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INVESTIGATION OF 75-MILLIMETER-BORE CYLINDRICAL ROLLER BEARINGS AT HIGH SPEEDS

II - LUBRICATION STUDIES - EFFECT OF OIL-INLET LOCATION, ANGLE, AND VELOCITY FOR SINGLE-JET LUBRICATION

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SUMMARY

An experimental investigation of the effect of several oil introduction parameters on the temperature of 75-millimeter-bore (size 215) cylindrical-roller, inner-race-riding cage-type bearings over a range of DN values (product of the bearing bore in mm and the shaft speed in rpm) from 0.3×10^6 to 1.43×10^6 and static radial loads to 368 pounds with single-jet circulating oil feed is reported herein.

For the oil-jet positions investigated, the inner- and outer-race bearing temperatures were found to be at a minimum when the oil was directed at the cage-locating surface perpendicular to the bearing face. In consequence of the foregoing findings, all subsequent investigations were conducted with the lubricant directed at the cage-locating surface and perpendicular to the bearing face; and therefore the following results and conclusions apply only under such a lubricating condition:

For all values of oil flow and all oil-jet diameters investigated, the outer-race maximum temperature increased linearly with an increase in DN value; however, the inner-race temperature increased with DN value at a rate slightly greater than linear.

An increase in oil inlet velocity at a given flow decreased both the outer-race and inner-race temperatures. It appeared that a point of diminishing returns regarding the effectiveness of an increase of oil inlet velocity occurred at about 60 feet per second for the operating conditions investigated.

Heat-transfer considerations indicated that to minimize the number of parameters when dealing with inner-race temperatures the variables to be plotted should be $\frac{(T_{IR}-T_{OI})}{(DN)^y}$ and $\left(\frac{d}{M}\right)^n$ where 1414
$(T_{IR}-T_{OI})$ is the difference between the inner-race bearing temperature and the oil inlet temperature, DN is the product of the bearing bore in millimeters and the shaft speed in rpm, d is the oil-jet diameter, M is the oil flow, and n and y are constants. It was found empirically, however, that the variable $(d^{0.8}/M)^n$ gave better correlation than did $(d/M)^n$.

When $\frac{(T_{IR}-T_{OI})}{(DN)^{1.2}}$ was plotted against $\left(\frac{d^{0.8}}{M}\right)^{0.4}$ for a given operating condition, a representative single straight line resulted regardless of whether bearing speed, oil flow, or jet diameter was varied. The results of this investigation are graphically presented herein.

INTRODUCTION

The oil supplied to high-speed rolling-contact bearings performs two principal functions: lubricating and cooling. The lubricating function of the oil in a high-speed roller bearing is critical in that lubrication failures occur at the cage-locating surfaces (reference 1 and Gurney's discussion in reference 2). The cooling action of the oil has two primary effects:

(1) To maintain bearing temperatures low enough to prevent softening by tempering of the bearing material thereby maintaining maximum hardness and maximum bearing life

(2) To keep the bearing surfaces cool enough to prevent the oil itself from breaking down excessively.

The heat generated within a rolling-contact bearing increases with an increase in operating speed. The DN value (bearing bore in mm multiplied by the shaft speed in rpm) is considered to be an indication of bearing operating severity with regard to the generation of heat within the bearing. It has been found in practice that bearings begin to overheat at DN values between 0.5×10^6 and 1×10^6 depending upon the application (reference 3). In these cases, cooling must be provided either by additional or more effective lubricant flow through the bearing or by external cooling, for example, by means of a cooling medium flowing through a passageway in the shaft or bearing housing. Heat flow to the bearing either through the shaft, the housing, or due to high ambient temperature requires additional coolant.

The cage-locating surface is the critical area in a high-speed roller-bearing (references 1 and 2). Although the cage may be located on the inner race, the rollers, or the outer race, this investigation was limited to the inner-race-riding cage. For race-riding cages, the locating surface is a sleeve bearing of extremely small length-diameter ratio and therefore of very low load capacity. Operation at the locating surface of the cage is likely to occur in the boundary region of lubrication. The cage material and the surface finish, as well as the lubricant, are therefore significant factors affecting the friction characteristics and hence the heat generated within the bearing. Many failures of high-speed roller bearings occur at the cage surfaces and are designated lubrication failures.

High-speed rolling-contact bearings in aircraft gas-turbine engines are currently lubricated with an oil-air mist or with oil flowing from a single jet directed between the bearing races. The investigation reported herein is a continuation of the work reported in reference 1, and was conducted at the NACA Lewis laboratory in order to compare and to evaluate the effects of oil-inlet location, angle, and velocity on outer- and inner-race bearing temperatures for single-jet lubrication over a wide range of variables.

Cylindrical-roller bearings currently employed as the turbine roller bearing in aircraft gas-turbine engines were used as test bearings in this investigation. These bearings were of 75-millimeter-bore (size 215), 25-millimeter width, and 130-millimeter outside diameter and were equipped with a one-piece, inner-race-riding brass cage. The ranges of controlled variables in this investigation were: loads, 113 and 368 pounds; DN values, 0.3×10^6 to 1.43×10^6; oil flows, 0.6 to 12.9 pounds per minute; and oil inlet velocities, 13 to 200 feet per second.

APPARATUS

Bearing rig. - The bearing rig (fig. 1) used for this investigation is the same as that used in reference 1. The bearing under investigation was mounted on one end of the test shaft, which was supported in a cantilever fashion, for purposes of observing bearing component parts and lubricant flow during operation. A radial load was applied to the experimental bearing by means of a lever and dead-weight system in such a manner that the outer race of the experimental bearing was essentially unaffected by small shaft deflection or by small shaft and load arm misalinements.

The support bearings were lubricated in the same manner as described in reference 1. For all runs reported herein, the oil was supplied to the support bearings at a temperature of 100° F and a pressure of 10 pounds per square inch through 0.180-inch-diameter jets.

The drive equipment is described in reference 1. The speed range of the test shaft is 800 to 50,000 rpm controllable to within ±30 rpm.

Test bearings. - Three test bearings (table I) were used for this investigation; these bearings were standard cylindrical turbine roller bearings of a conventional aircraft gas-turbine engine. The bearing dimensions were: bore, 75 millimeters; outside diameter, 130 millimeters; and width, 25 millimeters. The bearings were equipped with one-piece inner-race-riding brass cages. The operating conditions imposed on this type of bearing in engine service are as follows: DN range, 0.3×10^6 to 0.86×10^6; approximate gravity load, 375 pounds; oil flow, 0.8 to 2 pounds per minute.

The bearings investigated are numbered consecutively from the first part of the investigation (reference 1); bearing 6 of reference 1 is the same bearing as bearing 6, which is discussed herein.

Temperature measurement. - The method of temperature measurement is described in reference 1. Iron-constantan thermocouples were located at 60° intervals around the outer-race periphery at the axial center line of the bearing under investigation. A copper-constantan thermocouple was pressed against the bore of the inner race at the axial midpoint of the test bearing; the voltage was transferred from the rotating shaft by means of slip rings.

Lubrication system. - The lubricating system is described in reference 1 with the exception that changes were made to increase the upper limit of the oil-supply pressure to the test bearing from 90 to 400 pounds per square inch.

PROCEDURE

Lubricating of test bearing. - Lubricant at 100° F was supplied to the bearing under investigation through single jets of various sizes over a range of oil flows from 0.6 to 12.9 pounds per minute.

In a preliminary investigation, the radial location and the angle of the jet were varied with respect to the bearing in order to determine the optimum radial and angular positions with regard to bearing operating temperatures.

Two sets of jet size were then investigated with the oil directed according to the optimum positions determined in the preliminary investigation. The A series of jets (fig. 2(a)) had four diameter sizes: 0.180, 0.157, 0.129, and 0.089 inch and are hereinafter designated type-A jets. The B series of jets had six diameter sizes: 0.129, 0.089, 0.066, 0.048, 0.035, and 0.023 inch and are designated type-B jets (fig. 2(b)). The type-B jets had a length-dameter ratio of 1.

A discussion of the effect of oil viscosity on flow through the jets as well as a discussion concerning the cause for a discontinuity of the pressure-flow curves for certain jet sizes is presented in appendix A.

The properties of the oil used are given in figure 3. This oil was the same as that used in reference 1 and is a commercially prepared blend of a highly refined paraffin base with a small percentage of a polymer added to improve viscosity index.

Test-bearing measurements. - The test-bearing measurements were obtained in the same manner as described in reference 1 and are given in table I.

RESULTS AND DISCUSSION

Experimental Results

The results of the experimental investigation are presented in figures 4 to 13. Bearing temperature was chosen as the principal criterion of operation inasmuch as in the final analysis temperature is an over-all indication of the effects of all operating conditions.

Effect of radial location of oil jet. - The effect of oil-inlet radial location on the operating temperature of bearing 8 is shown in figure 4 for a DN of 1.2×10^6. Oil was directed perpendicular to the bearing face at the three radial locations shown in the inset, that is, at the space between the cage and the outer race, at the space between the cage and the inner race, and at the cage itself. The data show that of the three positions investigated, when the oil was directed at the cage-locating surface, that is, between the cage and the inner race for this bearing, (oil-jet location A, fig. 4) lower outer-race maximum and inner-race temperatures result. A more favorable outer-race circumferential temperature distribution also results over the entire flow range investigated.

The investigation was extended over a DN range from 0.3×10^6 to 1.43×10^6 to determine whether the marked difference observed at a DN of 1.43×10^6 also existed at the other speeds. The results showed that, when the oil is directed at the cage-locating surface, more uniform outer-race circumferential temperature distribution and lower temperatures exist over the entire speed range.

Effect of angle of impingement of oil jets. - In order to determine whether the angle of oil-jet impingement significantly influenced bearing operating temperature, an investigation was undertaken using bearing 9 with a single oil jet directed in the five positions shown in the inset of figure 5. In each position the jet was so adjusted that the stream of oil was directed exactly at the cage-locating surface. A plus angle indicates that the jet was directed opposite to the direction of shaft rotation and a minus angle indicates that the jet was directed in the direction of shaft rotation. The results are shown in figure 5 where maximum, mean, and minimum outer-race and the inner-race temperatures are plotted against oil-inlet angle over a range of DN from 0.3×10^6 to 1.43×10^6 at a load of 113 pounds.

The effect of oil-inlet angle is more pronounced at the higher DN values. The minimum temperatures occur when the oil is directed either perpendicularly to the bearing face or at a slight plus angle; therefore a jet perpendicular to the bearing appears to be the best solution. A greater rate of temperature increase occurs with negative angle than with positive angle; therefore any tolerance in mounting the jet should be in the positive direction. The subsequent lubrication studies reported herein were conducted with the oil directed at the cage-locating surface and perpendicular to the bearing face because of the foregoing results.

Effect of jet diameter. - Lubrication of the test bearing using a series of jets of various diameters was undertaken to determine if the jet size influenced the bearing operating temperatures at equal rates of oil flow. A preliminary study was made with type-A jets (fig. 2(a)). The results are shown in figure 6 for bearing 9 over a range of oil flows from 1.4 to 12.9 pounds per minute at a DN of 1.43×10^6 and a load of 113 pounds. The oil inlet pressure ranged from 5 to 90 pounds per square inch for the investigation of this group of jets.

In general the maximum, mean, and minimum outer-race temperatures as well as the inner-race temperatures decrease with a decrease in jet diameter over the entire flow range (fig. 6). It is therefore evident that the initial oil-stream velocity (or the oil inlet penetrating force) is a significant factor regarding the effectiveness of the stream of oil to cool high-speed roller bearings.

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In order to study further the effects of initial oil-stream velocity, a group of smaller jets (type B, fig. 2(b)) was investigated over a wide range of oil flows with oil inlet pressures up to 400 pounds per square inch. The original data are shown for bearing 6 in figure 7 at DN values of 0.3×10^6, 0.735×10^6, 0.995×10^6, and 1.2×10^6 for a load of 368 pounds. Covering the entire flow range for each jet size was impossible inasmuch as the pressure drop exceeded 400 pounds per square inch for the smaller jet sizes. The bearing temperatures decrease with a decrease in jet diameter over the entire flow range (fig. 7); this decrease in temperature with a decrease in jet diameter occurs at a faster rate at the higher DN values. The effect of jet size is almost negligible at a DN value of 0.3×10^6.

