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PISTON HEAT-TRANSFER COEFFICIENTS ACROSS AN OIL FILM IN A SMOOTH-WALLED PISTON RECIPROCATING-SLEEVE APPARATUS

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ADVANCE RESTRICTED REPORT

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SUMMARY

Tests were conducted with a heat-transfer apparatus that simulates the piston-cylinder-wall relation by means of a stationary, electrically heated, smooth-walled aluminum piston and a reciprocating steel sleeve separated by an oil film. Piston and sleeve temperatures were obtained for a range of heat inputs from 1.0 to 7.6 Btu per second, speeds from 200 to 1000 rpm, steady side thrusts from 10 to 150 pounds, and a range of piston-clearance oil-supply rates from 2 to 20 pounds per hour. The range of average temperatures observed was 200° F to 455° F for the piston and 150° F to 290° F for the sleeve.

The tests showed that the piston heat-transfer coefficient increased rapidly with an increase in the average oil-film temperature, increased with speed, and increased with an increase in the supply of oil to the piston clearance space. Variation of the steady side thrust over a range of 10 to 150 pounds had no significant effect on the piston heat-transfer coefficient.

A fair correlation of the piston heat-transfer coefficient as a function of the average oil-film temperature or the average piston temperature, the average sleeve velocity, and the piston-clearance oil-supply rate was obtained. The piston heat-transfer coefficient varied as the 1.15 power of the average oil-film temperature, directly with the average piston temperature, as the 0.27 power of the average sleeve velocity, and as the 0.35 power of the piston-clearance oil-supply rate for the ranges of conditions specified.

The piston heat-transfer coefficient could also be fairly well correlated as a function of a Reynolds and a Prandtl number based on the average or the maximum sleeve velocity, the piston clearance, and the physical properties of the lubricating oil; the Nusselt number varied as the 0.30 power of both the Reynolds and the Prandtl numbers.

INTRODUCTION

Adequate piston cooling has long been one of the critical factors limiting the specific output of aircraft engines. Satisfactory analysis of the piston-cooling problem has been hindered primarily because of the slight and uncertain knowledge of the factors controlling the heat-transfer processes between the piston and cylinder wall. These processes are complicated by the presence of an oil film and piston rings as well as by the occurrence of reciprocating motion, piston friction, and side thrust.

As part of a program for the study of piston cooling, the NACA in 1940 developed a satisfactory method of measuring piston temperatures at high speeds (reference 1) using thermocouples whose circuits were completed by contacts at bottom center. This method was then employed in an investigation of piston temperatures in an air-cooled engine in which the variations of piston temperature with various operating conditions were independently determined (reference 2). A satisfactory correlation of these test data could not be obtained because of the difficulty in evaluating the variation of the surface heat-transfer coefficient between the piston and the cylinder wall with the different engine operating conditions.

In order to obtain an insight into the factors affecting the piston heat-transfer coefficient, there was constructed by the NACA an apparatus that simulates the relation of the piston and the cylinder wall and provides controlled heat flux, operating speed, side thrust, and rate of supply of lubricating oil to the piston clearance space and permits variation of the number and type of piston rings. The piston in this apparatus is a stationary aluminum piston enclosing an electrical heater unit and the cylinder wall is a reciprocating steel sleeve.

The tests reported herein present the results of the first phase of an investigation of some of the factors affecting the heat-transfer coefficients of a smooth-walled piston, that is, a piston on which no rings were installed. The variation of average piston and reciprocating-sleeve temperatures with heat flux, operating speed, side thrust, and rate of piston-clearance oil supply was investigated. The piston heat-transfer coefficients were correlated as functions of average oil-film temperature or average piston temperature, average sleeve velocity, and rate of supply of lubricating oil to the piston clearance space.

SYMBOLS

A_p	heat-transfer area of the piston wall
c_p	specific heat of fluid at constant pressure
D	characteristic dimension or hydraulic diameter (piston clearance)
F	piston side thrust
H	heat flux from piston to sleeve through oil film
h	piston heat-transfer coefficient: rate of heat transfer per unit area per unit temperature difference between piston and cylinder or sleeve
k	thermal conductivity of fluid
T_f	average oil-film temperature, $\frac{1}{2}(T_p + T_s)$
T_p	average piston temperature
T_s	average cylinder wall or sleeve temperature
V_f	average fluid velocity
V_s	average reciprocating-sleeve velocity
W	rate of oil supply to piston clearance space
μ	absolute viscosity of fluid
ρ	density of fluid
a_1, a_2, a_3	constants
n,r,r', s,t,y	exponents

ANALYSIS

During engine operation, the piston receives heat from the hot combustion gases through its crown and transfers this heat to the cylinder wall through an oil film via the ring belt and skirt and to the crankcase air and oil from the internal surfaces of the piston. When only the heat transferred to the cylinder wall is considered, the piston heat-transfer coefficient may be written as

$$h = \frac{H}{A_p\ (T_p - T_s)} \tag{1}$$

If it is assumed that the transfer of heat from piston to cylinder wall through the oil film is effected by a mechanism similar to that controlling forced-convection heat transfer for the flow of fluids through tubes without phase change, the piston heat-transfer coefficient may be expressed by the familiar relation obtained from dimensional analysis

$$\frac{hD}{k} = f\left(\frac{DV_f\rho}{\mu}, \frac{c_p\mu}{k}\right) \tag{2}$$

Specific Apparatus Variables

The physical properties of the fluid (the lubricating oil) are functions of the average oil-film temperature T_f taken as the mean of the average piston temperature T_p and the average sleeve temperature T_s. The characteristic dimension, or piston clearance, D is taken as the difference between the piston and the sleeve diameters (hydraulic diameter of the clearance space based on the total wetted surface); the piston clearance is effectively a function of T_f.

An average fluid velocity V_f as usually employed in equation (2) does not exist in the present application. The average oil-film velocity is related to the average piston velocity or average sleeve velocity V_s of the subject apparatus proportionally to the operating speed and is therefore used instead of the average fluid velocity. Equation (2) then becomes

$$h = f\ (T_f, V_s) \tag{3}$$

The piston side thrust F and the rate of supply W of lubricating oil to the piston clearance space are two pertinent variables that may have an appreciable effect on the piston heat-transfer coefficient. Incorporating these variables as additional functions, equation (3) may be replaced by

$$h = f\ (T_f, V_s, F, W) \tag{4}$$

Assuming that the foregoing function of h with each variable takes the following form by means of which the effects of the independent variables considered may be evaluated, equation (4) may be written

$$h = a_1 \, (T_f)^r \, (V_s)^s \, (F)^t \, (W)^y \tag{5}$$

For convenience T_p may be used to approximate T_f in equation (5) as a measure of the effect of the physical properties and piston clearance; therefore,

$$h = a_2 \, (T_p)^{r'} \, (V_s)^s \, (F)^t \, (W)^y \tag{6}$$

Additional phenomena, such as friction heating between the piston and the cylinder wall and the reciprocating motion of the piston, further complicate the piston heat-transfer processes. As a result, neither equation (5) nor (6) may provide a complete correlation of the test data; the present tests were run to substantiate their applicability.

The method of evaluating the exponents in the assumed relation given in equation (5) is as follows: A log plot of h against T_f for constant V_s, F, and W will determine the exponent r on T_f. A second plot of $\frac{h}{(T_f)^r}$ against V_s for constant F and W will establish the exponent s on V_s. A subsequent plot of $\frac{h}{(T_f)^r (V_s)^s}$ against F for constant W will determine exponent t. A final plot of $\frac{h}{(T_f)^r (V_s)^s (F)^t}$ against W will serve to determine the exponent y on W. Fair correlation of the test data will verify the chosen parameters as those representing the piston heat-transfer processes. A similar procedure may be followed for equation (6) using T_p in place of T_f.

General Correlation

An alternative method of correlating the data using the nondimensional parameters in equation (2) may be applicable to the piston heat-transfer process. Although the flow of fluids through tubes, for which equation (2) is derived, is admittedly different in many respects from the reciprocating relative movement of the oil film, the piston, and the cylinder wall, there is some similarity between the two processes.

Values of Reynolds number $\frac{DV_s\rho}{\mu}$ calculated for the present apparatus using the average or the maximum sleeve velocity, piston clearance, and physical properties of the lubricating oil evaluated at T_f fall within the laminar region for flow through tubes. Correlations of heat-transfer data for laminar flow through tubes (reference 3) indicate that the Nusselt number $\frac{hD}{k}$ varies as the same power of the Reynolds and Prandtl $\frac{c_p\mu}{k}$ numbers (about the 1/3 power). Hence, a likely basis for correlation of the present data would be a plot of the expression

$$\frac{hD}{k} = a_3 \left(\frac{DV_s\rho}{\mu} \; \frac{c_p\mu}{k}\right)^n \tag{7}$$

Satisfactory correlation of the data at values of piston-clearance oil-supply rate sufficient to insure an adequate oil film will substantiate the use of equation (7) to represent the piston heat-transfer process in the present apparatus.

APPARATUS

A photograph of the test setup showing the general arrangement of the equipment is shown in figure 1. The piston reciprocating-sleeve apparatus has a $5\frac{3}{4}$-inch stroke and was mounted on a two-cylinder engine crankcase. The second cylinder contained a dummy piston and was used only for balancing purposes. The apparatus was driven by an adjustable-speed motor, the speed of which was indicated by an aircraft-type tachometer. Vibration of the setup limited the speed to about 1000 rpm. Power input to the piston-heater unit was controlled by a voltage regulator and was measured with a wattmeter; power inputs up to 8 kilowatts were obtained.

Heat-transfer apparatus. - The arrangement and construction details of the piston reciprocating-sleeve apparatus are shown in figure 2. The apparatus employs an inversion of the usual engine configuration; the moving piston was replaced by a stationary, electrically heated piston and the cylinder wall was replaced by a reciprocating steel sleeve. The reason for the inversion was the anticipated difficulty of reciprocating the electrically heated piston.

Piston. - An aluminum cylinder having a wall thickness of 29/64 inch, an outside diameter of $5\frac{29}{32}$ inches, and a length of $5\frac{29}{32}$ inches was used as the piston. An electric-heater coil centered in an aluminum casting was used to provide a heat flux; this heater core was fitted closely within the piston wall. The ends of the piston were sealed by steel plates and asbestos sheets interspersed with polished aluminum radiation shields. A predominantly outward radial flow of heat was insured by a dead-air space and a stainless-steel radiation shield within the piston core. The entire piston assembly was suspended by a fixed support rod through a pin bearing.

Reciprocating sleeve. - The reciprocating sleeve consisted of two thin steel cylinders: a cylinder 1/16 inch thick was shrunk over a cylinder 1/8 inch thick so as to enclose thermocouple wires. The piston clearance was 0.035 inch at a temperature of 75° F.

Piston-clearance oiling ring. - The piston and reciprocating sleeve were lubricated by oil supplied to a piston-clearance oiling ring mounted above the piston. (See fig. 2.) The oil was supplied to the oiling ring from the crankcase lubricating system as shown in figure 3(a). The oil entered the annular groove in the oiling ring through two 1/4-inch tubes. Two keystone rings, used as oil seals and wipers, permitted a flow of oil to the clearance space between the reciprocating sleeve and the piston.

Barrel. - A steel barrel enclosing both the piston and the sleeve served as a cross-head guide for the reciprocating sleeve and provided a cooling jacket with the cooling oil in direct contact with the outer surface of the sleeve. (See fig. 2.) Contracting rings in the cross-head guide were used as oil seals.

Side-thrust device. - The piston was suspended from the center of a beam mounted on self-alining ball bearings and supported by the barrel. The bracket that held the piston from the center of the beam was extended at right angles to the piston axis to form a bell crank; weights hung from the horizontal arm provided a steady side thrust of the piston on the sleeve. A pulley permitted the reversal of this steady side thrust. The thrust arm and pulley are shown in figure 1.

Thermocouple installation. - The locations and installation details of the piston and sleeve thermocouples are shown in figure 4. Temperatures were taken at 12 locations on the piston by means of chromel-constantan thermocouples peened into the piston 1/16 inch from the outer surface. The thermocouple wires were insulated from each other and from the aluminum up to the hot junction by flexible glass sleeving so that the temperatures measured were essentially surface temperatures.

Reciprocating-sleeve temperatures were obtained at 11 locations by thermocouples, the circuits of which were closed by contacts for 28 crank-angle degrees at bottom center. (See reference 1 for details.) The thermocouple wires were housed in helical grooves between the two shrunk cylinders composing the reciprocating sleeve. The wires were sealed in the grooves with vitreous cement and were soldered in the ends of the grooves with soft solder of high melting point 3/32 inch from the inner surface of the sleeve. Two of the 11 helical grooves contained complete chromel-constantan thermocouples; the other 9 contained only one thermocouple wire, the material of the steel sleeve being utilized as the other thermocouple element. Figure 4(c) shows the installation on the thrust surface of the sleeve; the complete thermocouple on this surface was used as a reference junction for the other thermocouples. The thermocouple wires were brought out to the contact blocks at the top of the sleeve.

Two thermopiles, consisting of four chromel-constantan thermocouples in series were used to measure the temperature of the cooling oil into and out of the cooling jacket. A single thermocouple indicated the temperature of the oil entering the rotameter.

The thermal electromotive forces of all thermocouples were measured by a portable, precision-type potentiometer in conjunction with an external spotlight galvanometer having a sensitivity of 0.007 microampere per millimeter. Temperature measurements are believed to be accurate within $\pm 1^{\circ}$ F.

Oil systems. - The lubricating and cooling-oil systems for the piston reciprocating-sleeve apparatus are schematically shown in figure 3; both systems employed SAE 30 oil. The cooling-oil flow rate was measured by a calibrated rotameter. Oil coolers were provided in both systems for temperature control; the larger exposed oil pipes were lagged with wool felt. The crankcase was kept dry by a scavenging pump.

METHODS AND TESTS

Tests were conducted on the piston reciprocating-sleeve apparatus for a range of values of heat input, operating speed, side thrust, piston-clearance oil-supply rate, and average sleeve temperature. A few series of tests were made in which the side thrust was completely reversed by means of the reverse-thrust pulley (fig. 1). A constant average sleeve temperature was difficult to maintain over the range of the other variables with the available range of control of cooling-oil temperature and flow rate; the cooling-oil temperature and flow rate were therefore arbitrarily kept constant.

Piston and sleeve temperatures were obtained over the following range of operating conditions:

Heat input, Btu per second	1.0-7.6
Speed, rpm	200-1000
Side thrust, pounds	10-150
Clearance-oil supply rate, pounds per hour	2-20
Cooling-oil temperature, °F	110-170
Cooling-oil flow rate, pounds per minute	10-85

With this range of conditions, the following range of temperatures was observed:

Average piston temperature, °F	200-455
Average sleeve temperature, °F	150-290

When each of the operating factors was separately varied, the other factors were kept approximately constant. Several series of tests were run for each variable with the other operating conditions at different constant values to confirm the trends at different temperature and speed levels. A summary of these test conditions is included with the test data in table I.

The physical properties of the oil (SAE 30) used in these tests are shown in figure 5 as functions of temperature. Specific-heat and thermal-conductivity data were taken from reference 4, density data from reference 5, and absolute-viscosity data from measurements made at the NACA Cleveland laboratory.

The variation of piston and sleeve diameters with average temperature is presented in figure 6 as calculated from the measured diameters at 75° F and the respective expansion coefficients of aluminum and steel. The curves provide means for evaluating the piston clearance under any condition of operation encountered in the tests. The piston clearance calculated from figure 6 at observed average piston and sleeve temperatures is shown to be a function of average oil-film temperature in figure 7, in which representative data at piston-clearance oil-supply rates of 5 and 12 pounds per hour are presented.

The piston-clearance oil-supply rate was kept constant at either approximately 12 or 5 pounds per hour except in those tests in which the piston-clearance oil-supply rate was varied. The flow to the oiling ring was controlled by varying the feed-line pressure by means of a needle valve. The pressure drops across the needle valve were calibrated against the piston-clearance oil-supply rates.

Above a piston-clearance oil-supply rate of about 20 pounds per hour, the space above the piston filled and overflowed, which indicated that, for the given apparatus, this flow was approximately the largest that would pass by the piston through the existing clearance space. A few runs were made, however, with piston-clearance oil-supply rates in excess of 20 pounds per hour.

The pressure of the oil entering the crankcase was kept at 30 pounds per square inch and the crankcase-oil temperature in the reservoir at approximately 110° F. Sufficient time was allowed after a change in operating conditions to insure equilibrium before readings were taken.

The average piston temperature T_p was taken as the average of the temperature indications of the 12 equally spaced thermocouples shown in figure 4(b). The average sleeve temperature T_s was taken as one-fourth of the sum of the averages of the temperature indications of the thermocouples located in each quadrant. The piston heat-transfer area was taken as 1.312 square feet. The piston heat-transfer coefficient between the piston and the reciprocating sleeve was calculated from equation (1) using the electrically measured heat input.

The heat rejected to the cooling oil was calculated for heat-balance purposes as the product of the cooling-oil flow, the temperature rise of the oil flowing through the cooling jacket, and the specific heat evaluated at the average cooling-oil temperature.

More tests than were required to establish the effect of the variables were made; test results are not presented for exploratory and check runs.

RESULTS AND DISCUSSION

A summary of the test results for all conditions is presented in table I.

