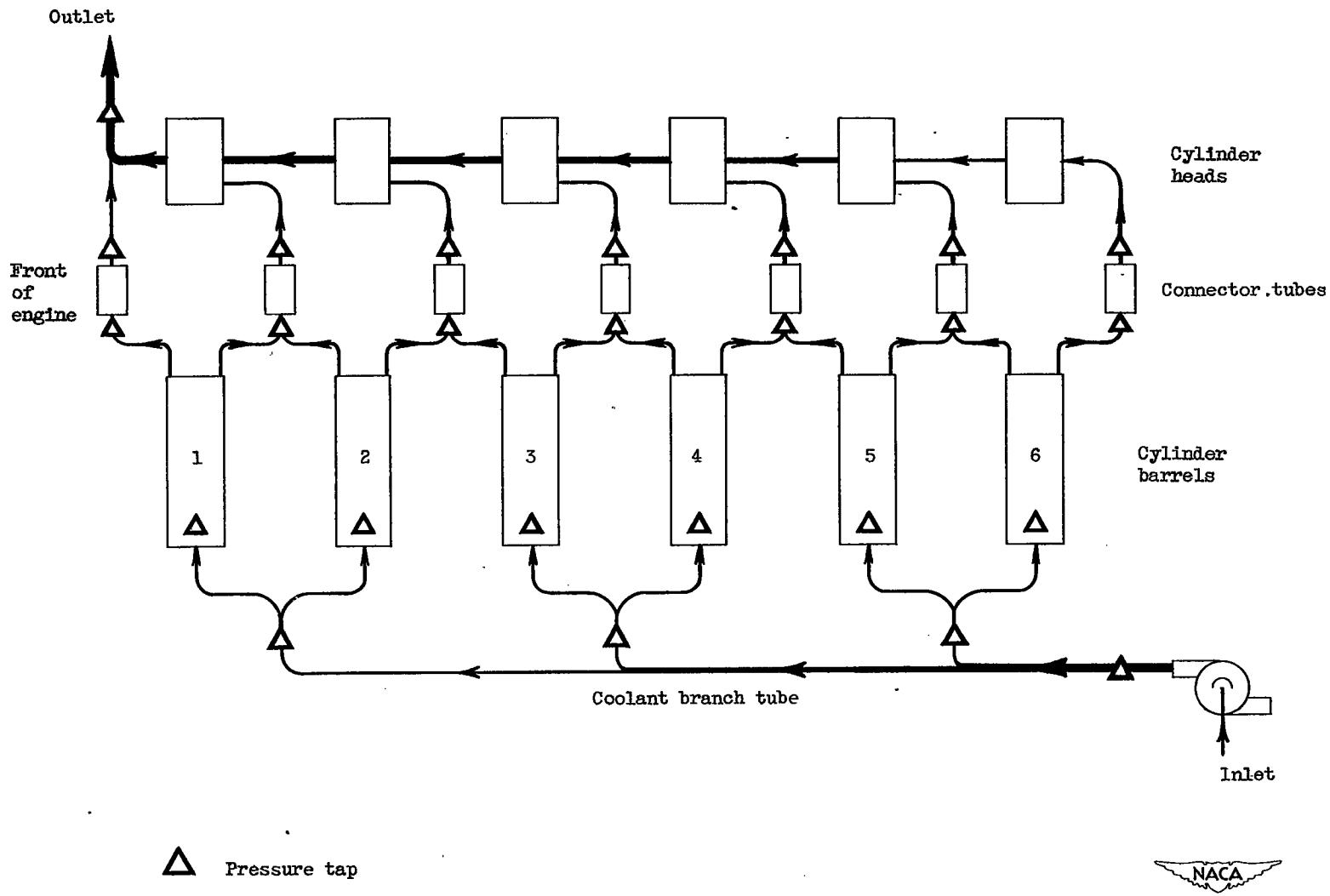

Figure 4. - Schematic diagram of engine-coolant flow passages.

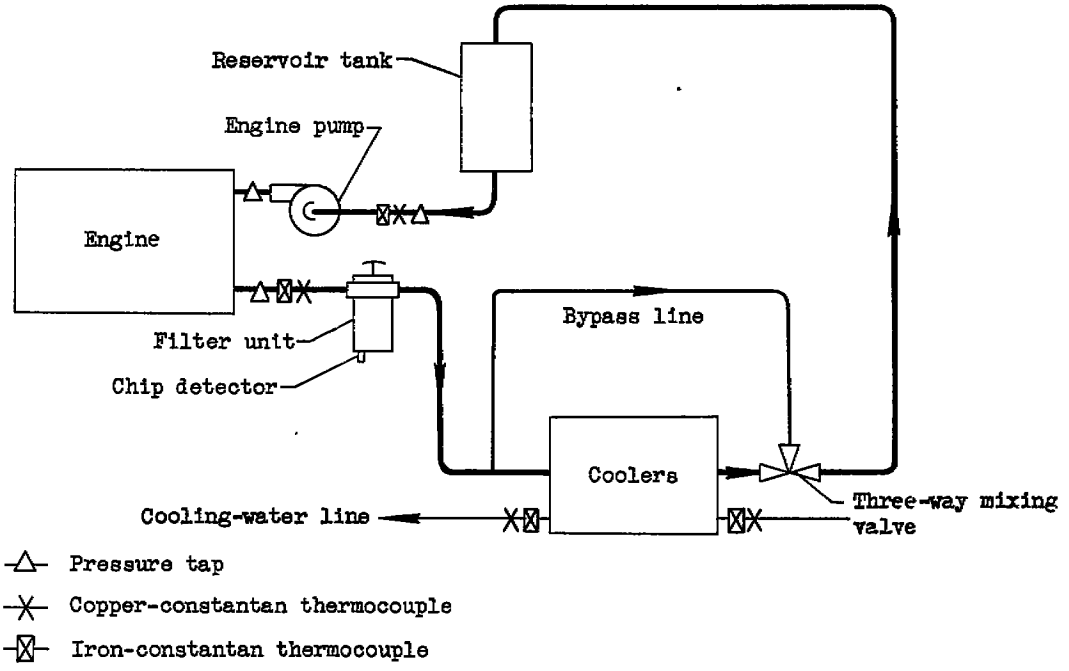

Figure 5. - Schematic diagram of lubricating oil system.

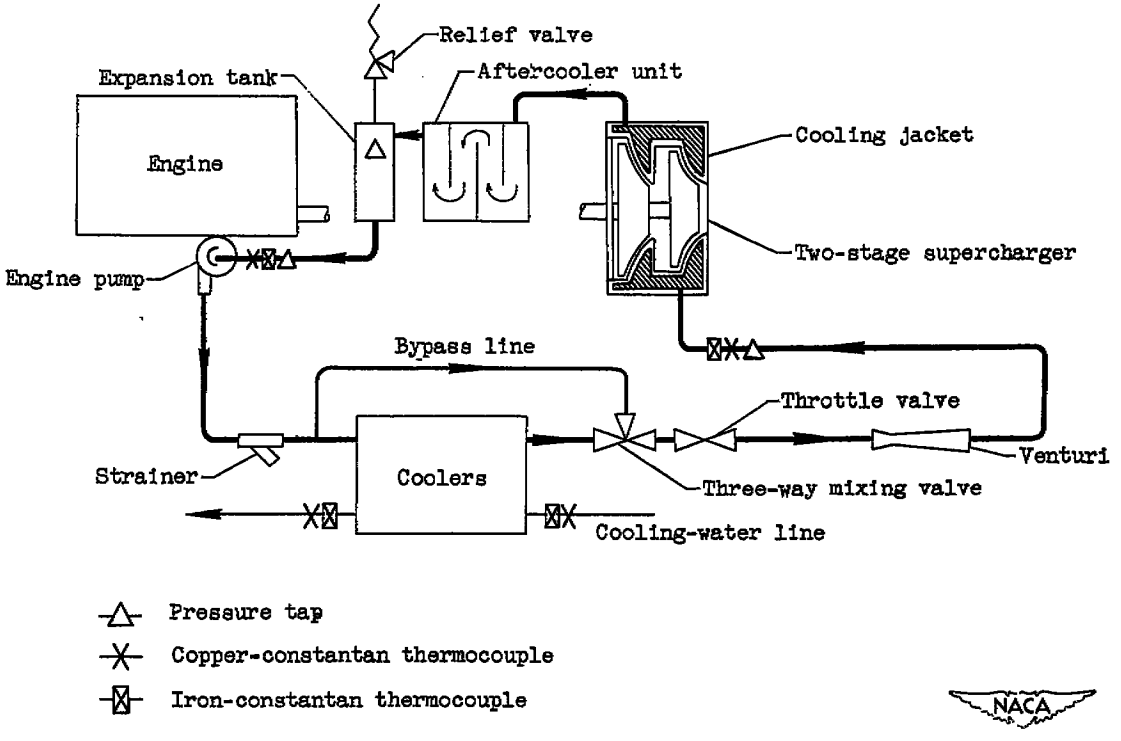

Figure 6. - Schematic diagram of aftercooler coolant system.

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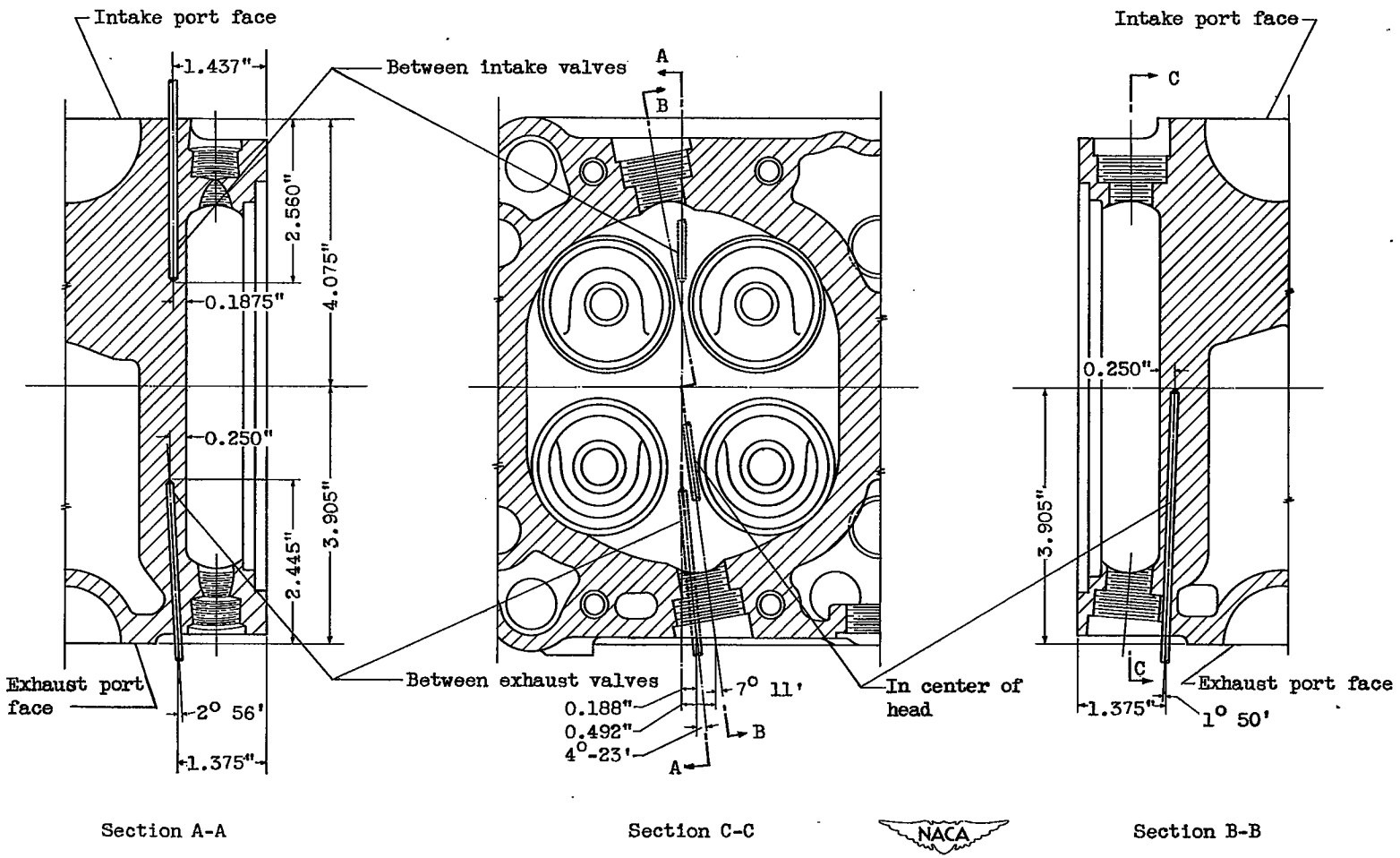

Figure 7. - Cylinder-head thermocouple installation.

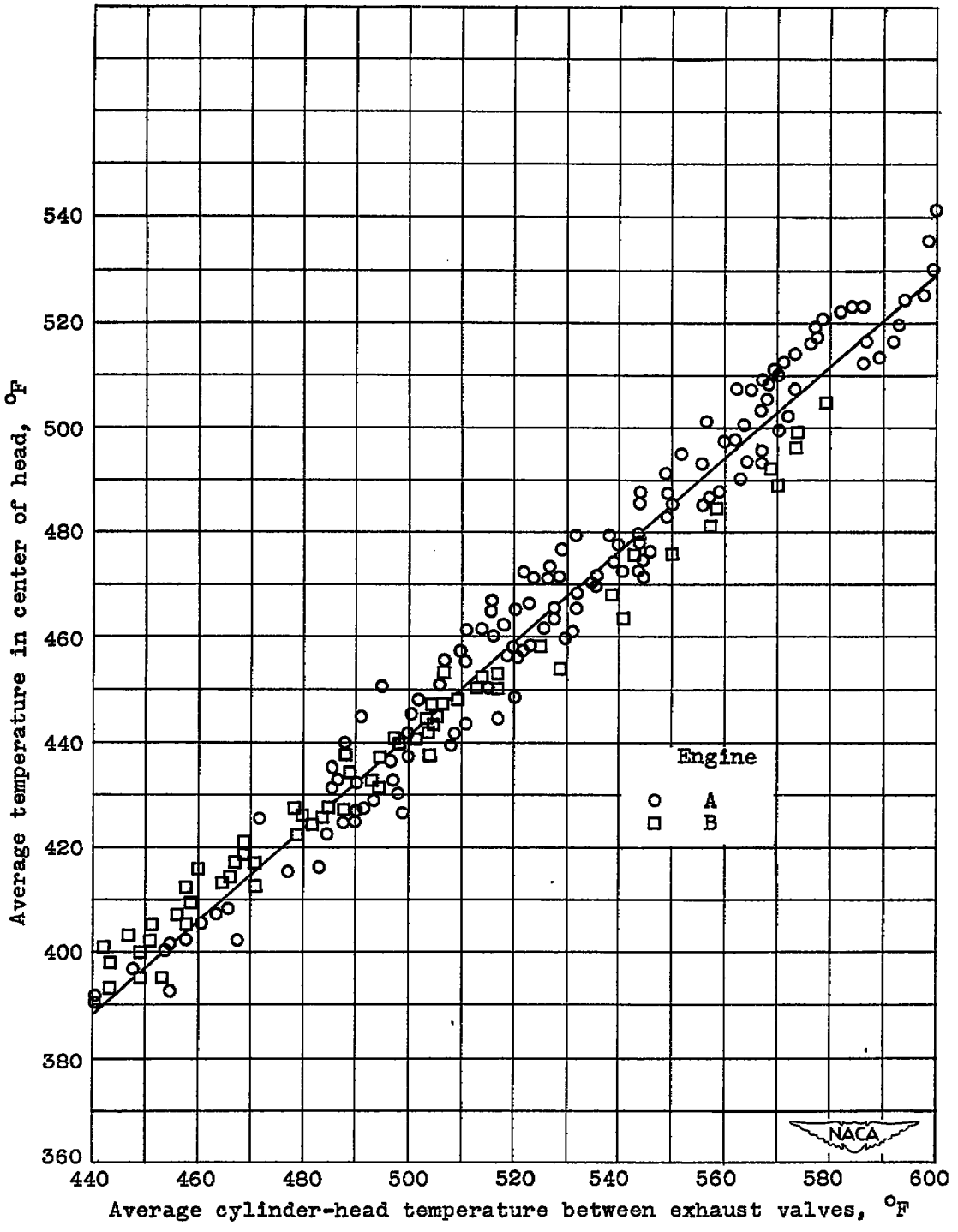

Figure 8. - Relation between average temperature in center of head and average temperature in cylinder head between exhaust valves.