The effect of jet size on the outer-race maximum temperature of bearing 6 with oil flow as the parameter for DN values of 0.735×10^6, 0.995×10^6, and 1.2×10^6 and a load of 368 pounds is shown in figure 8(a). A similar plot for inner-race temperature is given in figure 8(b). The faired curves of figure 8 have been obtained by cross-plotting the original data of figure 7. For a given flow, the outer-race maximum temperature decreases approximately as a straight line with a decrease in jet diameter for each DN value (fig. 8(a)). The inner-race temperature (fig. 8(b)) decreases with a decrease in jet diameter at a rate greater than linear; this rate increases for the higher DN values. These relations are further analyzed in the section "Analysis of Experimental Results."

The effect of DN value on outer-race maximum and inner-race temperatures of bearing 6 with oil flow as the parameter for each jet size is shown in figure 9. These faired curves have also been obtained by cross-plotting the original data of figure 7. For all jet sizes and all values of oil flow investigated, the outer-race maximum temperature and inner-race temperature increase essentially as straight-line functions with increase in DN value.

Power required for oil delivery. - The horsepower (hp) required to deliver oil (when an average specific gravity for oil of 0.854 is assumed) may be determined by the relation

$$hp = \frac{8.18 \times 10^{-5}\ pM}{\eta} \qquad (1)$$

where

p pump outlet pressure, pounds per square inch

M oil flow, pounds per minute

η pump efficiency

For example, an indication of the power required to deliver 2 pounds per minute of oil through a 0.048-inch jet is 0.0095 horsepower, as calculated using an oil pressure of 29 pounds per square inch and a pump efficiency of 50 percent. In order to deliver 4.5 pounds per minute of oil through a 0.035-inch jet, 0.18 horsepower is required as calculated using an oil pressure of 400 pounds per square inch and a pump efficiency of 80 percent.

ANALYSIS OF EXPERIMENTAL RESULTS

Although this investigation was conducted for the most part at a load of 368 pounds (the approximate gravity load acting upon this type of bearing in the aircraft gas-turbine engine), the results may be used for higher loads inasmuch as reference 1 shows that there is only a small increase in bearing operating temperature for the load range 368 to 1113 pounds.

The experimental results of the present investigation indicate that the cooling effect of the oil is a function of the velocity of the oil as well as of the amount of oil. An analysis of these effects with regard to bearing temperature follows.

Velocity effect. - The jet velocity may be calculated from the jet size for various oil flows from the relation

$$V = 0.0574 \frac{M}{d^2} \qquad (2)$$

in which an average specific gravity for oil of 0.854 is assumed, and where

V average stream velocity, feet per second

d jet diameter, inch

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For convenience in estimating the oil inlet velocity, equation (2) is plotted in figure 10 for jet diameters from 0.02 to 0.2 inch and oil flows from 0.5 to 12 pounds per minute.

An increase of oil inlet velocity at a given flow decreases both the outer-race maximum and the inner-race temperatures (fig. 11). The inner-race temperature decreases more rapidly than the outer-race maximum temperature with an increase in oil inlet velocity. This effect is particularly evident at the lower values of oil flow, for which engine designers always strive. The effect of oil inlet velocity is greater for velocity values below about 60 feet per second; therefore for satisfactory lubrication effectiveness, a minimum oil inlet velocity of approximately 60 feet per second is recommended for conditions similar to those investigated herein. For example, at a DN of 1.2×10^6, the same inner-race temperature (200° F) may be obtained with an oil flow of 2 pounds per minute as with an oil flow of 6 pounds per minute, if the oil inlet velocity is 120 feet per second for the low flow and 24 feet per second for the high flow.

The increase in bearing temperature, which occurs when the jet is directed at an angle other than zero (fig. 5), may be accounted for, in part, by the decrease in jet-stream velocity perpendicular to the bearing face. When the data of figures 5 and 11 are compared, it is found that the decrease in jet-stream velocity, perpendicular to the bearing face, approximately accounts for the increase in bearing temperature for plus angles of jet impingement. For minus angles of jet impingement, however, the bearing temperature is appreciably greater than can be attributed to decrease in velocity perpendicular to the bearing face.

Heat-transfer considerations. - The application of rigorous heat-transfer theory to obtain cooling correlations for high-speed roller bearings having jet lubrication requires considerably more theoretical development and experimental data than are at present available. By employing existing heat-transfer theory and simplifying assumptions, however, it is possible to minimize the number of variables when certain of the research data included herein are plotted. These heat-transfer considerations are given in appendix B.

As a first approximation, equation (B7) of appendix B implies that the variables to be plotted for a given oil inlet temperature and bearing load are $\frac{(T_{IR}-T_{OI})}{(DN)^y}$ and $\left(\frac{d}{M}\right)^n$, where $(T_{IR}-T_{OI})$ is the difference between the inner-race bearing temperature and the oil inlet temperature in $^{\circ}F$, DN is the bore in millimeters multiplied by the bearing speed in rpm, d is the jet diameter in inches, and M is the oil flow in pounds per minute. The constant and the exponent in equation (B7) are dependent upon the particular application. 1414

The inner-race-temperature data of figure 7 are shown replotted in figure 12 on log-log paper with $(T_{IR}-T_{OI})$ as the ordinate, $\left(\frac{d^x}{M}\right)^n$ as the abscissa (d is in inches and M in lb/min), and jet diameter as the parameter. (There is no basis in the analysis presented in appendix B for changing the value of the exponent x from one; however, this change was found to give empirically better correlation.)

The exponent x was determined by plotting d^x/M against $(T_{IR}-T_{OI})$ on log-log paper using various values of x for each of the DN values investigated. The final value of x for a given DN was taken as that value which resulted in the least scatter of data and the straightest line through the data. The value of x was found to be essentially constant for the various DN values investigated.

The exponent n was obtained for each DN value as the slope of the faired straight line drawn through the data of figure 12. It may be seen from figure 12 that the value of n did not vary greatly with DN.

The exponent y in equation (B7) was determined by plotting $\left(\frac{d^x}{M}\right)^n$ against $(T_{IR}-T_{OI})$ with oil flow as the parameter (fig. 13). Representative values of $x = 0.8$ and $n = 0.4$ were used for this correlation. Straight lines were faired through the oil-flow curves for each DN value to obtain mean values of $(T_{IR}-T_{OI})$ at various values of $\left(\frac{d^{0.8}}{M}\right)^{0.4}$; $\frac{(T_{IR}-T_{OI})}{(DN)^y}$ was then determined for various values of y. The final value of y was taken as that value which gave the least percentage difference between the four DN values over the range of $\left(\frac{d^{0.8}}{M}\right)^{0.4}$ investigated.

The final cooling-correlation curve is shown in figure 14, which is a plot of equation (B7). The slope of the straight line of figure 14 is the constant E'''' of equation (B7) for the conditions

1414 investigated. This curve indicates that, when $\frac{(T_{IR}-T_{OI})}{(DN)^{1.2}}$ is plotted against $\left(\frac{d^{0.8}}{M}\right)^{0.4}$ for a given oil inlet temperature and bearing load, a representative single straight line results regardless of whether bearing speed, oil flow, or jet diameter is varied.

Effect of surface finish at cage-locating surface and cage pockets. - The values of the surface finish (obtained by use of a Profilometer) of the component parts of a sample bearing similar to the test bearings used herein, are given in reference 1. Inasmuch as the cage of an inner-race-riding cage-type bearing rides on two lands, one on either side of the inner-race roller track, the surface finish of these inner-race lands, as well as the surface finish of the mating cage surface, is significant. These surface-finish values together with the cage-pocket finish are as follows:

	(microinches rms)
Inner-race lands:	
Left land, axially	10-12
Right land, axially	8- 9
Left land, circumferentially	3- 5
Right land, circumferentially	3- 4
Cage-locating surface:	
Left and right surfaces, axially	30-40
Left and right surfaces, circumferentially	20-25
Cage pockets:	
Axially	18-22
Circumferentially	18-22

Much can probably be done regarding closer tolerances of surface finishes at the critical locations particularly at the cage-locating surface in order to enable these surfaces to function with a minimum of frictional heat.

Concluding remarks. - The cage design of high-speed roller bearings could be modified so as to induce the flow of lubricant to the areas where it is most needed. The addition of expediently designed chamfers, grooves, and passageways should enhance the lubricating and cooling effectiveness of a given quantity of lubricant.

Inasmuch as operation at the cage-locating surfaces and in the cage pockets is thought to occur in the boundary region of lubrication, the friction properties of the cage material, with respect 1414
to the steel used, are also important considerations. Improved cage materials and improved surface finishes may offer marked advantages in this respect.

SUMMARY OF RESULTS

In an experimental investigation of single-jet lubrication of 75-millimeter-bore (size 215) cylindrical-roller, inner-race-riding cage-type bearings at high speeds over a range of oil flows and jet diameters, it was found that for the positions investigated, the inner- and outer-race bearing temperatures were a minimum when the oil was directed perpendicular to the bearing face at the cage-locating surface.

The following results were obtained when the oil was directed perpendicular to the bearing face at the cage-locating surface:

1. For all jet diameters and all values of oil flow investigated, the outer-race maximum temperature increased linearly with an increase in DN value. (DN is the product of the bearing bore in mm and shaft speed in rpm.) The inner-race temperature, however, increased with DN value at a rate slightly greater than linear.

2. An increase in oil inlet velocity at a given flow decreased both the outer-race and the inner-race temperatures. It appears that a point of diminishing returns, regarding the effectiveness of an increase of oil inlet velocity, occurs at about 60 feet per second for the conditions investigated.

3. Heat-transfer considerations indicated that to minimize the number of parameters when dealing with inner-race temperatures the variables to be plotted should be $\frac{(T_{IR}-T_{OI})}{(DN)^y}$ and $\left(\frac{d}{M}\right)^n$, where $(T_{IR}-T_{OI})$ is the difference between the inner-race bearing temperature and the oil inlet temperature, d is the oil-jet diameter, M is the oil flow and n and y are constants. It was empirically found, however, that the variable $(d^{0.8}/M)^n$ gave better correlation than did $(d/M)^n$.

4. When $\frac{(T_{IR}-T_{OI})}{(DN)^{1.2}}$ was plotted against $\left(\frac{d^{0.8}}{M}\right)^{0.4}$ for a given operating condition, a representative single straight line resulted regardless of whether bearing speed, oil flow, or jet diameter was varied.

Lewis Flight Propulsion Laboratory,
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APPENDIX A

CALIBRATION OF OIL JETS

During calibration of the jets, the flow did not significantly vary with a change in oil viscosity for a given jet. Dimensional analysis shows that this result is to be expected for a jet having a length-diameter ratio approaching that of a plate orifice (reference 4).

For certain of the jets investigated, the pressure-flow curve was discontinuous and hence two values of flow were obtained at a critical value of pressure. A determination of the Reynolds number at the critical pressures indicate values of Reynolds number below the critical range of 1200 to 2500 for the results covered in the investigation reported herein.

A calculation of the critical pressures (reference 4), above which a jet will not flow full, may be derived by means of the Bernoulli equation.

For jets having length-diameter ratios of 2.5 or greater, the values of the experimental and calculated critical pressures closely agree. For jets having length-diameter ratios of approximately 1, however, the experimental critical pressures were less than the calculated values. This effect is to be expected for very short jets, inasmuch as for values of pressure below the calculated critical pressure, the length of the oil-stream contraction may be greater than the jet length thus causing the stream of oil to flow free of the jet prematurely (that is, with regard to critical pressure). The discontinuity of the pressure-flow curve is thus probably due to the transition from full jet flow to flow free of the jet rather than due to a transition from laminar to turbulent flow.

In order for the oil jet to flow full over the entire pressure range, properly rounded inlets should be provided instead of sharp corners. Sharp-edged orifices will also have a continuous pressure-flow relation.

