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CYLINDER-TEMPERATURE CORRELATION OF A SINGLE-CYLINDER LIQUID-COOLED ENGINE

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SUMMARY

An analysis based on nonboiling forced-convection heat-transfer theory is made of the cooling processes in liquid-cooled engine cylinders. Semiempirical equations that relate the average head and barrel temperatures with the primary engine and coolant parameters are derived.

A correlation method based on these equations is applied to data obtained from previously reported investigations, which were conducted over large ranges of engine and coolant conditions with two liquid-cooled cylinders using water and various aqueous ethylene glycol solutions as coolants. Upon evaluation of empirical factors, an equation for the cylinder-head temperature as a function of the engine operating conditions and the flow rate, temperature, and physical properties of the coolants is obtained, which represents the data with good accuracy. A similar equation is obtained for the barrel-temperature data. The physical properties of the coolant appearing in these equations in order of their importance in determining the heat-transfer quality of the coolant are thermal conductivity, specific heat, and viscosity. The cooling performance of the various coolants investigated is adequately correlated by these physical properties in the correlation equation. The application of the correlation equation is, however, limited to conditions for which boiling of the coolant on the engine walls is an unimportant part of the cooling process.

INTRODUCTION

A knowledge of the cooling characteristics of reciprocating aircraft engines is of importance in order to insure satisfactory engine operation at extreme operating conditions.

In 1937 the heat-transfer processes in air-cooled engine cylinders were analyzed and a method based on forced-convection heat-transfer theory was developed for correlating cylinder temperatures with engine and cooling-air conditions (reference 1). This method was extended in 1939 to include multicylinder-engine data (reference 2) and since then has been used extensively in connection with many air-cooled engine investigations.

In 1943 an experimental investigation of the cooling characteristics of liquid-cooled engines was instituted at the NACA Cleveland laboratory. As the first phase of this investigation, tests were conducted with two liquid-cooled cylinders to isolate the effects of the different engine and coolant variables on cylinder temperatures and to obtain the data required for a study of the fundamental heat-transfer processes involved in the cooling of liquid-cooled engines. The cylinder temperatures, which were obtained over ranges of engine speeds, manifold pressures, carburetor-air temperatures, fuel-air ratios, spark advances, coolant-flow rates, coolant temperatures, and coolant pressures with water and with various ethylene glycol-water mixtures, are presented in reference 3.

In the investigation reported herein, the theory of heat transfer by nonboiling forced convection is applied to the cooling processes in a liquid-cooled engine cylinder. The method of analysis is essentially the same as that used in reference 1 for air-cooled engines with the additional complications: (1) that the variation of Prandtl number of the coolant with coolant temperature and composition, which was negligible for the air-cooled engine, becomes an important consideration for the liquid-cooled engine, and (2) that the effect of temperature drop through the cylinder-head wall must be accounted for in the correlation because it is an important part of the over-all temperature difference between wall and coolant.

The analysis results in semiempirical expressions for the average head and barrel temperatures as functions of the engine conditions and temperature, the flow rate, and the physical properties of the coolant. The physical properties of the coolant that appear in this equation are the thermal conductivity, the specific heat, and the viscosity; and in the region of operation where forced convection is the controlling cooling phenomenon, correlation of the cooling performance of various coolants on the basis of these coolant physical properties can be expected. In the test range where appreciable boiling of the coolant occurs on the engine walls, the correlation equation developed herein can be expected to be inaccurate.

The cooling-correlation method is applied to the data presented in reference 3.

SYMBOLS

The following symbols are used in this analysis:

$\bar{A}$	mean area of cylinder wall perpendicular to direction of heat flow, (sq ft)
A_l	cylinder-wall area in contact with coolant, (sq ft)
$B_1, B_2 \ldots B_{10}$	constants
c	specific heat of coolant, Btu/(lb)(°F)

D	hydraulic diameter, (ft)
H_b	heat rejected to barrel coolant, Btu/(min)
H_h	heat rejected to head coolant, Btu/(min)
h	heat-transfer coefficient, liquid side, Btu/(sec) (sq ft) (°F)
k	thermal conductivity of coolant, Btu/(sec) (sq ft) (°F/ft)
k_w	thermal conductivity of cylinder wall, Btu/(sec) (sq ft) (°F/ft)
m, n, s	exponents
$T_{b,o}$	average cylinder-barrel (liquid-side) temperature, (°F)
T_g	effective cylinder-gas temperature, (°F)
T_h	average cylinder-head (gas-side) temperature, (°F)
$T_{h,o}$	average cylinder-head (liquid-side) temperature, (°F)
T_l	average coolant temperature, (°F)
V	coolant velocity, (ft/sec)
W_c	charge (air plus fuel) flow rate, (lb/min)
W_l	coolant-flow rate, (lb/min)
x	mean thickness of cylinder head parallel to direction of heat flow, (ft)
Z	$B_1x/B_2k_w\overline{A}$
μ	absolute viscosity of coolant, (lb)/(ft) (sec)
ρ	density of coolant, (lb/cu ft)
$c\mu/k$	Prandtl number of coolant

ANALYSIS

The analysis of the heat-transfer processes in a liquid-cooled engine cylinder is separated into three processes: the heat transfer (1) from the cylinder gases to the gas-side cylinder walls, (2) through the cylinder walls, and (3) from the liquid-side cylinder walls to the coolant. A diagrammatic sketch of a liquid-cooled engine cylinder illustrating the various sections is shown in figure 1.

HEAT TRANSFER IN CYLINDER HEAD

Cylinder gases to gas-side cylinder wall.—The transfer of heat from the cylinder gases to the walls may be expected to be the same for liquid-cooled engines as for air-cooled engines; hence, an expression based on forced-convection heat-transfer theory similar to that derived in reference 1 may be written

$$H_h=B_1W_c^n(T_g-T_h) \tag{1}$$

Conduction through cylinder walls.—The heat transferred through the cylinder walls follows the law of conduction and may be expressed as

$$H_h=B_2\frac{k_w\overline{A}}{x}(T_h-T_{h,o}) \tag{2}$$

Liquid-side cylinder walls to coolant.—For a large part of the normal operating range, cooling of the engine walls occurs by nonboiling forced convection. In this range of

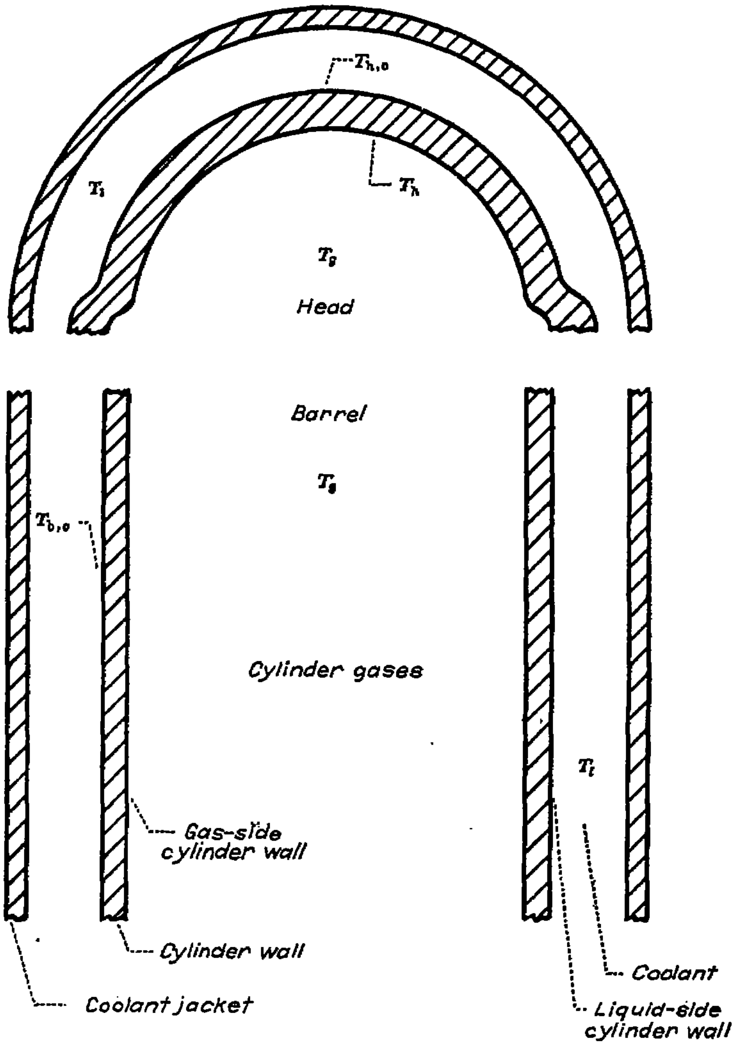


FIGURE 1.—Diagrammatic sketch of liquid-cooled engine cylinder illustrating various sections.

operation, the following familiar Nusselt relation may be written

$$\frac{hD}{k}=B_3\left(\frac{\rho VD}{\mu}\right)^m\left(\frac{c\mu}{k}\right)^s \tag{3}$$

When appreciable boiling of the coolant on the engine walls occurs, deviation of the heat-transfer coefficient from the value given by equation (3) may be expected. The analysis will, however, be continued on the basis of equation (3). The final correlation equation will then apply to operating conditions where nonboiling forced convection controls the heat-transfer process. Inasmuch as h is defined as $\frac{H_h}{B_2A_l(T_{h,o}-T_l)}$, D and A_l are constant for a given engine, and ρVD is proportional to W_l. The heat flow per unit time is

$$H_h=B_4k\left(\frac{W_l}{\mu}\right)^m\left(\frac{c\mu}{k}\right)^s(T_{h,o}-T_l) \tag{4}$$

Correlation equation.—The correlation equation is obtained by eliminating H_h and $T_{h,o}$ from equations (1), (2), and (4):

$$\left[\frac{T_h-T_l}{W_c^n(T_g-T_h)}-Z\right]\left(\frac{c\mu}{k}\right)^s k=B_6\left(\frac{W_l}{\mu}\right)^{-m} \tag{5a}$$

where $Z=B_1x/B_2k_w\overline{A}$ and, in effect, accounts for the temperature drop through the cylinder-head wall. Inasmuch as the thermal conductivity of the cylinder-head metal k_w is effectively constant for the temperature range encountered in normal engine operation, Z may be taken as a constant for a given engine.