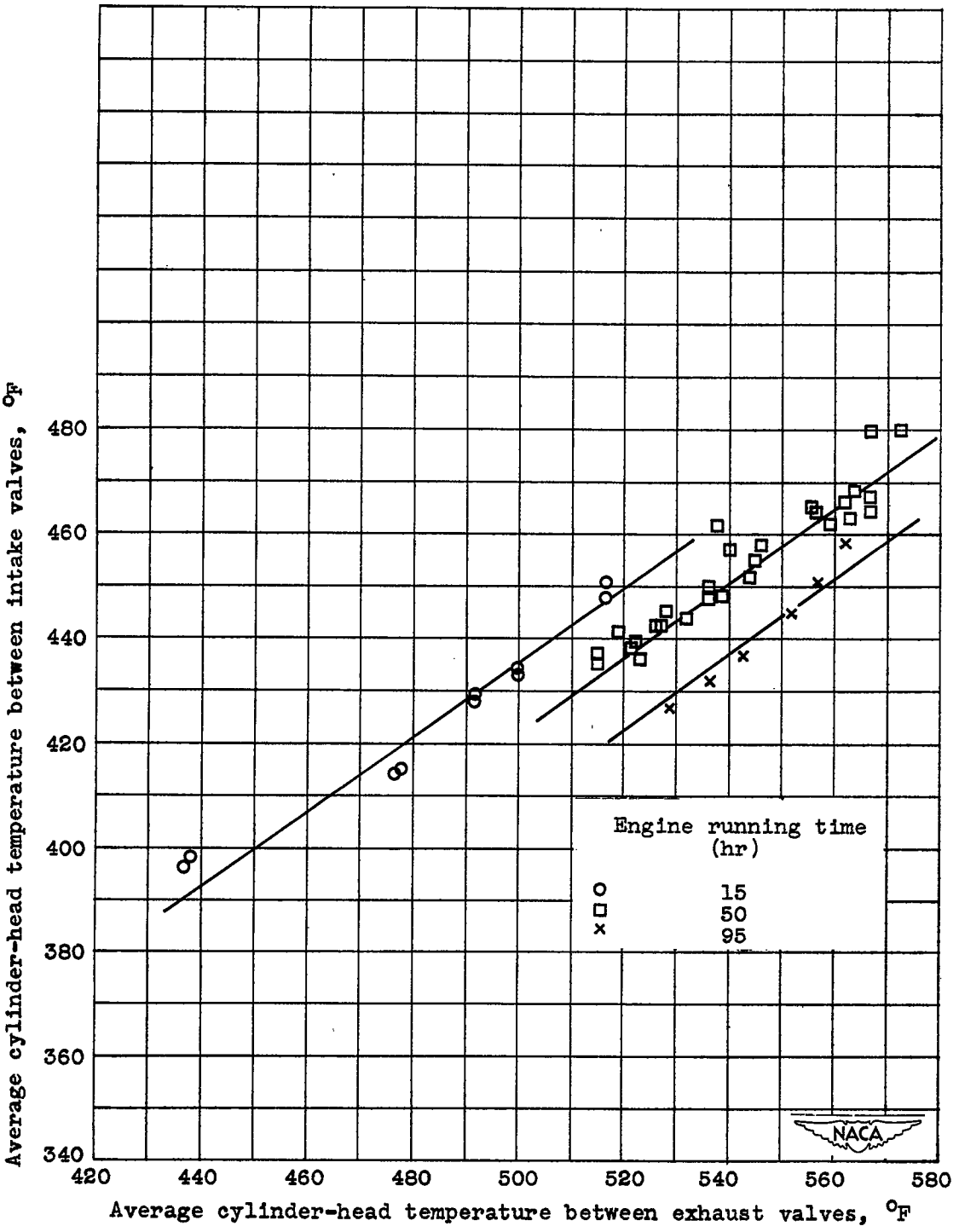

Figure 9. - Relation between average temperature in cylinder head between intake valves and average temperature in cylinder head between exhaust valves. Engine A.

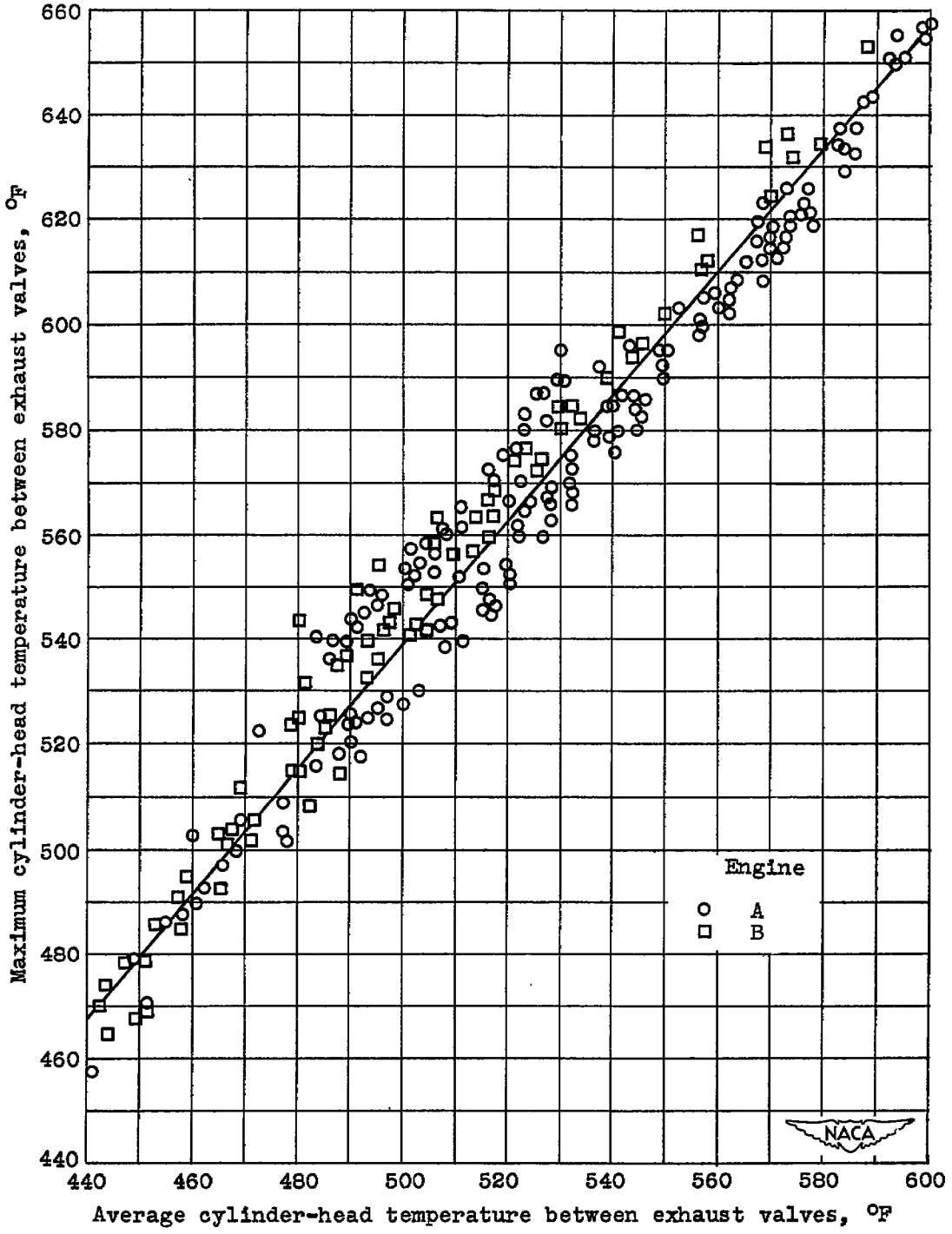

Figure 10. - Relation between maximum and average temperatures in cylinder head between exhaust valves.

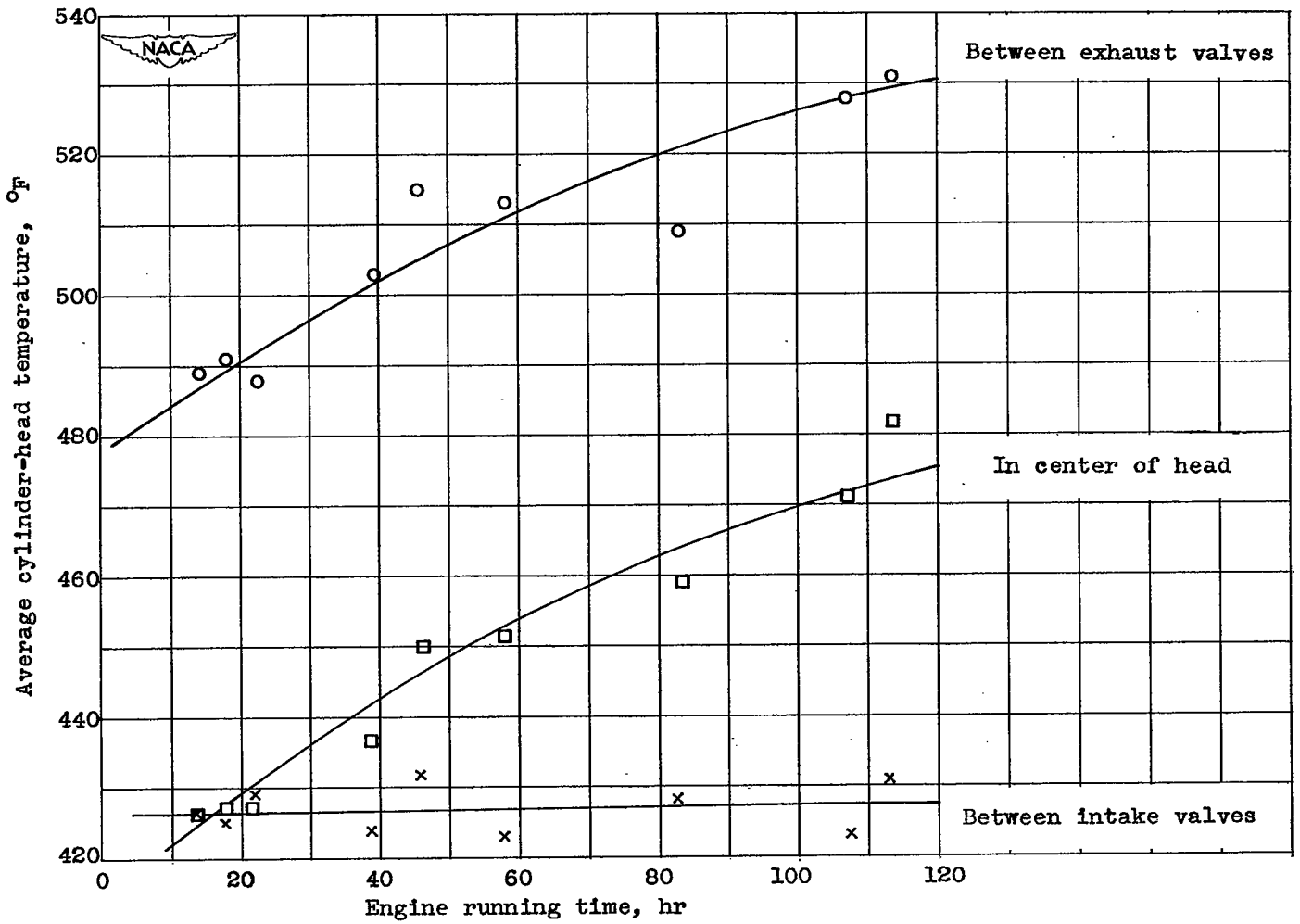

Figure 11. - Variation of average cylinder-head temperatures with engine running time. Engine power, 1100 brake horsepower; engine speed, 2700 rpm; manifold pressure, 46 inches mercury absolute; charge flow, 2.20 pounds per second; fuel-air ratio, 0.080; measured inlet-manifold temperature, 135° F; coolant, 30-70 ethylene glycol and water; coolant flow, 200 gallons per minute; coolant-outlet temperature, 245° F; coolant-outlet pressure, 35 pounds per square inch gage; engine A.