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APPENDIX B

CORRELATION OF INNER-RACE TEMPERATURE

WITH OPERATING VARIABLES

The heat-transfer coefficient for a liquid flowing perpendicular to a cylinder may be estimated by means of the following relation (reference 5):

$$\frac{hD}{k} = E\left(\frac{\rho D v}{\mu}\right)^n \left(\frac{\mu c_p}{k}\right)^m \tag{B1}$$

where

h coefficient of heat transfer, Btu/hr sq ft °F

D characteristic diameter, ft

k thermal conductivity, Btu/hr ft °F

E experimental constant

ρ density, slugs/cu ft

v velocity, ft/hr

μ viscosity, lb-hr/sq ft

c_p specific heat at constant pressure, Btu/slug °F

g gravity acceleration, ft/hr^2

n,m exponents

In order to correlate inner-raceway temperatures of high-speed inner-race-riding cage-type roller bearings having single-jet lubrication directed at the cage-locating surface with operating variables, the above relation may be employed, together with the following assumptions:

(a) For a constant oil inlet temperature, bearing speed, and load, ρ, μ, c_p, and k are effectively constant.

(b) The thickness of flow upon the cooled-surface area of the inner-raceway assembly is proportional to the oil-jet diameter at a given flow.

(c) At any instant the cooled-surface area (annular) of the inner-raceway assembly is proportional to the diameter of the oil jet.

(d) The oil inlet temperature is used in correlating the bearing inner-race operating temperature with the flow parameters rather than the mean oil temperature.

(e) As a first approximation, the heat developed within the bearing inner race is considered to be essentially independent of oil flow and to remain constant for a given speed and load.

(f) The majority of the heat removed from the inner-raceway assembly is removed by the oil impinging at the cage-locating surface.

(g) The power input to the inner race is proportional to $(DN)^y$.

By using assumption (a) equation (B1) can be reduced to

$$h = E' \left(V^n D^{n-1}\right) \tag{B2}$$

where

$$E' = E\rho^n c_p{}^m \left(k^{1-m}\right)\left(\mu^{m-n}\right)$$

When assumptions (b) and (c) are employed, equation (B2) becomes:

$$h = E'' \left(\frac{M^n}{d^{n+1}}\right) \tag{B3}$$

where

M flow, lb/hr

d jet diameter, ft

where

$$E'' = \frac{E'}{a^n b \rho^n}$$

a proportionality constant (see assumption (b))

b proportionality constant (see assumption (c))

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The heat flow from the inner race of the bearing may be estimated using assumption (d) and the following equation:

$$H = Ah\,(T_{IR} - T_{OI}) \tag{B4}$$

where

H heat flow, Btu/hr

A cooled area, sq ft

h coefficient of heat transfer, Btu/hr sq ft $^{\circ}F$

T_{IR} inner-race temperature, $^{\circ}F$

T_{OI} oil inlet temperature, $^{\circ}F$

By employing assumptions (c), (e), and (f), equation (B4) for a given speed and load becomes

$$H = P = (b)(d)(h)(T_{IR} - T_{OI}) = \text{constant} \tag{B5}$$

where

P power into inner race, Btu/hr

b proportionality constant (assumption (c))

When equation (B3 is substituted into equation (B5)

$$(T_{IR} - T_{OI}) = E''' \left(\frac{d}{M}\right)^n \tag{B6}$$

where

$$E''' = \frac{a^n P}{(E)(c_p)^m (k^{1-m})(\mu^{m-n})}$$

E''' is assumed to be essentially constant for given values of oil inlet temperature, bearing speed, and bearing load.

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When assumption (g) is introduced, equation (B6) becomes

$$\frac{T_{IR}-T_{OI}}{(DN)^y} = E'''' \left(\frac{d}{M}\right)^n \tag{B7}$$

where

$$E'''' = \frac{(a)^n c}{E(c_p)^m(k^{1-m})(\mu^{m-n})}$$

where

c proportionality constant (assumption (g))

The foregoing heat-transfer considerations indicate that as a first approximation the data for a given oil inlet temperature, bearing speed, and bearing load may be presented by plotting the variable $\frac{(T_{IR}-T_{OI})}{(DN)^y}$ against $\left(\frac{d}{M}\right)^n$.

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TABLE I - PHYSICAL CHARACTERISTICS OF TEST BEARINGS

Bearing number	8		9		6	
Construction	One-piece inner-race-riding cage					
Number of rollers	18		18		18	
Roller length-diameter ratio	1		1		1	
Pitch diameter of bearing, (in.)	4.036		4.036		4.036	
	Before	After	Before	After	Before	After
Total running time, (hr)	0	38.6	0	44.7	0	195.9
[a]Δ temperature (°F) x time (min)	0	3.22×10^5	0	1.38×10^5	0	11.64×10^5
Roller diameter, (in.)	[b]0.5513	0.5509	[b]0.5513	0.5510	[b]0.5513	0.5510
Roller length, (in.)	[b]0.5510	0.5509	[b]0.5510	0.5510	[b]0.5510	0.5507
Diametral clearance between cage and roller, (in.)	[b]0.0087	0.0076	[b]0.0087	0.0103	[b]0.0087	0.010
Axial clearance between roller and inner-race flange, (in.)	[b]0.002	0.002	[b]0.002	0.0022	[b]0.002	0.002
Axial clearance between roller and cage, (in.)	[b]0.006	0.006	[b]0.007	0.008	[b]0.007	0.009
Unmounted bearing:						
[c]Diametral clearance, (in.) Bearing	0.0018	0.0019	0.0016	0.0017	0.0020	0.0022
Cage	.021	.021	.014	.017	.015	.018
[d]Eccentricity, (in.)	.0000	.0002	.0000	.0000	.0000	.0002
Mounted bearing:						
[e]Diametral clearance, (in.) Bearing	0.0005	0.0005	0.0002	-------	0.0009	0.0009
Cage	.019	.022	.013	.016	.014	.017
[f]Eccentricity, (in.)	.0004	.0005	.00045	-------	.0005	.0004
Remarks	Satisfactory operation		Satisfactory operation		Satisfactory operation	

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[a]Severity factor (reference 1).

[b]Measurements obtained from sample bearing.

[c]Measurement obtained in fixture with dial gage.

[d]Measurement obtained in fixture with dial gage, inner race rotating and outer race stationary.

[e]Measurements obtained as mounted in test rig with dial gage.

[f]Measurements obtained as mounted in test rig with dial-gage inner race rotating and outer race stationary.

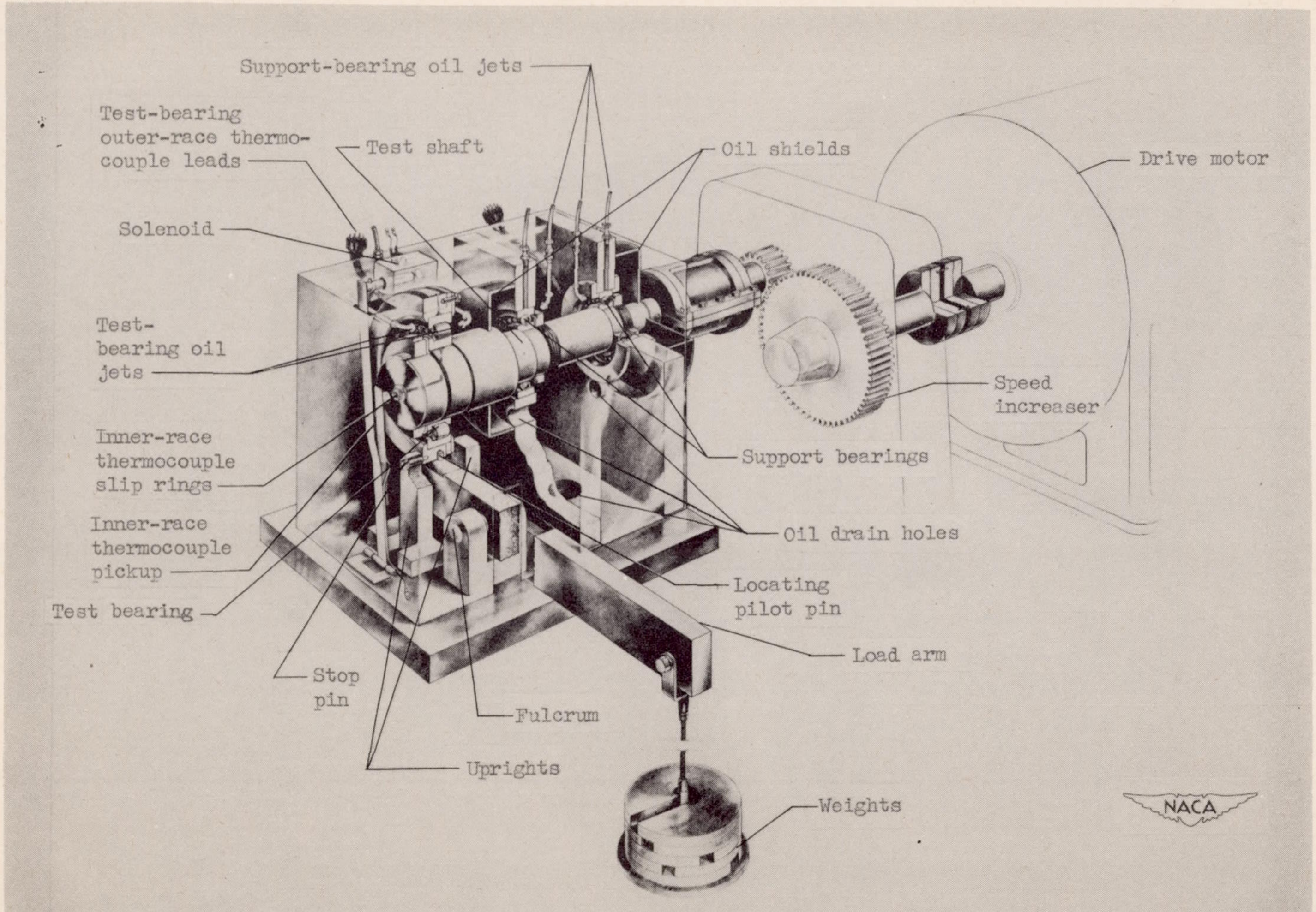
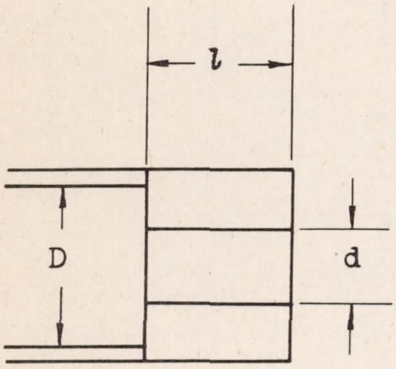


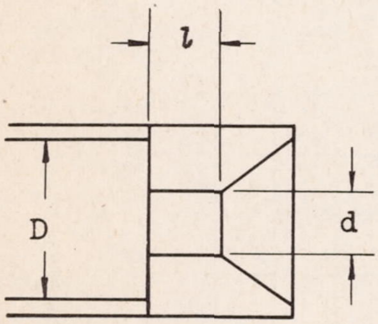
Figure 1. - Cutaway view of radial load rig.



Sizes investigated

d (in.)	l (in.)	l/d	D (in.)
0.180	∞	∞	0.180
.157	0.125	0.80	.180
.129	.218	1.70	.180
.089	.218	2.46	.180

(a) Type A.



d (in.)	l (in.)	l/d	D (in.)
0.129	0.129	1	0.173
.089	.089	1	.173
.066	.066	1	.173
.048	.048	1	.173
.035	.035	1	.189
.023	.023	1	.189

(b) Type B.

Figure 2. - Oil-jet types and sizes investigated.