An alternate form of the final equation (5a) involving both gas-side and liquid-side head temperatures can be obtained by eliminating H_h from equations (1) and (4):

$$\frac{T_{h,o}-T_l}{W_c^n(T_g-T_h)}\left(\frac{c\mu}{k}\right)^s k=B_5\left(\frac{W_l}{\mu}\right)^{-m} \tag{5b}$$

For illustrating the relative cooling quality of various coolants, equation (5a) is simplified, by rearranging and expressing c, μ, and k as a function of T_l, to a form that will result in a separate correlation curve for each coolant:

$$\left[\frac{T_h-T_l}{W_c^n(T_g-T_h)}-Z\right]W_l^m=B_6\frac{\mu^{(m-s)}}{c^s k^{(1-s)}}=B_6 f(T_l) \tag{5c}$$

HEAT TRANSFER IN CYLINDER BARREL

The heat-transfer processes in the cylinder barrel are expected to be the same as those in the head except that: (a) The temperature drop through the barrel wall is so relatively small that the Z factor may be omitted; and (b) the simple application of the theory of forced convection to the heat transfer from the cylinder gases to the barrel wall, as shown in reference 1, is only a rough approximation because the process is complicated by the generation and the transfer of heat by the moving piston. In the interest of simplicity, consideration of the complications introduced by the piston is neglected and the relation for the over-all heat transfer in the cylinder barrel is written in a form similar to that for the cylinder head:

$$\frac{T_{b,o}-T_l}{W_c^n(T_g-T_{b,o})}\left(\frac{c\mu}{k}\right)^s k=B_6\left(\frac{W_l}{\mu}\right)^{-m} \tag{6a}$$

Equation (6a) is simplified as follows for obtaining a separate correlation curve for any one coolant

$$\frac{T_{b,o}-T_l}{W_c^n(T_g-T_{b,o})}W_l^m=B_6\frac{\mu^{(m-s)}}{c^s k^{(1-s)}}=B_5 f'(T_l) \tag{6b}$$

SIGNIFICANCE OF EQUATIONS AND CONSTANTS

Equations (5a), (5b), and (6a) are the final equations for correlating the cylinder-temperature data with the engine and coolant variables. When the left-hand member of any of these equations is plotted on logarithmic coordinates against W_l/μ, a single curve should be obtained for a given engine regardless of the engine conditions or the temperature, flow rate, and composition of the coolant used, provided that modes of heat transfer other than nonboiling forced convection are of negligible magnitude and provided that no pertinent variables have been omitted in the analysis. Systematic deviation of the test points from the common correlation curve will indicate the presence and the importance of other modes of heat transfer or of variables not included in the final equations.

The effective gas temperature T_g, as indicated in references 1 and 2, is a function of fuel-air ratio, carburetor-air temperature, spark advance, and exhaust back pressure. The values of T_g at reference operating conditions for the head and for the barrel, as well as the variation of T_g with fuel-air ratio, carburetor-air temperature, and spark advance, is obtained from engine tests, as will subsequently be described. The variation of T_g with exhaust back pressure is not discussed inasmuch as data on variable exhaust back pressure were not available.

The factors Z, B_5, and B_6 and the exponents m, n, and s are also determined from engine tests. The details of evaluating these terms are presented in a later section. Although the symbols used in the cylinder-head equations for the effective gas temperature, the average coolant temperature, the coolant-flow rate, and the exponents are the same as those used in the barrel equations, the numerical values of these factors are not necessarily the same for the head and the barrel.

The physical properties c, μ, and k of the coolants are functions of coolant composition and temperature. Curves of c, μ, and k as functions of temperature for water and for ethylene glycol-water mixtures are presented subsequently.

APPARATUS AND TESTS

The data used herein were obtained from two modified Lycoming 0–1230 liquid-cooled cylinders, which, for convenience, are designated cylinder A and cylinder B. A detailed description of the setup, the methods, and the results obtained is given in reference 3. The general arrangement of the engine and the auxiliary equipment is shown in figure 2.

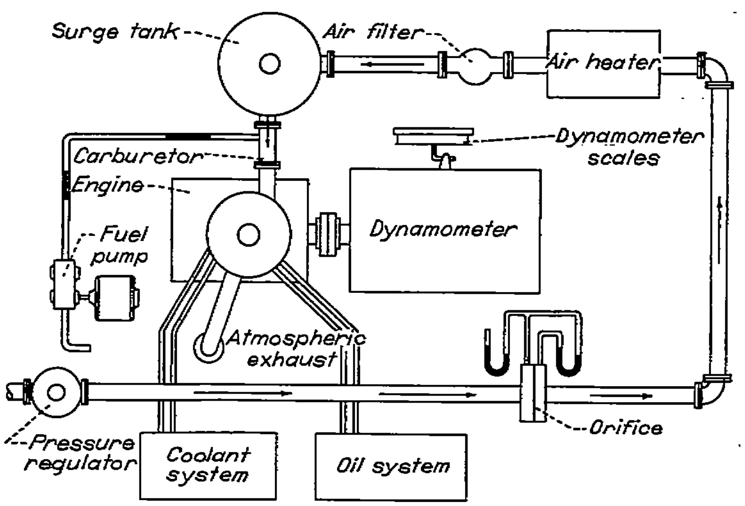
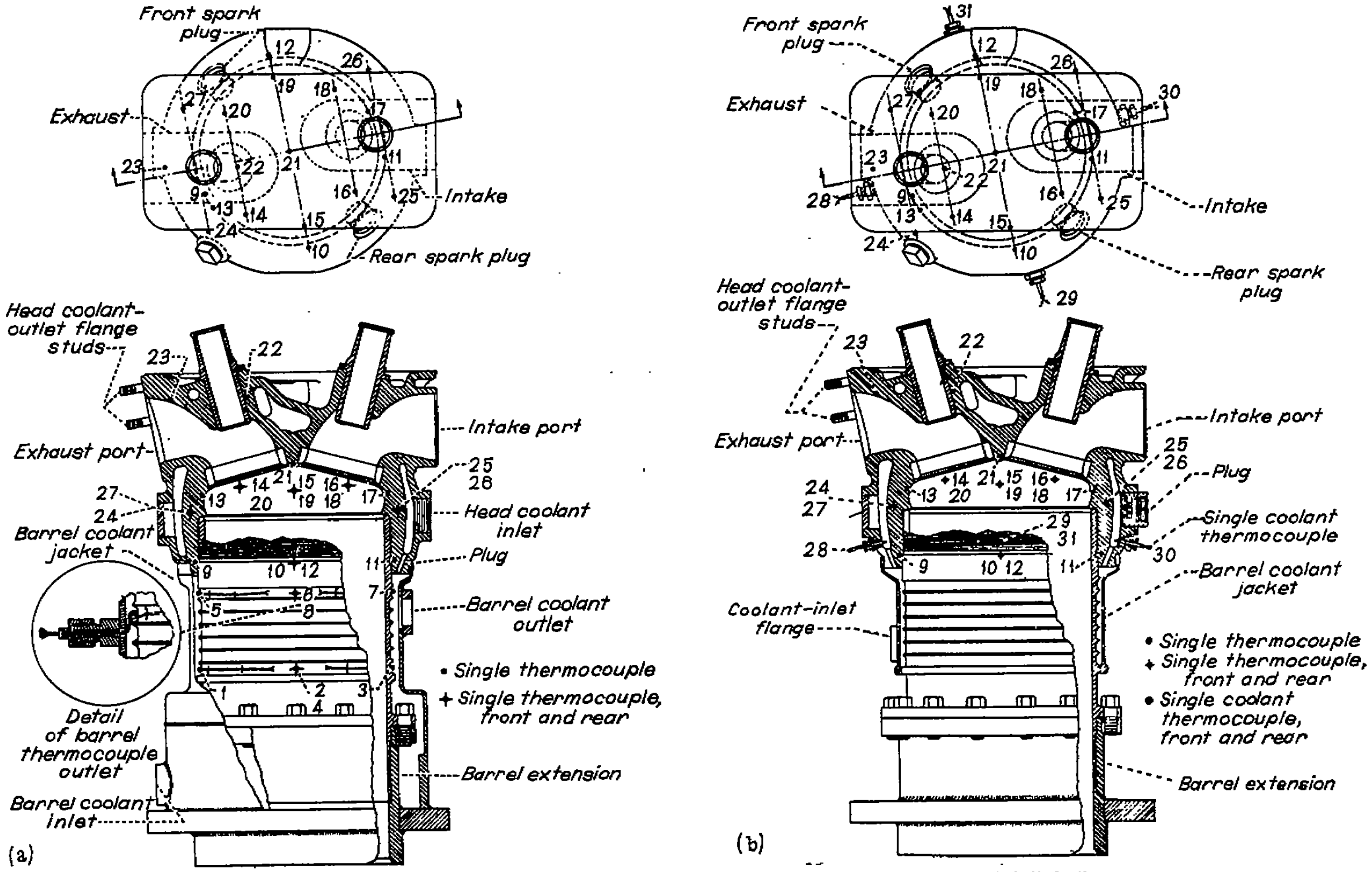


FIGURE 2.—Diagrammatic layout of engine-investigation setup.



(a) Cylinder A. (b) Cylinder B.

FIGURE 3.—Thermocouple installation and details of cylinders A and B.