Heat balance. - A plot of the heat rejection to the cooling oil against the electrical heat input to the piston is shown in figure 8 for speed ranges of 200 to 600 and 600 to 1000 rpm. The generally lower heat rejection to the cooling oil is considered, for the most part, to be due to a heat loss from the reciprocating sleeve to the air. Thermal losses from the ends of the piston are estimated to be less than 2 percent of the electrical heat input.

The question arises of whether the circulation of oil through the piston clearance space carries off an appreciable portion of the total heat flux, thereby decreasing the actual amount of heat transferred to the sleeve and making the calculated heat-transfer coefficients based on electrical heat input fictitiously high. Conservative estimates of the heat carried away by the lubricating oil circulating through the piston clearance space, assuming a temperature rise from the reservoir-oil temperature of 110° F to the average oil-film temperature and an average specific heat of 0.50 Btu per pound per °F, indicate that these losses for most of the tests employing piston-clearance oil-supply rates of 5 and 12 pounds per hour could not exceed 3 and 6 percent of the electrical heat input, respectively. The largest portion of the electrical heat input is therefore transferred across the oil film to the reciprocating sleeve.

Figure 8 shows that more heat was rejected to the cooling oil in the higher speed range than in the lower speed range for the same electrical heat input. This condition was undoubtedly the result of increased friction heating occurring in the higher speed range. The largest part of the friction heating is developed between the outer sleeve surface and the barrel and compression oil-sealing rings. Although this friction may have considerable effect on the heat balance, it should not appreciably affect the the calculated heat-transfer coefficients between the piston and the inner sleeve surface. The scatter of the data at any one speed was probably due to varying thermal losses from the exterior of the barrel to the atmosphere with different cooling-oil temperatures and flows and to the difficulty of accurately measuring the small temperature rise of the cooling oil at the higher rates of flow.

Temperature distribution. - The temperature distribution for two typical runs that are representative of the range of powers, speeds, and piston-clearance oil-supply rates encountered in the tests is presented in figures 9 and 10. The peripheral distribution of the temperature around the piston and the reciprocating sleeve is shown in figure 9(a); the plotted temperatures are the averages of the thermocouple indications in each quadrant. The temperature difference between the piston and sleeve is greatest at the anti-thrust surface and decreases to a minimum at the thrust surface.

Figure 9(b) shows the axial variation of temperature along the thrust surface of the sleeve. The fact that the temperature was highest at the center of the sleeve was expected, inasmuch as this point is always in contact with the hot piston surface; the ends of the sleeve, on the other hand, are alternately heated by the piston and cooled by the surrounding air.

Isothermal patterns for both the piston and the sleeve for the two representative runs just discussed are presented in figure 10; the piston and sleeve surface developments are drawn to the same scale as shown in figure 4. Perpendiculars to isothermals indicate heat-flow paths and, if these are visualized, it may be seen that in addition to a radial flow across the piston clearance space there is a secondary circumferential heat flow in both the piston and sleeve walls. The heat flow in the piston is from the antithrust to the thrust side; in the sleeve, the flow is from the thrust to the antithrust side. An estimate of the circumferential flow of heat in the piston was obtained from simple calculations based on the cross-sectional area of the piston wall, the thermal conductivity of the aluminum, the average temperature difference measured between the antithrust and thrust side of the piston, and the two parallel flow paths, each of a length equal to half the piston circumference. The calculations indicated that the heat conducted circumferentially through the piston walls is less than 3 percent of the total heat input. Accordingly, the temperature data shown in figures 9 and 10 may be used as approximate measures of the local heat-transfer coefficients. The circumferential variation of the local heat-transfer coefficient may be attributed to the variations in the clearance space around the piston resulting from steady side thrust.

Heat input. - The variation of average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient with electrical heat input is shown in figure 11. The temperature level at which the apparatus is operated was controlled primarily by heat input. Results show an increase in piston heat-transfer coefficient with an increase in heat input; this variation will be shown to be mainly an effect of a variation with temperature of the physical properties of the lubricating oil and the clearance between the piston and the sleeve.

Speed. - In figure 12, h, T_p, T_f, and T_s are plotted against average sleeve velocity. (A scale of speed values is given in the figure for convenience.) An increase in piston heat-transfer coefficient with increase in speed was obtained. Figure 12 presents the combined effect of speed and average oil-film temperature on h, inasmuch as both conditions varied; the fact that h appreciably leveled off at a value of V_s of 16 feet per second may have been due to the decrease in temperature with increase in speed. The independent effect of speed on h is isolated in a subsequent plot.

Average sleeve temperature. - The variation of h, T_p, and T_f with T_s is presented in figure 13. Data are shown in which T_s was varied by varying both cooling-oil temperature and flow rate. The increase noted in h is attributed to the increase in T_f.

Side thrust. - The effect of a steady side thrust on the average piston, oil-film, and sleeve temperatures and on piston heat-transfer coefficient is shown in figure 14. The results show a slight decrease in piston temperature with an increase in side thrust to about 50 pounds; at greater side thrusts, T_p is constant. The sleeve temperature is practically constant for the entire range of side thrusts tested. For all practical purposes, therefore, T_p, T_f, T_s, and h are independent of a steady piston side thrust as measured in the test apparatus.

Piston-clearance oil-supply rate. - The variation of average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient with the rate of supply of oil to the piston-clearance oiling ring is shown in figure 15. When the other operating conditions are constant, h may be seen to increase as the piston-clearance oil-supply rate is increased. The trend shown is not the pure effect of piston-clearance oil-supply rate, inasmuch as the average oil-film temperature also varied; the independent variation of h with W is determined in a later plot. At a piston-clearance oil-supply rate of 12 pounds per hour, h levels off appreciably as a result of the decrease in temperature with increase in supply rate.

As previously indicated, the maximum possible amount of heat that could be removed by the clearance oil at a supply rate of 12 pounds per hour was 6 percent of the electrical heat input. At this flow rate, therefore, the apparent increase in h due to the heat removal by the clearance oil would not exceed 6 percent, whereas the indicated increase in figure 12 is 60 percent above the value at the lowest observed flow rate of 2 pounds per hour. Most of the increase may therefore be attributed to an actual improvement in the heat-transfer coefficient across the oil film with increased piston-clearance oil-flow rate.

By way of explanation of the improvement in h with increase in W, the variation of average temperature difference between the piston and the sleeve with W is plotted in figure 16 for four peripheral positions: thrust, antithrust, and two intermediate positions as indicated in the cross-sectional sketch. The data are the same as those shown in figure 15. The temperature differences on the thrust surface drop 10° F over the entire range of W; on the other hand, the temperature differences on the antithrust surface, where the clearance space is a maximum, decrease 100° F over the range of W. A decrease in the temperature differences of about 60° F at the intermediate peripheral positions is also observed.

The improvement in the average piston heat-transfer coefficient may therefore be attributed to a reduction of the thermal resistance of the clearance space at the antithrust and the two intermediate surfaces. It would appear that the increased rate of supply of oil establishes and maintains a more completely oil-filled clearance space with attendant improved heat-transfer properties.

CORRELATION OF RESULTS

Specific Variable Correlation

As indicated in the ANALYSIS, h is fundamentally a function of T_f that expresses the clearance and physical-properties effects of the lubricating oil on the heat transfer from the piston to the sleeve. The variation of h with T_f is shown in figure 17(a) for an average sleeve velocity of approximately 8.5 feet per second, a side thrust of 100 pounds, and a piston-clearance oil-supply rate of 12 pounds per hour. The plotted data include runs for variable electrical heat input and variable cooling-oil temperature. It may be seen that plotting h as a function of T_f to the 1.15 power provides a fair correlation of these test data.

For convenience, T_p may be used to approximate T_f as a basis for correlating the test data. Furthermore, inasmuch as the observed spread of T_p was greater than the spread of T_f for the range of operating conditions encountered in the tests, the use of T_p provides a more sensitive index of the variation of h. The variation of h with T_p for the same data presented in figure 17(a) is shown in figure 17(b). The trend of the data is best represented by a line of unity slope; hence, the exponent $r' = 1.00$.

In figure 14 it had been shown that h was practically independent of side thrust so that the effect of side thrust, as varied in the tests, is constant.

Figures 18(a) and 18(b), respectively, show the variation of $h/(T_f)^{1.15}$ and h/T_p with average sleeve velocity V_s. The slope of the line that best fits the data is 0.27, so that the exponent s equals 0.27. The piston heat-transfer coefficient, measured for stationary operation of the apparatus (with the sleeve at bottom center), is about one-half the heat-transfer coefficient measured under comparable operating conditions of average oil-film temperature, piston clearance, piston-clearance oil-supply rate, side thrust, and an average sleeve velocity of about 8 feet per second. The 0.27 power variation of h with V_s, which if extrapolated would predicate zero h at zero speed, is therefore restricted to the range of speeds tested.

The variation of $\dfrac{h}{(T_f)^{1.15}(V_s)^{0.27}}$ with W is shown in figure 19(a); figure 19(b) shows the variation of $\dfrac{h}{T_p(V_s)^{0.27}}$ with W. For the range of piston-clearance oil-supply rate from 2 to 20 pounds per hour, a line of slope 0.35 fits the data quite well. As previously mentioned, greater values of W cause the space above the piston to fill and overflow, indicating a maximum rate of oil circulation through the piston clearance. Values of $\dfrac{h}{(T_f)^{1.15}(V_s)^{0.27}}$ or $\dfrac{h}{T_p(V_s)^{0.27}}$ for the larger rates of oil supply are about the same as those observed at a W of 20 pounds per hour, verifying this value as approximately the maximum oil flow rate by the piston for the existing clearance. The value 0.35 for the exponent y on W is therefore limited to piston-clearance oil-supply rates below 20 pounds per hour for the data of the subject apparatus.

The logarithmic correlation plots presented (figs. 17 to 19) separate the effects of the variables on the piston heat-transfer coefficient. The previous curves (figs. 11 to 15) did not show pure trends because T_f varied during tests in which other variables were investigated.

The final correlation curve of h against the established parameters $(T_f)^{1.15}(V_s)^{0.27}(W)^{0.35}$ or $(T_p)(V_s)^{0.27}(W)^{0.35}$ is shown in figure 20. All the data presented in table I are plotted against these parameters. Included in figure 20(a) and 20(b) are series of runs with the thrust arm reversed so as to interchange the thrust and antithrust surfaces. The temperature distributions and the heat-flow paths were altered, but the effect of the variables on the piston heat-transfer coefficient was not changed.

The solid line in figure 20(a) represents the relationship

$$h = 1.78\ (T_f)^{1.15}\ (V_s)^{0.27}\ (W)^{0.35} \times 10^{-5} \qquad (8)$$

and in figure 20(b), the equation of the solid line is

$$h = 3.39\ (T_p)\ (V_s)^{0.27}\ (W)^{0.35} \times 10^{-5} \qquad (9)$$

in which T_f and T_p are expressed in $^{\circ}F$, V_s in feet per second, and W in pounds per hour.

Approximately the same degree of correlation is obtained with the average oil-film temperature as with the average piston temperature as the correlation basis over the range of operating conditions encountered in the tests. Dashed lines representing a ±10-percent deviation from the correlation curve show that, with the exception of a few runs, the data fall within these limits. Either equation (8) or equation (9), therefore, sums up all the effects of the controllable factors on the piston heat-transfer coefficient within the specified limits.

General Correlation

The general correlation involving the nondimensional parameters is presented in figure 21(a) and 21(b), where $\frac{hD}{k}$ is plotted against the product $\left(\frac{DV_s\rho}{\mu}\right)\left(\frac{c_p\mu}{k}\right)$ for all the test data at piston-clearance oil-supply rates of 5 and 12 pounds per hour, respectively. Physical properties, evaluated at the average oil-film temperature T_f, were taken from figure 5, the piston clearance was calculated from figure 6 at the observed average piston and sleeve temperatures, and h and V_s were taken as before. Reynolds numbers for the data of figure 20 based on average sleeve velocity, range from 70 to 660. Reynolds numbers based upon the maximum velocity occurring in the stroke, which is about 1.5 V_s range from 105 to 990.

A line of slope 0.30 fits the data fairly well; dashed lines representing ±10 percent deviation from the correlation curve are included. The tailed points which fall well below the curve in figure 21(b) are for runs at the lowest heat input (0.95 Btu/sec), where the precision of measurement is poor. The fact that the absolute values of $\frac{hD}{k}$ are lower for a piston-clearance oil-supply rate of 5 pounds per hour than for 12 pounds per hour may be attributed to less complete filling of the clearance space with oil at the lower supply rate and hence a reduction in effective heat-transfer area. The region of the piston and the cylinder separated by an air gap is considered to be an ineffective heat-transfer area because of the decidedly inferior heat-transfer properties of air as compared with oil.

Although a fair correlation of the data is obtained through use of equation (7), it is recognized that the amount and the scope of data obtained is insufficient to place too much confidence in the validity of this type of correlation.

CONCLUSIONS

From tests of a heat-transfer apparatus simulating the usual relation between piston and cylinder wall by means of an electrically heated smooth-walled aluminum piston and a reciprocating steel sleeve separated by an oil film, it was found that the piston heat-transfer coefficient:

1. Increased with speed but began to level off at an average sleeve velocity of 16 feet per second as a result of the reduced oil-film temperatures occurring with increased speed.

2. Was not significantly affected by a variation in steady side thrust over a range of 10 to 150 pounds.

3. Increased with an increase in the piston-clearance oil-supply rate, but approached constancy with an increase in oil supply above about 12 pounds per hour as a result of the attendant decreasing oil-film temperature on the anti- and non-thrust surfaces.

4. Could be correlated fairly well as functions of the average oil-film temperature or the average piston temperature, the average sleeve velocity, and the piston-clearance oil-supply rate; the piston heat-transfer coefficient varied as the 1.15 power of the average oil-film temperature, directly with the average piston temperature, as the 0.27 power of the average sleeve velocity, and as the 0.35 power of the piston-clearance oil-supply rate within the range of conditions tested.

5. Could be correlated fairly well as functions of a Reynolds and a Prandtl number based on the average or the maximum sleeve velocity, the piston clearance, and the physical properties of the lubricating oil; the Nusselt number varied as the 0.30 power of both the Reynolds and Prandtl numbers.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio. October 3, [illegible]

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NATIONAL ADVISORY
COMMITTEE FOR AERONAUTICS

TABLE I - SUMMARY OF DATA AND RESULTS FOR PISTON RECIPROCATING-SLEEVE APPARATUS

Run	Electrical heat input H (Btu/sec)	Operating speed (rpm)	Piston-clearance oil-supply rate W (lb/hr)	Piston side thrust F (lb)	Cooling-oil flow rate (lb/min)	Average cooling-oil temperature (°F)	Specific heat of cooling oil (Btu)/(lb)(°F)	Heat rejection to oil (Btu/sec)	Heat-balance ratio (percent)	Average piston temperature T_p (°F)	Average sleeve temperature T_s (°F)	Average oil-film temperature T_f (°F)	Piston heat-transfer coefficient h (Btu)/(sec)(sq ft)(°F)	Correlation parameter $(T_f)^{1.15}(V_s)^{0.27}(W)^{0.35}$	Correlation parameter $T_p(V_s)^{0.27}(W)^{0.35}$
Variable heat input															
80	0.95	490	12	100	20	130	0.467	0.89	94	206	156	181	0.0249	1637	856
81	1.90	490			20	132	.468	1.43	75	239	168	203	.0351	1869	995
82	2.84	490			20	130	.467	2.17	76	278	183	230	.0392	2156	1157
83	3.79	490			20	130	.467	3.06	81	305	199	252	.0469	2392	1269
84	4.74	490			20	130	.467	3.90	82	341	212	276	.0482	2659	1419
85	5.69	490			20	130	.467	4.74	83	370	232	301	.0541	2935	1538
125	2.84	950			42	159	.480	2.76	97	292	205	249	.0428	2824	1452
126	3.41	950			40	152	.477	3.27	96	297	205	251	.0486	2848	1476
127	4.74	950			40	150	.476	4.51	95	337	221	279	.0536	3219	1674
128	5.35	950			40	154	.478	4.73	88	348	230	289	.0594	3349	1729
129	6.07	950			39	154	.478	5.34	88	366	241	304	.0637	3550	1820
130	6.64	950			39	153	.478	5.74	86	378	247	313	.0665	3673	1879
131	6.98	950			39	155	.478	6.06	87	387	250	319	.0668	3755	1925
161	.95	940			40	132	.468	1.67	176	199	156	178	.0290	1911	985
162	1.90	940			40	131	.467	2.34	123	227	167	197	.0415	2147	1123
163	2.84	940			39	132	.468	2.95	104	266	177	222	.0418	2466	1317
164	3.79	940			38	130	.467	3.62	96	292	187	240	.0473	2692	1445
165	4.74	940			41	132	.468	4.35	92	317	200	259	.0531	2945	1569
166	.95	540			41	150	.476	.94	99	210	171	191	.0319	1790	897
167	1.90	540			42	149	.476	1.65	87	244	184	214	.0415	2037	1042
168	2.84	540			43	153	.478	2.38	84	290	198	244	.0405	2369	1238
169	3.79	540			41	150	.476	3.05	80	325	209	267	.0428	2627	1388
170	5.69	540			39	150	.476	4.63	81	377	236	307	.0529	3088	1610
171	1.90	540			20	151	.477	1.48	78	258	186	222	.0346	2127	1102
172	3.79	540			20	154	.478	2.79	74	323	212	268	.0448	2638	1379
173	5.69	540			21	152	.477	4.19	74	376	238	307	.0541	3086	1605
249	5.69	940			60	142	.473	5.18	91	348	223	286	.0597	3310	1724
250	4.74	940			60	142	.472	4.37	92	323	211	267	.0555	3046	1600
251	3.79	940			59	141	.472	3.44	91	296	196	246	.0497	2775	1467
252	2.84	940			60	144	.474	2.63	93	272	187	230	.0438	2567	1348
253	1.90	940			60	139	.471	2.14	113	235	176	206	.0422	2264	1164
254	1.42	940			61	144	.474	1.54	108	219	175	197	.0423	2150	1085
255	.95	940			61	140	.472	1.34	141	202	164	183	.0328	1979	1002
405	1.90	560			60	140	.472	1.65	87	237	165	201	.0346	1913	1021
406	2.84	560			60	140	.472	2.25	79	273	180	227	.0400	2205	1178
407	3.79	560			60	141	.472	3.02	80	307	189	248	.0421	2437	1324
408	4.74	560			60	142	.472	3.78	80	339	205	272	.0464	2713	1462
409	5.69	560			60	140	.472	4.49	79	367	216	292	.0494	2948	1584
410	6.64	560			60	141	.472	5.21	78	393	234	314	.0547	3198	1696
411	7.58	560			59	141	.472	5.88	78	416	245	331	.0581	3398	1794
267	6.64	535	5	[a]100	60	140	.472	5.00	75	423	242	332	.0481	2476	1326
268	5.69	535			59	138	.471	4.44	78	395	228	312	.0447	2307	1238
269	6.64	530			60	141	.472	5.00	75	415	241	328	.0500	2440	1298
270	5.69	530			60	137	.470	4.54	80	394	228	311	.0449	2294	1233
271	4.74	530			60	142	.472	3.85	81	368	212	290	.0398	2116	1150
272	3.79	530			60	140	.471	3.16	83	331	195	263	.0365	1894	1036
273	2.84	530			63	142	.473	2.37	83	294	183	239	.0335	1699	920
274	1.90	530			62	140	.471	1.81	95	253	169	211	.0297	1471	792
285	3.79	950			59	140	.471	3.58	94	309	184	247	.0397	2061	1131
286	2.84	950			59	137	.470	2.84	100	276	176	227	.0365	1869	1017
287	1.90	950			60	139	.471	2.07	109	247	168	206	.0315	1691	904
363	7.58	560		100	61	142	.472	5.79	76	455	252	354	.0490	2706	1445
364	6.64	560			60	140	.471	5.47	82	434	234	334	.0435	2527	1378
365	5.69	560			60	142	.472	4.74	83	406	220	313	.0401	2343	1289
366	4.74	565			61	142	.472	3.94	83	376	203	290	.0359	2153	1196
367	3.79	565			60	140	.472	3.11	82	344	191	268	.0325	1969	1094
368	2.84	565			61	140	.472	2.19	77	299	182	241	.0318	1738	950
369	2.37	570			60	140	.471	1.89	80	278	171	225	.0290	1614	887
370	1.90	570			62	140	.472	1.35	71	256	164	210	.0271	1490	817
371	1.42	570			63	138	.471	1.13	80	232	158	195	.0252	1366	739
372	.95	560			61	138	.471	.72	76	206	151	179	.0226	1233	653