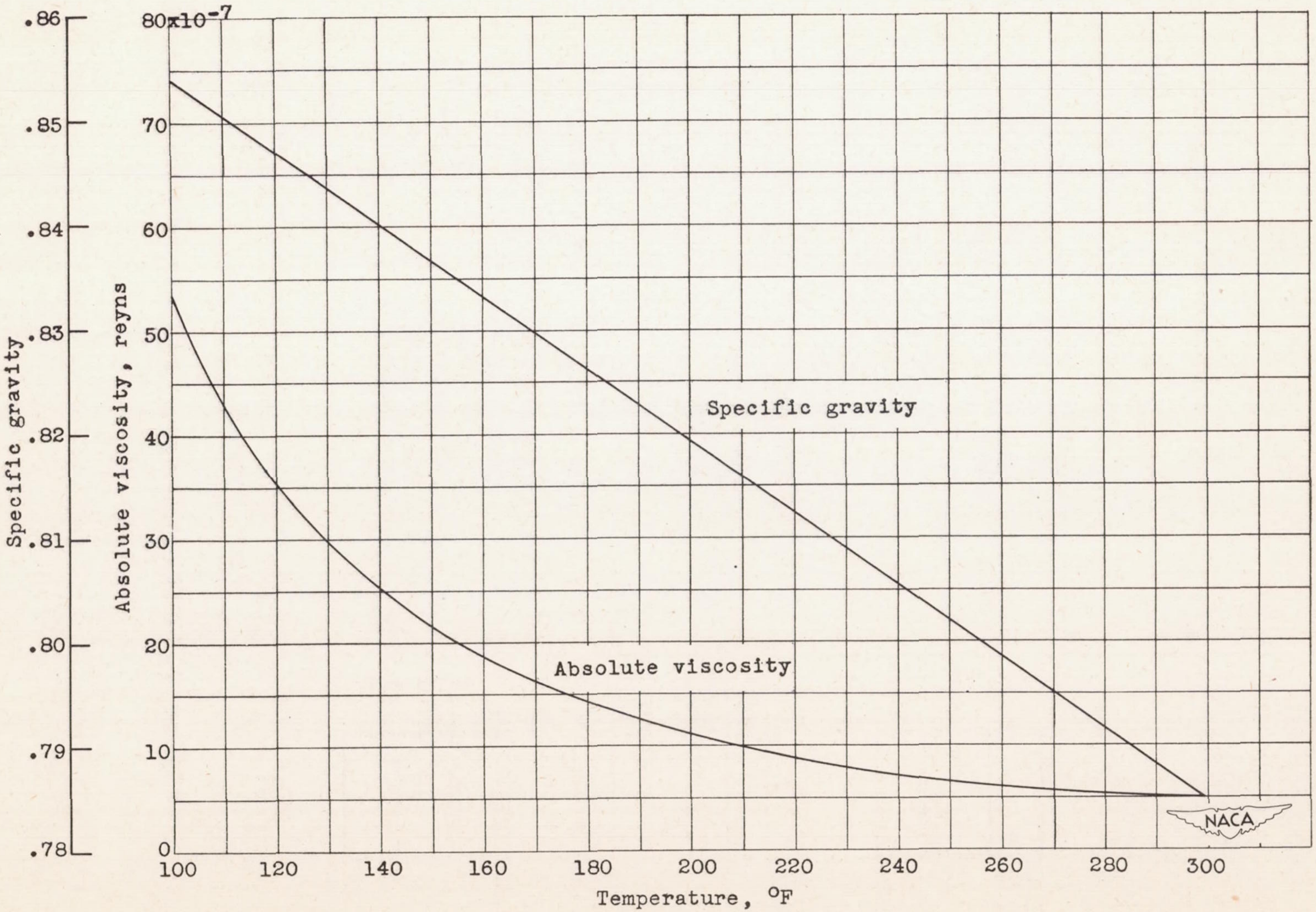


Figure 3. - Absolute viscosity and specific gravity of oil. Pour point, -50° F; flash point, 310° F; viscosity index, 150.

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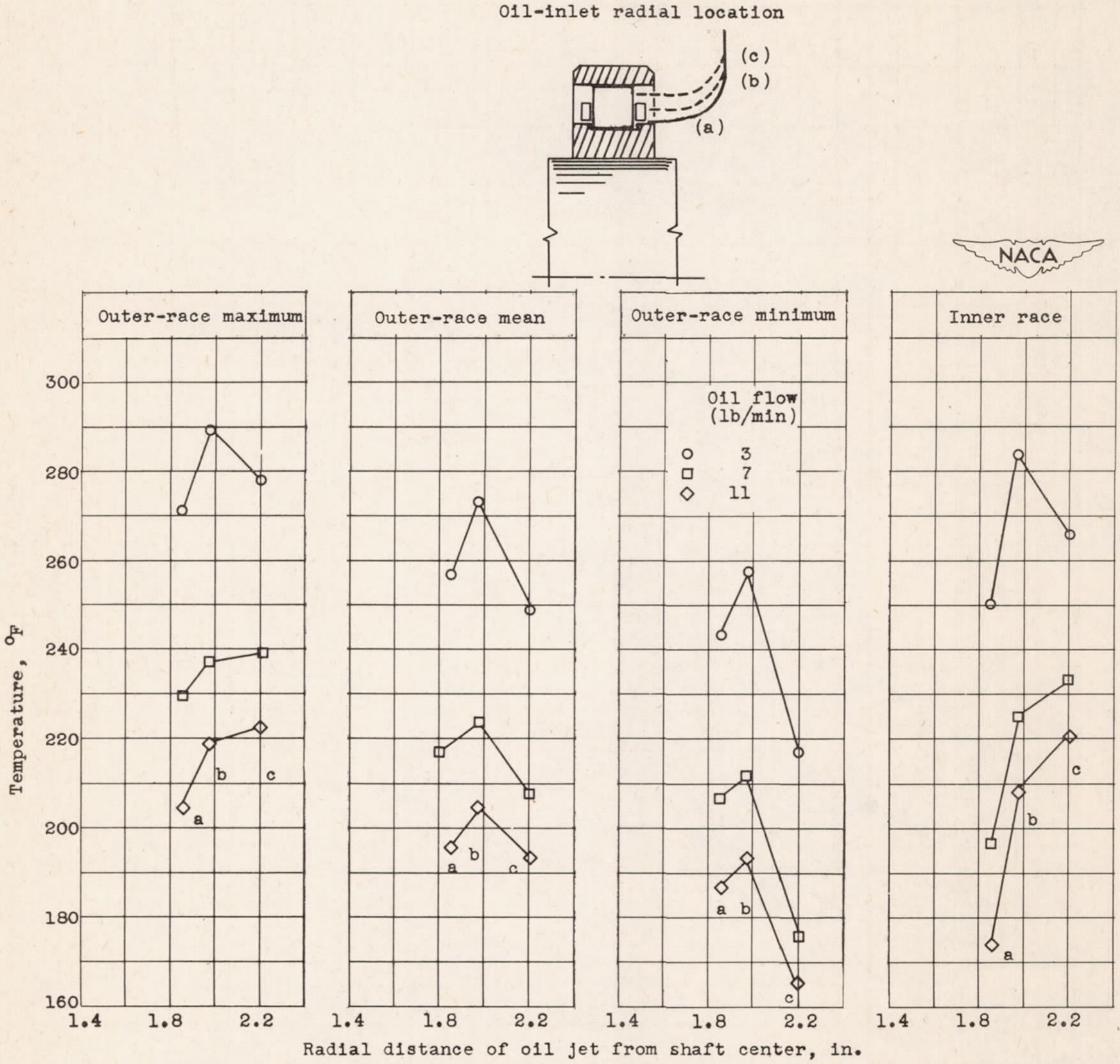


Figure 4. - Effect of oil-inlet radial location on operating temperature of bearing 8 with type-B single-jet circulatory oil feed over flow range from 3 to 11 pounds per minute. Load, 368 pounds; DN, 1.2×10^6; oil-jet diameter, 0.089 inch; oil inlet temperature, 100° F.

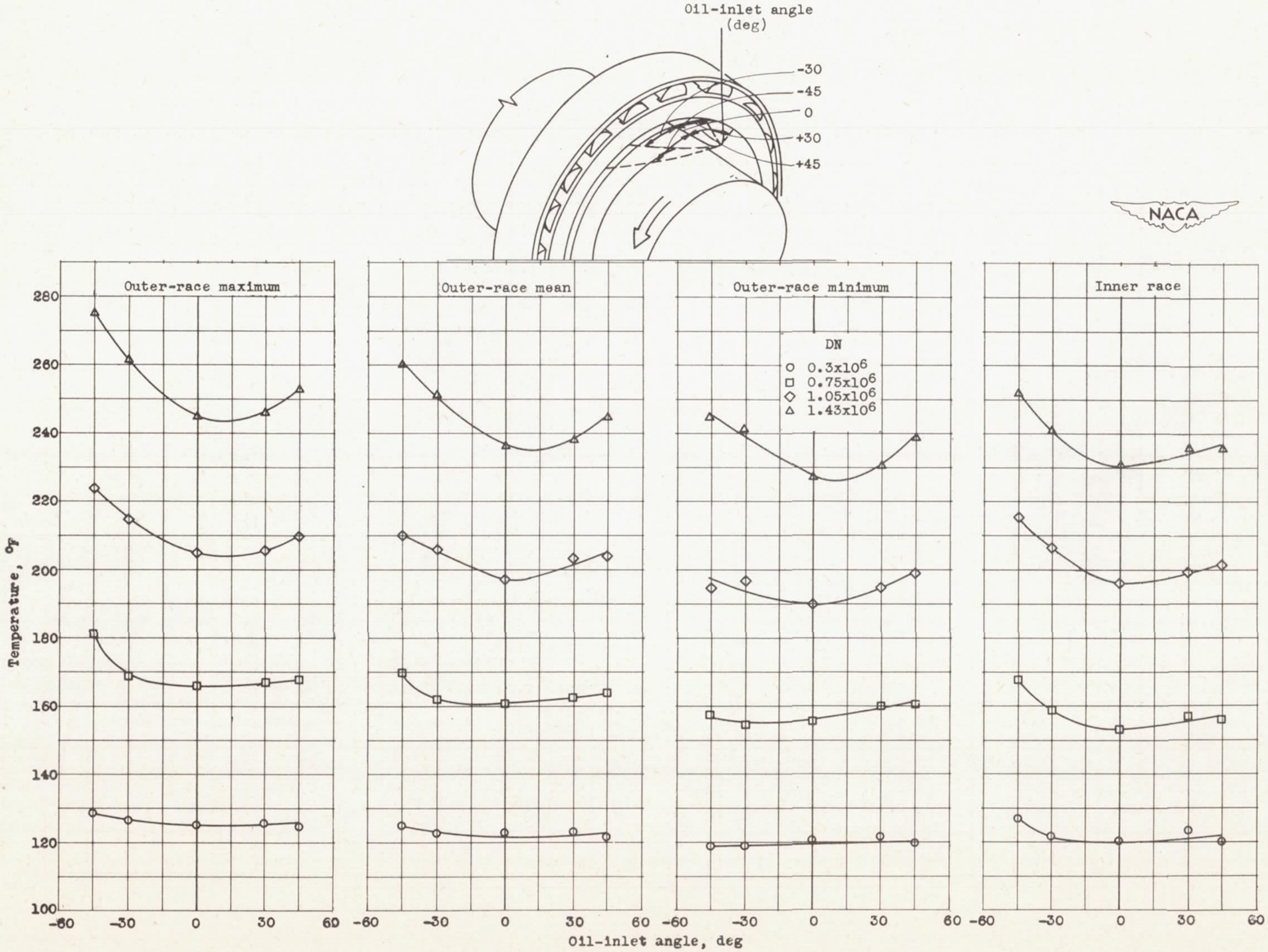


Figure 5. - Effect of impingement angle of oil jet on operating temperature of bearing 9 with type-A single-jet circulatory oil feed. Load, 113 pounds; DN, 0.3×10^6 to 1.43×10^6; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 10 pounds per square inch; oil inlet temperature, 100° F.