Diagrams of cylinder A and cylinder B, which are the same except for the cylinder-barrel instrumentation and the coolant-flow paths, are presented in figure 3. In cylinder A, the barrel-coolant jacket was modified and eight thermocouples were spot-welded around the barrel. In addition, the coolant-transfer passages between the head and barrel jackets were plugged, thus separating the coolant flow to the head and barrel. In cylinder B, thermocouples were not installed on the barrel and the coolant-flow paths were not altered; hence, the coolant flowed first through the barrel and then through the head. The table opposite is a compilation of the conditions investigated.

In each of the runs, only one item was varied whereas all the others were held approximately constant. (In one run both coolant temperature and flow rate were varied simultaneously.) The following reference conditions were held approximately constant throughout the investigation except for the cases in which the particular condition was the primary variable:

Coolant pressure, pounds per square inch absolute 19
Exhaust back pressure, inches mercury absolute 30
Carburetor-air temperature, °F .... 80±5
Fuel-air ratio .... 0.077±0.001
Spark advance, degrees B. T. C. 28

Several runs conducted with cylinder A were repeated at higher coolant pressures and in the tests of cylinder B, each run was repeated at a higher coolant pressure.

Variable	Range of variable	Coolant, glycol-water (approximate percent by volume)[a]	Cylinder
Engine speed, rpm	1050 to 2760	[b]97-3 0-100	A and B B
Manifold pressure, in. Hg absolute	21.0 to 39.0	[b]97-3 0-100	A and B B
Carburetor-air temperature, °F	80 to 222	[b]97-3	A
Fuel-air ratio	0.048 to 0.121	[b]97-3 0-100	A B
Spark advance, deg B. T. C.	12 to 42	[b]97-3	A
Coolant-flow rate, lb/min	10.0 to 128.3	[b]97-3 70-30 30-70 0-100	A and B A and B A and B A and B
Average coolant temperature, °F	90.0 to 311.0	[b]97-3 70-30 30-70 0-100	A and B A and B A and B A and B
Coolant pressure, lb/sq in. absolute	17 to 75	0-100	A

[a] The ethylene glycol-water mixture used in several tests conducted with cylinder A was actually 38-62 percent instead of 30-70 percent.
[b] AN-E-2 ethylene glycol.

METHODS

EVALUATION OF FACTORS FOR CORRELATION EQUATIONS

In most cases the data used herein for evaluating the exponents and constants are from cylinder A inasmuch as all the variables were investigated with this cylinder and, in addition, barrel temperatures were obtained.

Average temperatures.—The average temperatures used are those given in reference 3. The average head (gas-side) temperature T_h was taken as the average of values obtained from nine thermocouples (13 to 21, fig. 3) located in the cylinder head, approximately ⅛ inch from the combustion-chamber surface. The average head (liquid-side) temperature $T_{h,o}$ was obtained from the average of values determined by four thermocouples (24 to 27, fig. 3) located on the liquid-side surface of the combustion-chamber wall. The average barrel (liquid-side) temperature $T_{b,o}$ was taken as the average of readings obtained from eight thermocouples (1 to 8, fig. 3 (a)) spot-welded on the outside of the barrel in two rows of four thermocouples each. The average of the coolant temperatures measured in the inlet and outlet coolant passages of each portion of the cylinder was used as the average coolant temperature T_l.

Physical properties of coolants.—The thermal conductivity k, the absolute viscosity μ, and the specific heat c of water and of the ethylene glycol-water solutions are shown as functions of temperature in figures 4 and 5, respectively (data from reference 4). The tests reported in reference 5 indicate that satisfactory correlation of forced-convection heat-transfer data for water and for aqueous ethylene glycol solutions is obtained using these values of the physical properties. The physical properties used in the final equations are evaluated at the value of the average coolant temperature T_l.

Effective gas temperature T_g.—The values of T_g for the head and the barrel are obtained from tests at reference conditions of fuel-air ratio, carburetor-air temperature, and spark advance in which only W_l and T_l were varied. For these conditions, T_g and W_c are constant and equation (1) reduces to

$$\frac{H_h}{T_g - T_h} = \text{constant}$$

The values of the heat rejected to the head and barrel coolant are plotted against T_h and $T_{b,o}$, respectively, and the curves

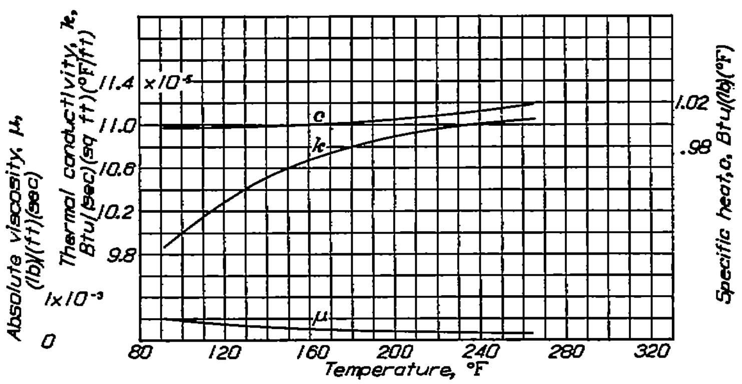
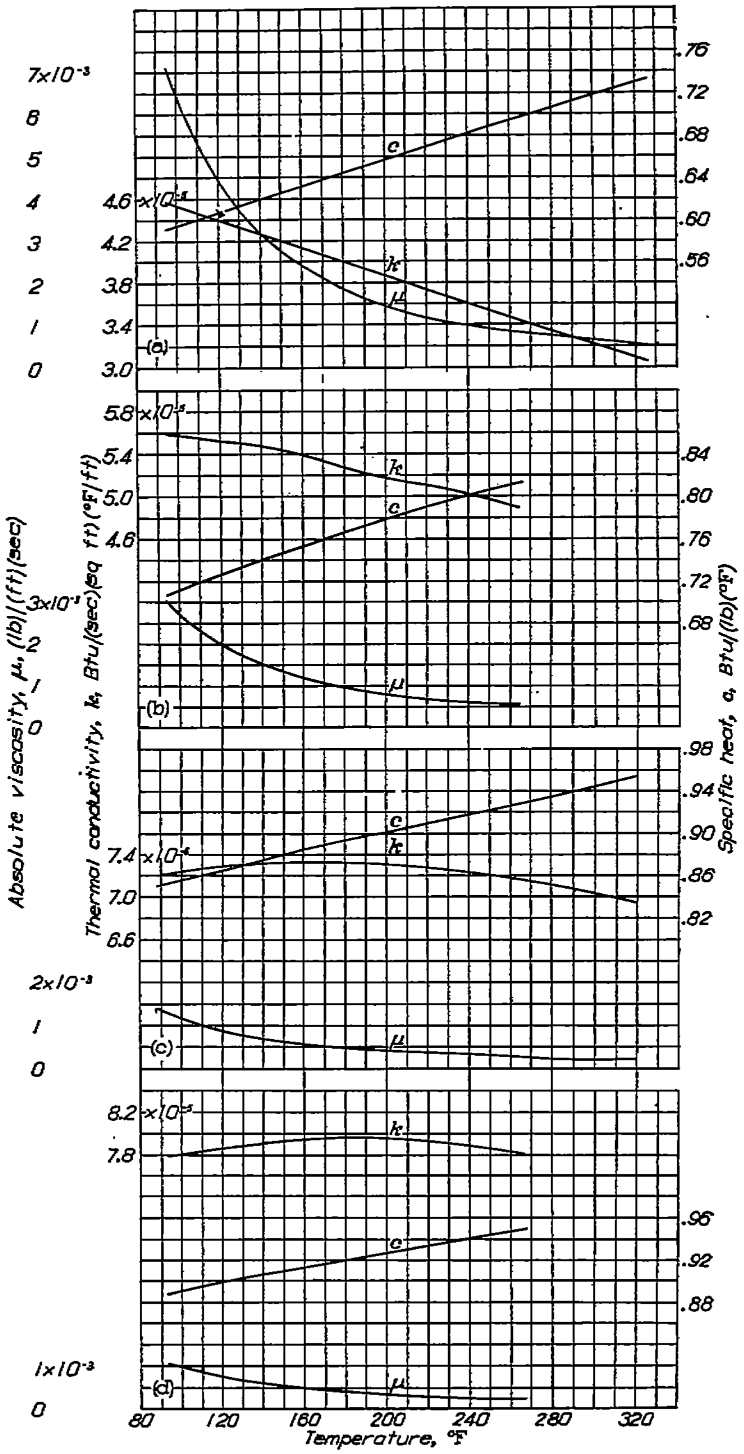


FIGURE 4.—Variation of thermal conductivity, specific heat, and viscosity of water with temperature. (Data from reference 4.)



(a) 97-3 percent (by volume) glycol-water solution.
(b) 70-30 percent (by volume) glycol-water solution.
(c) 38-62 percent (by volume) glycol-water solution.
(d) 30-70 percent (by volume) glycol-water solution.

FIGURE 5.—Variation of thermal conductivity, specific heat, and viscosity of aqueous ethylene-glycol solutions with temperature. (Data from reference 4.)

are extrapolated to zero heat rejection, at which points the average head and barrel temperatures are equal to their respective effective gas temperatures.

The variation of T_g with fuel-air ratio, carburetor-air temperature, and spark advance is obtained from the results of the runs in which each of these factors was independently varied while the engine charge flow and the coolant conditions were held constant. For such conditions, equations (5a) and (6a) reduce, respectively, to

$$\frac{T_h - T_l}{T_g - T_h} = \text{constant}$$

and

$$\frac{T_{b,o} - T_l}{T_g - T_{b,o}} = \text{constant}$$

These constants are evaluated from the cylinder-temperature and coolant-temperature data at the reference operating conditions for which T_g has already been determined. The value of T_g at other than the reference conditions is then calculated from the values of the constant, the constant coolant temperature, and the cylinder temperature obtained at the operating conditions in question.

Exponent n of charge-flow rate W_c.—The values of the exponent n of charge-flow rate W_c are obtained from runs at constant coolant conditions in which engine speed or manifold pressure, hence W_c, was varied.