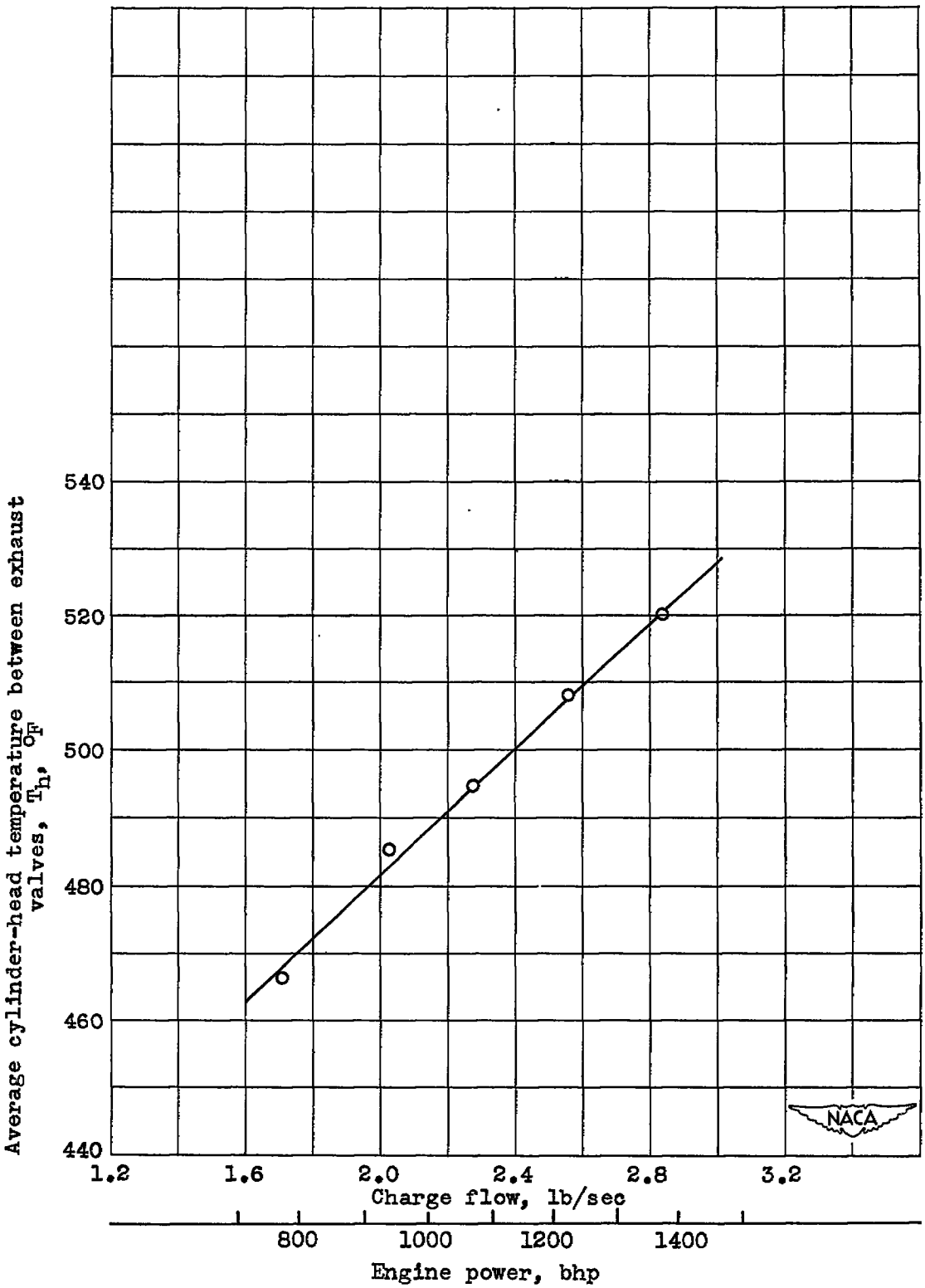

Figure 12. - Variation of average cylinder-head temperature between exhaust valves with charge flow. Engine speed, 3000 rpm; manifold pressure, 35 to 55 inches mercury absolute; fuel-air ratio, 0.085; measured inlet-manifold temperature, 175° F; engine A.

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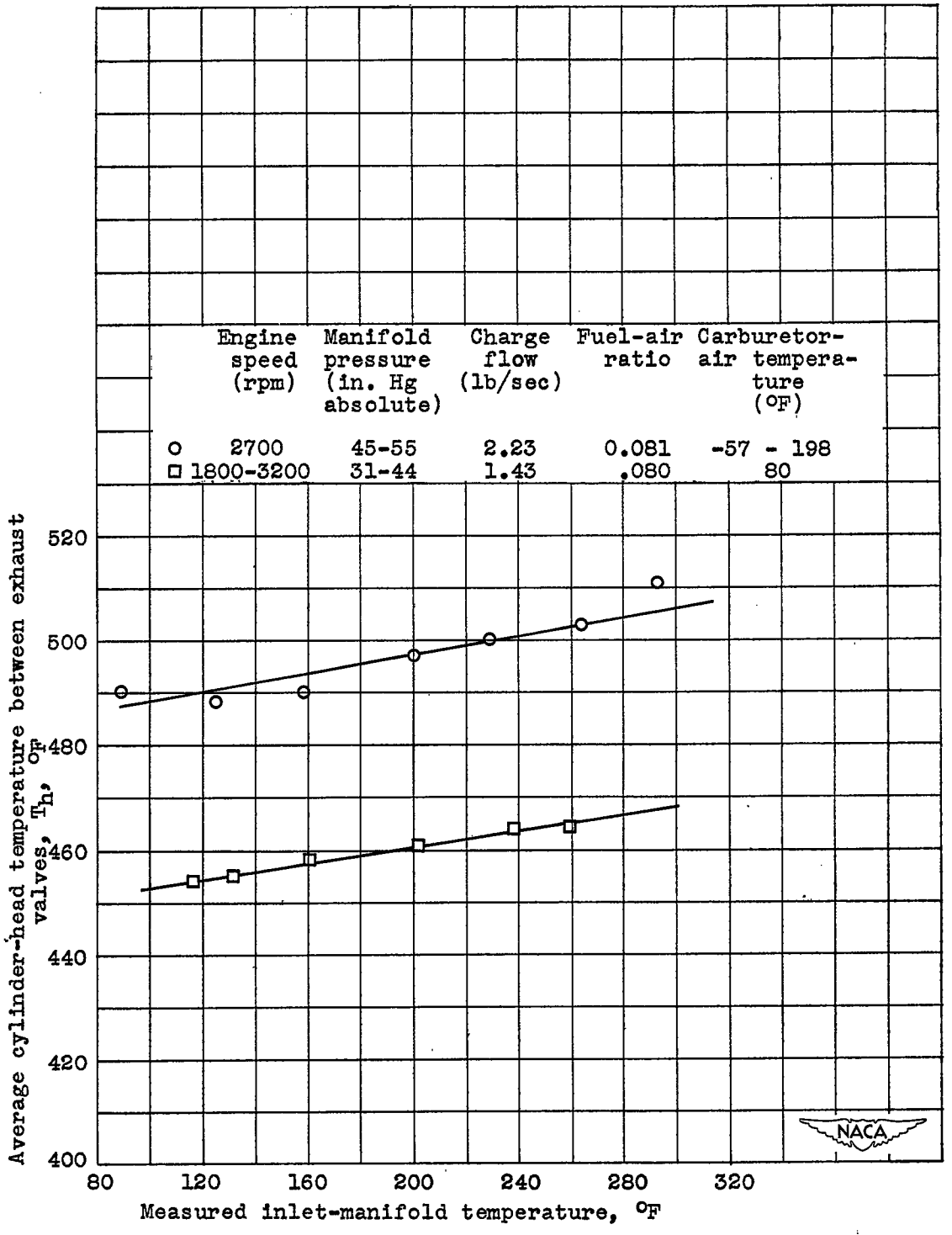

Figure 13. - Variation of average cylinder-head temperature between exhaust valves with measured inlet-manifold temperature. Engine A.

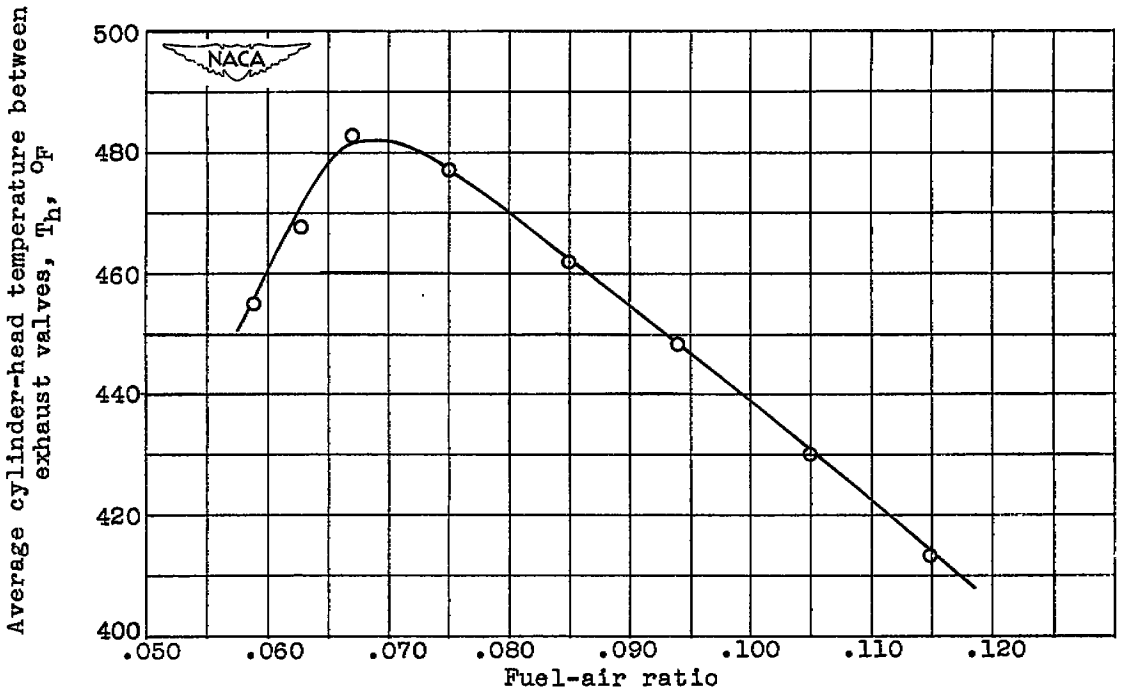

Figure 14. - Variation of average cylinder-head temperature between exhaust valves with fuel-air ratio. Engine speed, 2400 rpm; manifold pressure, 38 to 40 inches mercury absolute; charge flow, 1.68 pounds per second; measured inlet-manifold temperature, 116° to 122° F; engine A.

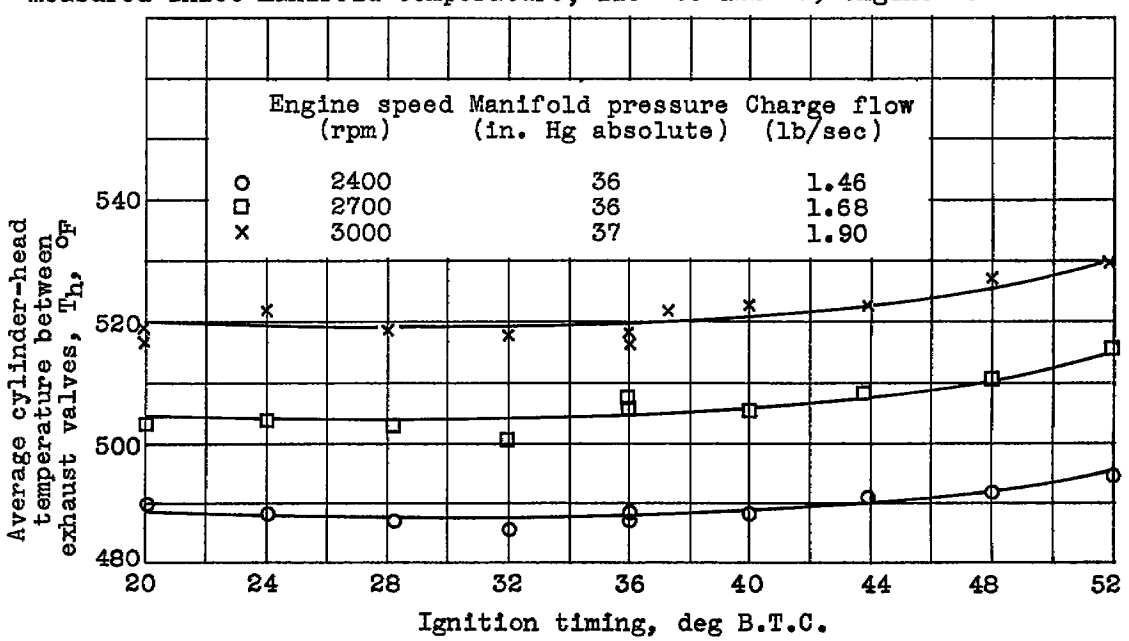

Figure 15. - Variation of average cylinder-head temperature between exhaust valves with ignition timing. Fuel-air ratio, 0.080; measured inlet-manifold temperature, 140° to 142° F; engine A.

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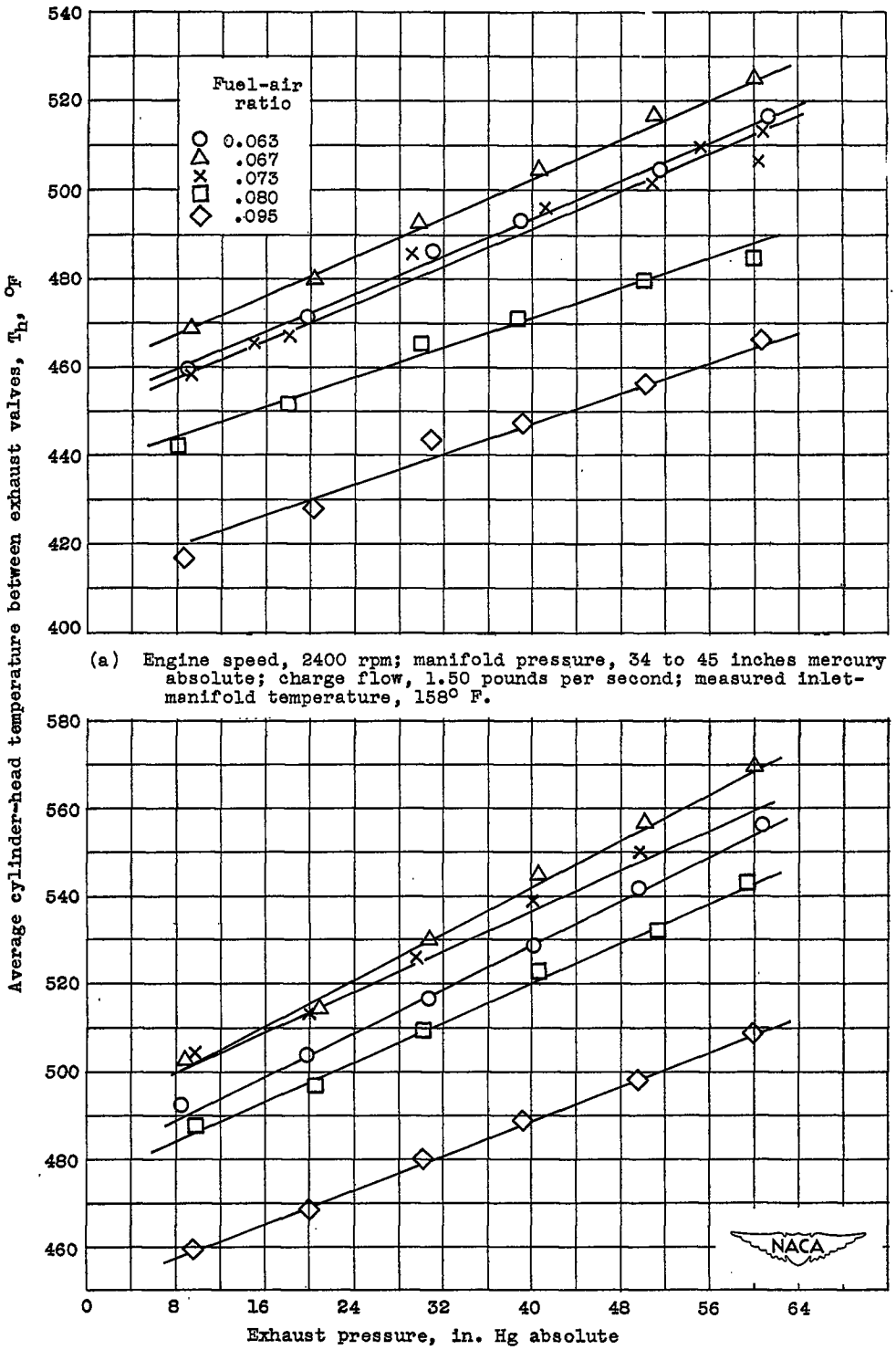


(a) Engine speed, 2400 rpm; manifold pressure, 34 to 45 inches mercury absolute; charge flow, 1.50 pounds per second; measured inlet-manifold temperature, 158° F.