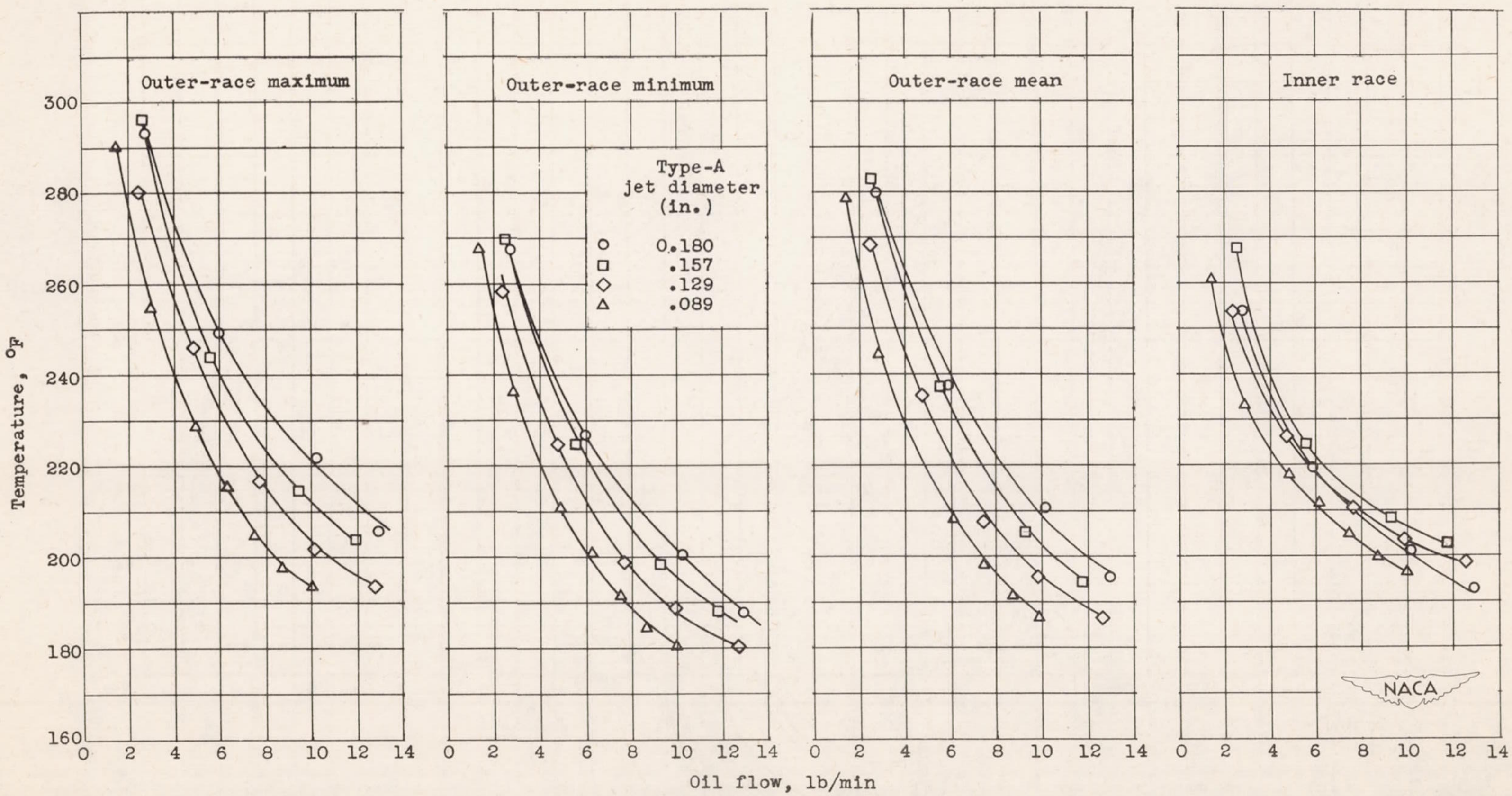


Figure 6. - Effect of oil flow on temperatures of bearing 9 for type-A jets. Load, 113 pounds; DN, 1.43×10^6; oil inlet temperature, 100° F.

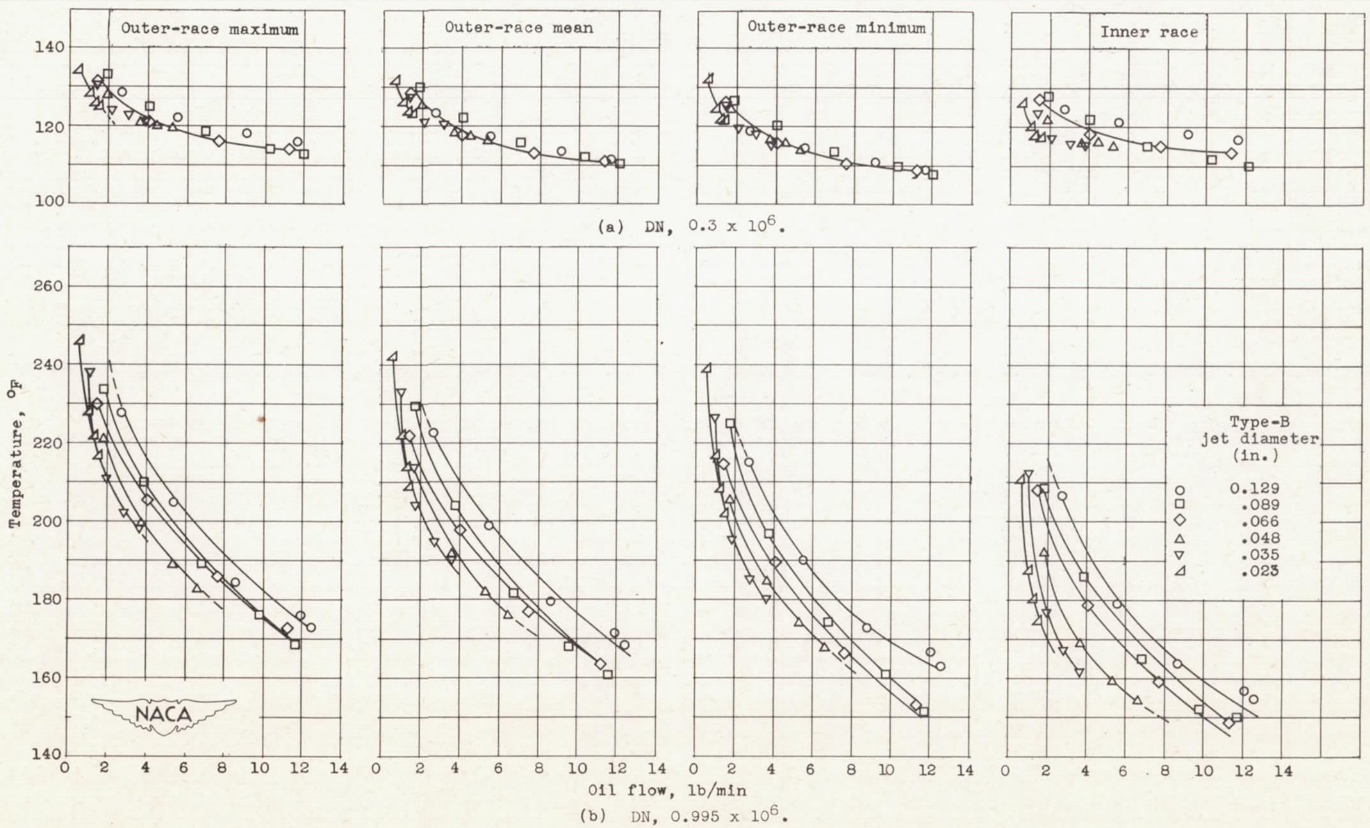


Figure 7. - Variation of temperature of bearing 6 with oil flow for type-B jets. Load, 368 pounds; oil inlet temperature, 100° F.