[a]Steady side thrust reversed.

TABLE I - SUMMARY OF DATA AND RESULTS FOR PISTON RECIPROCATING-SLEEVE APPARATUS - Continued

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

Run	Electrical heat input H (Btu/sec)	Operating speed (rpm)	Piston-clearance oil-supply rate W (lb/hr)	Piston side thrust F (lb)	Cooling-oil flow rate (lb/min)	Average cooling-oil temperature (°F)	Specific heat of cooling oil (Btu)/(lb)(°F)	Heat rejection to oil (Btu/sec)	Heat-balance ratio (percent)	Average piston temperature T_p (°F)	Average sleeve temperature T_s (°F)	Average oil-film temperature T_f (°F)	Piston heat-transfer coefficient h (Btu)/(sec)(sq ft)(°F)	Correlation parameter $(T_f)^{1.15}(V_s)^{0.27}(W)^{0.35}$	Correlation parameter $T_p(V_s)^{0.27}(W)^{0.35}$
Variable cooling-oil flow rate															
86	3.79	485	12	100	12	130	0.467	3.22	85	320	207	263	0.0440	2507	1326
87	3.79	515			13	131	.467	3.26	86	314	202	258	.0444	2500	1324
88	3.79	515			19	130	.467	3.18	84	313	198	255	.0432	2465	1319
89	3.79	520			30	130	.467	3.31	87	315	198	256	.0425	2476	1331
90	3.79	520			41	131	.467	3.12	82	309	197	253	.0444	2449	1307
91	3.79	525			58	131	.467	3.02	80	306	197	251	.0456	2425	1295
92	3.79	525			77	130	.467	2.11	56	303	193	248	.0452	2392	1283
147	6.64	940			85	153	.478	5.76	87	376	241	309	.0645	3607	1863
148	6.64	940			66	151	.477	5.78	87	374	240	307	.0650	3582	1853
149	6.64	940			48	153	.478	5.67	85	377	244	311	.0654	3630	1867
150	6.64	940			22	153	.478	5.28	80	385	248	317	.0635	3717	1908
Variable piston side thrust															
27	1.82	260	12	50	15	117	0.461	1.46	80	245	156	201	0.0268	1551	856
28	1.82	260		100	15	116	.461	1.49	82	246	161	204	.0281	1578	859
29	1.82	260		150	15	118	.461	1.43	79	245	159	202	.0277	1562	856
30	1.82	265		10	15	118	.461	1.44	79	245	159	202	.0277	1571	861
31	1.82	215		10	15	117	.461	1.39	76	252	162	207	.0265	1525	837
32	1.82	215		100	15	117	.461	1.36	75	251	166	209	.0281	1541	832
33	1.82	215		50	15	118	.462	1.33	73	251	163	207	.0271	1521	832
34	1.82	215		150	15	116	.461	1.42	78	250	162	206	.0271	1521	832
35	1.82	295		10	14	116	.463	1.39	76	242	156	199	.0277	1587	875
36	1.82	300		50	14	115	.462	1.41	77	239	155	197	.0284	1576	868
37	1.82	300		100	14	115	.462	1.39	76	239	154	197	.0281	1576	868
38	1.82	300		150	14	116	.463	1.41	77	239	157	198	.0291	1587	868
39	1.86	300		30	15	116	.461	1.47	79	242	154	198	.0277	1589	880
40	1.86	360		50	15	117	.461	1.48	80	238	156	197	.0297	1661	911
41	1.88	360		100	15	118	.461	1.49	79	238	156	197	.0300	1661	911
42	1.88	360		30	15	117	.461	1.50	80	238	154	196	.0293	1650	911
43	1.86	360		150	14	117	.461	1.45	78	235	157	196	.0312	1649	899
44	1.86	360		10	15	117	.461	1.50	81	235	153	194	.0297	1630	899
45	1.90	405		50	15	118	.461	1.45	76	235	158	197	.0323	1714	928
46	1.90	405		10	15	119	.462	1.51	79	234	157	196	.0323	1704	925
47	1.90	405		150	15	119	.462	1.44	76	234	159	197	.0331	1716	925
48	1.90	405		30	15	118	.461	1.47	77	236	157	197	.0315	1719	935
49	1.90	405		100	15	118	.461	1.44	76	236	159	198	.0323	1731	935
55	1.90	365		10	21	132	.468	1.26	66	246	168	207	.0319	1761	944
56	1.90	365		50	21	132	.468	1.36	72	244	169	206	.0332	1755	937
58	1.90	365		100	20	132	.468	1.51	79	247	170	208	.0323	1775	949
59	1.90	365		150	20	131	.467	1.49	78	246	168	207	.0319	1761	944
66	2.84	485		10	23	132	.468	2.38	84	276	183	229	.0400	2141	1145
67	2.84	485		50	21	132	.468	2.13	75	274	179	226	.0392	2104	1135
68	2.84	485		100	21	132	.468	2.18	77	276	184	230	.0405	2149	1145
69	2.84	485		150	20	132	.468	2.08	73	267	186	226	.0460	2106	1107
74	3.79	490		50	20	130	.467	3.08	81	311	198	254	.0440	2416	1293
75	3.79	490		100	20	130	.467	3.08	81	310	197	253	.0440	2409	1290
76	3.79	490		150	20	132	.468	3.09	82	310	199	254	.0448	2418	1290
77	3.79	490		10	20	130	.467	3.11	82	313	192	252	.0411	2392	1302
78	3.79	490		100	20	130	.467	3.13	83	309	195	252	.0436	2393	1266
79	3.79	490		50	20	130	.467	3.07	81	309	199	254	.0452	2418	1286
121	6.64	960		10	40	152	.477	5.50	83	386	249	318	.0635	3744	1922
122	6.64	960		50	39	151	.477	5.53	83	380	246	313	.0650	3683	1894
123	6.64	960		100	39	152	.477	5.47	82	379	247	313	.0659	3683	1889
124	6.64	960		150	39	152	.477	5.57	84	380	249	315	.0665	3708	1894
143	6.64	940		150	40	152	.477	5.91	89	381	245	313	.0640	3660	1887
144	6.64	940		100	39	152	.477	5.75	87	381	243	312	.0633	3645	1887
145	6.64	940		50	39	152	.477	5.72	86	380	244	312	.0640	3645	1882
146	6.64	940		10	38	150	.476	5.53	83	378	243	311	.0645	3630	1872

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TABLE I - SUMMARY OF DATA AND RESULTS FOR PISTON RECIPROCATING-SLEEVE APPARATUS - Continued

Run	Electrical heat input H (Btu/sec)	Operating speed (rpm)	Piston-clearance oil-supply rate W (lb/hr)	Piston side thrust F (lb)	Cooling-oil flow rate (lb/min)	Average cooling-oil temperature (°F)	Specific heat of cooling oil (Btu)/(lb)(°F)	Heat rejection to oil (Btu/sec)	Heat-balance ratio (percent)	Average piston temperature T_p (°F)	Average sleeve temperature T_s (°F)	Average oil-film temperature T_f (°F)	Piston heat-transfer coefficient h (Btu)/(sec)(sq ft)(°F)	Correlation parameter $(T_f)^{1.15} (V_s)^{0.27} (W)^{0.35}$	Correlation parameter $T_p(V_s)^{0.27} (W)^{0.35}$
Variable piston-clearance oil-supply rate															
313	3.79	560	6	100	60	140	0.471	3.42	90	325	191	258	0.0371	2003	1105
314	3.79		20		59	138	.471	3.25	86	295	186	241	.0456	2820	1522
315	3.79		7		59	139	.471	3.25	86	312	190	251	.0407	2046	1117
316	3.79		12		60	140	.471	3.23	85	298	187	243	.0448	2380	1280
317	3.79		4		60	140	.471	3.09	82	331	192	262	.0357	1767	969
324	1.90		3		59	153	.478	1.67	88	267	173	220	.0264	1309	708
325	1.90		5		60	152	.477	1.70	89	260	174	217	.0289	1539	827
326	1.90		6		61	153	.478	1.68	88	254	173	214	.0307	1614	863
327	1.90		11		60	152	.477	1.66	87	245	173	209	.0345	1941	1028
328	1.90		15		61	155	.479	1.52	80	239	173	206	.0377	2131	1114
329	1.90		20		60	155	.479	1.37	72	232	174	203	.0429	2315	1196
330	1.90		30		60	151	.477	1.25	66	229	170	200	.0421	-----------	-----------
331	1.90		45		60	156	.479	.99	52	228	170	199	.0429	-----------	-----------
332	1.90		4		61	155	.479	1.47	77	263	174	219	.0279	1438	769
394	3.79		2		60	143	.473	3.23	85	376	196	286	.0277	1534	867
395	3.79		2		61	140	.472	3.34	88	369	195	282	.0286	1511	850
396	3.79		9		59	138	.471	3.17	84	312	191	252	.0410	2245	1218
397	3.79		5		61	140	.472	3.24	85	337	197	267	.0355	1951	1072
398	3.79		19		60	139	.471	3.02	80	299	189	244	.0452	2811	1515
399	1.90		3		59	152	.477	1.65	87	282	177	230	.0237	1375	750
400	1.90		5		66	152	.477	1.68	88	264	182	223	.0304	1587	839
401	1.90		9		60	151	.477	1.63	86	250	177	214	.0341	1859	890
402	1.90		19		60	151	.477	1.49	78	232	174	203	.0429	2274	1173
403	1.90		30		61	153	.477	1.37	72	234	172	203	.0402	-----------	-----------
404	1.90		14		60	153	.478	1.42	75	243	177	210	.0377	2125	1106
Variable operating speed															
50	1.90	405	12	50	15	117	0.461	1.44	76	234	157	196	0.0323	1704	925
51	1.90	360			15	118	.461	1.45	76	239	157	198	.0303	1671	916
52	1.90	300			15	118	.461	1.46	77	243	159	201	.0296	1617	885
53	1.90	250			15	118	.461	1.46	77	248	159	204	.0279	1566	859
54	1.90	215			15	118	.461	1.40	74	252	163	208	.0279	1534	837
106	3.79	225		100	20	132	.468	2.87	76	349	215	282	.0371	2201	1173
107	3.79	225			20	130	.467	3.10	82	345	212	279	.0374	2177	1159
108	3.79	300			20	130	.467	3.17	84	333	206	270	.0391	2268	1212
109	3.79	800			17	128	.466	3.63	96	303	196	250	.0464	2710	1438
110	3.79	1015			19	131	.467	3.70	98	290	197	244	.0534	2808	1467
132	6.64	950			40	154	.478	5.73	86	382	243	313	.0626	3672	1898
133	6.64	800			40	154	.478	5.70	86	389	256	323	.0655	3635	1846
134	6.64	650			39	155	.478	5.47	82	396	253	325	.0609	3464	1777
135	6.64	500			39	154	.478	5.47	82	404	258	331	.0596	3294	1689
136	6.64	860			39	153	.478	5.54	83	386	250	318	.0640	3637	1867
137	6.64	730			39	152	.477	5.48	83	391	253	322	.0631	3537	1810
190	.95	450			61	127	.466	1.07	113	189	143	166	.0270	1447	768
191	.95	660			61	131	.467	1.17	123	194	148	171	.0270	1661	873
192	.95	820			63	135	.469	1.25	132	199	159	179	.0311	1855	949
193	.95	1020			61	140	.472	1.22	129	196	162	179	.0366	1969	992
194	5.69	460			60	140	.472	4.42	78	370	221	296	.0500	2832	1512
195	5.69	465			62	140	.472	4.41	78	375	224	300	.0494	2887	1538
196	5.69	640			61	141	.472	4.79	84	365	220	293	.0514	3061	1631
197	5.69	825			63	142	.472	5.10	90	359	225	292	.0556	3262	1717
198	5.69	1010			63	140	.472	5.13	90	345	218	282	.0587	3309	1743