(b) Engine speed, 3000 rpm; manifold pressure, 38 to 48 inches mercury absolute; charge flow, 2.04 pounds per second; measured inlet-manifold temperature, 156° F.

Figure 16. - Variation of average cylinder-head temperature between exhaust valves with exhaust pressure. Engine B.

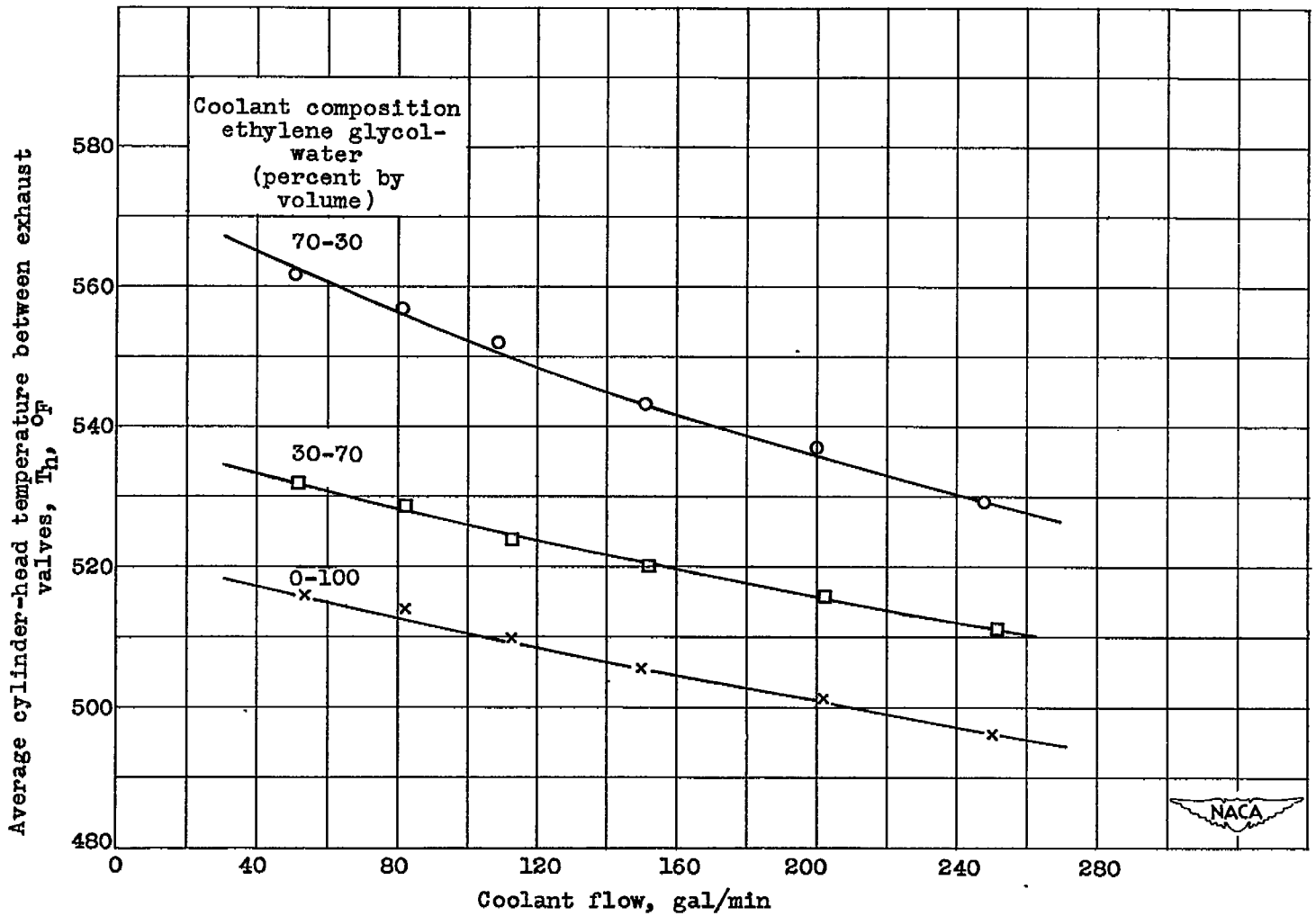
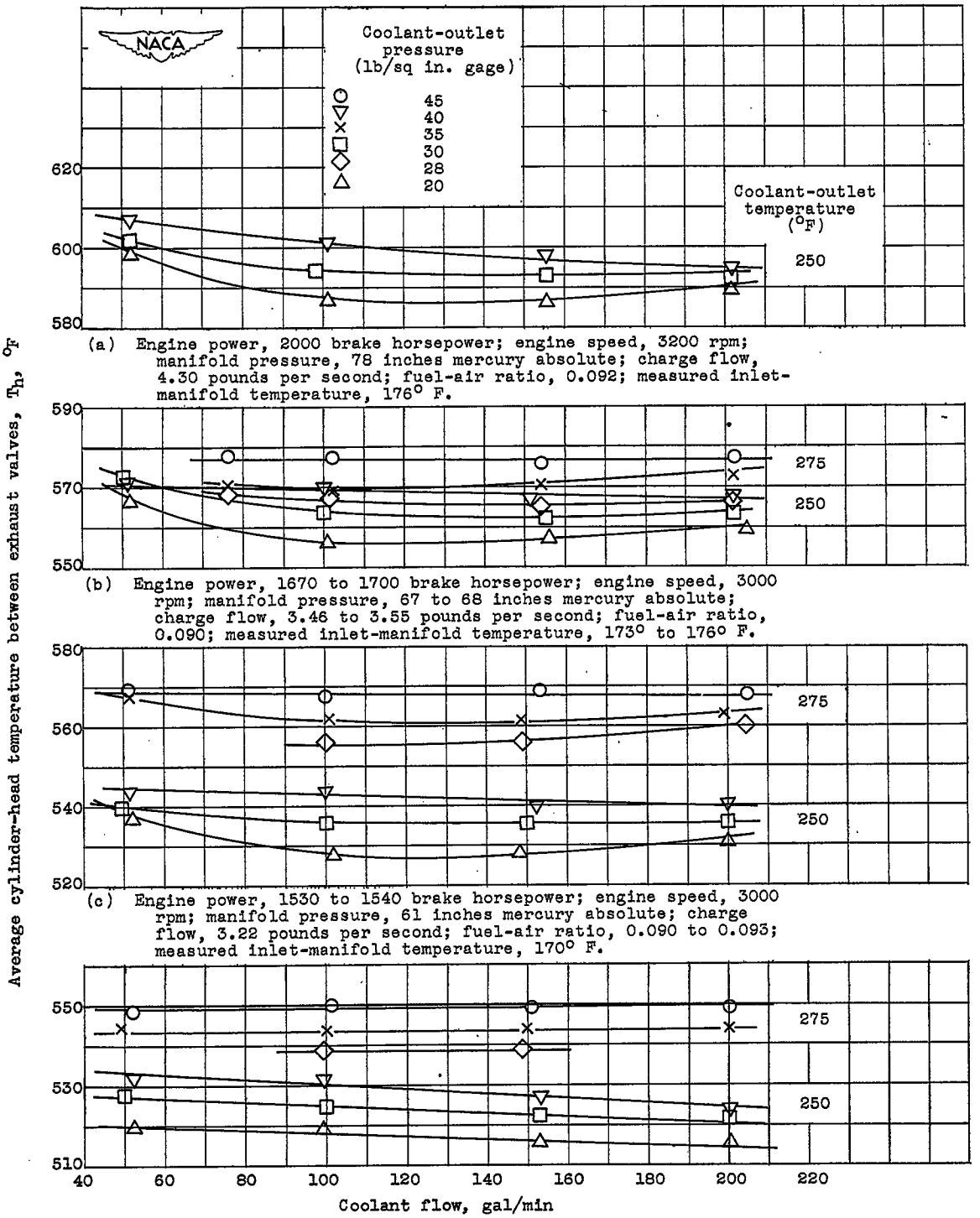


Figure 17. - Variation of average cylinder-head temperature between exhaust valves with coolant flow for several coolants. Engine speed, 2700 rpm; manifold pressure, 40 inches mercury absolute; charge flow, 1.87 pounds per second; fuel-air ratio, 0.080; measured inlet-manifold temperature, 140° F; average coolant temperature, 240° F; coolant-outlet pressure, 35 pounds per square inch gage; engine A.



(a) Engine power, 2000 brake horsepower; engine speed, 3200 rpm; manifold pressure, 78 inches mercury absolute; charge flow, 4.30 pounds per second; fuel-air ratio, 0.092; measured inlet-manifold temperature, 176° F.

(b) Engine power, 1670 to 1700 brake horsepower; engine speed, 3000 rpm; manifold pressure, 67 to 68 inches mercury absolute; charge flow, 3.46 to 3.55 pounds per second; fuel-air ratio, 0.090; measured inlet-manifold temperature, 173° to 176° F.

(c) Engine power, 1530 to 1540 brake horsepower; engine speed, 3000 rpm; manifold pressure, 61 inches mercury absolute; charge flow, 3.22 pounds per second; fuel-air ratio, 0.090 to 0.093; measured inlet-manifold temperature, 170° F.

(d) Engine power, 1085 to 1115 brake horsepower; engine speed, 2700 rpm; manifold pressure, 46 inches mercury absolute; charge flow, 2.16 to 2.19 pounds per second; fuel-air ratio, 0.077; measured inlet-manifold temperature, 136° to 157° F.

Figure 18. - Variation of average cylinder-head temperature between exhaust valves with coolant flow. Coolant, 30-70 ethylene glycol and water; engine A.

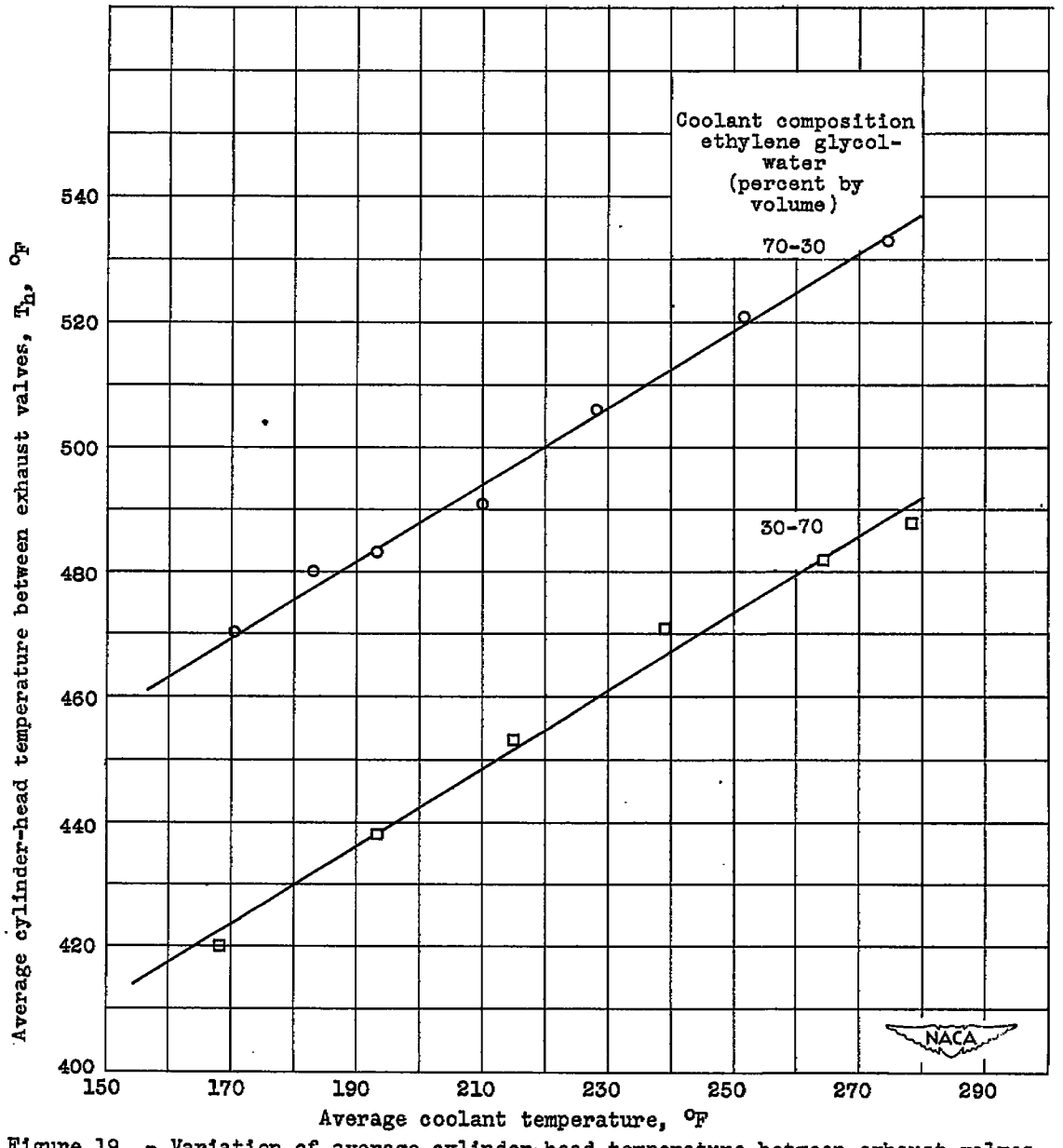


Figure 19. - Variation of average cylinder-head temperature between exhaust valves with average coolant temperature. Engine speed, 2700 rpm; manifold pressure, 34 to 36 inches mercury absolute; charge flow, 1.57 pounds per second; fuel-air ratio, 0.080; measured inlet-manifold temperature, 135° to 140° F; coolant flow, 200 to 205 gallons per minute; coolant-outlet pressure, 35 pounds per square inch gage; engine B.