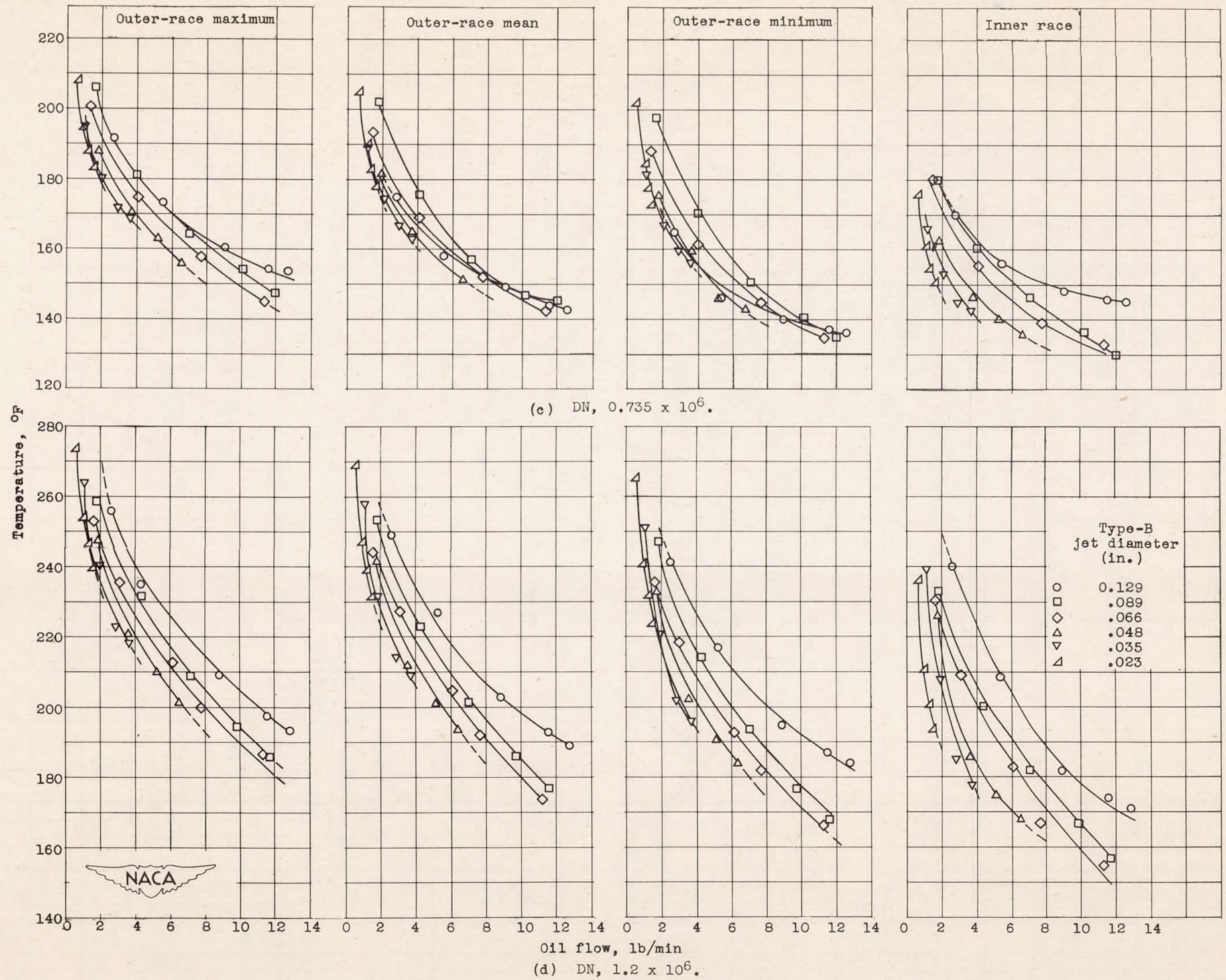
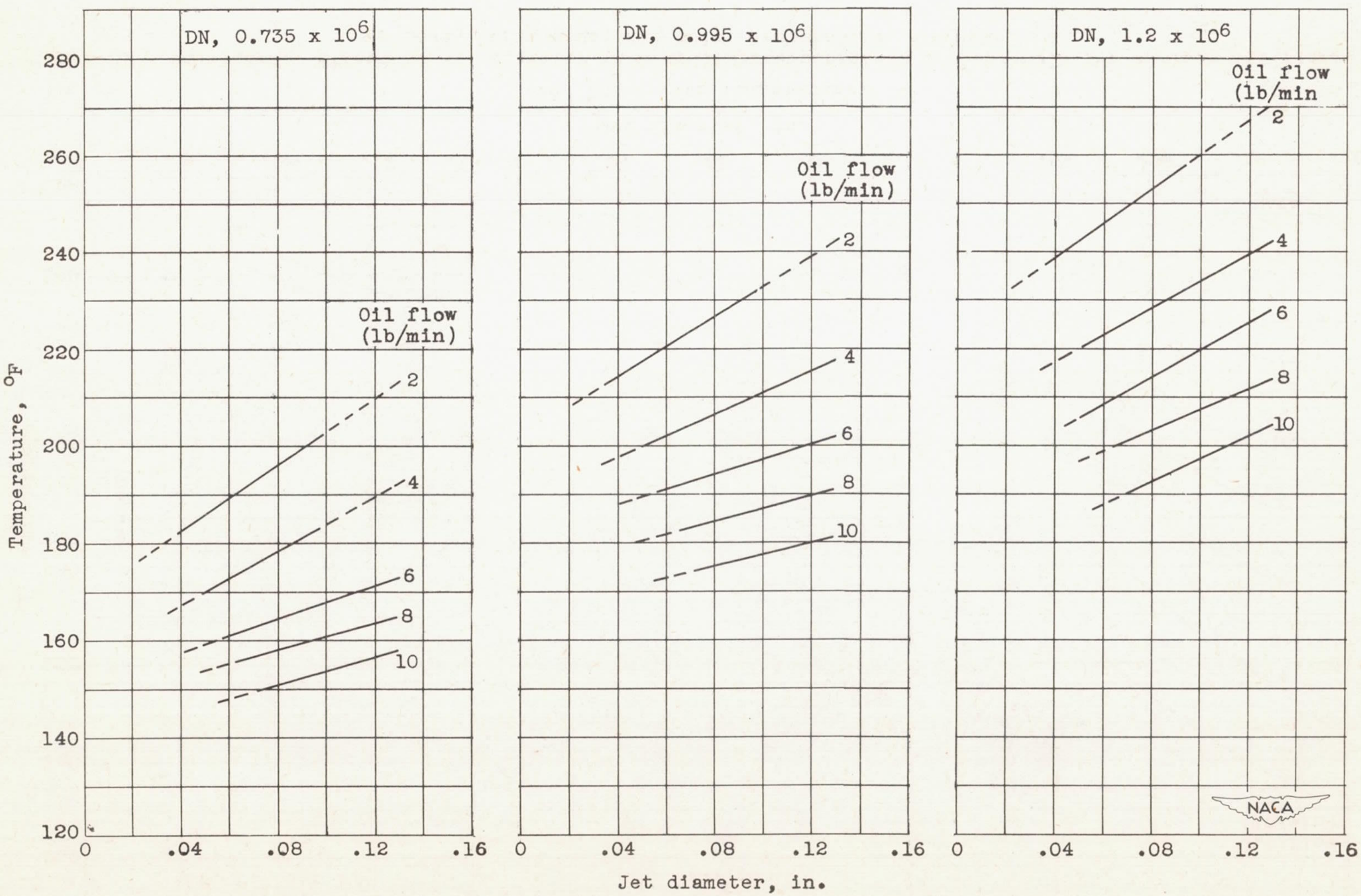
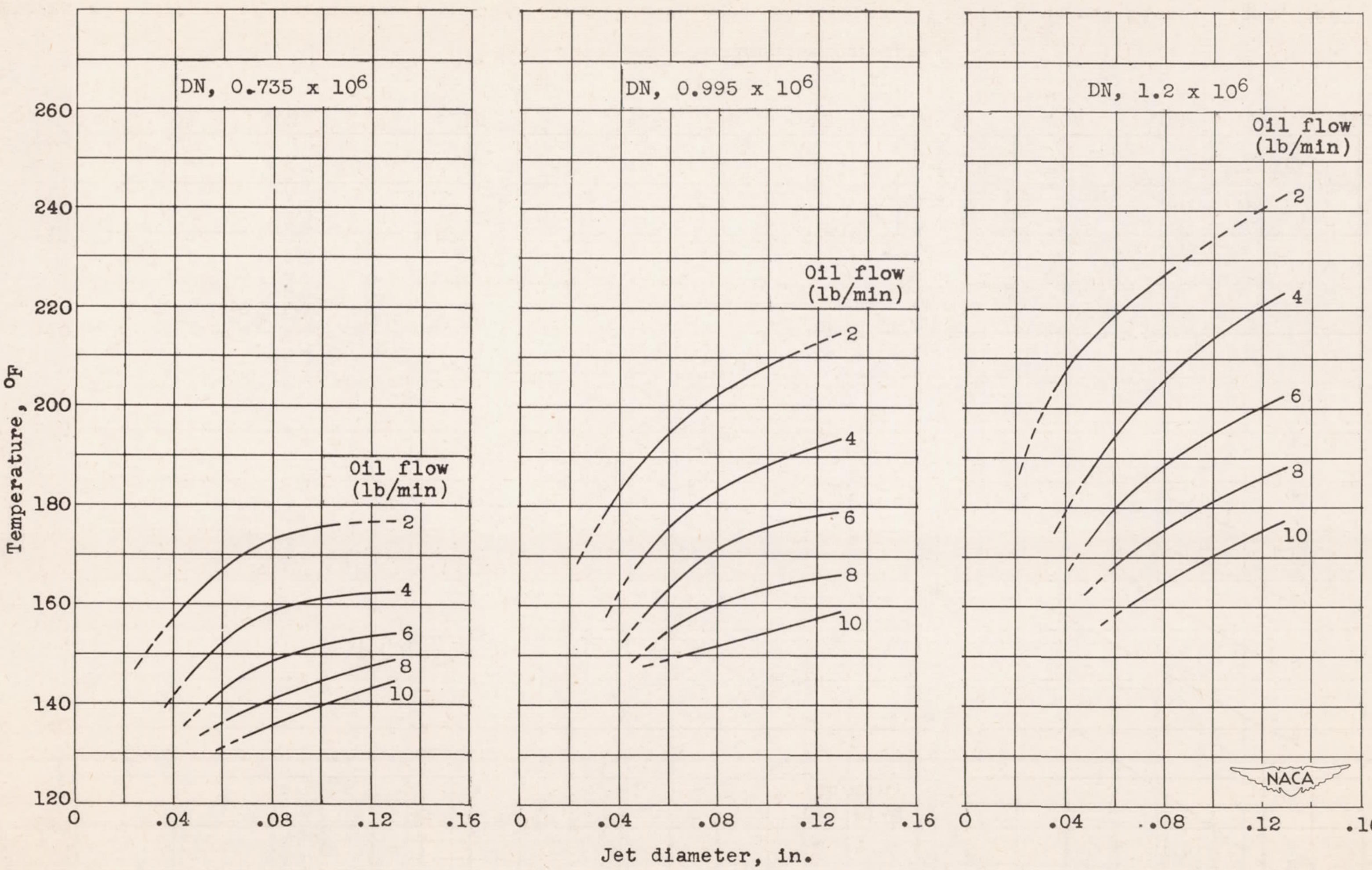


Figure 7. - Concluded. Variation of temperature of bearing 6 with oil flow for type-B jets. Load, 368 pounds; oil inlet temperature, 100° F.



(a) Outer-race maximum temperature.

Figure 8. - Effect of jet diameter on operating temperatures of bearing 6 for various oil flows. Load, 368 pounds; oil inlet temperature, 100° F.



(b) Inner-race temperature.

Figure 8. - Concluded. Effect of jet diameter on operating temperatures of bearing 6 for various oil flows. Load, 368 pounds; oil inlet temperature, 100° F.

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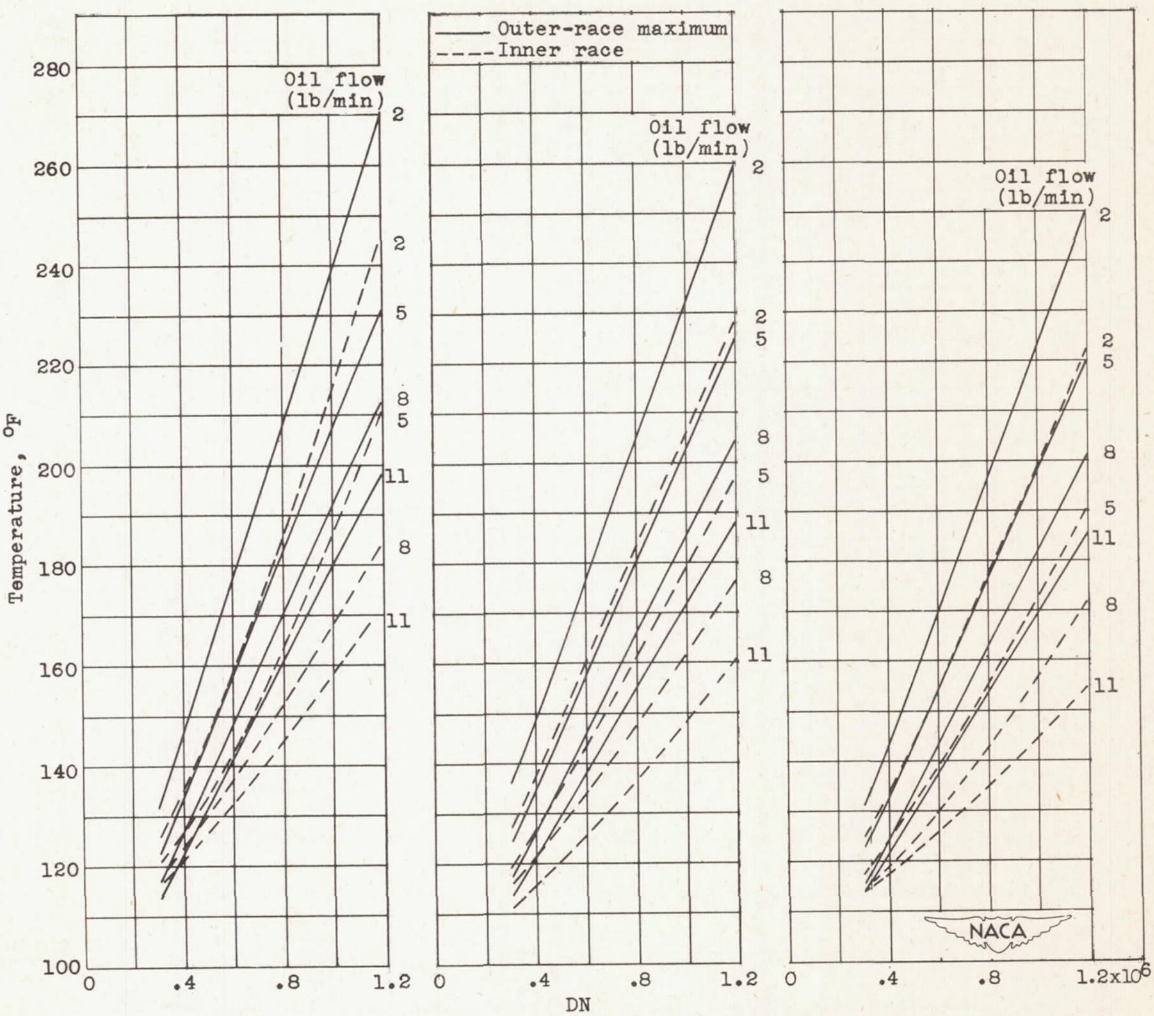
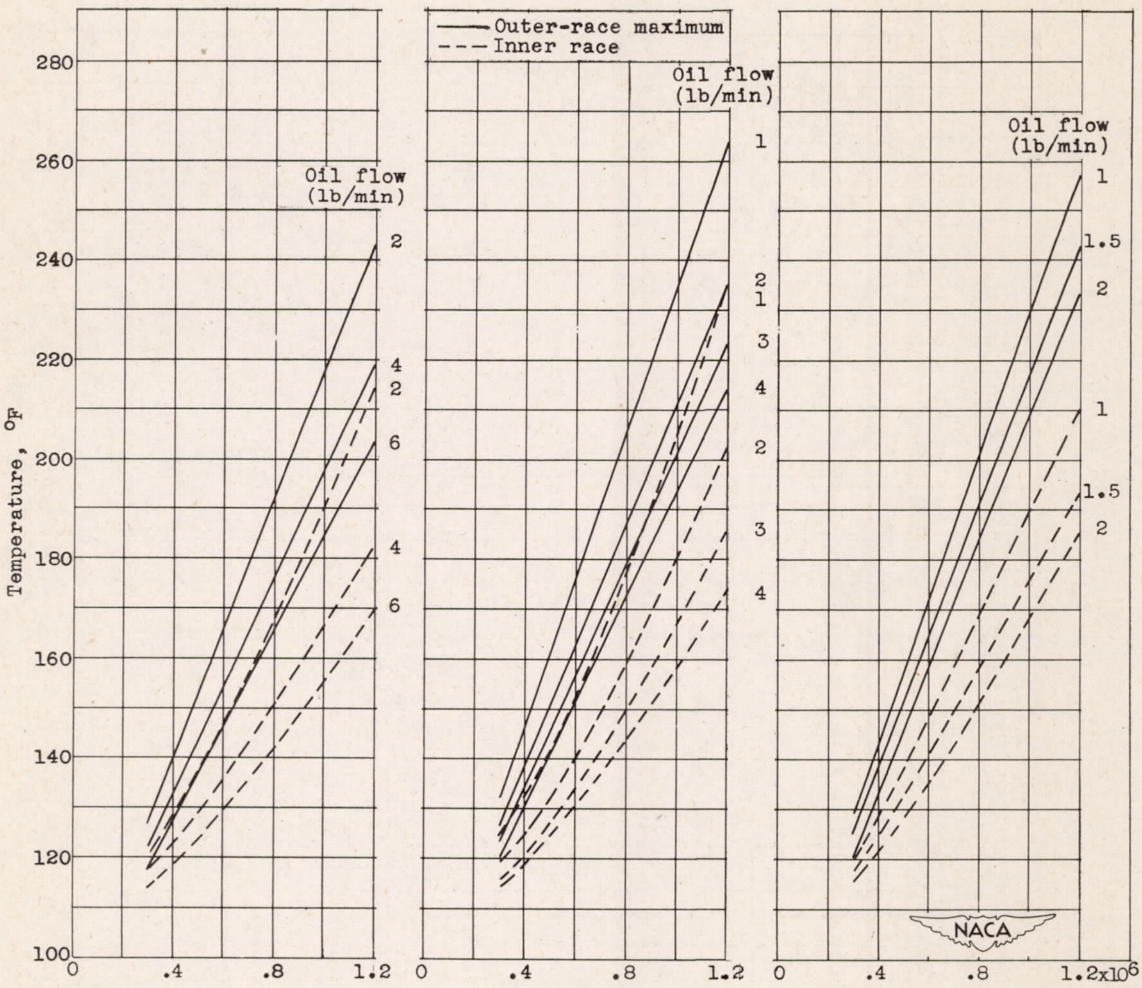