TABLE I - SUMMARY OF DATA AND RESULTS FOR PISTON RECIPROCATING-SLEEVE APPARATUS - Continued

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

Run	Electrical heat input H (Btu/sec)	Operating speed (rpm)	Piston-clearance oil-supply rate W (lb/hr)	Piston side thrust F (lb)	Cooling-oil flow rate (lb/min)	Average cooling-oil temperature (°F)	Specific heat of cooling oil (Btu)/(lb)(°F)	Heat rejection to oil (Btu/sec)	Heat-balance ratio (percent)	Average piston temperature T_p (°F)	Average sleeve temperature T_s (°F)	Average oil-film temperature T_f (°F)	Piston heat-transfer coefficient h (Btu)/(sec)(sq ft)(°F)	Correlation parameter $\frac{(T_f)^{1.15}}{(V_s)^{0.27} (W)^{0.35}}$	Correlation parameter $\frac{T_p(V_s)^{0.27}}{(W)^{0.35}}$
Variable operating speed - Concluded															
233	7.58	300	12	100	62	142	0.473	5.76	76	437	275	356	0.0613	3114	1588
234	7.58	460			61	140	.472	5.88	78	426	264	345	.0613	3376	1741
235	7.58	560			60	140	.471	6.09	80	418	254	336	.0606	3459	1803
236	7.58	650			60	139	.471	6.25	82	413	251	332	.0613	3544	1853
237	7.58	800			60	138	.471	6.39	84	407	246	327	.0617	3688	1932
238	7.58	965			59	138	.471	6.48	85	397	234	316	.0610	3729	1982
243	6.64	340			60	146	.474	4.99	75	425	259	342	.0524	3089	1603
244	6.64	460			60	150	.476	5.32	80	413	250	332	.0534	3231	1689
245	6.64	680			61	149	.476	5.89	89	404	244	324	.0544	3491	1834
246	6.64	840			61	149	.476	5.86	88	386	239	313	.0592	3553	1856
247	6.64	575			60	148	.476	5.47	82	402	245	324	.0554	3340	1746
248	6.64	965			60	149	.476	5.93	89	383	238	311	.0600	3360	1910
256	7.39	1010			61	150	.476	6.65	90	399	249	324	.0646	3884	2015
257	7.39	930			61	150	.476	6.21	84	400	247	324	.0633	3801	1977
258	7.39	790			61	152	.477	6.07	82	406	258	332	.0655	3736	1920
259	7.39	645			62	150	.476	5.86	79	415	258	337	.0617	3600	1858
260	7.39	565			61	150	.476	5.85	79	423	259	341	.0591	3524	1827
261	7.39	430			61	150	.476	5.59	76	425	267	346	.0613	3330	1705
262	7.39	340			61	150	.476	5.41	73	442	280	361	.0598	3289	1667
263	7.39	245			61	151	.477	5.87	79	456	292	374	.0591	3117	1567
343	5.69	300			60	150	.476	3.70	65	390	242	316	.0504	2718	1419
344	5.69	420			60	152	.477	4.13	73	377	231	304	.0511	2847	1503
345	5.69	560			60	150	.476	4.37	77	372	223	298	.0501	3011	1605
346	5.69	660			59	150	.476	4.26	75	367	215	291	.0491	3063	1653
347	5.69	850			60	151	.477	4.74	83	347	219	283	.0583	3179	1674
412	3.79	250			61	140	.472	2.81	74	335	209	272	.0394	2176	1159
413	3.79	415			60	139	.471	2.87	76	316	201	259	.0432	2363	1257
414	3.79	560			60	140	.472	2.97	78	308	191	250	.0425	2462	1328
415	3.79	680			60	139	.471	3.14	83	303	188	246	.0432	2543	1376
416	3.79	875			60	140	.472	3.12	82	298	184	241	.0436	2658	1448
417	3.79	1000			62	142	.472	3.08	81	285	181	233	.0478	2650	1436
275	3.79	250	5	[a]100	60	142	.472	3.02	80	360	218	289	.0350	1714	917
276	3.79	415			60	139	.471	2.99	79	344	207	276	.0363	1872	1006
277	3.79	535			60	140	.472	3.21	85	341	199	270	.0350	1953	1069
278	3.79	650			60	140	.472	3.18	84	334	194	264	.0355	2005	1103
279	3.79	800			62	140	.471	3.59	95	321	190	256	.0379	2048	1120
280	3.79	925			61	142	.472	3.50	92	313	186	250	.0391	2076	1138
373	3.79	250		100	60	140	.472	3.00	79	372	209	291	.0305	1729	946
374	3.79	415			60	140	.472	2.78	73	347	197	272	.0331	1840	1015
375	3.79	560			59	140	.472	3.25	86	336	192	264	.0345	1926	1066
376	3.79	685			59	140	.472	3.11	82	327	193	260	.0371	2001	1096
377	3.79	875			59	140	.471	3.22	85	315	184	250	.0379	2043	1127
378	3.79	1000			61	141	.472	3.25	86	313	186	250	.0391	2118	1161
379	6.64	1000			60	139	.471	5.28	80	383	215	299	.0518	2601	1421
380	6.64	870			61	142	.472	5.23	79	398	219	309	.0486	2601	1422
381	6.64	670			60	141	.472	5.18	78	416	224	320	.0453	2524	1385
382	6.64	575			61	139	.471	4.77	72	421	233	327	.0463	2482	1345
383	6.64	410			60	139	.471	3.88	58	428	242	335	.0468	2329	1249
384	6.64	270			60	140	.471	4.49	68	455	263	359	.0453	2237	1182

[a]Steady side thrust reversed.

TABLE I - SUMMARY OF DATA AND RESULTS FOR PISTON RECIPROCATING-SLEEVE APPARATUS - Concluded

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

Run	Electrical heat input H (Btu/sec)	Operating speed (rpm)	Piston-clearance oil-supply rate W (lb/hr)	Piston side thrust F (lb)	Cooling-oil flow rate (lb/min)	Average cooling-oil temperature (°F)	Specific heat of cooling oil (Btu)/(lb)(°F)	Heat rejection to oil (Btu/sec)	Heat-balance ratio (percent)	Average piston temperature T_p (°F)	Average sleeve temperature T_s (°F)	Average oil-film temperature T_f (°F)	Piston heat-transfer coefficient h (Btu)/(sec)(sq ft)(°F)	Correlation parameter $(T_f)^{1.15} (V_s)^{0.27} (W)^{0.35}$	Correlation parameter $T_p (V_s)^{0.27} (W)^{0.35}$
Variable average cooling-oil temperature															
61	1.90	360	12	100	21	111	0.458	2.00	105	237	150	194	0.0286	1629	906
62	1.90	360			20	118	.461	1.82	96	242	156	199	.0290	1678	925
63	1.90	360			20	127	.466	1.62	85	250	167	209	.0300	1778	956
64	1.90	360			20	143	.473	1.36	72	257	178	218	.0315	1867	983
65	1.90	360			22	154	.478	1.51	79	265	191	228	.0337	1963	1014
100	3.79	525			19	123	.464	3.24	85	308	192	250	.0428	2419	1305
101	3.79	525			20	136	.470	2.78	73	314	203	258	.0448	2505	1331
102	3.79	525			19	127	.466	3.12	82	314	197	255	.0425	2480	1331
103	3.79	525			20	142	.472	2.84	75	316	211	263	.0473	2562	1338
104	3.79	525			20	152	.477	2.68	71	324	225	274	.0502	2687	1371
105	3.79	525			20	160	.482	2.68	71	332	233	282	.0502	2776	1407
138	6.64	935			40	129	.466	6.03	91	370	223	297	.0592	3441	1829
139	6.64	940			41	138	.470	5.92	89	372	231	302	.0617	3514	1844
140	6.64	940			44	148	.475	6.27	94	379	242	311	.0635	3630	1877
141	6.64	940			39	152	.477	5.95	90	385	247	316	.0631	3702	1908
142	6.64	940			41	162	.482	5.90	89	389	250	320	.0626	3755	1927
385	5.69	560	5		58	172	.487	3.94	69	415	245	330	.0439	2491	1317
386	5.69	560			61	164	.483	4.10	72	407	237	322	.0439	2422	1292
387	5.69	560			60	151	.477	4.32	76	401	229	315	.0434	2362	1273
388	5.69	560			61	138	.471	4.51	79	389	217	303	.0434	2256	1234
389	5.69	560			57	125	.464	4.23	74	382	204	293	.0419	2175	1213
390	1.90	560			61	121	.463	1.67	88	240	148	194	.0271	1353	762
391	1.90	560			61	136	.470	1.21	64	245	161	203	.0297	1426	778
392	1.90	560			60	149	.476	1.14	60	253	170	212	.0300	1496	802
393	1.90	560			61	158	.480	1.15	61	261	177	219	.0297	1556	829

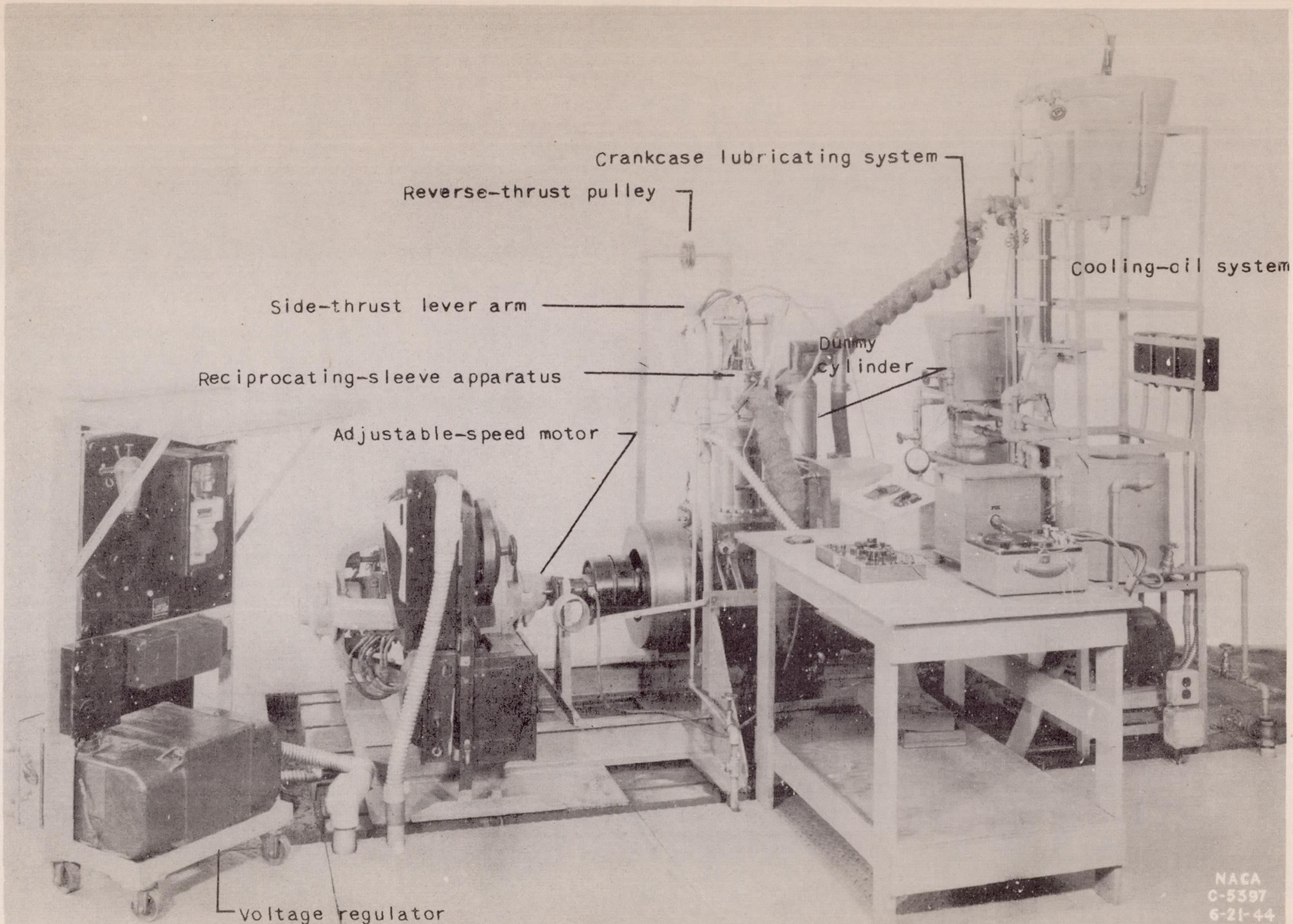


Figure 1. - Piston-reciprocating-sleeve test setup.

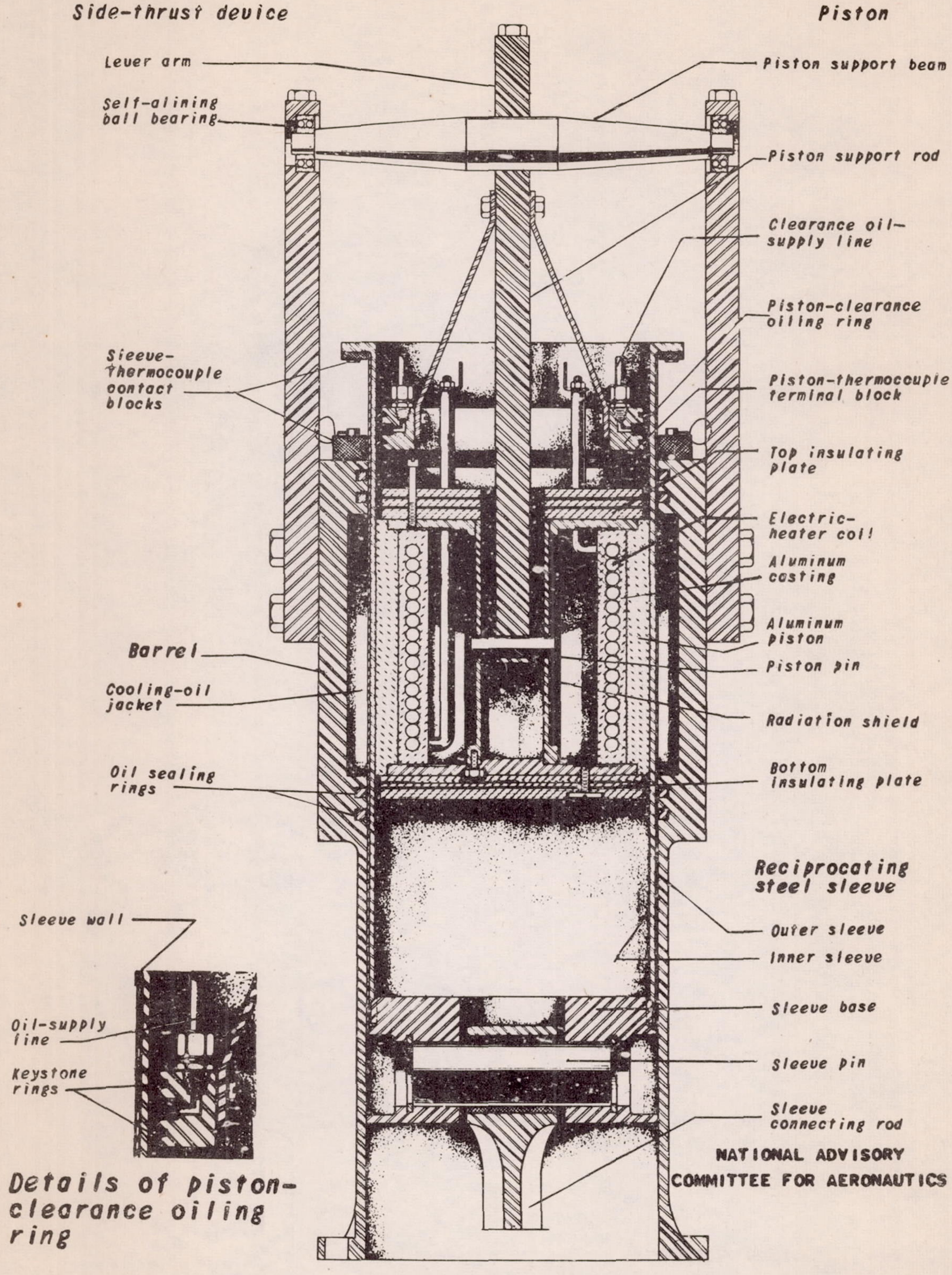
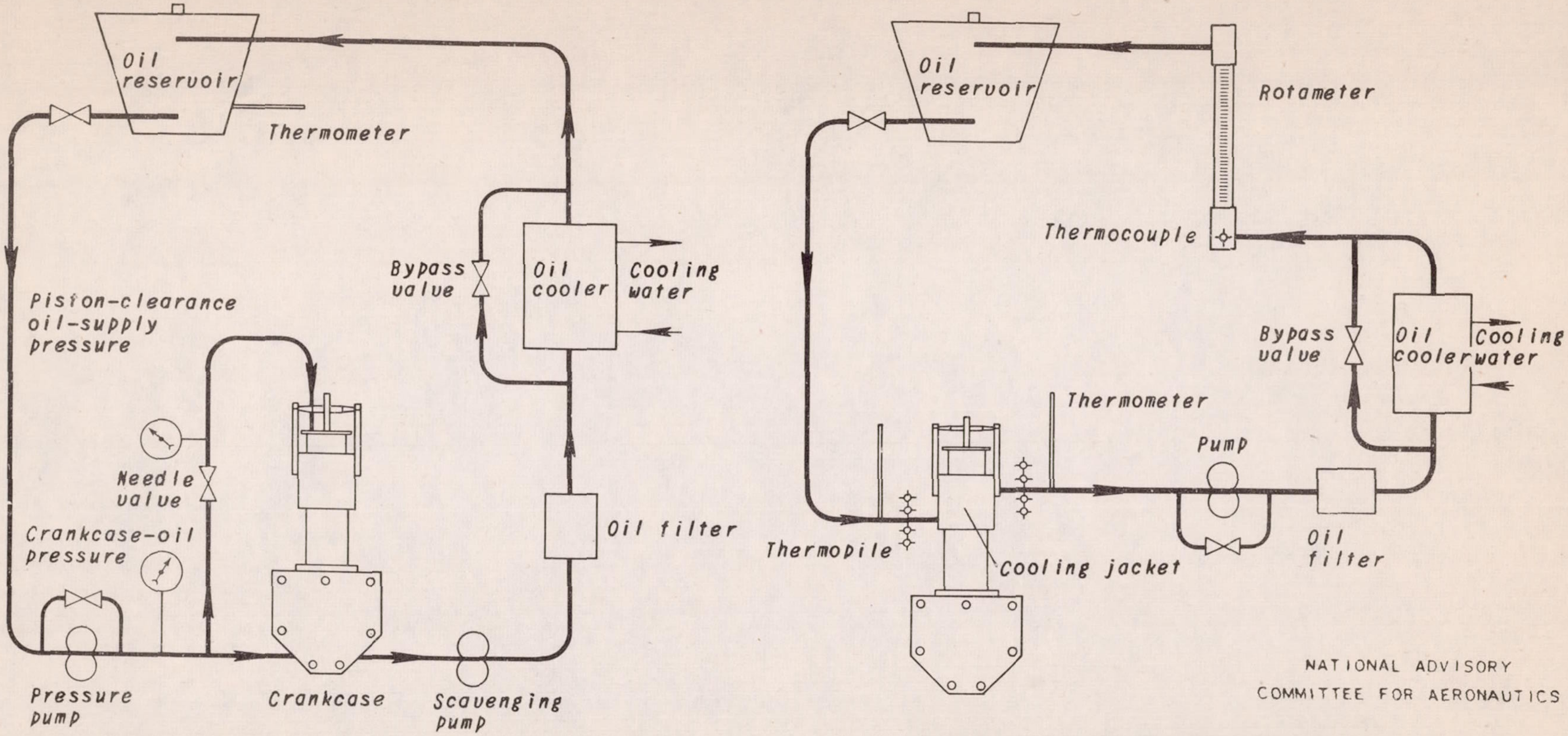