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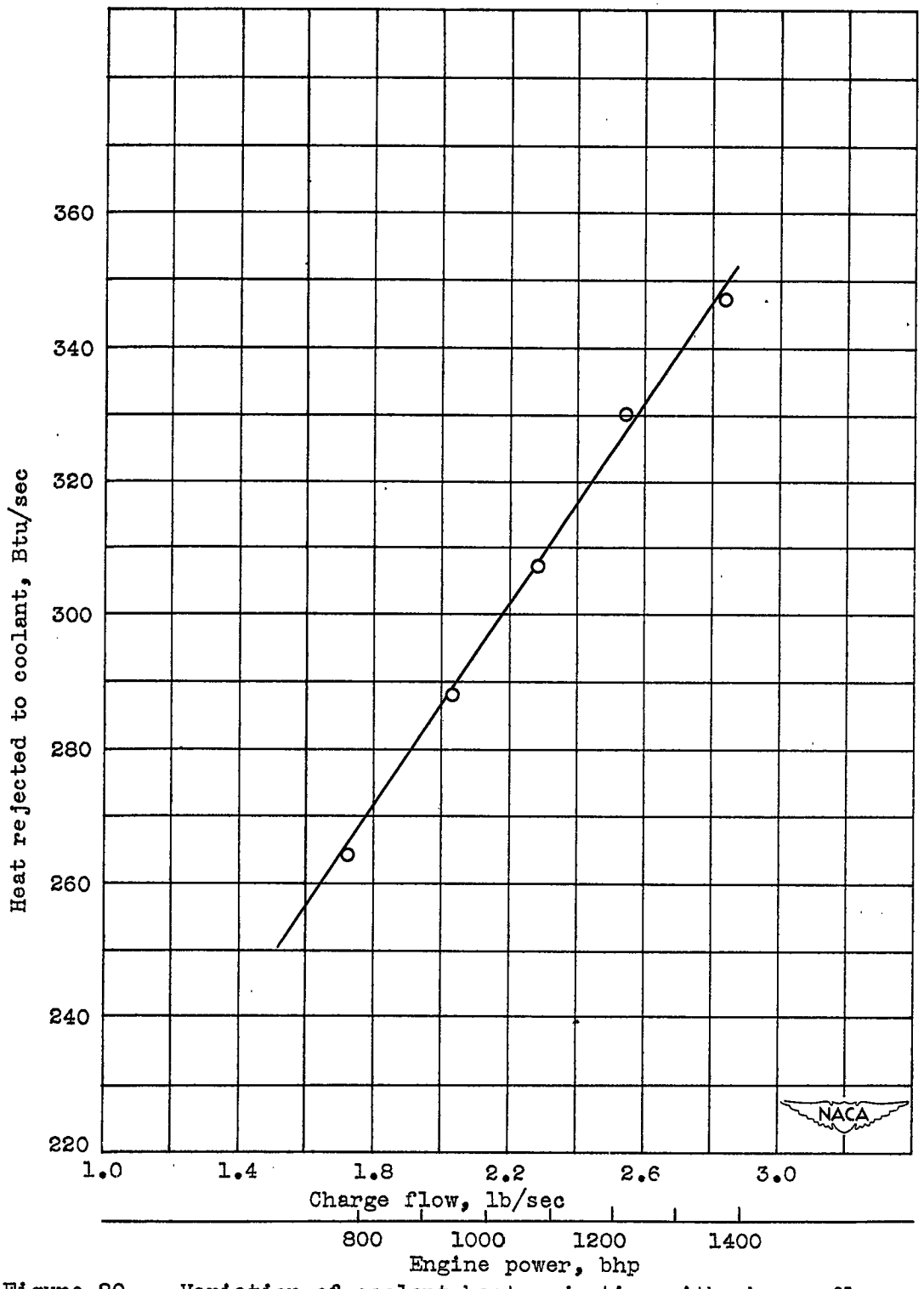

Figure 20. - Variation of coolant heat rejection with charge flow. Engine speed, 3000 rpm; manifold pressure, 35 to 55 inches mercury absolute; fuel-air ratio, 0.085; measured inlet-manifold temperature, 175° F; engine A.

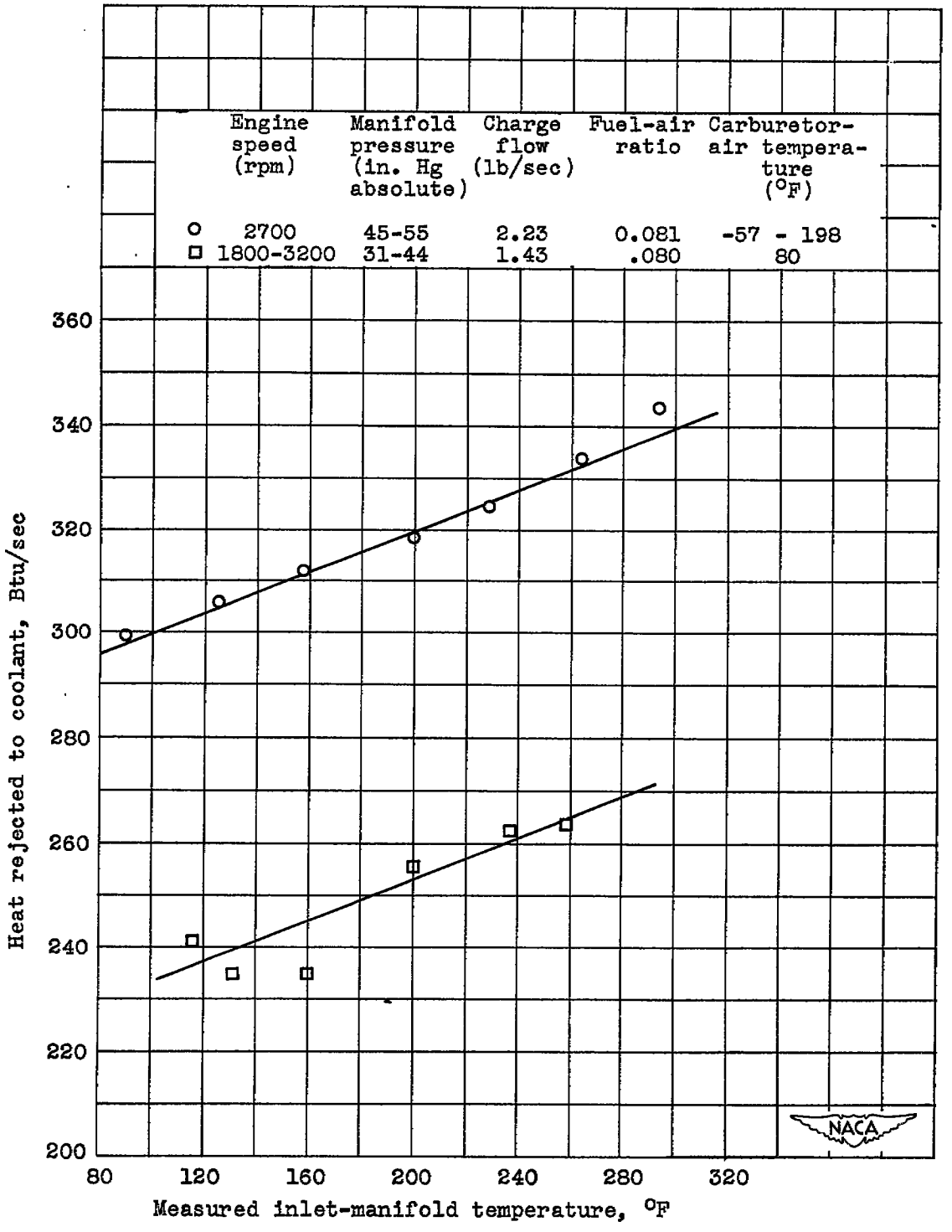

Figure 21. - Variation of coolant heat rejection with measured inlet-manifold temperature. Engine A.

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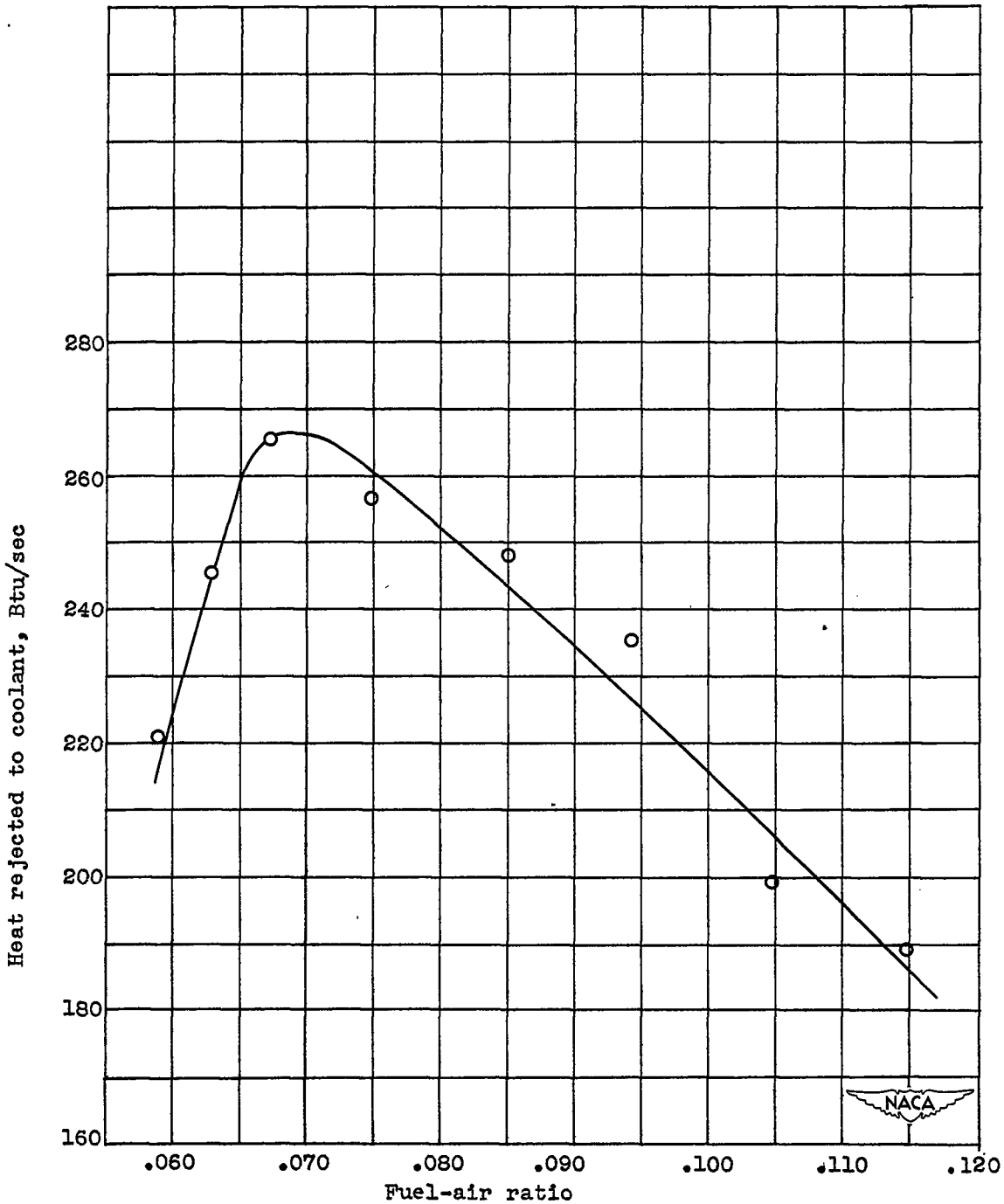


Figure 22. - Variation of coolant heat rejection with fuel-air ratio. Engine speed, 2400 rpm; manifold pressure, 38 to 40 inches mercury absolute; charge flow, 1.68 pounds per second; measured inlet-manifold temperature, 116° to 122° F; engine A.

