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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 3003

INVESTIGATION OF 75-MILLIMETER-BORE DEEP-GROOVE BALL BEARINGS UNDER RADIAL LOAD AT HIGH SPEEDS

II - OIL INLET TEMPERATURE, VISCOSITY, AND GENERALIZED COOLING CORRELATION

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SUMMARY

Two 75-millimeter-bore (size 215) inner-race-riding cage-type ball bearings were used in an experimental investigation of the effects of oil inlet temperature and viscosity on bearing operating characteristics over a range of DN values (bearing bore in mm times shaft speed in rpm) from 0.3×10^{6} to 2.4×10^{6}, static radial loads from 7 to 1113 pounds, oil flows from 1.6 to 8 pounds per minute, and oil inlet temperatures of 100° and 205° F. Absolute viscosity at the inlet temperatures varied from 2.18×10^{-7} to 42.6×10^{-7} reyns (kinematic viscosities of 1.77 to 34.5 centistokes).

A previously developed cooling-correlation analysis for cylindrical-roller-bearing temperatures was found to be applicable to ball bearings. The effect of load, although small, was included. A similar cooling correlation was developed for the power rejected to the oil. This correlation makes it possible to predict either the inner- or outer-race bearing temperature or the power rejected to the oil from single curves regardless of whether speed, load, oil flow, oil inlet temperature, oil inlet viscosity, or any combination of these parameters is varied.

An increase in oil viscosity produced increased bearing temperatures and caused the power rejected to the oil to be greater with constant DN, load, oil flow, and oil inlet temperature.

With an increase in oil inlet temperature (at constant DN, load, oil flow, and oil inlet viscosity), the bearing temperatures increased nearly 1° F for each 1° F increase in oil inlet temperature and the power rejected to the oil remained unchanged. For a specific oil, the bearing temperature rise was from 0.6° to 1° F for each 1° F increase in oil inlet temperature because of decreasing oil viscosity with

increasing oil inlet temperature. Power rejected to the oil decreased with increasing oil inlet temperature for the same reason.

In a speed run, the maximum DN reached before bearing failure was 2.4×10^6 (32,000 rpm). The bearing was lubricated with a medium viscosity oil at an oil inlet temperature of 100° F. Failure was caused by cage breakage.

INTRODUCTION 2730

The influence of oil viscosity and inlet temperature on the effectiveness of cooling and lubricating high-speed rolling-contact bearings is of particular significance in turbojet and turbine-propeller engine design, because the high speeds, loads, and ambient temperatures encountered produce high bearing operating temperatures. In high-speed bearings, a large portion of the bearing heat is removed by the lubricant, and, in turn, the heat absorbed by the lubricant must be removed to maintain the desired oil inlet temperature. The amount of heat to be removed from the lubricant, which is determined by the degree of cooling of the bearing desired, is limited by the capacity of the available coolant. In order to minimize the heat to be removed from the lubricant, heat generation in the bearing should be held to a minimum.

With single, small-diameter jet lubrication, a portion of the oil is deflected from the bearing and a portion of the oil is transmitted through the bearing. The transmitted oil serves as both coolant and lubricant (ref. 1) and may also be a source of heat due to churning. Greater cooling is obtained with increased transmitted-oil flow but only at the expense of greater power loss due to churning.

Very little information is available in the literature on the effect of lubricant viscosity and oil inlet temperature on high-speed ball-bearing performance characteristics. Some information on the effect of oil viscosity and oil flow on operating temperature, friction torque, and power dissipated at relatively low speeds is contained in references 2 to 5. The only reported results for high speeds deal with cylindrical-roller bearings (ref. 6).

The investigation reported herein is a continuation of the work reported in reference 1 and was conducted at the NACA Lewis laboratory. Reference 1 shows that the amount of oil which flows through the bearing is an important variable regarding the bearing operating temperature and the lubrication-system heat load; also, the effects of different operating variables and of two methods of lubrication on the oil flow through the bearing and on bearing outer-race and inner-race temperatures are described. These studies were made with a single oil (oil B of this report) at an oil inlet temperature of 100° F. The operating variables included DN, total oil flow, load, and oil-jet radial position, while the two methods of lubrication were single jet and puddling.