For these test conditions, equations (5a) and (6a) reduce to

$$\frac{T_h - T_l}{T_g - T_h} = B_7 W_c^n$$

and

$$\frac{T_{b,o} - T_l}{T_g - T_{b,o}} = B_8 W_c^n$$

Logarithmic plots of $\frac{T_h - T_l}{T_g - T_h}$ and $\frac{T_{b,o} - T_l}{T_g - T_{b,o}}$ against W_c are made and the values of the exponent obtained directly from the slopes of the resulting lines.

Factor Z.—The factor Z is evaluated from runs in which W_l was varied for different values of constant coolant temperature and composition. These conditions of constant coolant temperature and composition predicate constant c, μ, and k, in which case equation (5a) may be written

$$\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z = B_9 \left(\frac{1}{W_l}\right)^m$$

Plots of $\frac{T_h - T_l}{W_c^n (T_g - T_h)}$ against $1/W_l$ are made and the curves obtained are extrapolated to zero $1/W_l$ (infinite W_l), at which point the value of $\frac{T_h - T_l}{W_c^n (T_g - T_h)}$ is equal to Z.

Exponent s of Prandtl number $c\mu/k$.—The value of the Prandtl number exponent s is determined from data for a constant value of W_l/μ. For this condition, equation (5a) may be reduced to

$$\left[\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z\right] k = B_{10} \left(\frac{c\mu}{k}\right)^s$$

The slope of the line determined by a logarithmic plot of the left-hand term of this equation against $c\mu/k$ would then establish the value of the exponent s. A wide range of the Prandtl number for this plot may be obtained if data for several different coolants are used. In order to construct this plot, it is convenient to first plot $\left[\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z\right] k$ against W_l/μ using data obtained from variable W_l/μ tests conducted with different coolants at several temperature levels. Values of $\left[\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z\right] k$ are then selected from this plot at a constant value of W_l/μ and are cross-plotted against the Prandtl number of the different coolants.

FINAL CORRELATION

Final correlation plots of the engine-cooling data for all the coolants tested are obtained by plotting the following parameters against W_l/μ on logarithmic coordinates:

(a) Head (gas-side) temperatures $\left[\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z\right] \left(\frac{c\mu}{k}\right)^s k$

(b) Head (gas- and liquid-side) temperatures

$$\frac{T_{h,o} - T_l}{W_c^n (T_g - T_h)} \left(\frac{c\mu}{k}\right)^s k$$

(c) Barrel (liquid-side) temperatures $\frac{T_{b,o} - T_l}{W_c^n (T_g - T_{b,o})} \left(\frac{c\mu}{k}\right)^s k$

Data are included from the tests in which each of the various engine and coolant variables were investigated. The values of the exponent m of the coolant-flow-rate variable W_l/μ and of the constants B_5 and B_6, which complete the determination of the final correlation equations (5a), (5b), and (6a), are obtained from the slopes and the coordinates of the points of the resulting lines.

In addition, separate correlation curves for each of the coolants tested are obtained by plotting

$$\left[\frac{T_h - T_l}{W_c^n (T_g - T_h)} - Z\right] W_l^m$$

against T_l and $\frac{T_{b,o} - T_l}{W_c^n (T_g - T_{b,o})} W_l^m$ against T_l. (See equations (5c) and (6b).)

RESULTS AND DISCUSSION

The final correlation equations are obtained by evaluating the exponents and constants. The final equations are presented with the data to show the agreement.

FACTORS FOR CORRELATION EQUATIONS

Effective gas temperature T_g.—Typical plots used for determining the reference values of T_g for the head and barrel are shown in figure 6. The heat rejected to the head and barrel coolant is plotted against T_h and $T_{b,o}$ from the results of tests conducted at a fuel-air ratio of 0.077, a spark advance of 28° B. T. C., and a carburetor-air temperature of 80° F. Values of T_g of 1170° F for the head and 610° F for the barrel are indicated by extrapolating the curves to zero heat rejection. Admittedly, a considerable amount of latitude exists in the choice of these values in view of the extrapolated distance as compared with the range covered

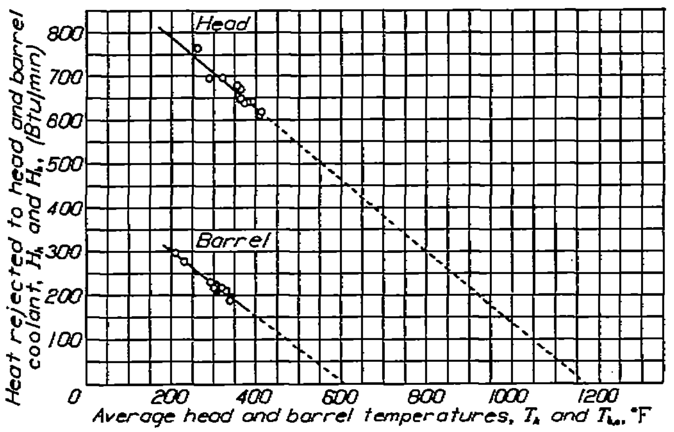


FIGURE 6.—Determination of T_g from variation of heat rejected to head and barrel coolant with average head and barrel temperatures. Cylinder A; coolant, AN-E-2 ethylene glycol; average coolant temperature: head, 125° to 293° F; barrel, 126° to 311° F; coolant-flow rate: head, 49 to 135 pounds per minute; barrel, 15 to 47 pounds per minute; engine speed, 2000 rpm; charge-flow rate, 5.4 pounds per minute; fuel-air ratio, 0.077; spark advance, 28° B. T. C.; carburetor-air temperature, 80° F.

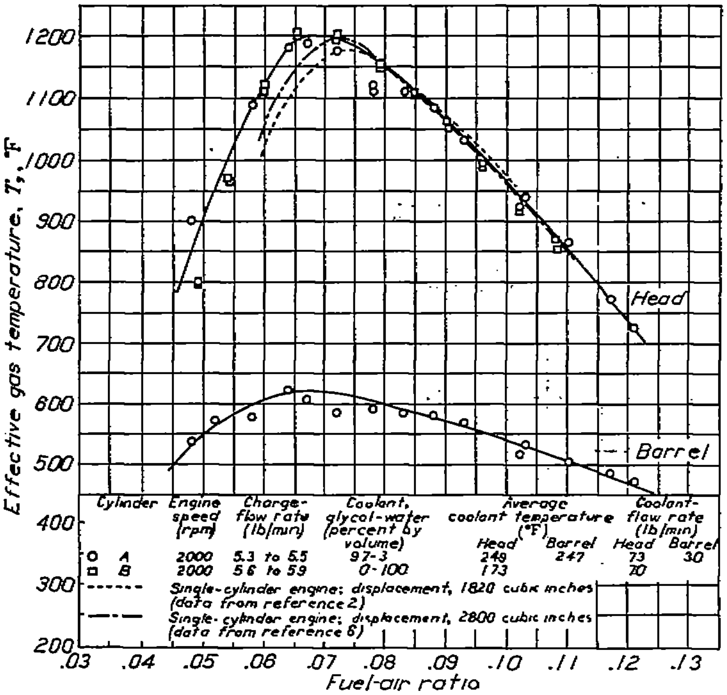


FIGURE 7.—Variation of effective gas temperature with fuel-air ratio for head and barrel. Spark advance, 28° B. T. C.; carburetor-air temperature, 80° F.

by the data; however, consideration of a large number of these curves for air-cooled engines (reference 1) led to the values chosen. Appreciable differences in the choice of the initial or reference value to T_g do not seriously change the accuracy of the correlation. It is of greater importance, once the initial value of T_g is chosen, that the variation of T_g from the initial value with variation in fuel-air ratio, carburetor-air temperature, and spark advance be accurate.

The variation of T_g for the head and the barrel with fuel-air ratio is presented in figure 7, which includes data for the head from both cylinders A and B. Also shown in the figure are typical head T_g curves for air-cooled engines obtained from single-cylinder tests of a radial engine with an 1820-cubic-inch displacement (reference 2) and one with a 2800-cubic-inch displacement (reference 6). Good agreement exists between the curves for the liquid-cooled and the air-cooled engines, especially in the rich-mixture region. The curves intersect at values of T_g of 1150° F for the head and 600° F for the barrel at a fuel-air ration of 0.080.

The effect of carburetor-air temperature on T_g is shown in figure 8. The data were obtained from tests conducted at fuel-air ratios of approximately 0.077 and 0.080; the data obtained at a fuel-air ratio of 0.080 are corrected to the reference value of 0.077. An increase in T_g of approximately 0.5° and 0.15° F per ° F increase in carburetor-air temperature is obtained for the head and the barrel, respectively.

The variation of T_g with spark advance at a fuel-air ratio of 0.077 is shown in figure 9. The value of T_g for the head decreases about 300° F as the spark is retarded through a crank angle from 42° to 12° B. T. C.; the rate of change decreases slightly at the retarded position. The effect of spark setting on T_g for the barrel is not so pronounced as for the head and a small reversal is obtained at the late setting.

Exponent n of charge-flow rate W_c.—Logarithmic plots of $\frac{T_h-T_l}{T_g-T_h}$ and $\frac{T_{b,o}-T_l}{T_g-T_{b,o}}$ against W_c are shown in figure 10, where the values of T_g are the previously determined reference values of 1170° and 610° F (fig. 6) for the head and the barrel, respectively. The slope of the average line through the data for both head and barrel, which is equal to the exponent n, is 0.65. This value is in agreement with values obtained for several air-cooled cylinders. Although slightly varying slopes would be obtained from plots of individual runs, the average line results in a good correlation of the data.