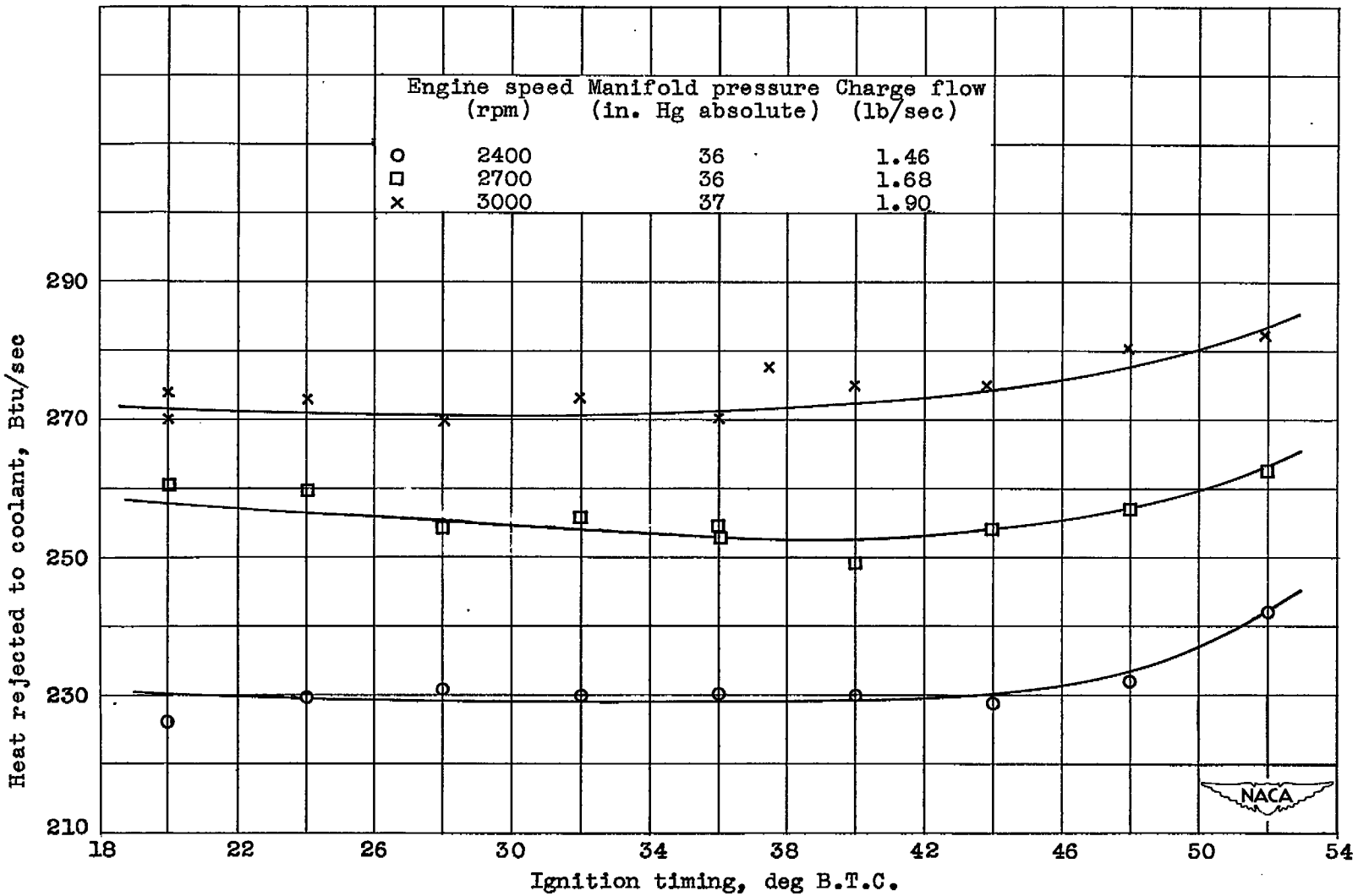
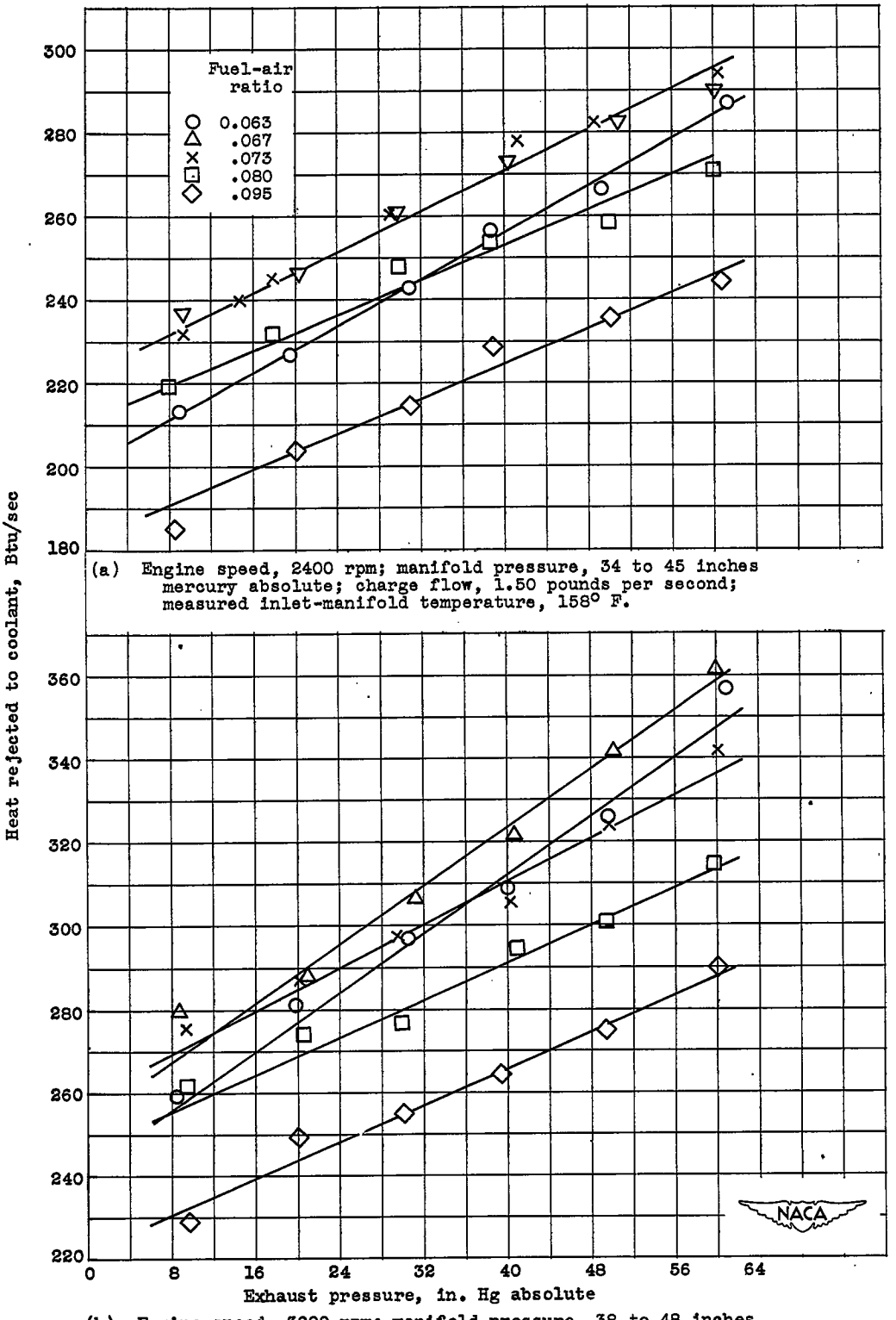


Figure 23. - Variation of coolant heat rejection with spark timing. Fuel-air ratio, 0.080; measured inlet-manifold temperature, 140° to 142° F; engine A.



(a) Engine speed, 2400 rpm; manifold pressure, 34 to 45 inches mercury absolute; charge flow, 1.50 pounds per second; measured inlet-manifold temperature, 158° F.

(b) Engine speed, 3000 rpm; manifold pressure, 38 to 48 inches mercury absolute; charge flow, 2.04 pounds per second; measured inlet-manifold temperature, 156° F.

Figure 24. - Variation of coolant heat rejection with exhaust pressure. Engine B.

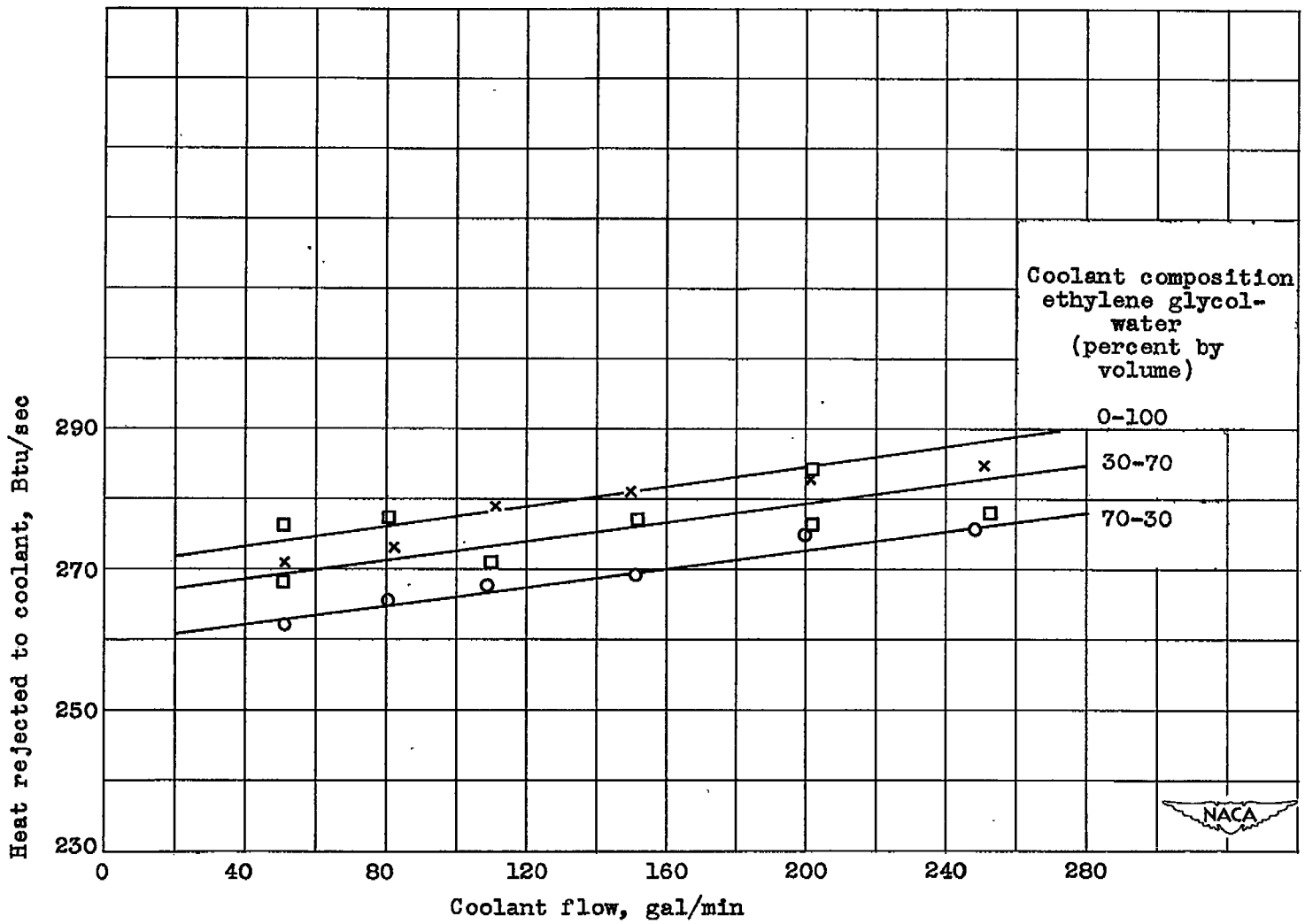


Figure 25. - Variation of coolant heat rejection with coolant flow for several coolants. Engine speed, 2700 rpm; manifold pressure, 40 inches mercury absolute; charge flow, 1.87 pounds per second; fuel-air ratio, 0.080; measured inlet-manifold temperature, 140° F; average coolant temperature, 240° F; coolant-outlet pressure, 35 pounds per square inch gage; engine A.

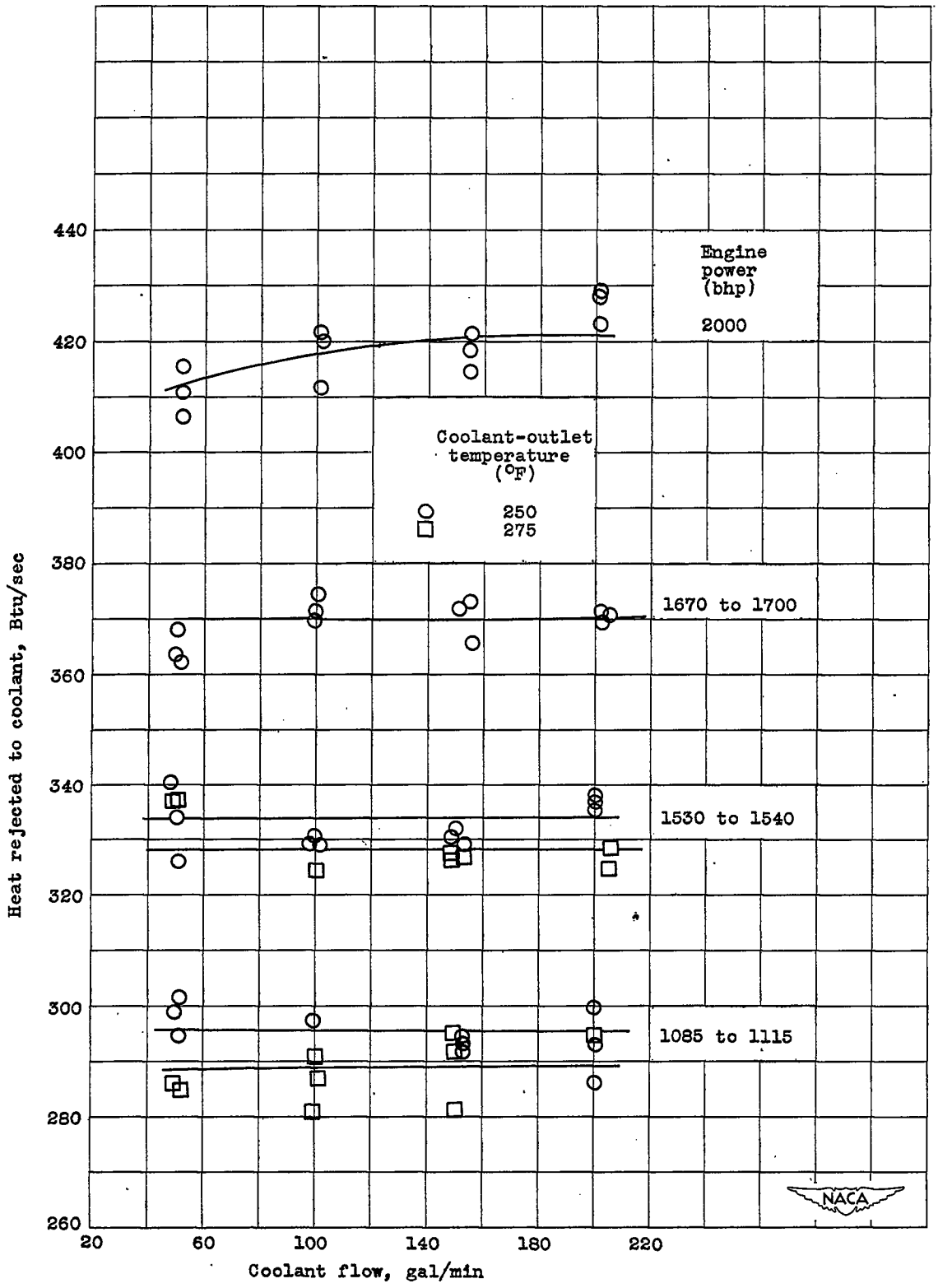


Figure 26. - Variation of coolant heat rejection with coolant flow. Coolant, 30-70 ethylene glycol and water; engine A.