The objectives of this investigation were: (1) To study the effects of oil inlet temperature and viscosity on the operating characteristics of conventional ball bearings at high speeds under radial load. The effects of oil viscosity, oil inlet temperature, DN, load, and oil flow on the bearing operating temperature and the power rejected to the oil are reported. (2) To provide a means for estimating the bearing temperatures and the horsepower rejected to the oil for any combination of the operating variables by use of a correlation of results similar to the one presented in reference 7 for roller bearings.

Two ABEC-5 deep-groove ball bearings (size 215) were investigated over the following ranges of controlled variables: DN, 0.3×10^6 to 2.4×10^6 (corresponding to 4,000 to 32,000 rpm); loads, 7 to 1113 pounds; oil flows, 1.6 to 8 pounds per minute; oil inlet temperatures, 100° and 205° F. The absolute viscosities of the two oils used in this investigation at inlet temperatures of 100° and 205° F varied from 2.18×10^{-7} to 42.6×10^{-7} reyns (kinematic viscosities from 1.77 to 34.5 centistokes).

The operating conditions imposed on a cylindrical-roller bearing of the same size (215) in a present-day turbojet engine are as follows: DN, 0.3×10^6 to 0.86×10^6; approximate gravity load, 375 pounds, and oil flow, 0.8 to 2 pounds per minute through a jet of 0.052-inch diameter.

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APPARATUS

Bearing rig. - The bearing rig (fig. 1) used in this investigation is the same as that used in the investigations reported in references 1, 6, 7, and 8. 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 test bearing by means of a lever and dead-weight system in such a manner that the alinement of the outer race of the test bearing was essentially unaffected by small shaft deflections or by small shaft and load-arm misalinements.

The support bearings were lubricated in the manner described in reference 9. The oil was supplied to the support bearings at a pressure of 10 pounds per square inch through a 0.180-inch-diameter jet and at a temperature equal to that of the oil supplied to the test bearing (either 100° or 205° F).

The drive equipment is described in reference 9. The available speed range of the test shaft is 800 to 50,000 rpm.

Test bearing. - Two test bearings were used for this investigation, although most of the data reported herein were obtained by use of only one of the bearings. These bearings were conventional aircraft-grade ball bearings (ABEC-5). The bearing dimensions were the following: bore, 75 millimeters; outside diameter, 130 millimeters; and width, 25 millimeters. The bearings were equipped with a two-piece riveted retainer (cage) of laminated cloth-base phenolic material and eleven 11/16-inch-diameter balls. The retainer was guided by the inner race.

The bearings investigated have been numbered consecutively; bearings 19 and 20 of reference 1 are the same as bearings 19 and 20, respectively, of this report. 2730

Temperature measurement. - The method of temperature measurement is described in reference 9. 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 transmitted from the rotating shaft by means of small slip rings located on the end of the test shaft (ref. 10).

Lubrication system. - The general make-up of the lubrication system is described in reference 9, and subsequent alterations of the lubrication system are described in reference 1.

PROCEDURE

Lubrication of test bearing. - Lubricant was supplied to the test bearing through a single jet having a 0.050-inch-diameter orifice and an orifice length-diameter ratio of 1. The oil was directed normal to the bearing face at the space between the cage and the inner race directly opposite the load zone.

Two oils, designated herein as oils A and B, were used to lubricate the test bearing. Oil A was used in reference 6; oil B was used in references 1, 6, and 7. The properties of the two oils are given in figure 2. Oil A was a highly refined, nonpolymer, petroleum-base lubricating oil. Oil B was a commercially prepared blend of a highly refined paraffin base with a small percentage of polymer added to improve the viscosity index.