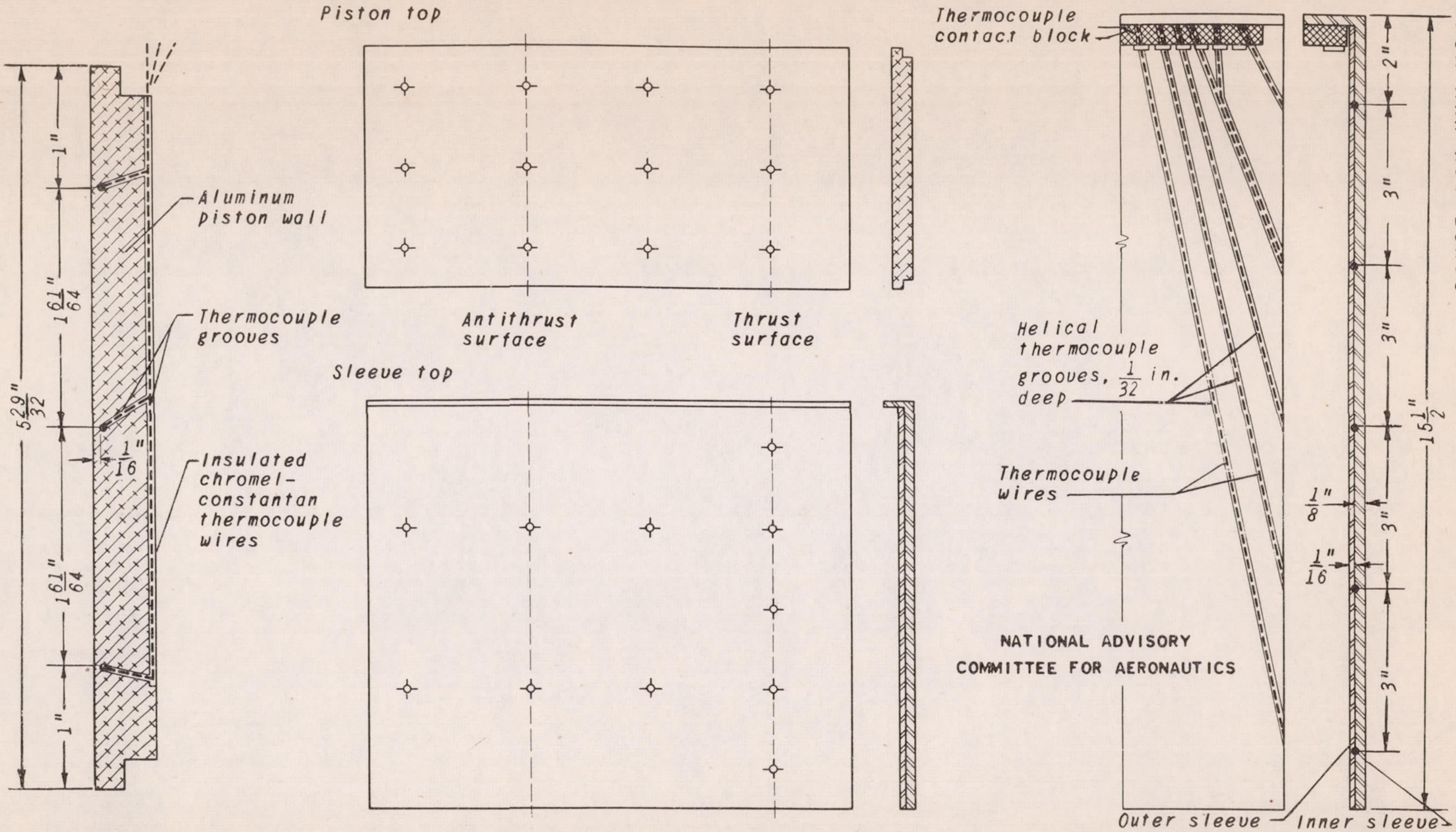
Figure 2. - Construction details of the piston reciprocating sleeve heat-transfer apparatus.



(a) Lubricating system for crankcase and reciprocating sleeve.

(b) Cooling-oil system for reciprocating sleeve.

Figure 3. - Schematic diagram of lubricating and cooling-oil systems for the piston reciprocating-sleeve heat-transfer apparatus.



(a) Piston-wall thermocouple installation.

(b) Piston and sleeve developments showing thermocouple locations.

(c) Sleeve-wall thermocouple installation on thrust surface.

Figure 4. - Locations and installation details of piston and reciprocating-sleeve thermocouples.

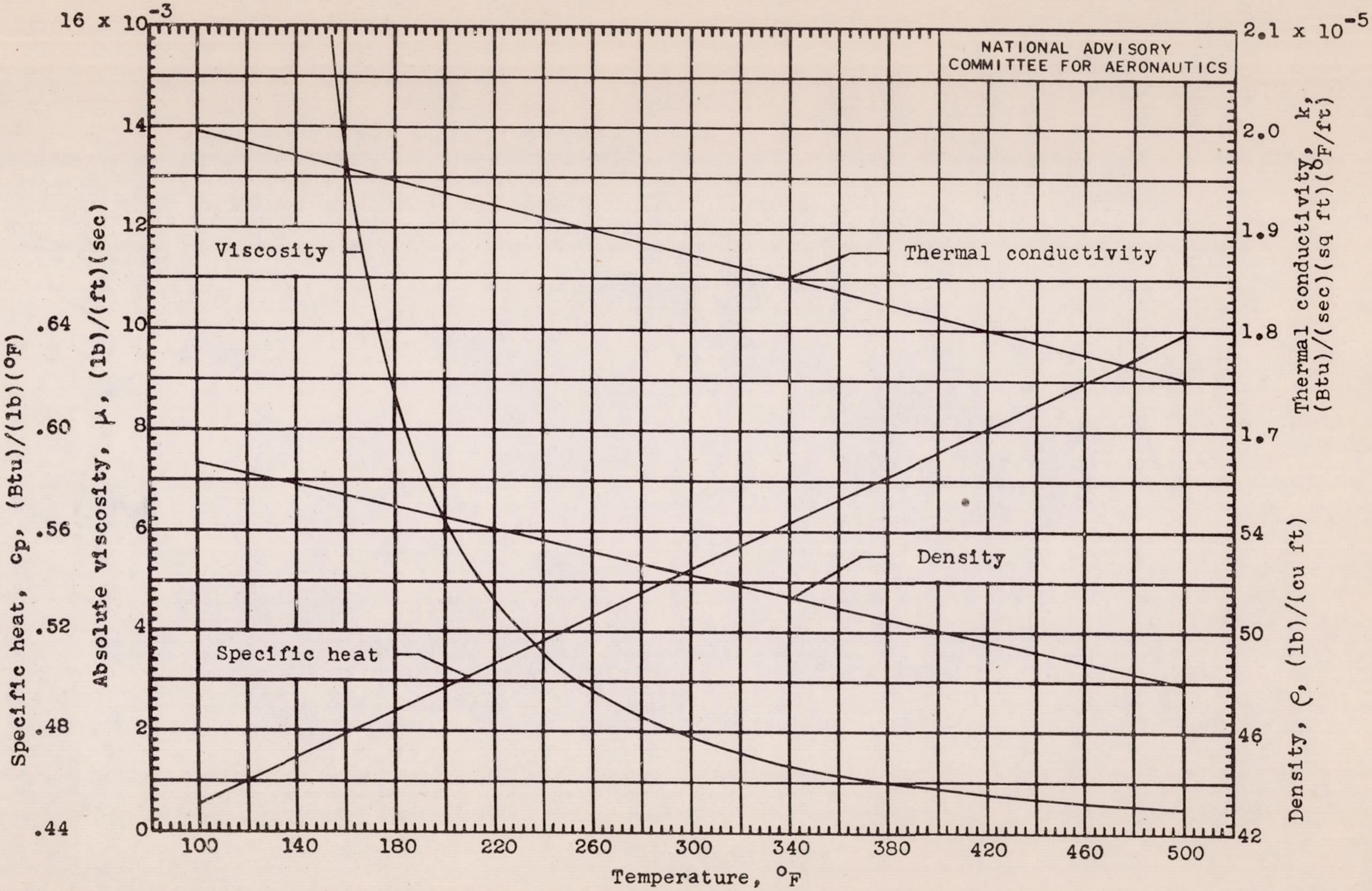


Figure 5.- Physical properties of SAE 30 oil as functions of temperature.

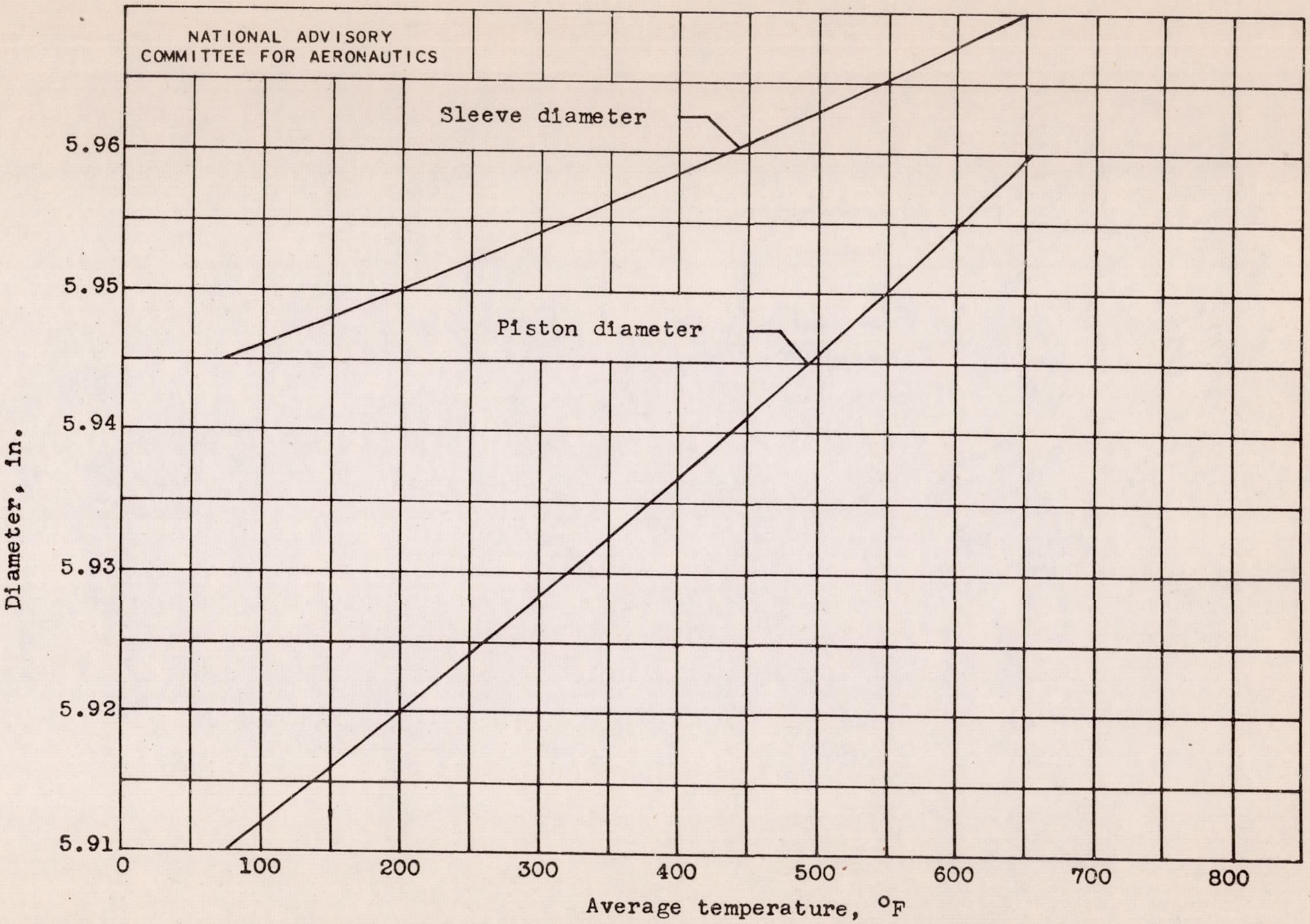


Figure 6.- Variation of piston and sleeve diameters with average temperature.

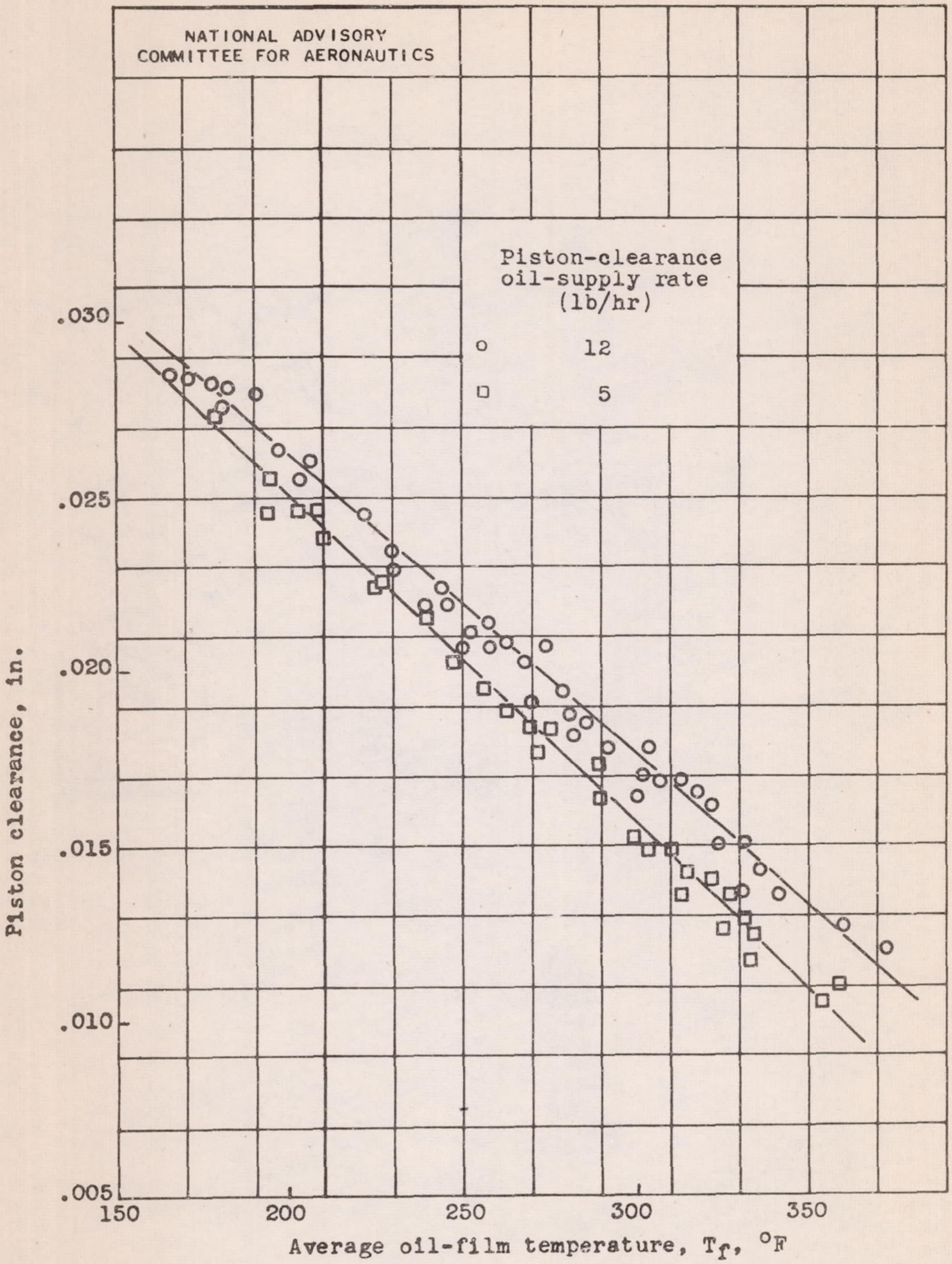


Figure 7.- Variation of calculated piston clearance with average oil-film temperature.