(a) Jet diameter, 0.129 inch. (b) Jet diameter, 0.089 inch. (c) Jet diameter, 0.066 inch.

Figure 9. - Effect of DN on outer-race maximum and inner-race temperatures of bearing 6 for various oil flows. Load, 368 pounds; oil inlet temperature, 100° F.



(d) Jet diameter, 0.048 inch.
(e) Jet diameter, 0.035 inch.
(f) Jet diameter, 0.023 inch.

Figure 9. - Concluded. Effect of DN on outer-race maximum and inner-race temperatures of bearing 6 for various oil flows. Load, 368 pounds; oil inlet temperature, 100° F.

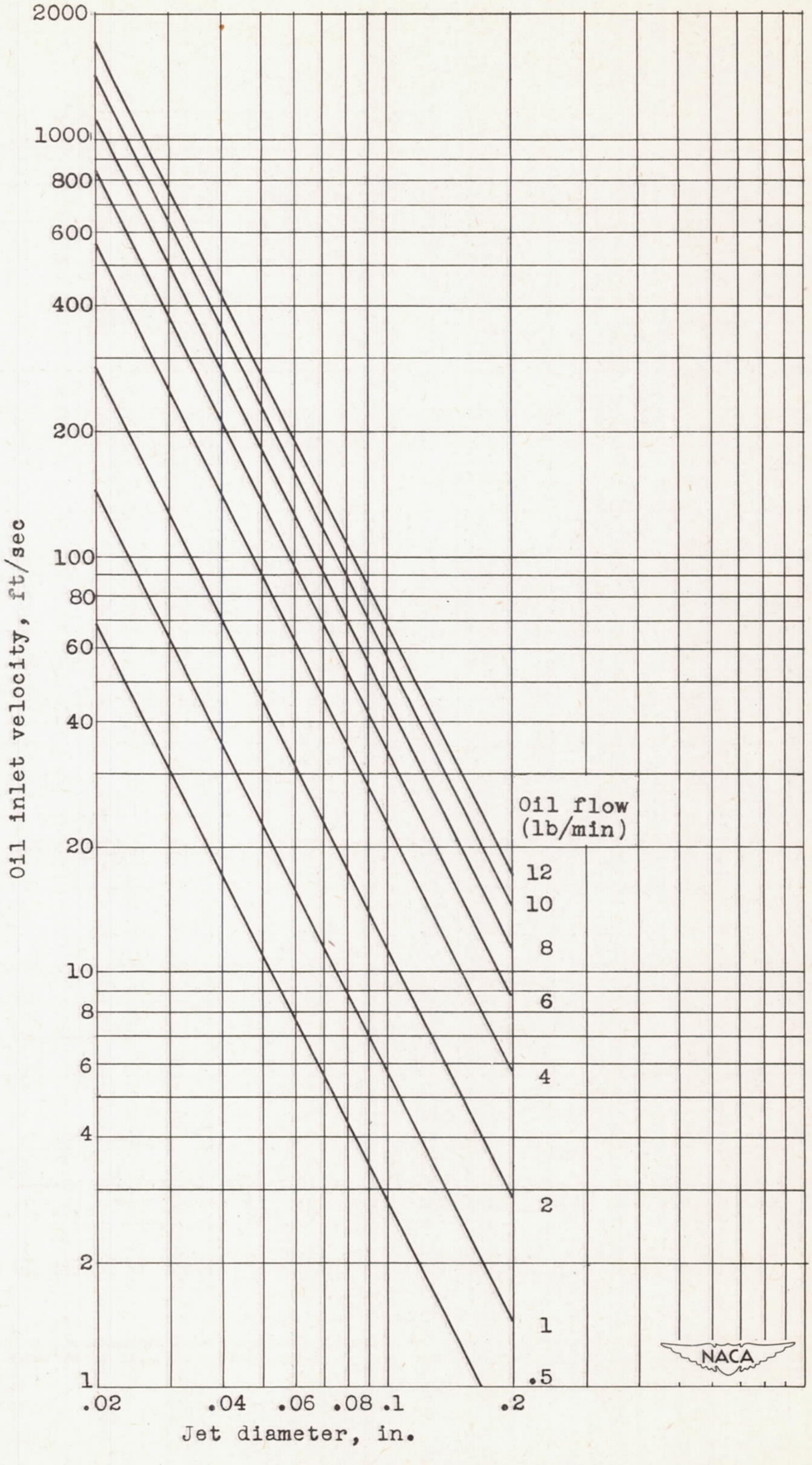


Figure 10. - Variation of jet diameter with oil inlet velocity for various oil flows.

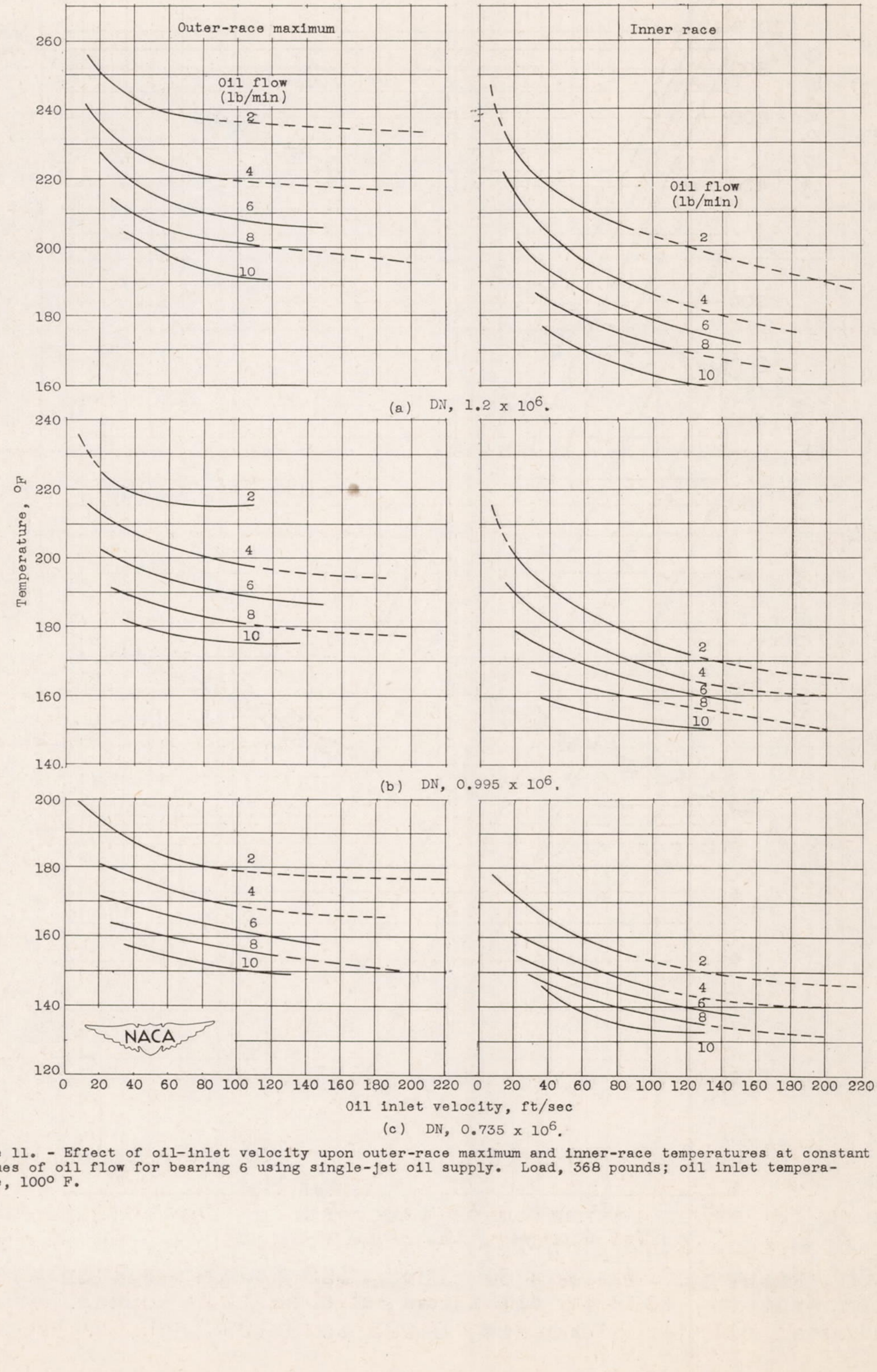


Figure 11. - Effect of oil-inlet velocity upon outer-race maximum and inner-race temperatures at constant values of oil flow for bearing 6 using single-jet oil supply. Load, 368 pounds; oil inlet temperature, 100° F.