An oil sample was taken at the start of each oil run, at the start of each day's run, and at the conclusion of tests. Viscosities were obtained by standard laboratory procedures, and the data plotted in figure 2 represent the average of these samples.

The variation in viscosity for each of the two oils as found in the laboratory tests was as follows:

Oil	Kinematic viscosity at 100° F, centistokes			Maximum variation from mean, percent
	Minimum	Mean	Maximum	
A	5.11	5.54	5.80	8
B	29.96	34.5	40.20	18

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The oils were supplied to the test bearing at temperatures of 100° and 205° F and pressures from 20 to 400 pounds per square inch, which correspond to oil flows of 1.6 to 8 pounds per minute.

When tests of one oil were completed, the oil system was pumped and drained dry; it was then thoroughly flushed and drained twice with clean petroleum solvent. A quantity of fresh test oil was then circulated through the system and allowed to drain completely. The test oil was then introduced into the system.

Test-bearing measurements. - The test-bearing measurements were obtained in the manner described in reference 9 and are reported in table I of reference 1.

Surface finishes of the bearing component parts (obtained using a profilometer) are given in table I, and the hardnesses of the component parts before and after running are given in table II.

RESULTS AND DISCUSSION

The results of the experimental investigation are presented in figures 3 to 14. Bearing temperature was chosen as the principal criterion of operation, because it gives a qualitative indication of the severity of operating conditions and of the effectiveness of lubrication.

Effect of Independent Variables on Ratio of

Deflected-Oil Flow to Transmitted-Oil Flow

In reference 1, it is shown that the ratio of deflected-oil flow to transmitted-oil flow is an important variable regarding ball-bearing lubrication and that the outer-race temperature, the power rejected to the oil, and (to a lesser extent) the inner-race temperature are determined by the portion of the total flow which is transmitted through the bearing. Oil B at an oil inlet temperature of 100° F was used in reference 1. Data are presented herein for oil B at an oil inlet temperature of 205° F and for oil A at inlet temperatures of 100° and 205° F. These data are generally similar to those presented in reference 1.

The effect of DN on the ratio of deflected flow to transmitted flow is shown in figure 3 for bearing 19 (curve from ref. 1 included for comparison). The ratio of deflected flow to transmitted flow increases with increasing DN at a load of 368 pounds with both oils. The increase of the flow ratio with DN is more marked at an inlet temperature of 100° F than at 205° F.

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The effect of oil flow on the ratio of deflected flow to transmitted flow is shown in figure 4 for bearing 19. In general, the flow ratio decreases with increasing oil flow for oils A and B at the two oil inlet temperatures, 100° and 205° F.

The effect of load on the ratio of deflected to transmitted flow is shown in figure 5. In general, the flow ratio is independent of load. The data for the load curves were obtained in supplementary tests run after all other data had been obtained. This fact accounts for the difference between the viscosities shown in figures 5 and 8 and the viscosities shown in the other figures. Viscosities shown in figures 5 and 8 were not used to determine the averages given in the table in the section entitled "Lubrication of test bearing." In reference 1, an increase in the flow ratio with increasing load is reported; the bearing was lubricated with oil B at an oil inlet temperature of 100° F. This increase in the flow ratio with increase in load was the result of shaft movement which changed the impingement point of the oil jet on the bearings; this condition may be peculiar to the test rig and not representative of engine operation. Therefore, the data shown in figure 5 were obtained with support bearings having small radial clearances in order to reduce shaft movement under changing loads. The possibility exists that better positioning of the jet relative to the bearing was obtained.

There are no really significant trends in the flow ratios over the range of viscosities and oil inlet temperatures investigated. The more viscous oil sometimes results in lower flow ratios, perhaps because of a greater "clinging" ability. Similar results are reported for cylindrical-roller bearings in reference 6. In reference 6, it is also shown that, for roller bearings, the ratio of deflected flow to transmitted flow is a function mainly of oil jet velocity and rotational velocity of the bearing. To a lesser extent, when the jet is directed at the cage-locating surface, the flow ratio is also a function of the clearance between the cage-locating surface and the inner race at the point of impingement. Similarly, for ball bearings, the cage clearance can be expected to be a complex function of DN and continuously varying cage loads so that its effect on the ratio of flows varies with time.