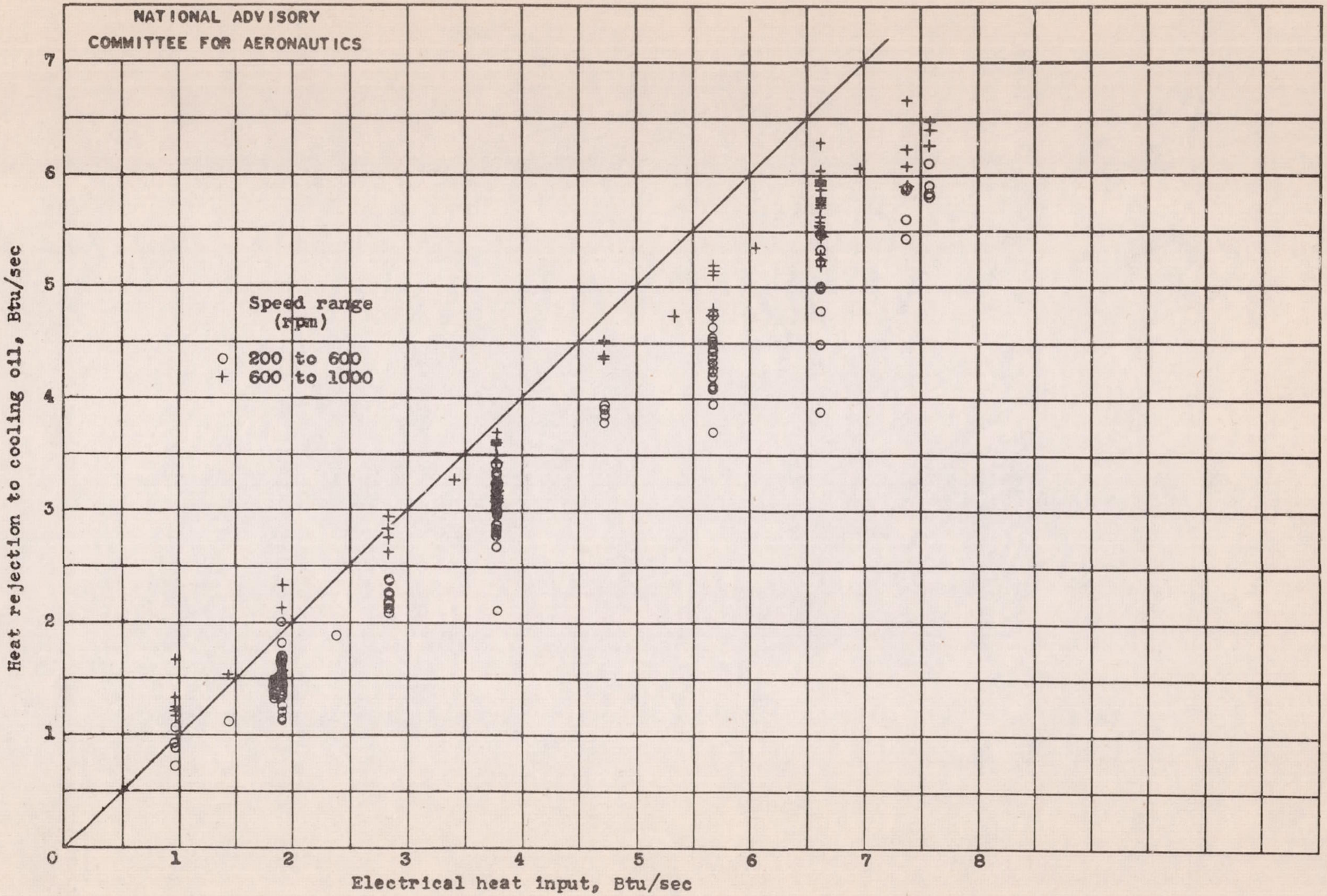
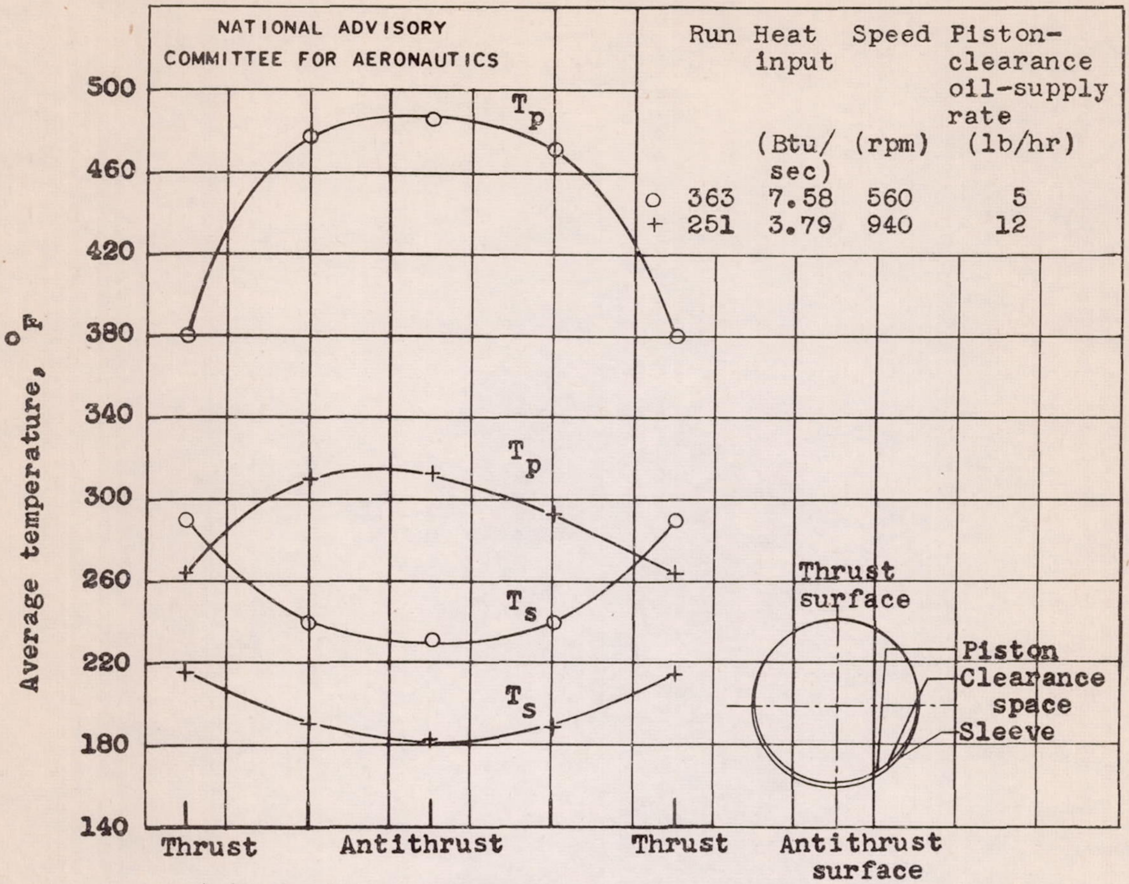
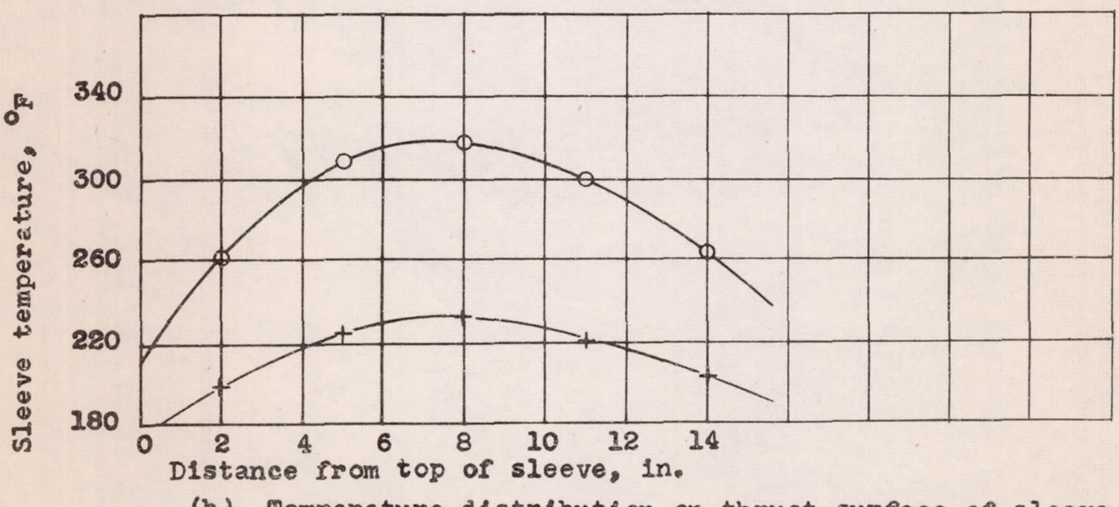


Figure 8. - Comparison of electrical heat input to piston with heat rejection to cooling oil.

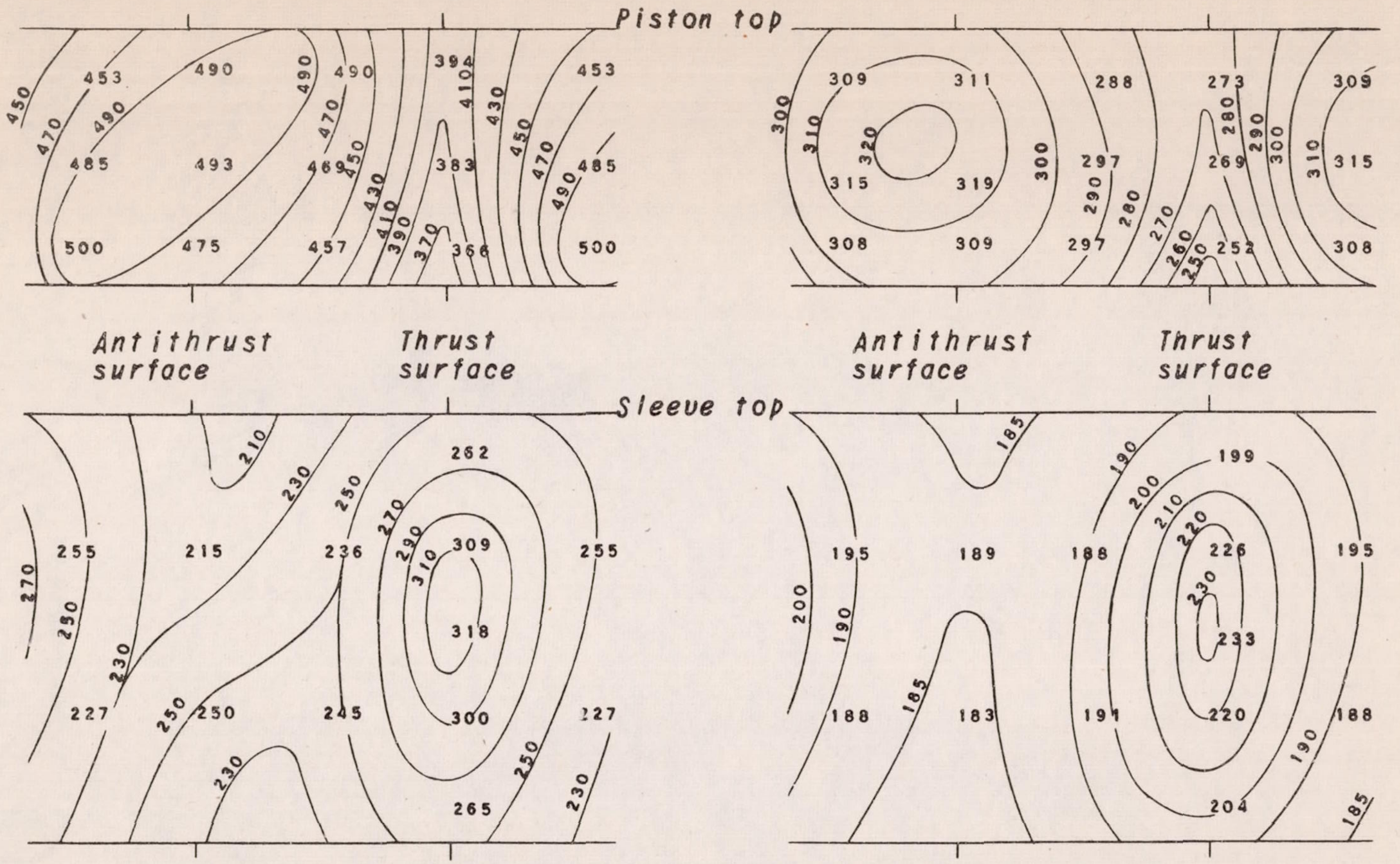


(a) Peripheral distribution of average piston and sleeve temperatures.



(b) Temperature distribution on thrust surface of sleeve.

Figure 9.- Piston and sleeve temperature distribution for representative runs on piston reciprocating-sleeve apparatus. Side thrust, 100 pounds; cooling-oil temperature, 140° F; cooling-oil flow, 60 pounds per minute.



(a) Heat input, 7.58 Btu per second; speed, 560 rpm; side thrust, 100 pounds; clearance-oil supply, 5 pounds per hour; cooling-oil temperature, 142° F; cooling-oil flow, 61 pounds per minute; run 363.

(b) Heat input, 3.79 Btu per second; speed, 940 rpm; side thrust, 100 pounds; clearance-oil supply, 12 pounds per hour; cooling-oil temperature, 141° F; cooling-oil flow, 58 pounds per minute; run 251.

NATIONAL ADVISORY
COMMITTEE FOR AERONAUTICS

Figure 10. - Isothermal patterns on piston and sleeve surfaces of the piston reciprocating-sleeve heat-transfer apparatus. Temperatures are in °F.

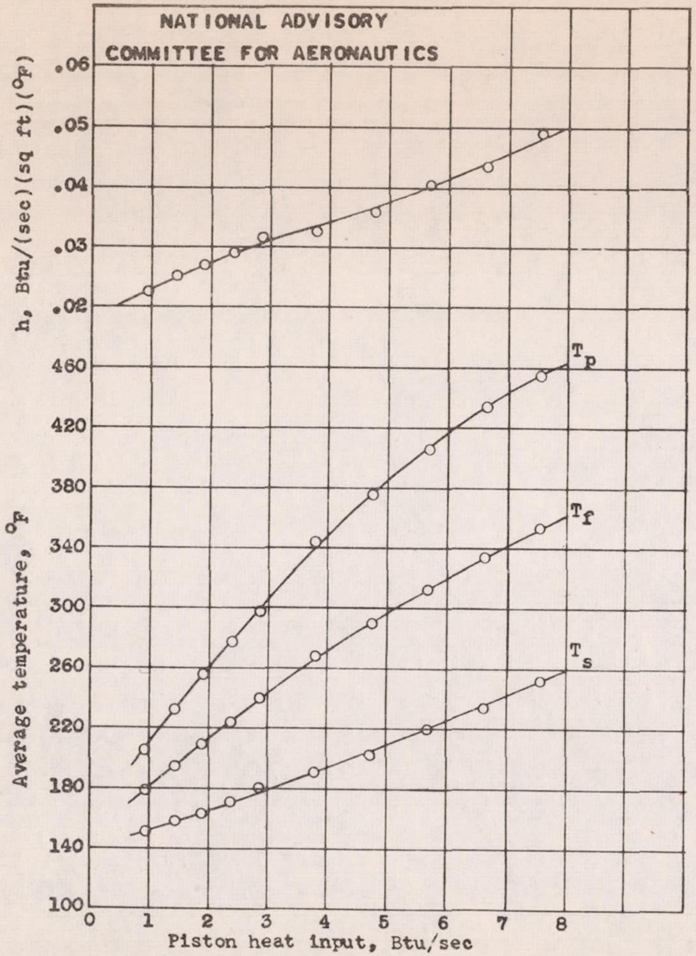


Figure 11.- Effect of heat input on average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient for runs 363 to 372. Speed, 565 rpm; piston-clearance oil-supply rate, 5 pounds per hour; side thrust, 100 pounds; cooling-oil temperature, 140° F; cooling-oil flow, 60 pounds per minute.

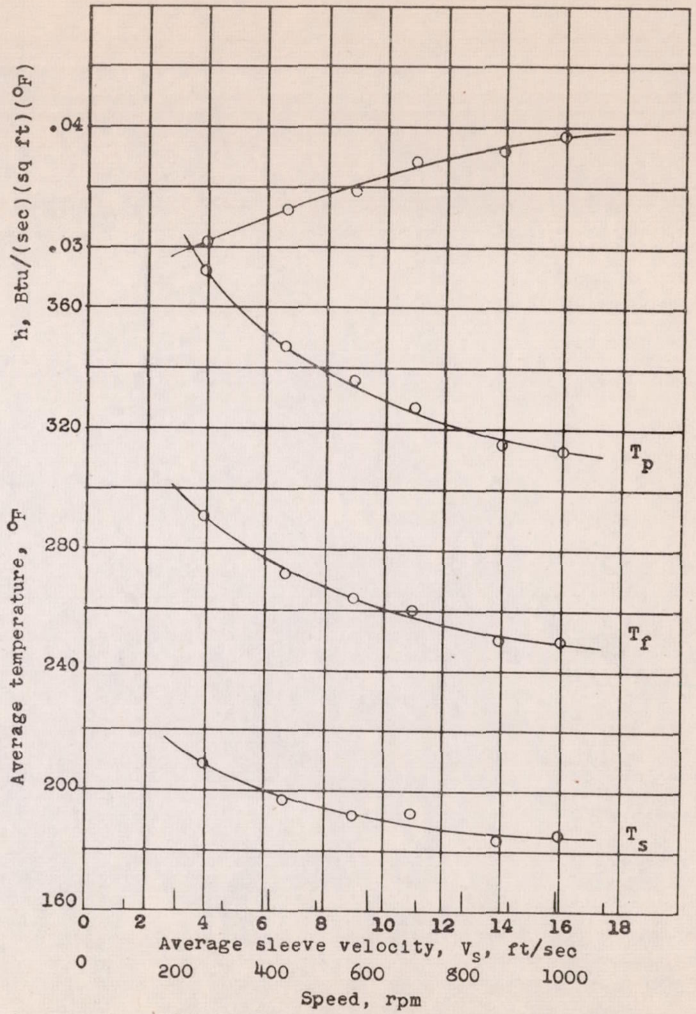


Figure 12.- Effect of average sleeve velocity on average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient for runs 373 to 378. Heat input, 3.79 Btu per second; piston-clearance oil-supply rate, 5 pounds per hour; side thrust, 100 pounds; cooling-oil temperature, 140° F; cooling-oil flow, 60 pounds per minute.