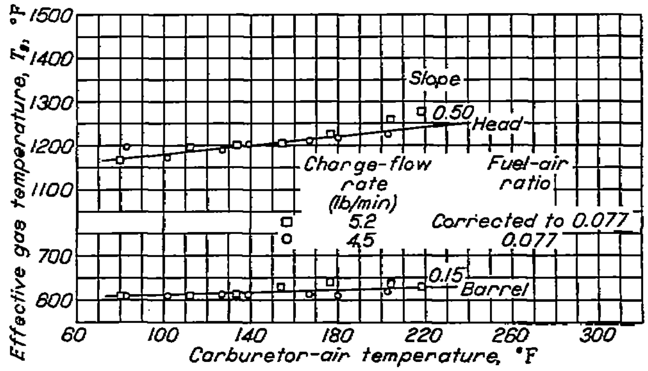


FIGURE 8.—Variation of effective gas temperature with carburetor-air temperature for head and barrel. Cylinder A; spark advance, 28° B. T. C.

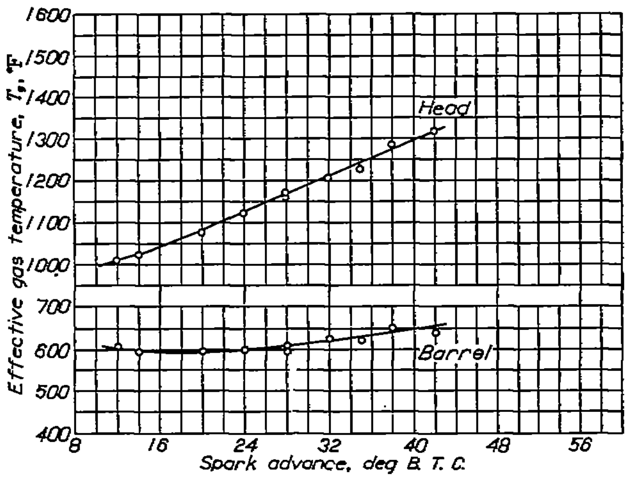


FIGURE 9.—Variation of effective gas temperature with spark advance for head and barrel. Cylinder A; fuel-air ration, 0.077; carburetor-air temperature, 83° F.

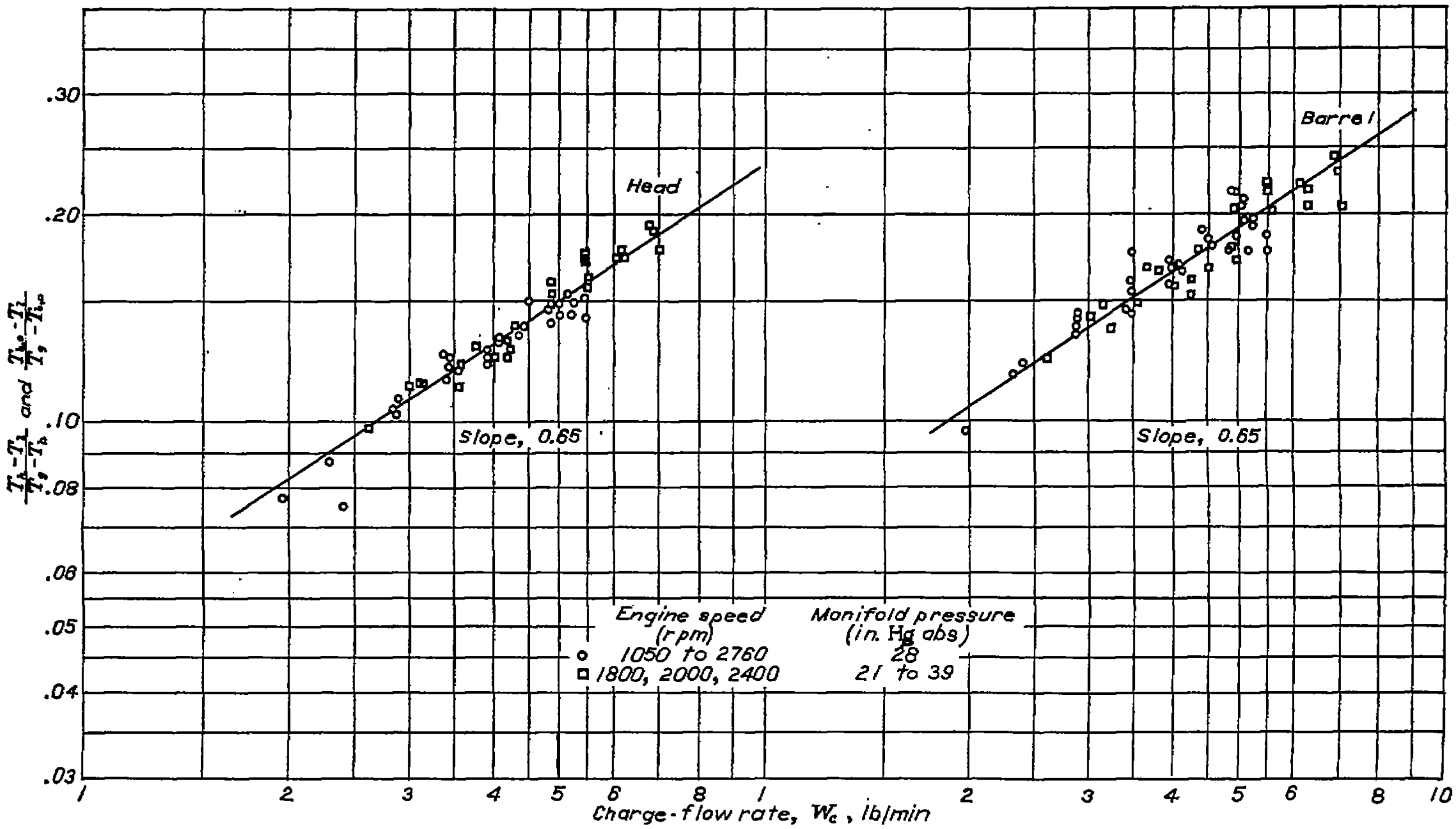


FIGURE 10.—Determination of exponents of charge-flow rate from variation of $(T_h-T_l)/(T_g-T_h)$ and $(T_{b,o}-T_l)/(T_g-T_{b,o})$ with W_c. Cylinder A; coolant, AN-E-2 ethylene glycol; average coolant temperature, 247° F; coolant-flow rate: head, 73 pounds per minute; barrel, 30 pounds per minute.