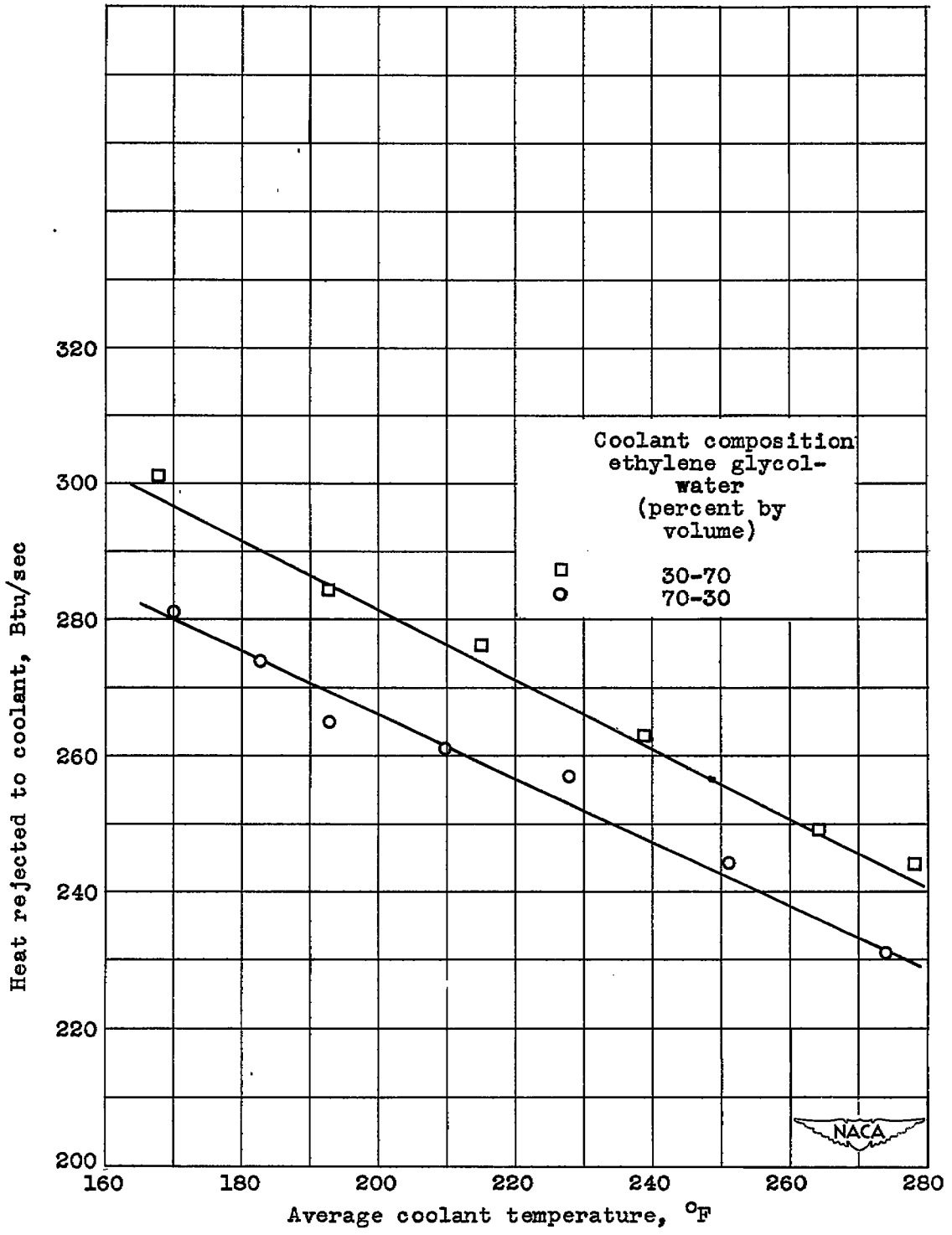


Figure 27. - Variation of coolant heat rejection with average coolant temperature. Engine speed, 2700 rpm; manifold pressure, 34 to 36 inches mercury absolute; charge flow, 1.57 pounds per second; fuel-air ratio, 0.080; measured inlet-manifold temperature, 135° to 140° F; coolant flow, 200 to 205 gallons per minute; coolant-outlet pressure, 35 pounds per square inch gage; engine B.

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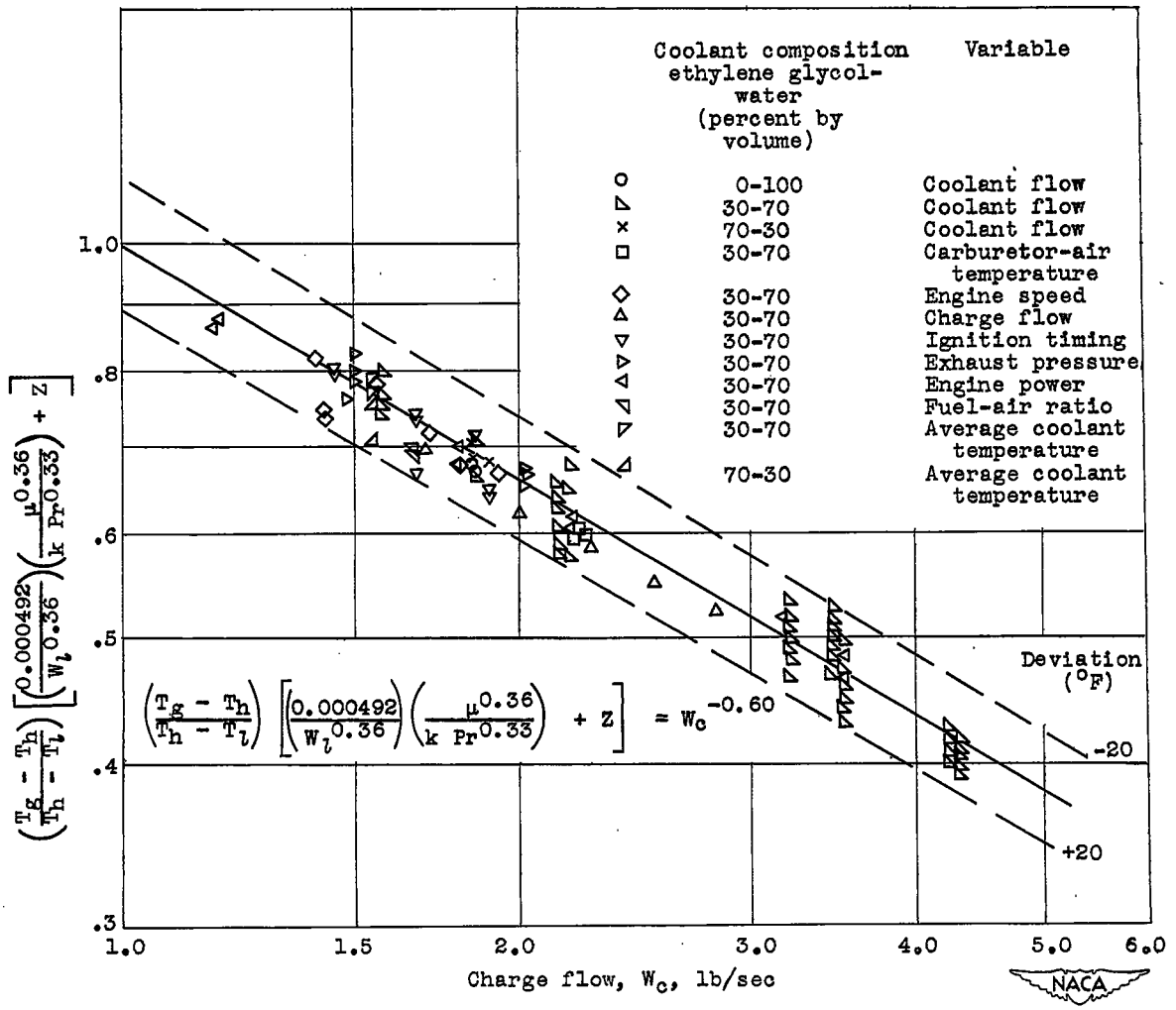

Figure 28. - Correlation of cylinder-head-temperature data.

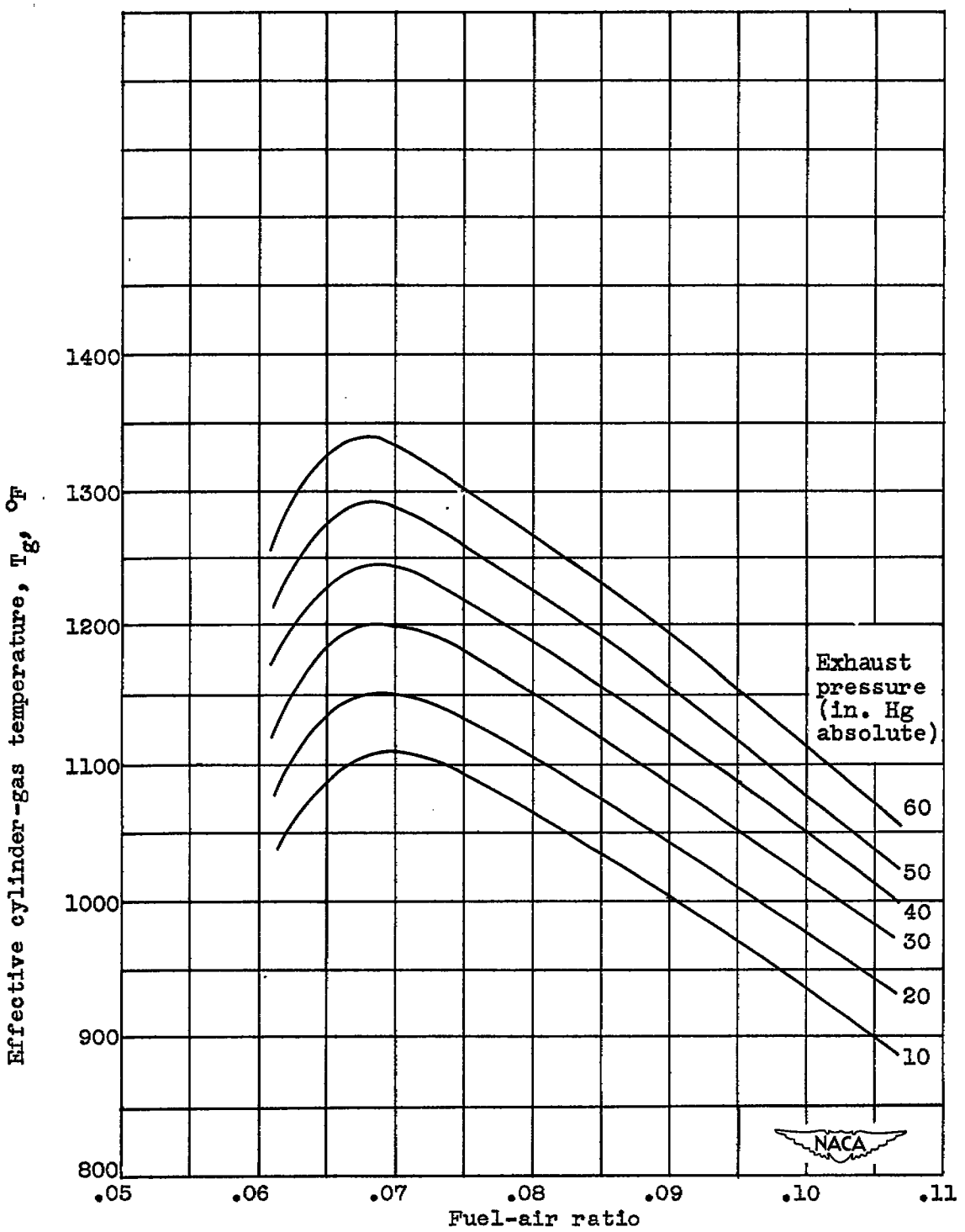

Figure 29. - Variation of effective cylinder-gas temperature with fuel-air ratio for various exhaust pressures for use in cylinder-head-temperature correlation. Data corrected to a calculated dry inlet-manifold temperature of 80° F.

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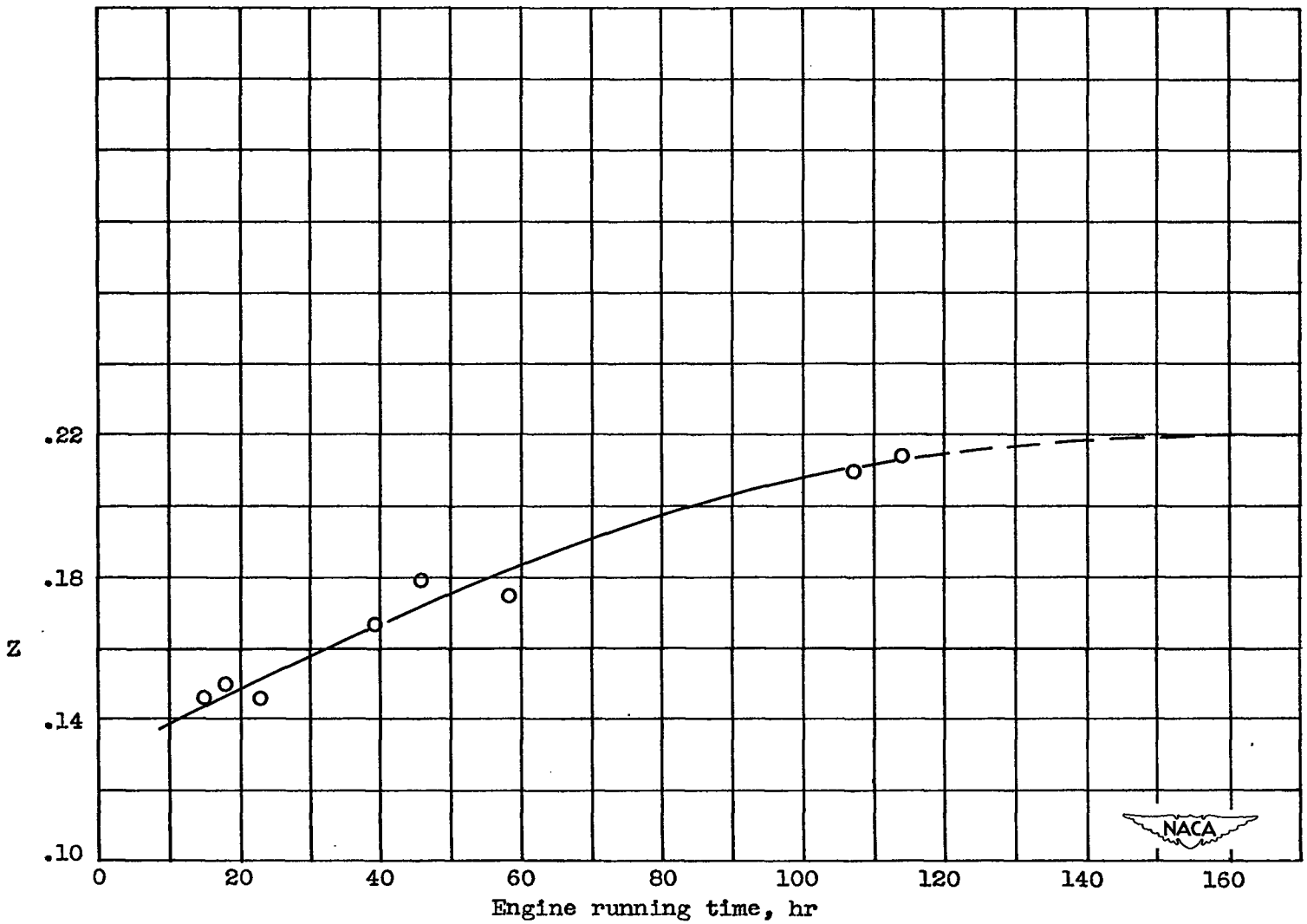

Figure 30. - Variation of Z factor with engine running time.