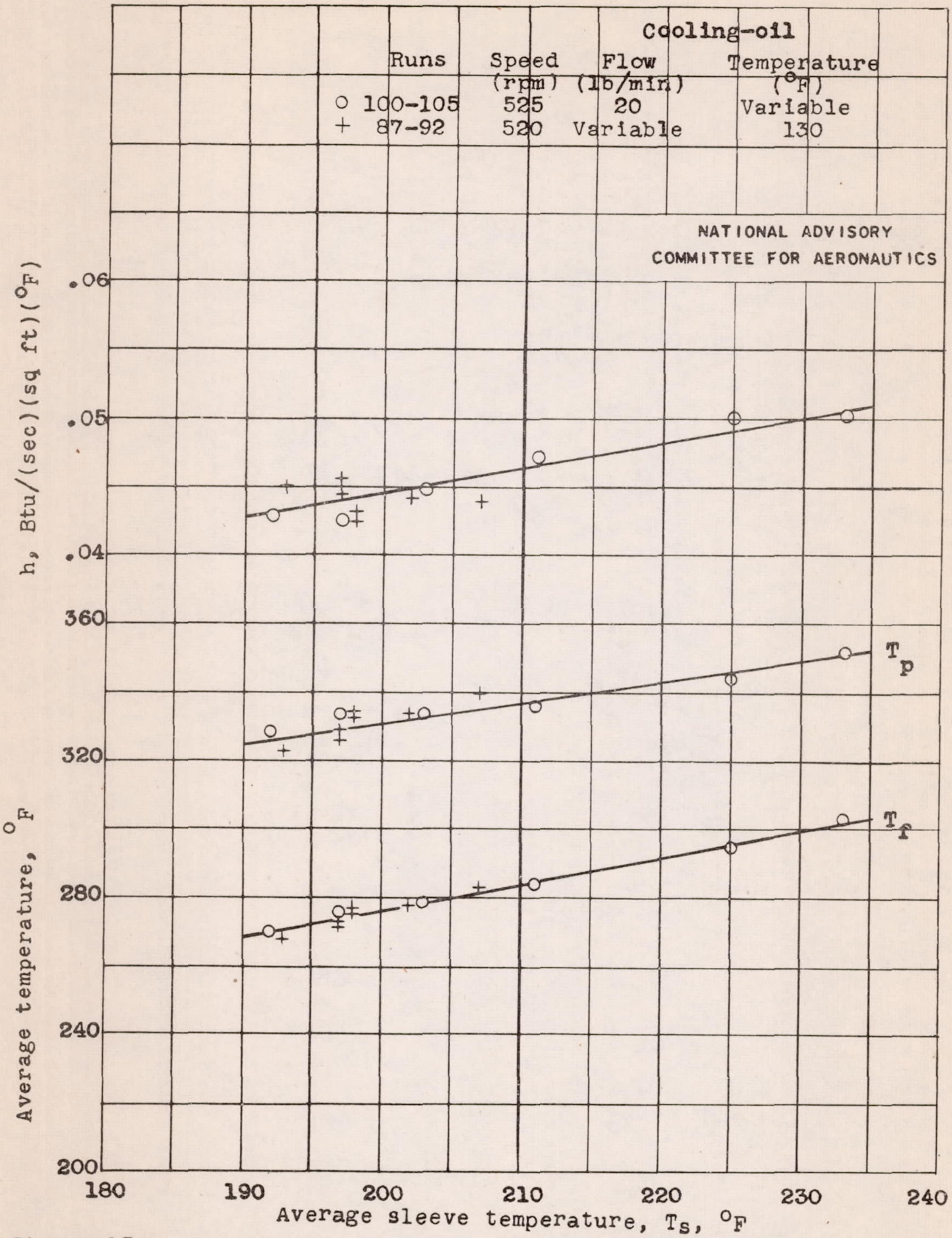


Figure 13.- Variation of average piston and oil-film temperatures and piston heat-transfer coefficient with average sleeve temperature. Heat input, 3.79 Btu per second; side thrust, 100 pounds; piston-clearance oil-supply rate, 12 pounds per hour.

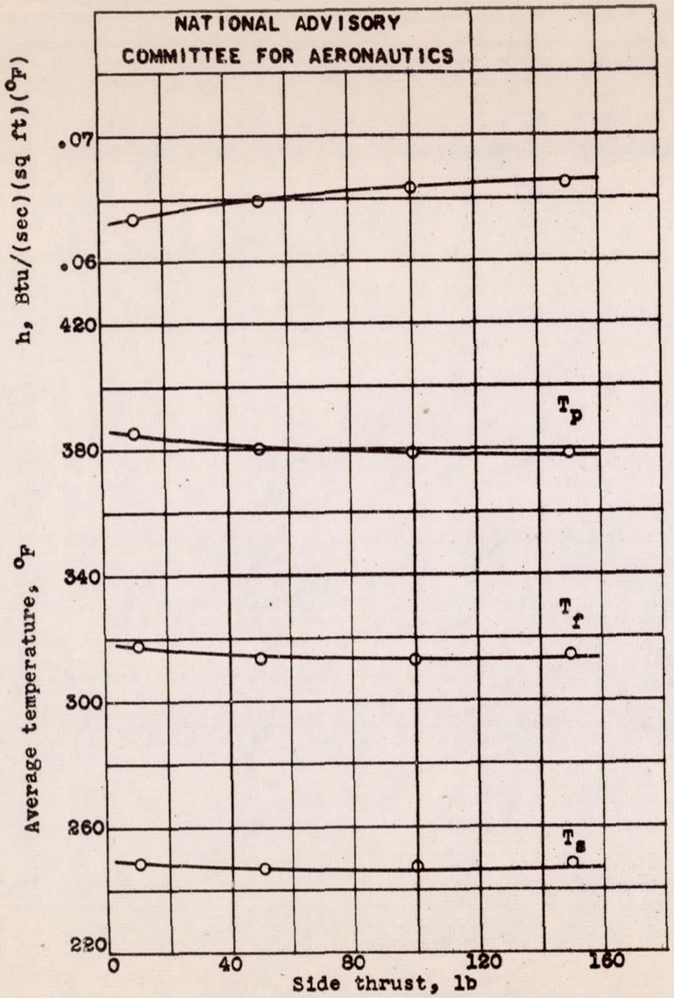


Figure 14.- Effect of side thrust on average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient for runs 121 to 124. Heat input, 6.64 Btu per second; speed, 960 rpm; piston-clearance oil-supply rate, 12 pounds per hour; cooling-oil temperature, 150° F; cooling-oil flow, 40 pounds per minute.

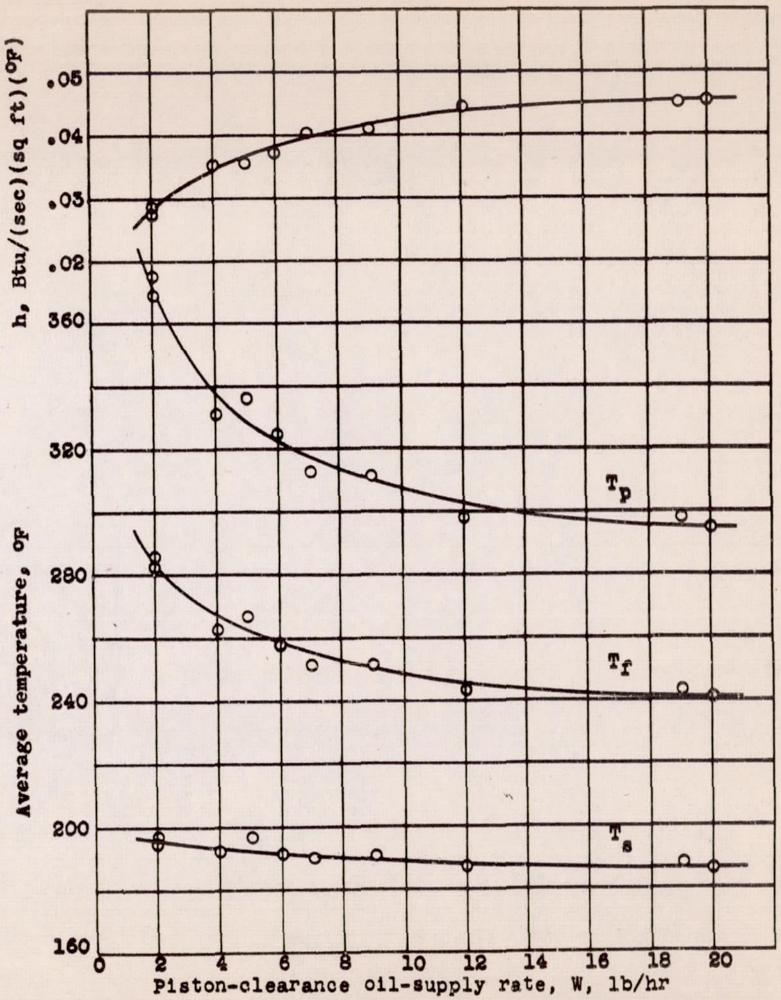


Figure 15.- Effect of piston-clearance oil-supply rate on average piston, oil-film, and sleeve temperatures and piston heat-transfer coefficient for runs 313 to 317 and 394 to 398. Heat input, 3.79 Btu per second; speed, 560 rpm; side thrust, 100 pounds; cooling-oil temperature, 140° F; cooling-oil flow, 80 pounds per minute.

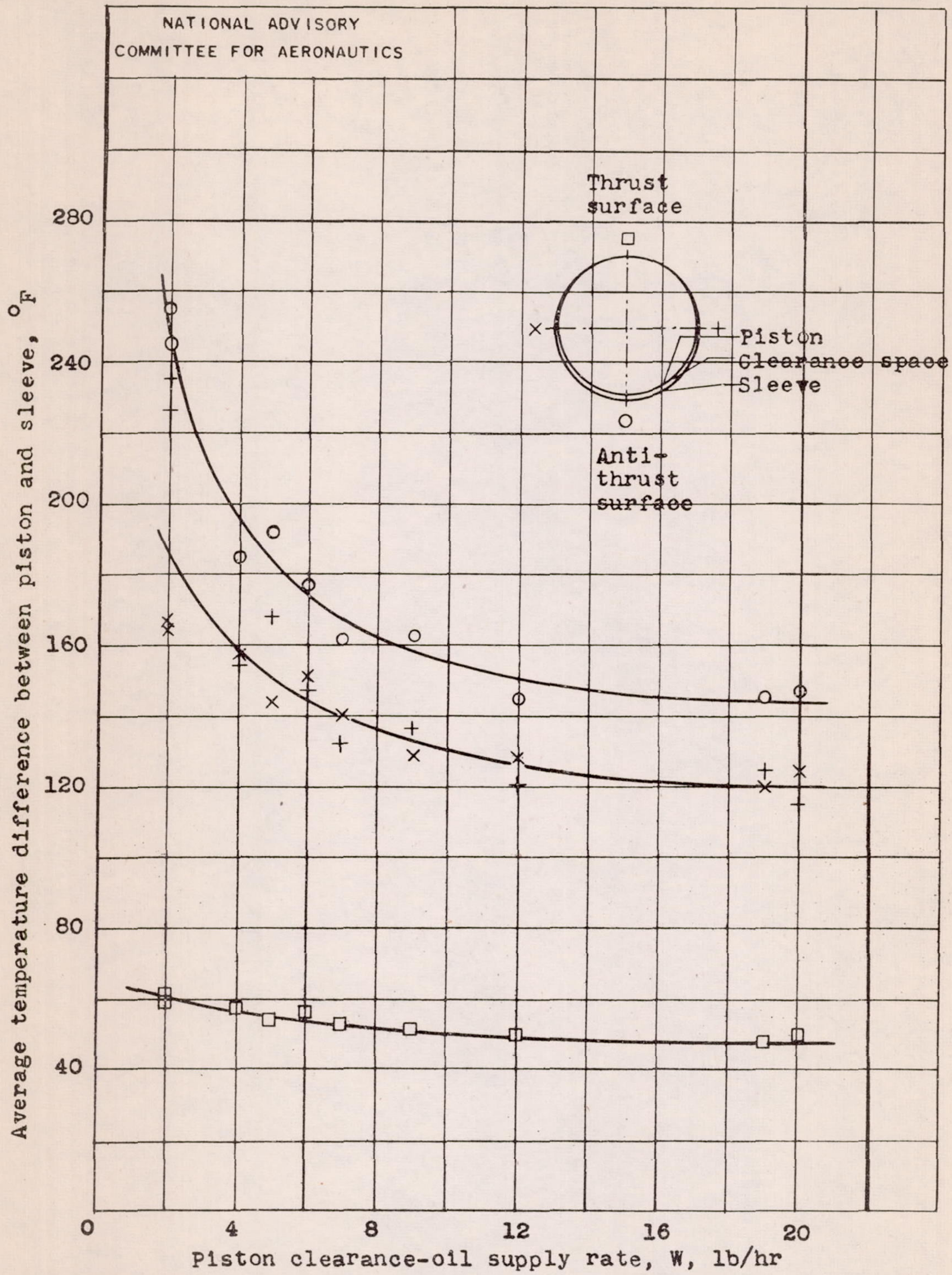
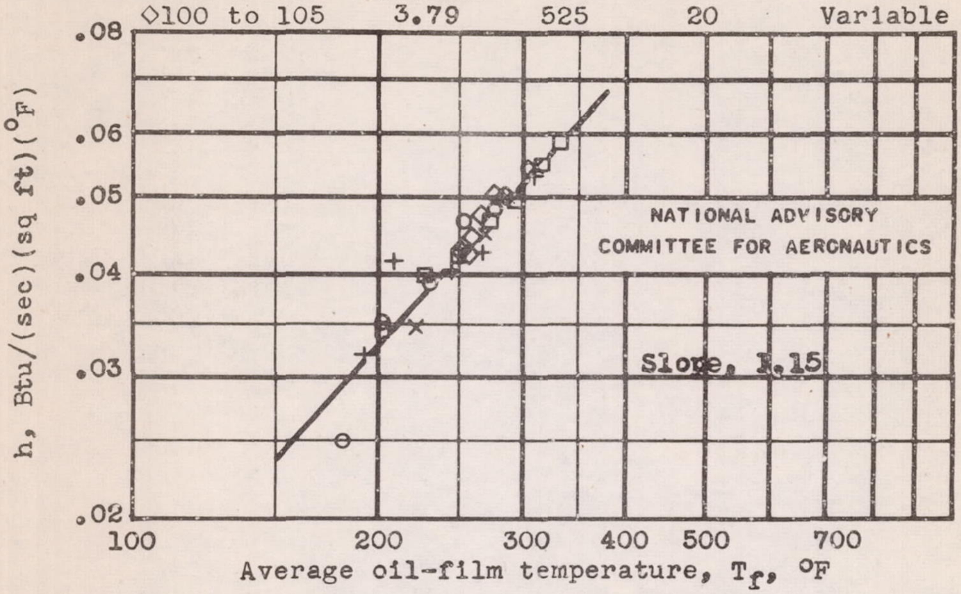
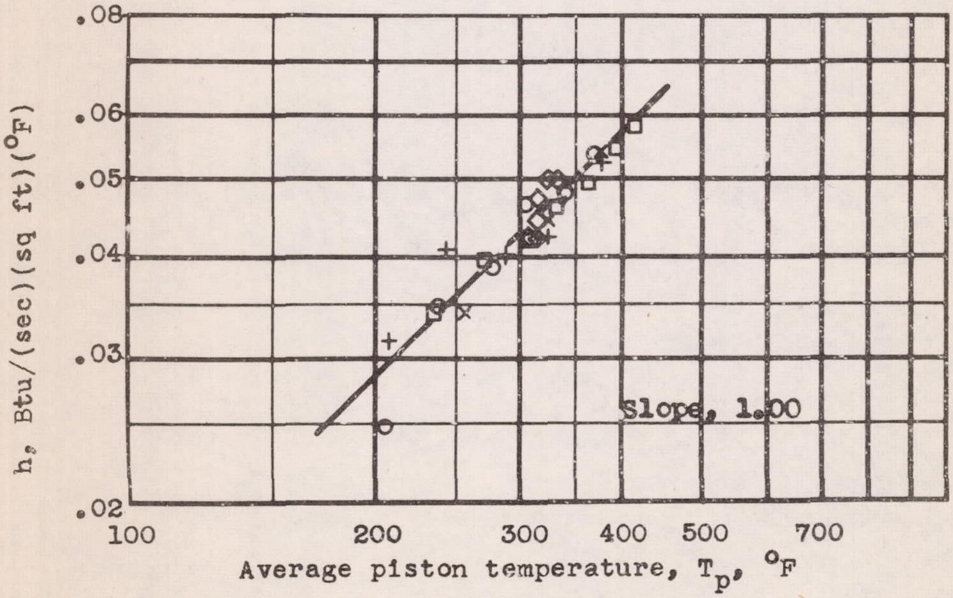


Figure 16.- Variation of peripheral average-temperature difference between piston and sleeve with rate of piston-clearance oil-supply rate for runs 313-317 and 394-398. Heat input, 3.79 Btu per second; speed, 560 rpm; side thrust, 100 pounds; cooling-oil temperature, 140° F; cooling-oil flow, 60 pounds per minute.