Effect of Independent Variables on Bearing Operating Temperatures

Effect of DN. - The effect of DN on the outer-race-maximum and inner-race temperatures of bearing 19 is shown in figure 6. The results are qualitatively the same as those for oil B at an oil inlet temperature of 100° F reported in reference 1 (curves from ref. 1 included for comparison). The bearing temperature at a given DN is higher for an oil of higher viscosity; this result is in agreement with the results reported in reference 6 for roller bearings. For the two oils, the
2730 difference in bearing temperatures at a given DN was greater at an oil inlet temperature of 100° than at 205° F; this is accounted for by the difference in the viscosities of the oils at these temperatures. The increases in bearing-temperature difference with increasing DN may be caused by increased churning at higher DN values. Had the transmitted-oil flows been equal for the two oils at the lower temperature, this temperature spread would have been even greater (ref. 1).

Effect of oil flow. - The effect of oil flow on the outer-race-maximum and inner-race temperatures of bearing 19 is shown in figure 7. These results again are qualitatively similar to those for oil B at an inlet temperature of 100° F reported in reference 1. As in figure 6 for a given oil flow, bearing temperature was higher for oil B than for oil A because of the higher viscosity.

Effect of load. - The effect of load on the outer-race-maximum and inner-race temperatures of bearing 19 is shown in figure 8 for two oils. Both outer-race-maximum and inner-race temperatures increase slightly with load at both oil inlet temperatures. As explained in the discussion of figure 5, the data reported in figure 8 were obtained in supplementary tests. In reference 1, it is reported that bearing temperatures increased appreciably with load at high DN values when bearing 19 was lubricated with oil B at an inlet temperature of 100° F. The cause of this result and the reason for obtaining new data are explained in the discussion of figure 5.

For a given value of load, the bearing temperatures were again higher for the more viscous oil, and the difference in bearing temperatures was greater at the lower oil inlet temperature.

Bearing failure. - A limiting speed run was made to determine the maximum DN value at which a 75-millimeter-bore (size 215) ball bearing would operate without failure under the following conditions of operation: load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F, and oil-jet diameter, 0.050 inch. Bearing number 20, lubricated with oil B, was used for this test. The test was stopped as soon as equilibrium bearing-temperature operation ceased. Bearing failure occurred at a DN of 2.4×10^{6}, $3\frac{1}{2}$ minutes after equilibrium temperature had been established (see fig. 9); equilibrium bearing temperatures were established in 6 minutes. At failure, the outer-race-maximum and inner-race temperatures increased rapidly from about 340° and 336° F to above 500° and 400° F, respectively. The rig was shut down and the test bearing inspected. The phenolic cage was broken and had been riding on the outer race (fig. 10). In figure 10(c) may be seen the extent to which the cage had been worn away by rubbing on the outer race of the bearing in the time interval between the initial cage fracture and the shut-down of the rig. The ends of the rivets holding the two halves of the cage together also show wear. 2730

The test rig was shut down before the bearing races and balls were damaged; thus, the cause of failure was not obscured. Damage to the races and balls was probably prevented by the fact that, at the instant of fracture, the cage fragments began sliding on the outer race, causing the outer race to expand more than the inner race and the bearing clearance to increase. The races and balls showed no evidence of wear after running for 13 hours at DN values from 1.2×10^6 to 2.4×10^6.

Causes of bearing failure. - Bearing failure, as evidenced by cage failure, may be attributed to several factors. The contribution of each factor cannot be determined, but two of these possible causes are worthy of mention. First, the embrittlement and loss of strength of the phenolic cage material because of repeated cooling and heating to temperatures above 300° F may have resulted in the weakening of the cage material and so may have hastened the final cage breakage, which was caused by the high inertia stresses at the extreme speeds encountered. Second, bearing preloading may have occurred as a result of differential thermal expansion (see fig. 9(b)) or a permanent change in bearing dimensions caused by the very severe operating conditions.