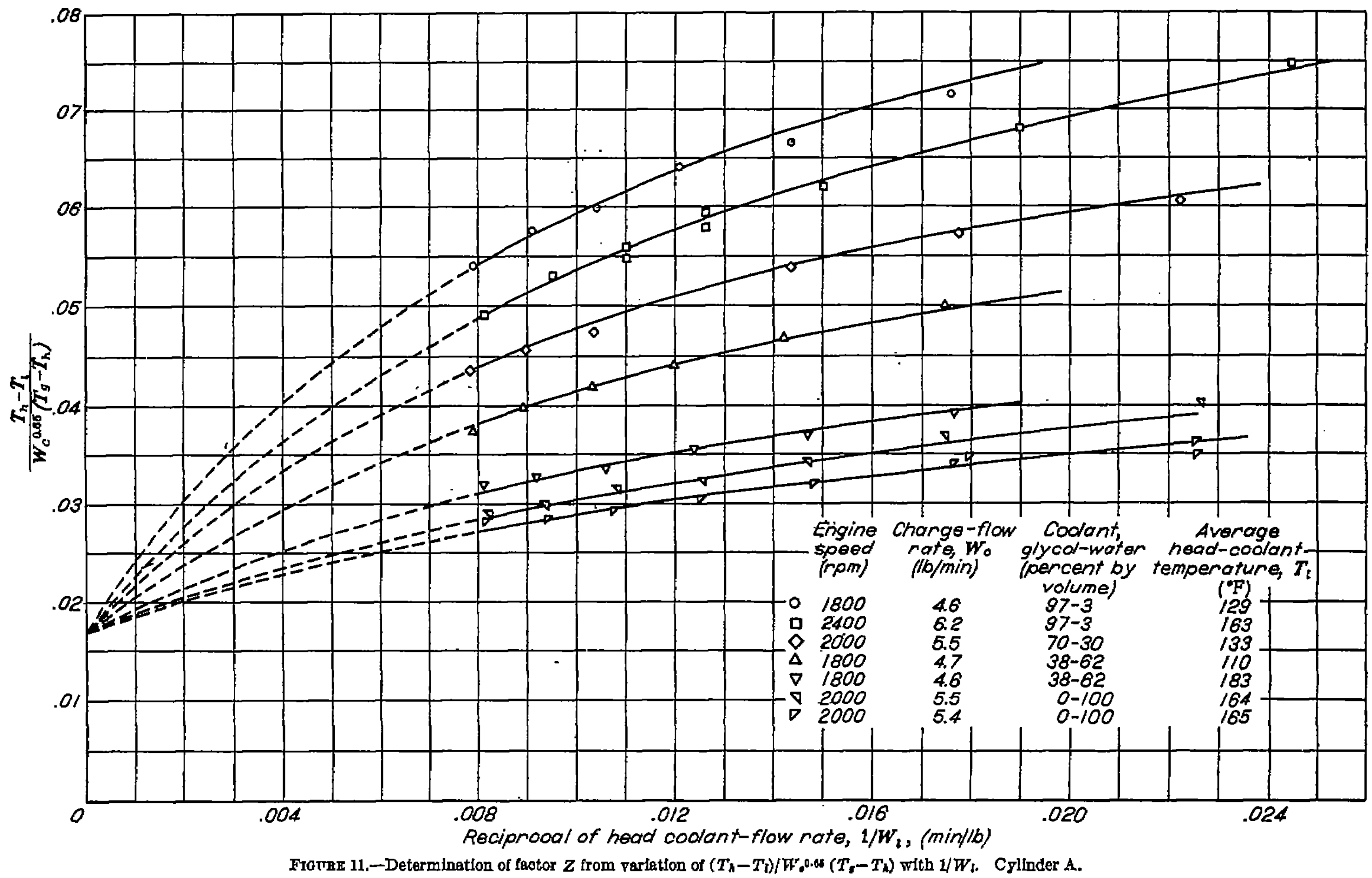


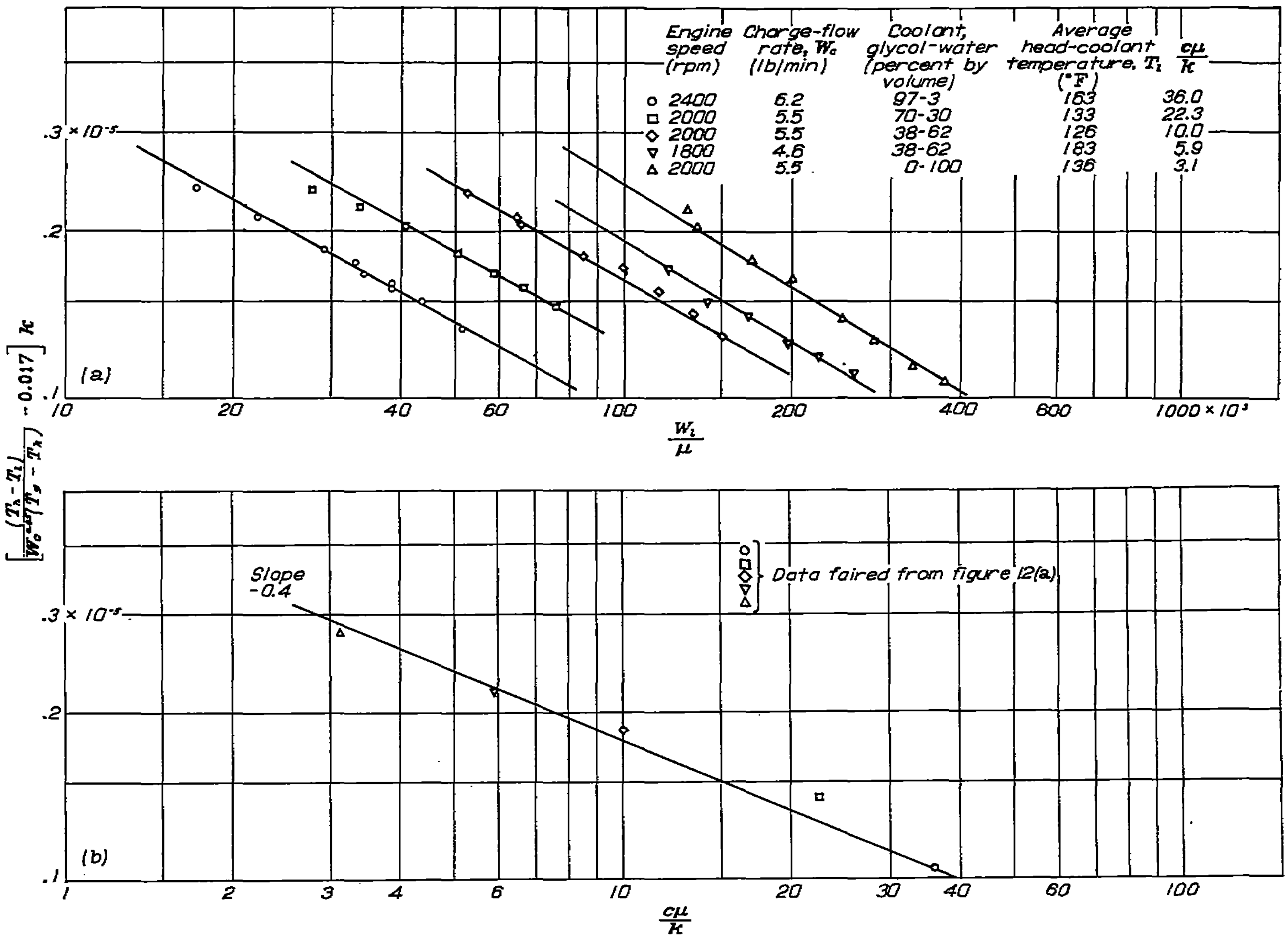
FIGURE 11.—Determination of factor Z from variation of $(T_h-T_l)/W_c^{0.65}(T_g-T_h)$ with $1/W_l$. Cylinder A.

Factor Z.—A plot of $\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}$ against $1/W_l$ is presented in figure 11 from the results of several tests conducted at different engine conditions using various coolants. Extrapolation of the different curves to zero $1/W_l$ results in a single value for Z equal to 0.017. The extrapolation, particularly of the upper curves in the figure, may be considered somewhat arbitrary inasmuch as it was necessarily made over a fairly wide range of flow rates. The value of 0.017 was chosen, however, after plotting final correlations using different values of Z covering the range of reasonable extrapolation of the curves of figure 11. The value of 0.017 gave the most satisfactory correlation and hence was accepted as the value of Z.

Exponent s of Prandtl number $c\mu/k$.—The two logarithmic plots used for determining the exponent of the Prandtl number are presented in figure 12. Figure 12 (a) shows $\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]k$ as a function of W_l/μ for several coolants at different constant average coolant temperatures. A family of approximately parallel lines is obtained. Figure 12 (b) shows the variation of $\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]k$ with $c\mu/k$ as determined by cross-plotting from figure 12 (a) at a value of W_l/μ equal to 80,000. A value for the exponent of $c\mu/k$ of 0.4 for the head is directly obtained from the absolute value of the slope of the resulting line. This value of the exponent is equal to that generally recommended for forced-convection heat transfer (reference 7); therefore, a value of 0.4 will also be used as the exponent for the barrel without further evaluation.

FINAL CORRELATION

Generalized correlation for all coolants.—The final generalized correlation plots, which include data for wide ranges of engine operating conditions, coolant temperatures, flow rates, and compositions, are presented in figures 13 to 16. In figures 13 and 14, the correlation parameters involving the gas-side head temperature for cylinders A and B, respectively,



(a) Variation of $\left[\frac{(T_h-T_l)}{W_c^{0.65}(T_g-T_h)}-0.017\right]k$ with $\frac{W_l}{\mu}$.

(b) Determination of exponent of Prandtl number from variation of $\left[\frac{(T_h-T_l)}{W_c^{0.65}(T_g-T_h)}-0.017\right]k$ with $\frac{c\mu}{k}$ as obtained by cross-plotting from figure 12 (a) at value of $\frac{W_l}{\mu}$ of 80,000.

FIGURE 12.—Determination of exponent of $\frac{c\mu}{k}$. Cylinder A.

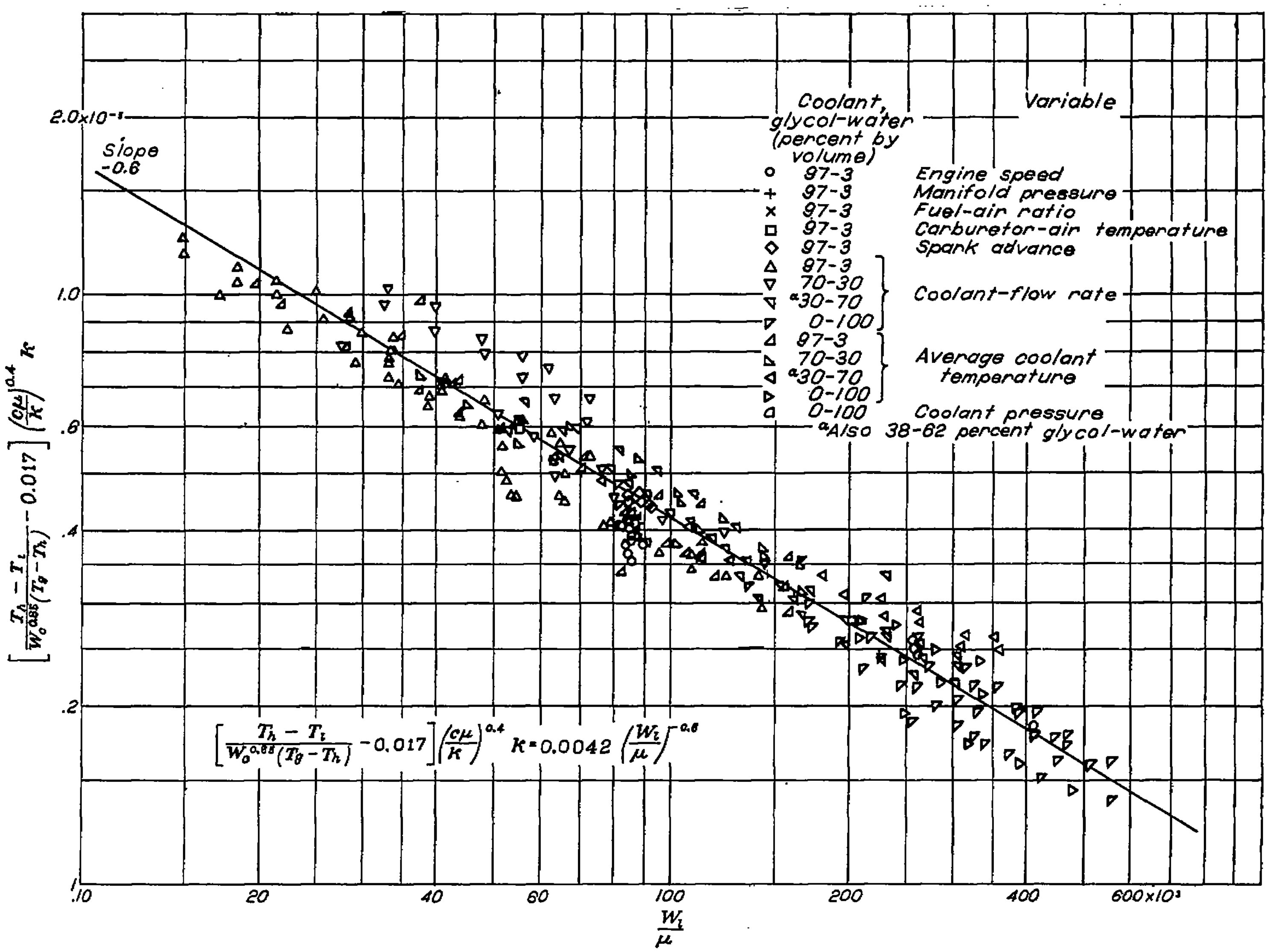


FIGURE 13.—Final correlation of average gas-side head temperatures for cylinder A.