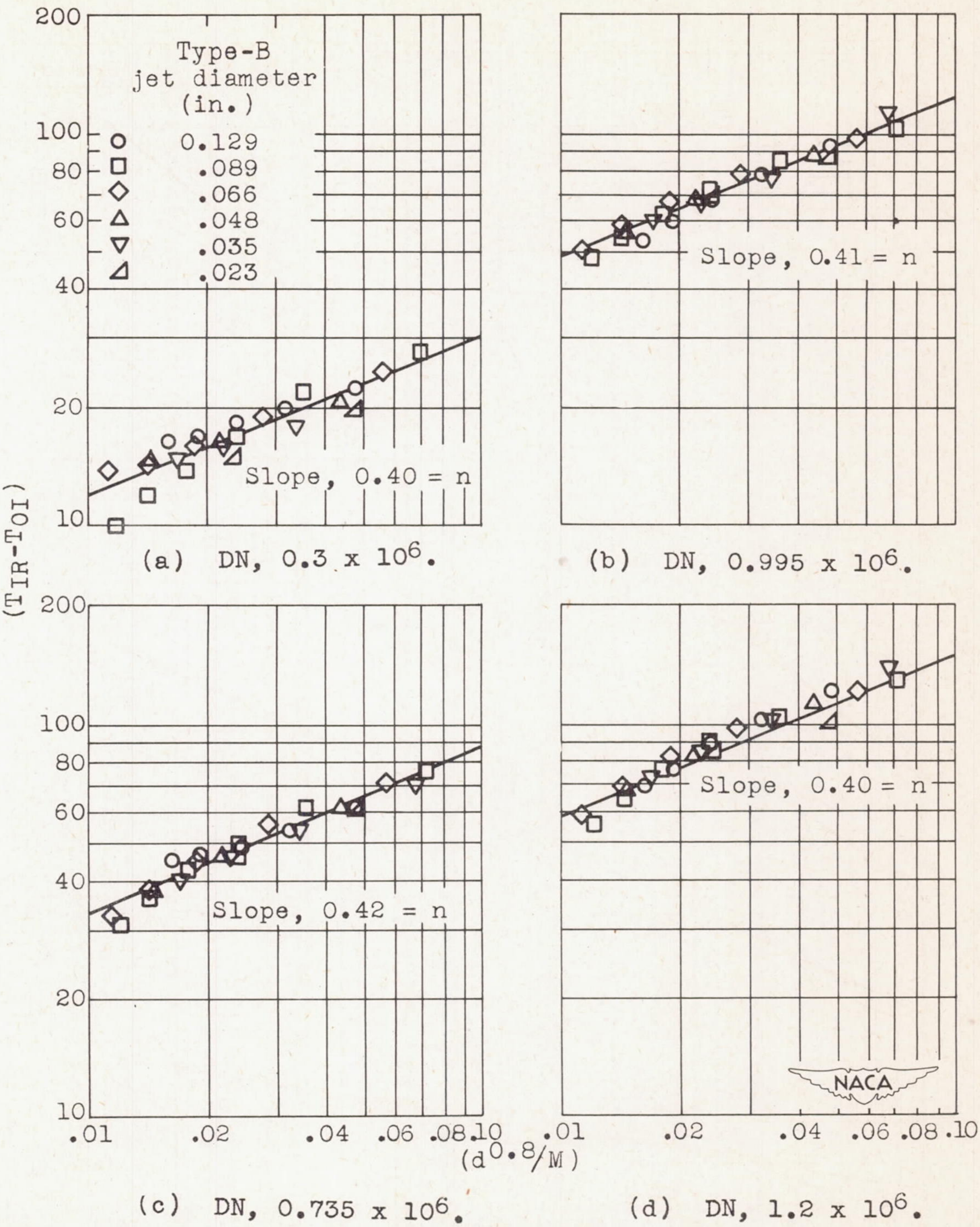


Figure 12. - Determination of exponent n for cooling correlation equation $\left[(T_{IR}-T_{OI}) = E'''(d^{0.8}/M)^n\right]$ for inner-race temperature of bearing 6. Load, 368 pounds; oil inlet temperature, 100° F; oil flows, 0.6 to 12.9 pounds per minute; oil-jet diameters, 0.023 to 0.129 inch.

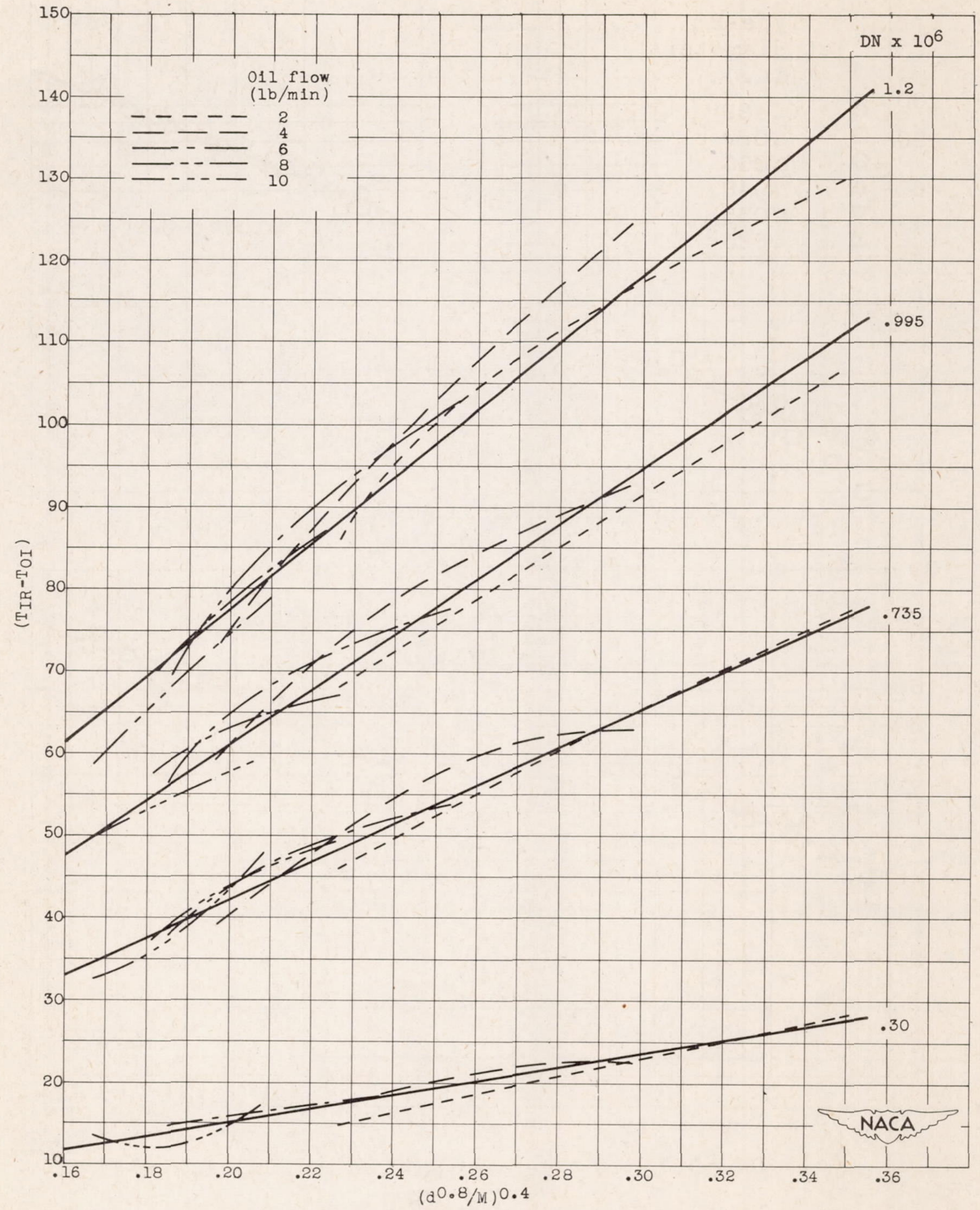


Figure 13. - Cooling correlation curves for inner-race temperature of bearing 6 at DN values 0.3 x 10^6, 0.735 x 10^6, 0.995 x 10^6, and 1.2 x 10^6. Load, 368 pounds; oil inlet temperature, 100° F; oil flows, 0.6 to 12.9 pounds per minute; oil-jet diameters, 0.023 to 0.129 inch.

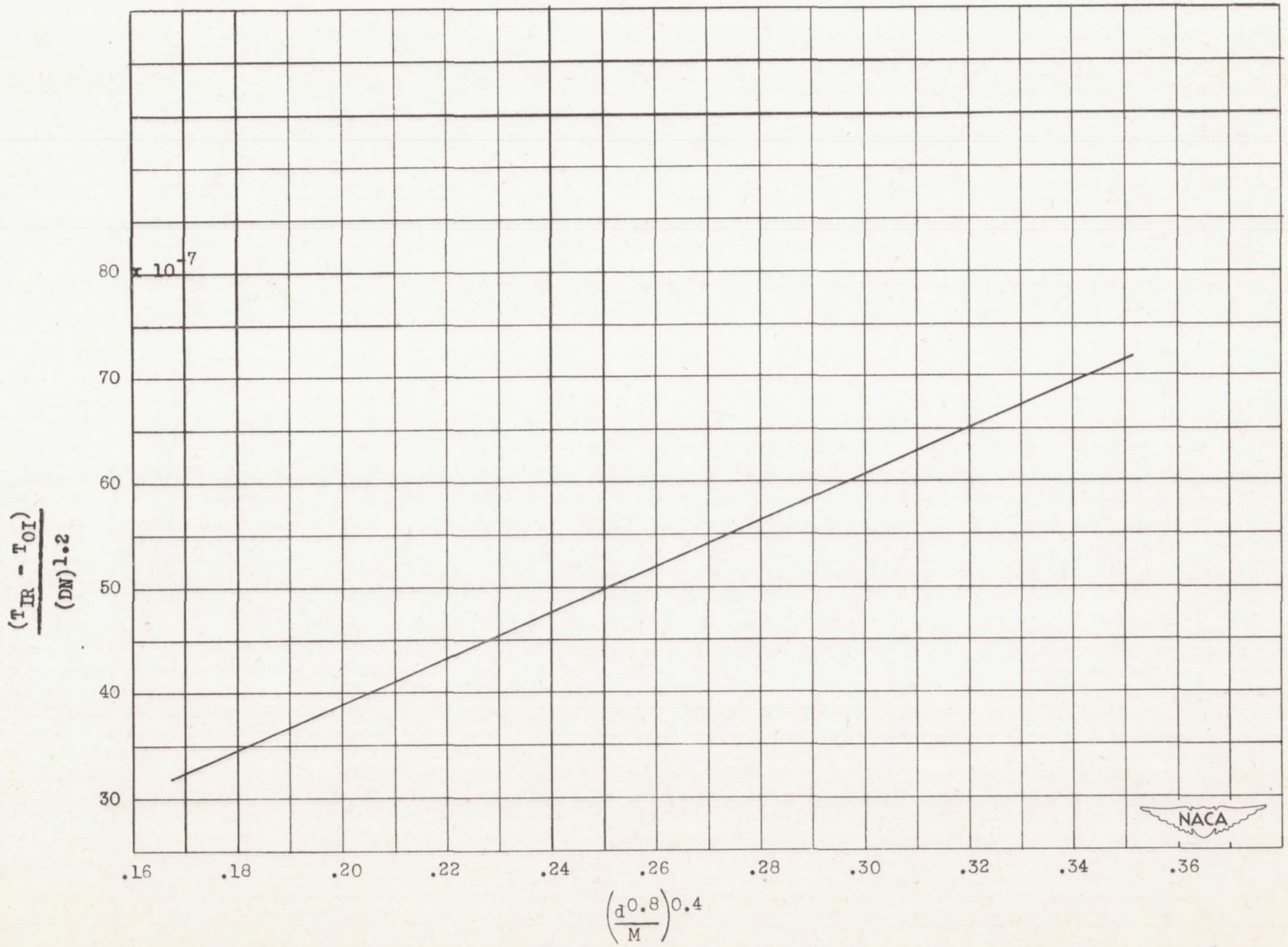


Figure 14. - Final cooling correlation curve for inner-race temperature of bearing 6. DN values, 0.3×10^6 to 1.2×10^6; oil flows, 0.6 to 12.9 pounds per minute; oil-jet diameters, 0.023 to 0.129 inch; load, 368 pounds; oil inlet temperature, 100° F.

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