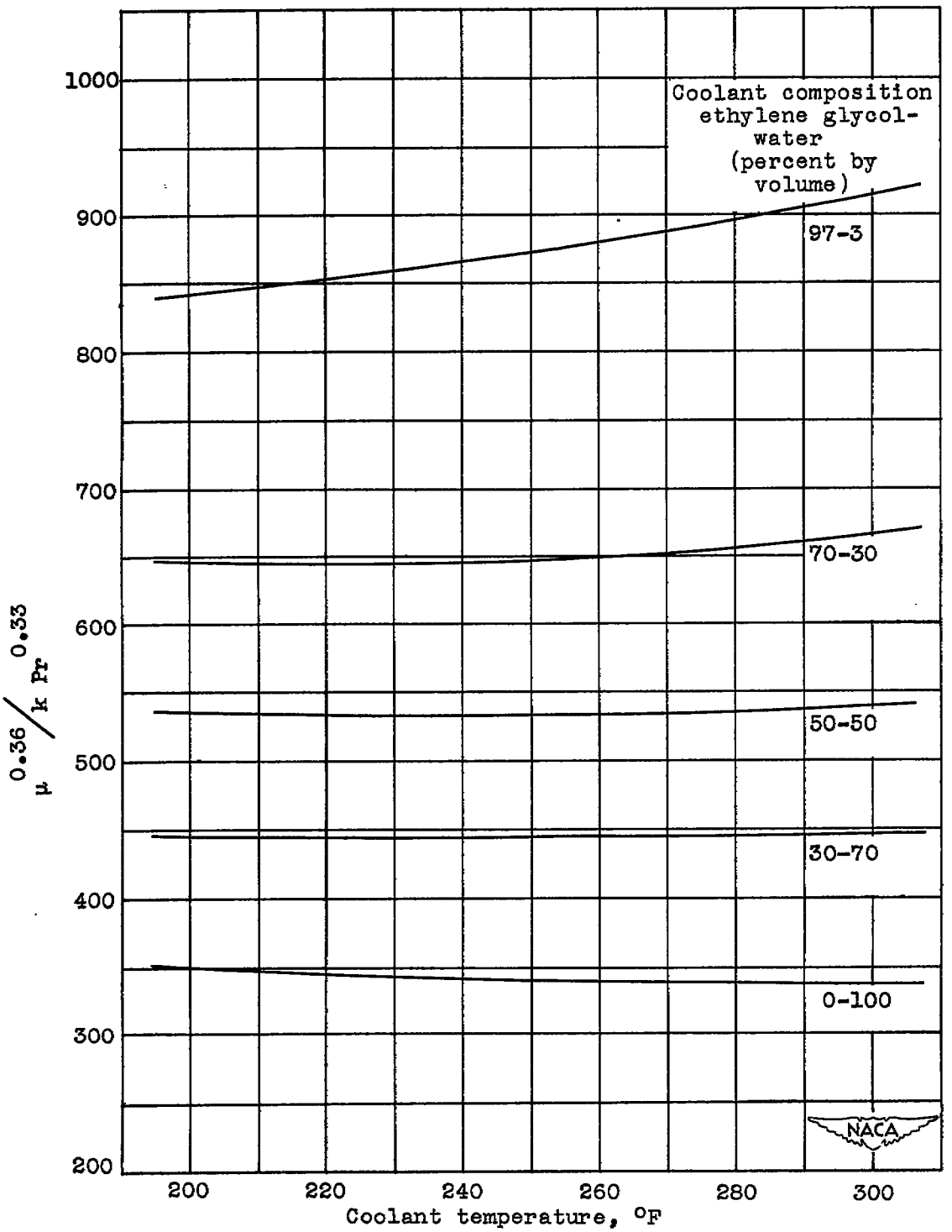

Figure 31. - Variation of coolant-property parameter $\mu^{0.36}/k\ Pr^{0.33}$ with coolant temperature for various aqueous ethylene-glycol solutions.

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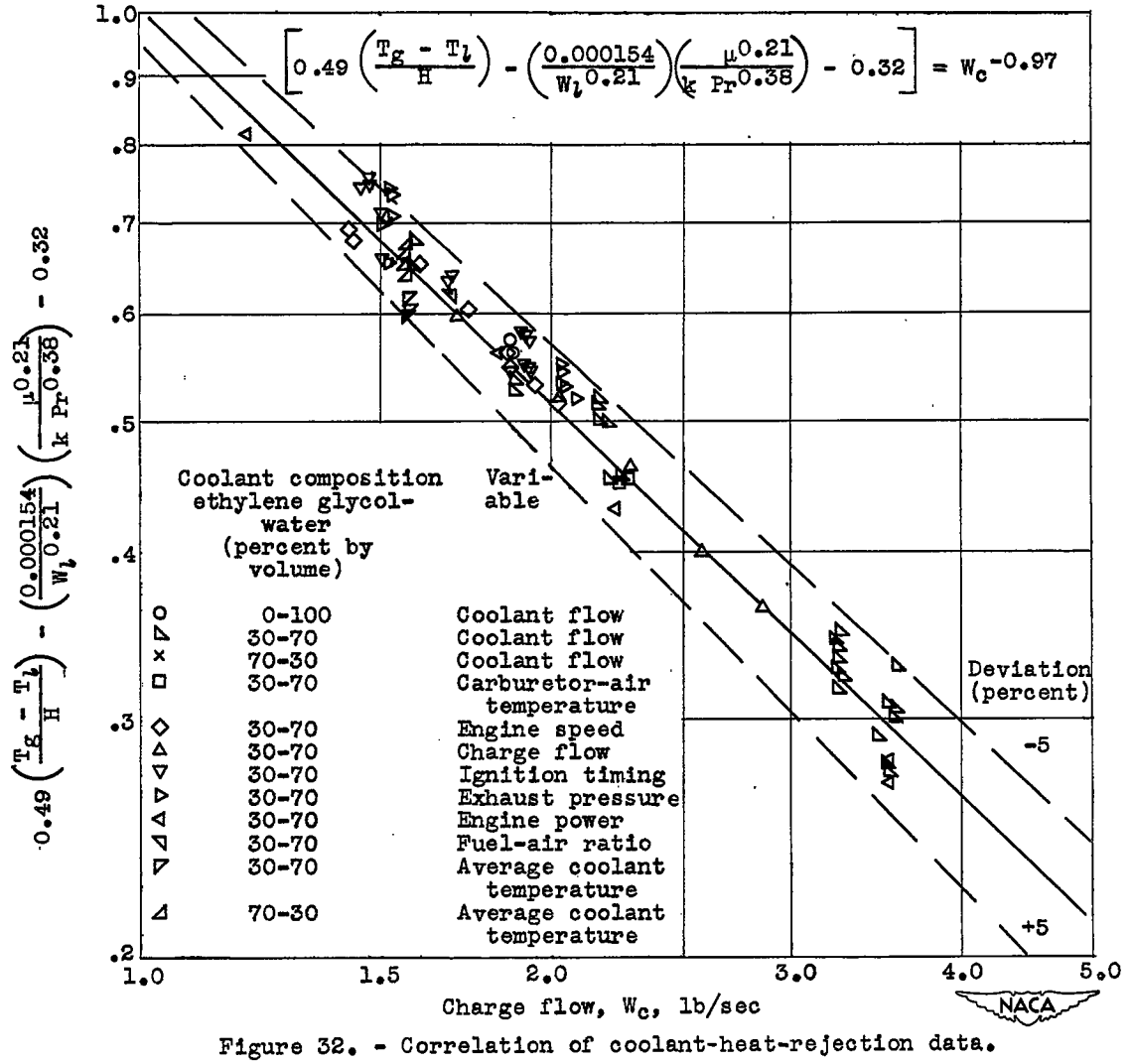

Figure 32. - Correlation of coolant-heat-rejection data.

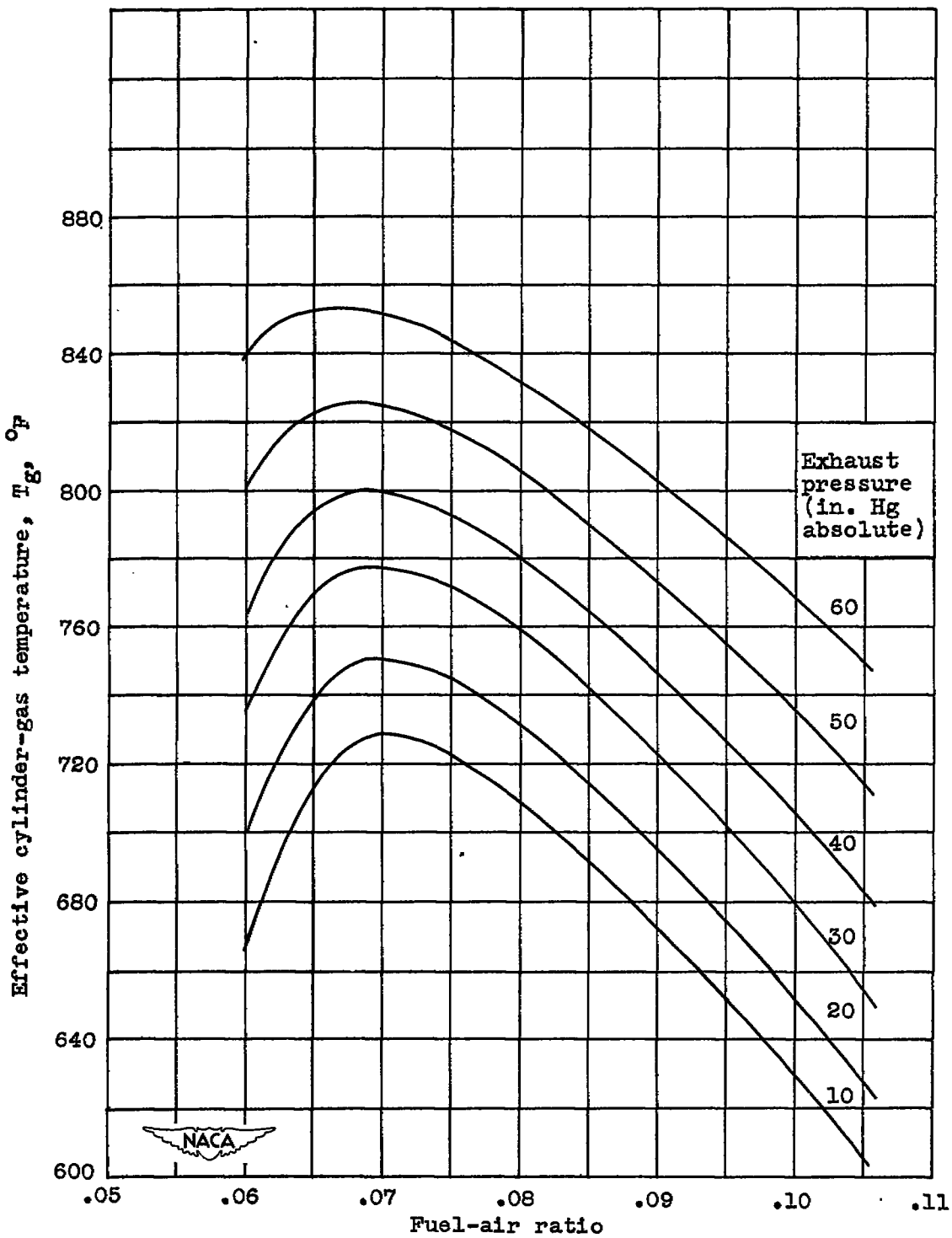

Figure 33. - Variation of effective cylinder-gas temperature with fuel-air ratio for various exhaust pressures for use in coolant-heat-rejection correlation. Data corrected to calculated dry inlet-manifold temperature of 80° F.

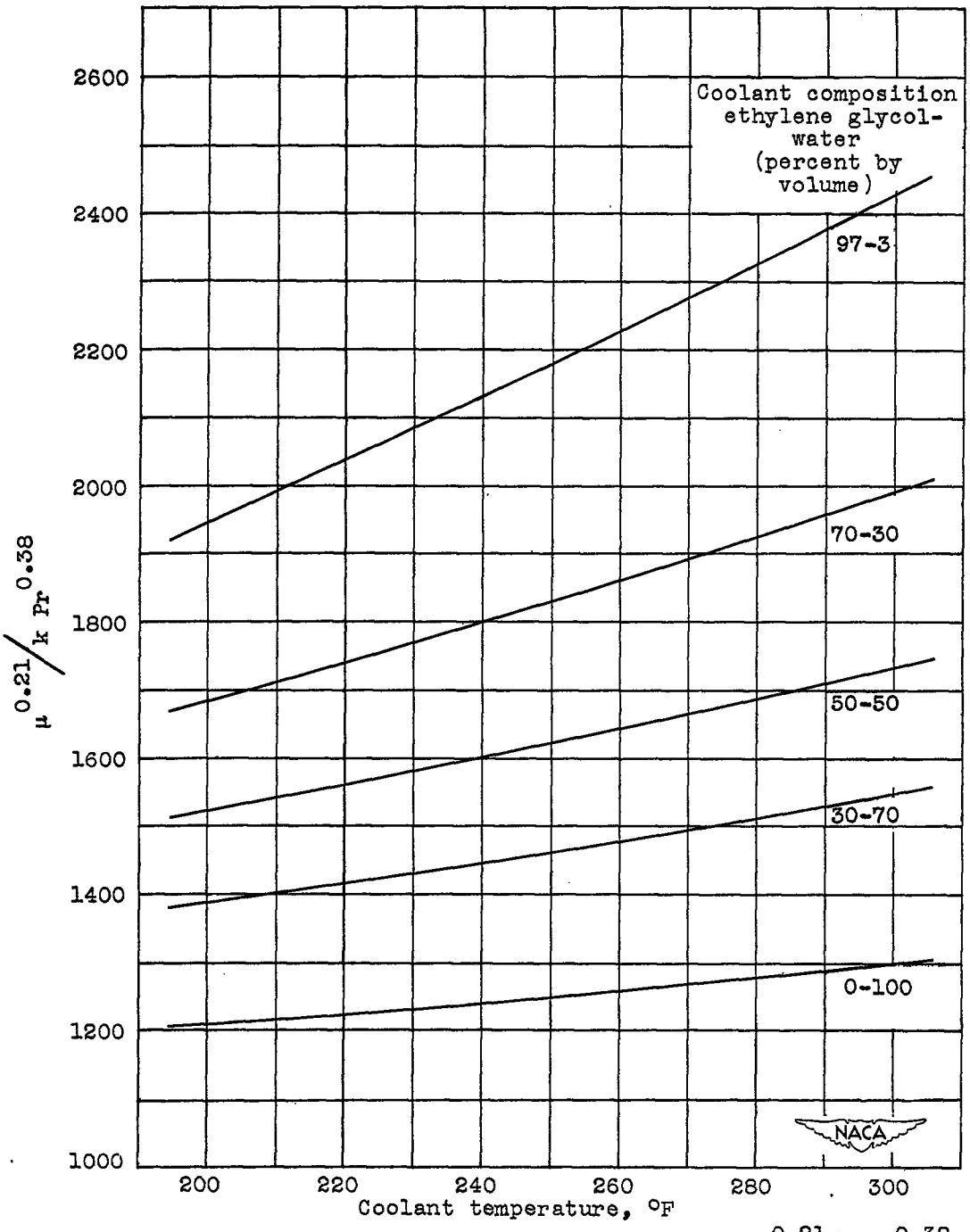

Figure 34. - Variation of coolant-property parameter $\mu^{0.21}/k\ Pr^{0.38}$ with coolant temperature for various aqueous ethylene-glycol solutions.