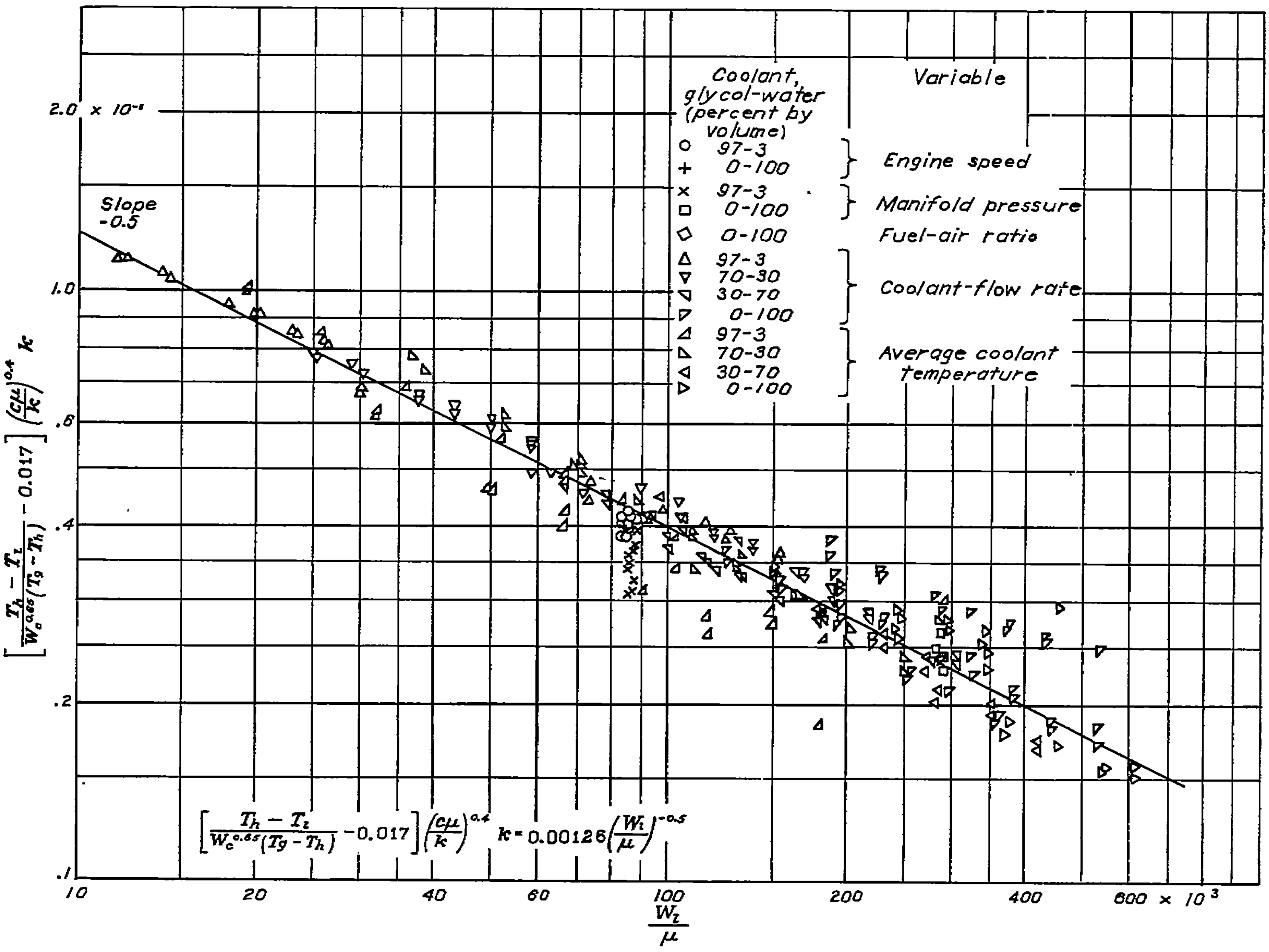


FIGURE 14.—Final correlation of average gas-side head temperatures for cylinder B.

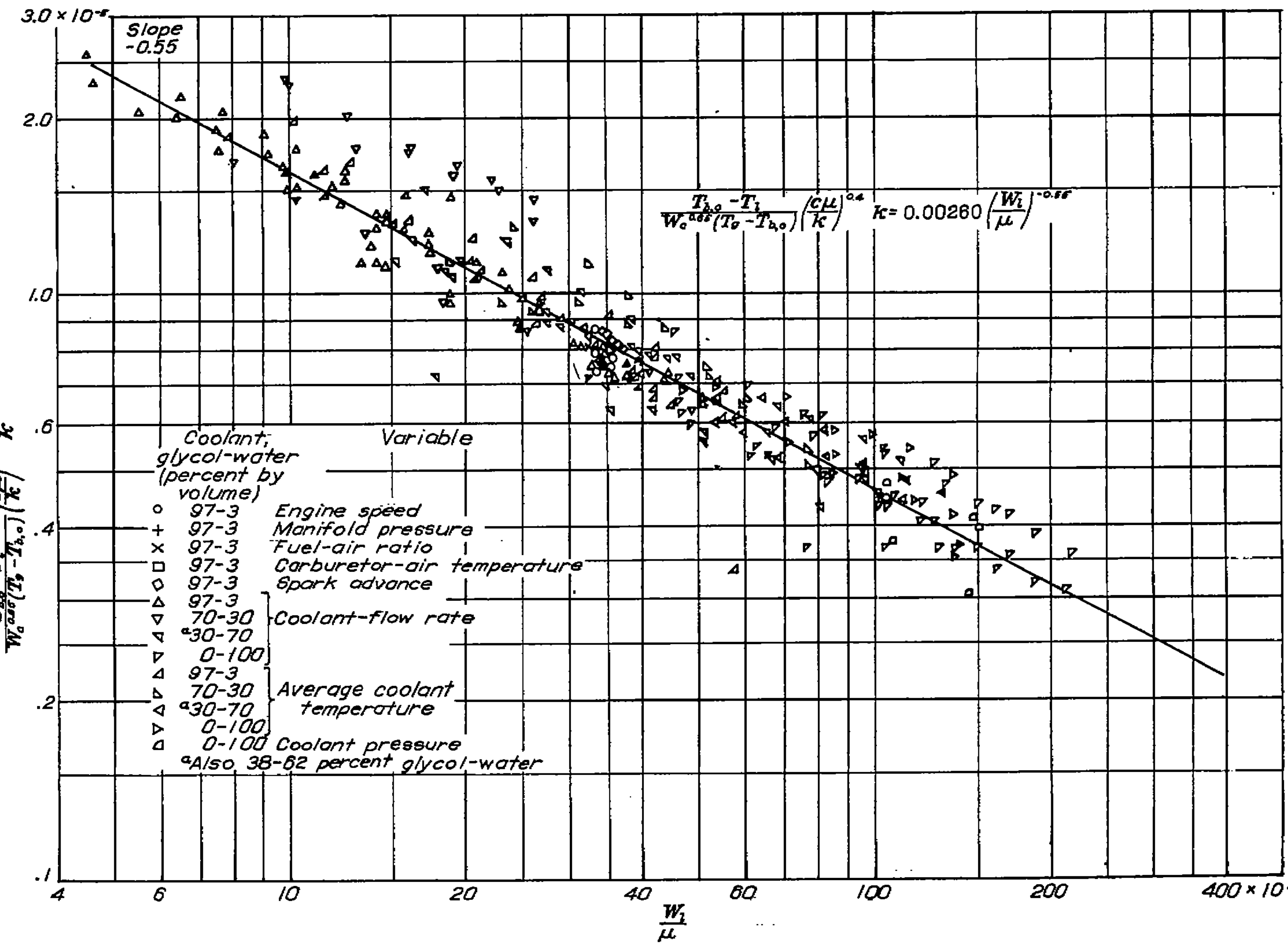


FIGURE 15.—Final correlation of average liquid-side barrel temperatures for cylinder A.