Runs	Heat input (Btu/sec)	Speed (rpm)	Cooling-oil Flow (lb/min)	Cooling-oil Temperature (°F)
○ 80 to 85	Variable	490	20	130
+ 166 to 170	Variable	540	40	150
× 171 to 173	Variable	540	20	152
□ 405 to 411	Variable	560	60	140
◇ 100 to 105	3.79	525	20	Variable



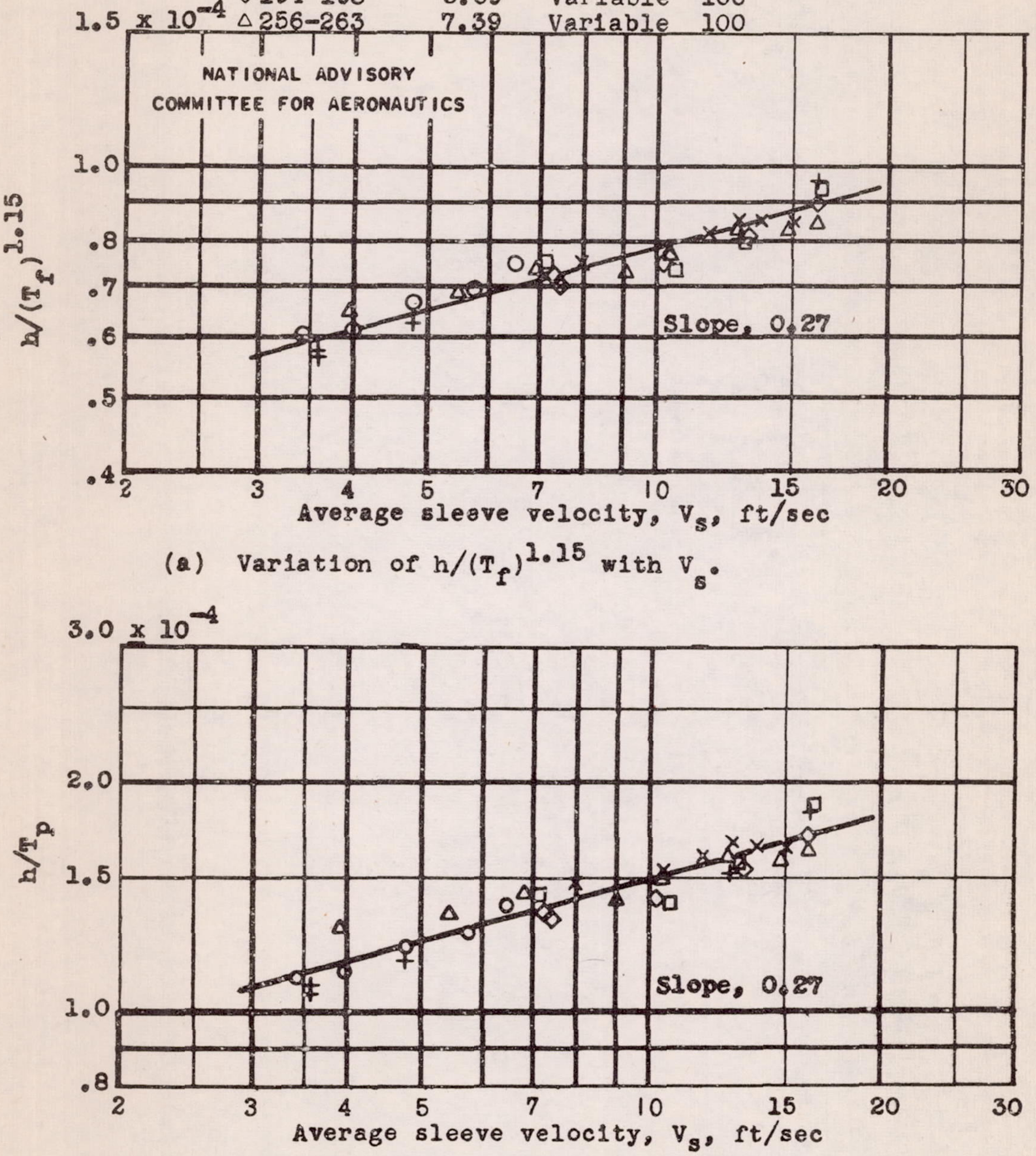
(a) Variation of h with T_f.



(b) Variation of h with T_p.

Figure 17. - Variation of piston heat-transfer coefficient with average oil-film and piston temperatures. Side thrust, 100 pounds; piston-clearance oil-supply rate, 12 pounds per hour.

Runs	Heat input (Btu/sec)	Speed (rpm)	Side thrust (lb)
○ 50-54	1.90	Variable	50
+ 106-110	3.79	Variable	100
× 132-137	6.64	Variable	100
□ 190-193	0.95	Variable	100
◇ 194-198	5.69	Variable	100
△ 256-263	7.39	Variable	100



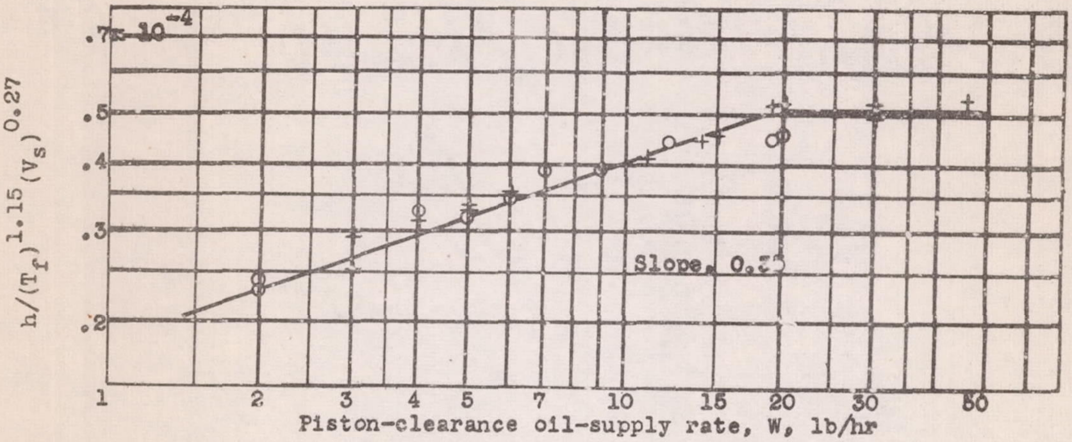
(a) Variation of $h/(T_f)^{1.15}$ with V_s.

(b) Variation of h/T_p with V_s.

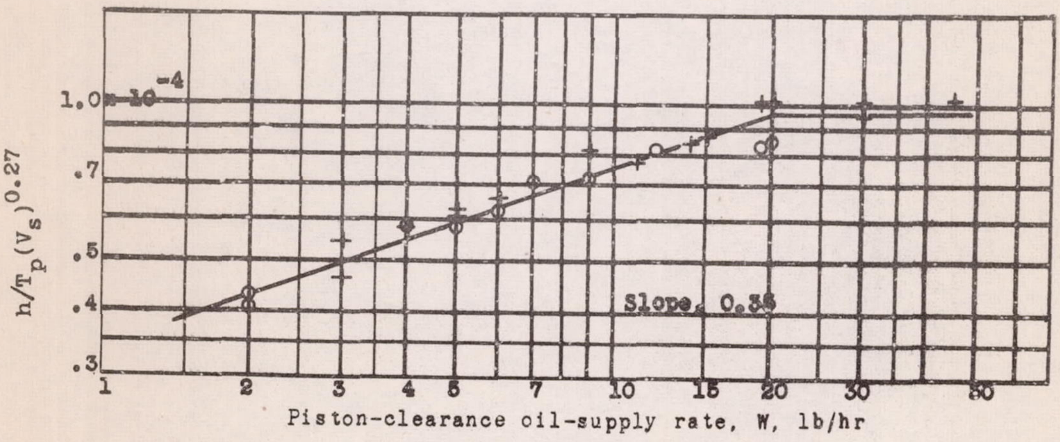
Figure 18.- Variation of $h/(T_f)^{1.15}$ and h/T_p with average sleeve velocity. Piston-clearance oil-supply rate, 12 pounds per hour.

NATIONAL ADVISORY
COMMITTEE FOR AERONAUTICS

	Runs	Heat input (Btu/sec)
o	313-317, 394-398	3.79
+	324-332, 399-404	1.90

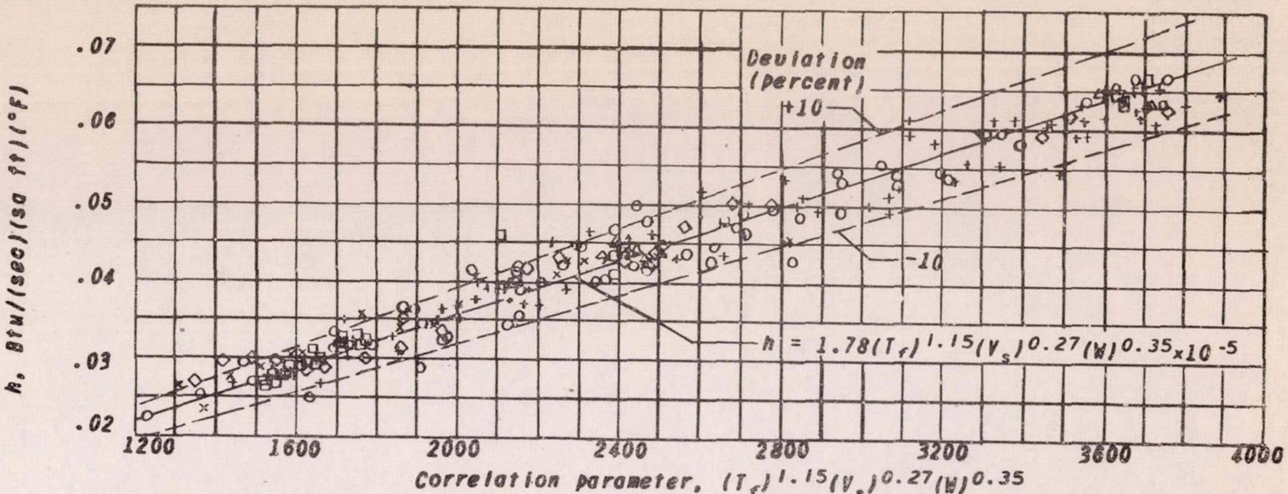


(a) Variation of $h/(T_f)^{1.15}\ (V_s)^{0.27}$ with W.

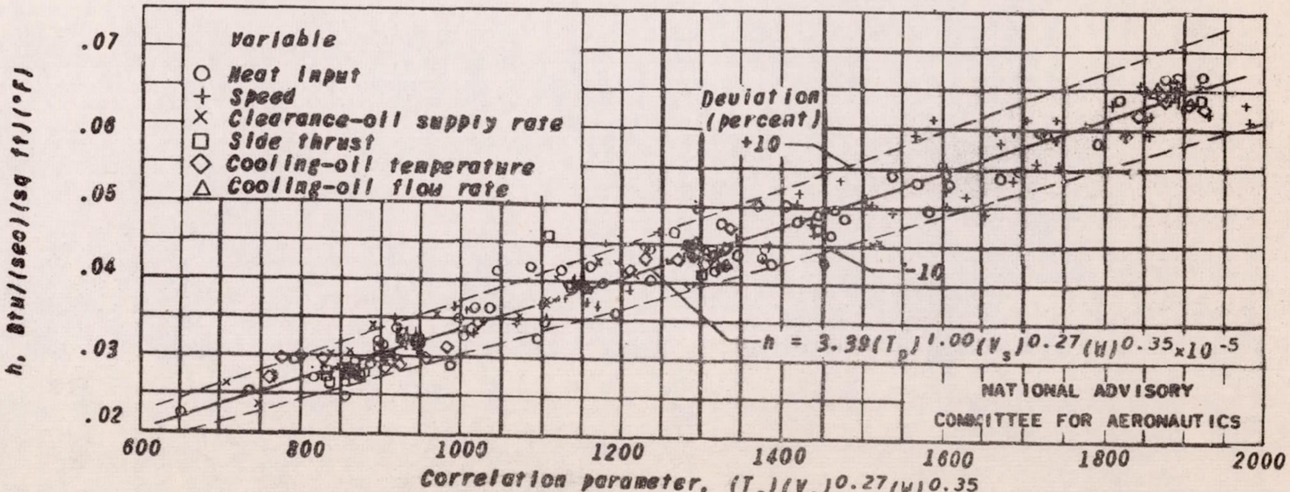


(b) Variation of $h/T_p\ (V_s)^{0.27}$ with W.

Figure 19. - Variation of $h/(T_f)^{1.15}\ (V_s)^{0.27}$ and $h/T_p\ (V_s)^{0.27}$ with piston-clearance oil-supply rate. Speed, 530 rpm; side thrust, 100 pounds.

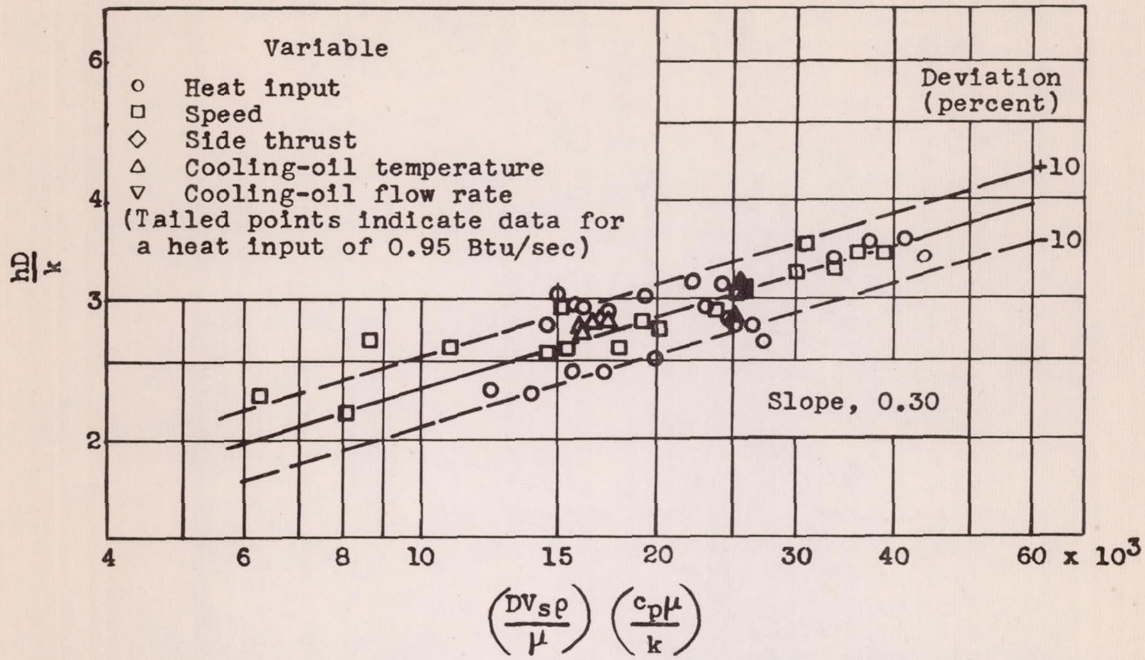


(a) Average oil-film temperature, T_f, used as correlation basis.

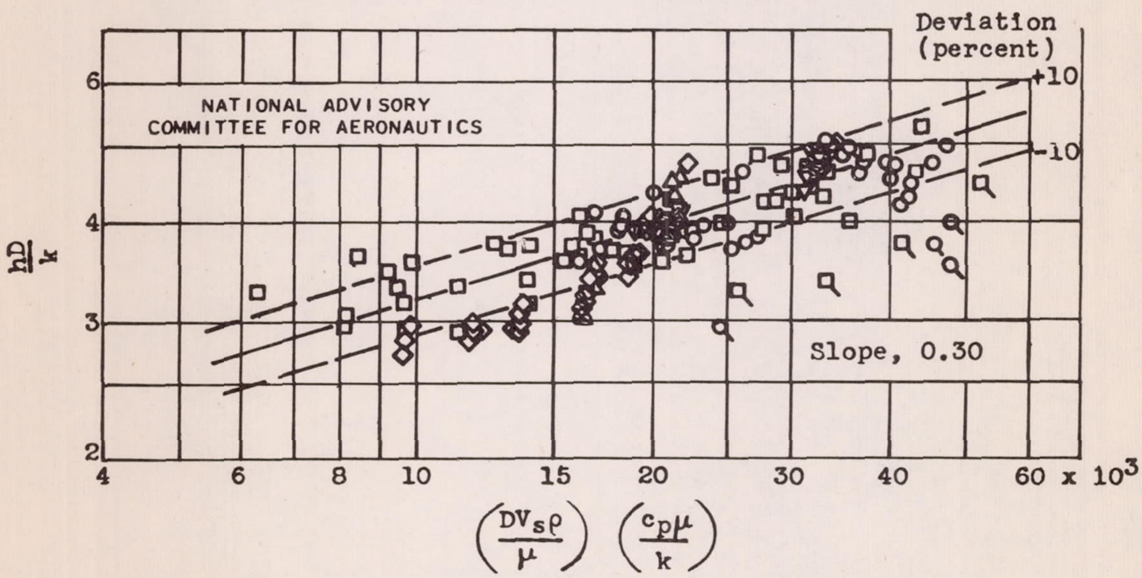


(b) Average piston temperature, T_p, used as correlation basis.

Figure 20.- Correlation curves for test results of piston reciprocating-sleeve heat-transfer apparatus. Piston clearance oil-supply rate limited to 20 pounds per hour.



(a) Piston-clearance oil-supply rate, 5 pounds per hour.



(b) Piston-clearance oil-supply rate, 12 pounds per hour.

Figure 21.- General correlation curves for test results of piston reciprocating-sleeve heat-transfer apparatus.