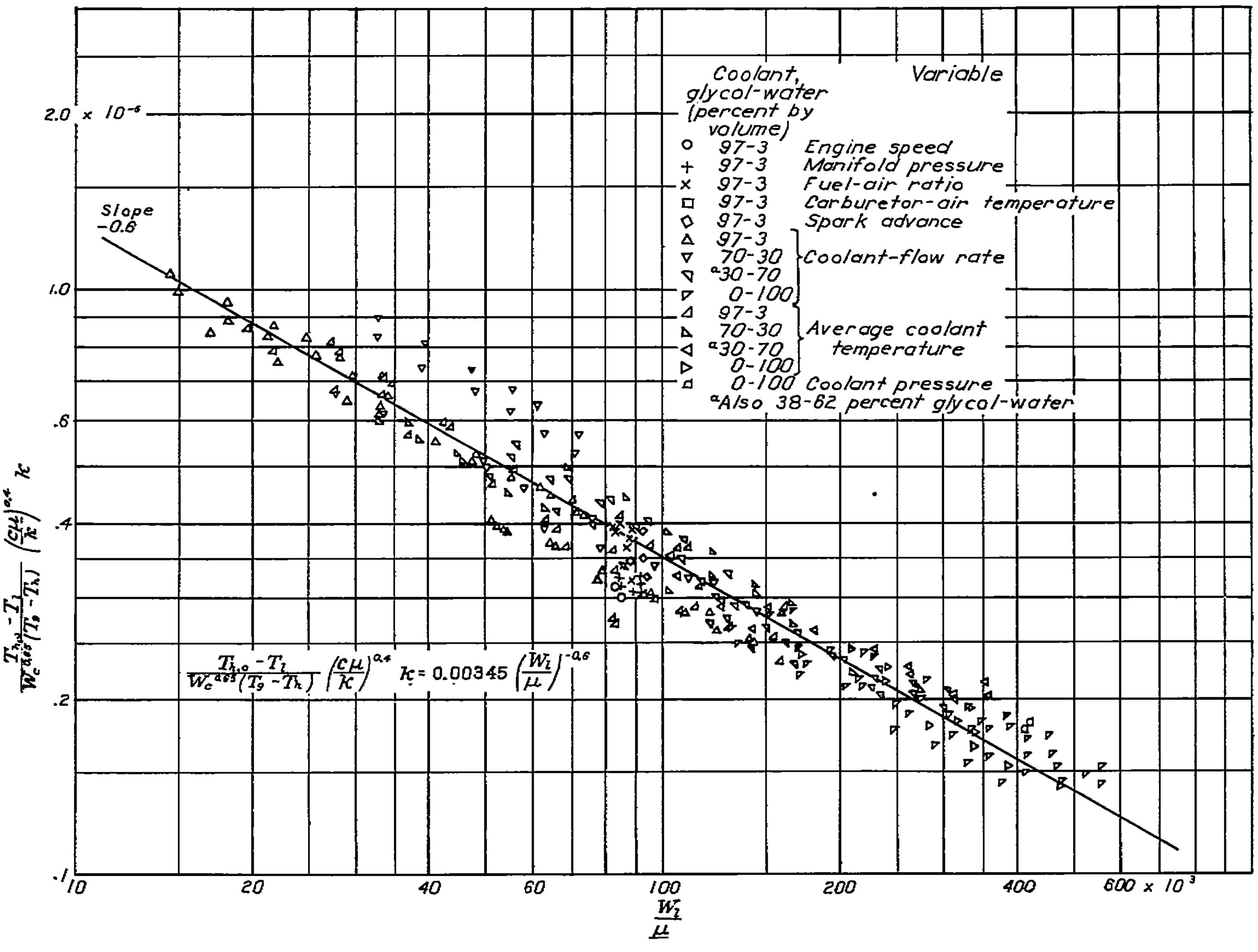


FIGURE 16.—Final correlation of average head temperatures for cylinder A on basis of T_h and h, o.

are plotted according to equation (5a). The slope of the average line through the data, which is equal to the exponent $-m$ of the coolant-flow-rate variable W_l/μ, is -0.6 for cylinder A and -0.5 for cylinder B. The difference in slope for cylinders A and B may be attributed to the difference in the coolant-flow conditions for the two cylinders. Except for a few cases, the deviations of the data points from the line represent differences in average head temperature of less than $\pm 10°$ F.

The correlation plot for the barrel of cylinder A (equation (6a)) is shown in figure 15 and the slope of the resulting line is -0.55. Slightly more scatter is obtained for the barrel than for the head; the correlation, however, is considered good.

The equations for the lines obtained in the preceding plots are:

Head, cylinder A (fig. 13)

$$\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]\left(\frac{c\mu}{k}\right)^{0.4}k=0.0042\left(\frac{W_l}{\mu}\right)^{-0.6} \qquad (7)$$

Head, cylinder B (fig. 14)

$$\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]\left(\frac{c\mu}{k}\right)^{0.4}k=0.00126\left(\frac{W_l}{\mu}\right)^{-0.5} \qquad (8)$$

Barrel, cylinder A (fig. 15)

$$\frac{T_{b,o}-T_l}{W_c^{0.65}(T_g-T_{b,o})}\left(\frac{c\mu}{k}\right)^{0.4}k=0.00260\left(\frac{W_l}{\mu}\right)^{-0.55} \qquad (9)$$

An alternative form of the correlation plot for the head involving both gas-side and liquid-side head temperatures (equation (5b)) is presented for cylinder A in figure 16. The slope of the average line through the data is -0.6, which agrees with that obtained in figure 13 for this cylinder. The equation for the line is

$$\frac{T_{h,o}-T_l}{W_c^{0.65}(T_g-T_h)}\left(\frac{c\mu}{k}\right)^{0.4}k=0.00345\left(\frac{W_l}{\mu}\right)^{-0.6} \qquad (10)$$

The apparent difference in the value of the constant B_5 in equations (7) and (10) is attributed to the fact that the values of $T_{h,o}$ used in this correlation are probably not representative of an average liquid-side head temperature that is consistent with the values of the average gas-side head temperature T_h, inasmuch as $T_{h,o}$ was obtained from the readings of only four thermocouples. The values of $T_{h,o}$ do, however, bear a linear relation to a consistent average temperature. The data of reference 3 show that individual temperatures give a straight-line relation with average temperatures over the entire range of engine and coolant conditions.

Relative cooling quality of coolant.—For illustrating the relative cooling quality of the individual coolants and indicating the relative importance of the individual physical properties of the coolant, the generalized correlation equations (7) and (9) for cylinder A are rewritten in the form of equations (5c) and (6b), respectively.

$$\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]W_l^{0.6}=0.0042\,\frac{\mu^{0.2}}{c^{0.4}k^{0.6}} \qquad (11)$$

$$\left[\frac{T_{b,o}-T_l}{W_c^{0.65}(T_g-T_{b,o})}\right]W_l^{0.6}=0.0026\,\frac{\mu^{0.15}}{c^{0.4}k^{0.6}} \qquad (12)$$

For a given engine operating condition, coolant-flow rate, and coolant temperature, the lower the left-hand term of equations (11) and (12), the lower are the cylinder-wall temperatures T_h and $T_{b,o}$, and hence the better is the cooling quality of the coolant. It is seen that the left-hand terms of equations (11) and (12) are equal to functions of the physical properties of the coolant (μ, c, and k), which, in turn, depend upon the composition and the temperature of the coolant. The coolant physical-property factors (right-hand terms) of equations (11) and (12) have been calculated from the data in figures 4 and 5 for each coolant and are plotted against coolant temperatures in figures 17 and 18, respectively. Comparison of the separate curves for each coolant indicates the better cooling quality of water and the less concentrated solutions of ethylene glycol. From equations (11) and (12),

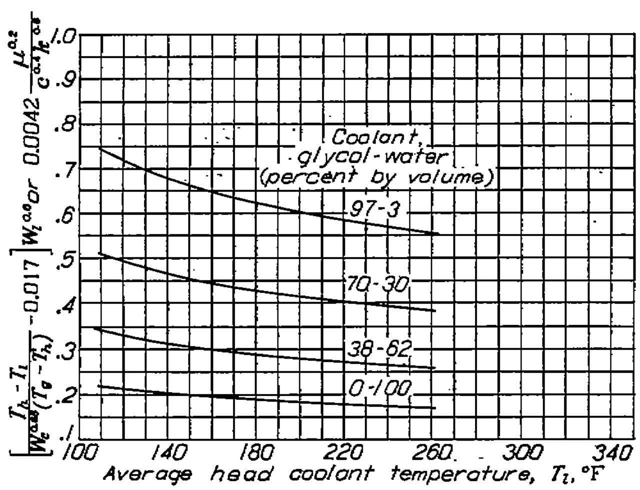


FIGURE 17.—Effect of coolant composition and temperature on $\left[\frac{T_h-T_l}{W_c^{0.65}(T_g-T_h)}-0.017\right]W_l^{0.6}$ or $0.0042\frac{\mu^{0.2}}{c^{0.4}k^{0.6}}$. Cylinder A.

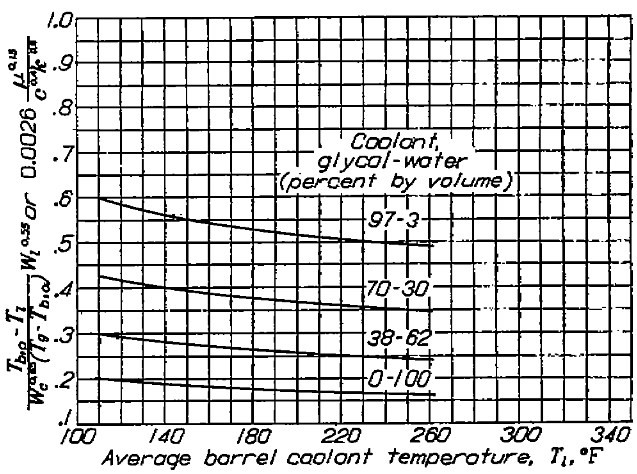


FIGURE 18.—Effect of coolant composition and temperature on $\frac{T_{b,o}-T_l}{W_c^{0.65}(T_g-T_{b,o})}W_l^{0.55}$ or $0.0026\frac{\mu^{0.15}}{c^{0.4}k^{0.6}}$. Cylinder A.

it is evident that high thermal conductivity and specific heat and low viscosity are desirable from forced-convection cooling considerations and that, for a given percentage change in these properties, the cooling is affected most by the change in thermal conductivity, somewhat less by the change in specific heat, and only slightly by the change in viscosity. The relative magnitude of the effects of these properties as indicated by the exponents of equations (11) and (12) may, of course, be expected to vary somewhat with different engines.

CONCLUSIONS

For the ranges of conditions investigated in previously reported tests conducted with two liquid-cooled cylinders using water and aqueous ethylene glycol solutions as coolants, it has been found that:

1. The equation derived from an analysis based on nonboiling forced-convection heat-transfer theory of the cooling processes in liquid-cooled engine cylinders provide satisfactory methods of correlating the average head and barrel temperatures with the engine operating conditions and the flow rate, the temperature, and the physical properties of the coolants.

2. The physical properties of the coolant appearing in these equations in order of their importance in determining the heat-transfer quality of the coolants are the thermal conductivity, the specific heat, and the viscosity. The cooling performance of the various coolants investigated is adequately correlated by these physical properties in the correlation equation.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, *October 1, 1946.*

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