Changes in Test Bearings with Running Time

Comparison of the test-bearing measurements taken before and after running is questionable, since the data were not obtained from the same bearing. Inasmuch as nondestructive disassembly and measurement of the bearings investigated is impossible, unused sample bearings were disassembled to obtain the necessary initial measurements. However, since high-speed aircraft-grade bearings are manufactured to very close tolerances, it is possible to make the following observations from the data obtained:

Changes in bearing dimensions. - The bearing dimensions are reported in table I of reference 1. No significant dimensional changes in bearing component parts was evident for either bearing except that the phenolic cage of bearing 20, which was run to a speed failure, broke in several places and wore quite appreciably on its outer diameter after coming into contact with the outer race. Also, the diametral clearance of bearing 20 decreased from 0.0006 to 0 inch because of a permanent change in bearing dimensions as a result of the high temperatures to which the bearing was subjected as the consequence of the failure.

<u>Changes in surface finish</u>. - Surface-finish values of the component parts, obtained from a disassembled sample bearing as well as from disassembled test bearings 19 and 20 after running, are given in table I.

The values of surface finish of new bearings are consistently lower in the circumferential direction than in the axial direction, which is normal to the direction of cut; however, after long running periods the difference in surface finish between circumferential and axial directions is not generally so great. In general, the surface finishes of both bearings after running were smoother than before.

<u>Changes in hardness</u>. - The hardness values of the component parts, obtained from a disassembled sample bearing as well as from disassembled test bearings 19 and 20 after running, are given in table II. The balls are the hardest parts of the new bearing; however, running tempered the balls, and they lost some of their hardness. There was very little difference in hardness of the raceways before and after operation even though the cage of bearing 20 failed.

ANALYSIS OF EXPERIMENTAL RESULTS

Cooling-Correlation Theory

The dimensional method of reasoning (ref. 11) may be used to obtain a correlation of the results obtained in this investigation. Such a correlation is presented in reference 7 for temperatures of cylindrical-roller bearings. The first step in applying this technique to the solution of a problem is to examine the physical situation and to determine the variables involved. The bearing may be recognized as a system involving heat-transfer and fluid-dynamic considerations; hence, the significant variables are those presented in the list that follows. The dimensional formulas of the variables are given in the mass M, length L, time θ, and temperature T system of dimensions.

Symbol	Variable	Dimensional formula
N	Bearing speed	θ^{-1}
D	Bearing bore	L
μ	Viscosity based on oil inlet temperature	$ML^{-1}\theta^{-1}$
C	Specific heat of oil at oil inlet temperature	$L^{2}\theta^{-2}T^{-1}$
ΔT	Temperature rise of bearing above oil inlet temperature	T
d	Oil-jet diameter	L
w	Mass rate of oil flow	$M\theta^{-1}$
W	Bearing load	$ML\theta^{-2}$
k	Thermal conductivity of oil at oil inlet temperature	$ML\theta^{-3}T^{-1}$
ρ	Mass density of oil at oil inlet temperature	ML^{-3}

2730

The dimensionally independent variables, the maximum number of variables that cannot be combined to form a dimensionless group, were chosen as speed, bearing bore, viscosity and specific heat. The dimensionless groups chosen are given in the following equation:

$$\frac{\Delta TC}{(DN)^2} = \Omega\left[\frac{\rho D(DN)}{\mu}, \frac{k}{\mu C}, \frac{d}{D}, \frac{w}{\mu D}, \frac{W}{\mu D(DN)}\right] \tag{1}$$

where $\frac{\Delta TC}{(DN)^2}$ is some function Ω of the several dimensionless groups.

Equation (1) contains a complete set of independent dimensionless groups. The groups are complete inasmuch as the equation contains the number of groups (six) required (ref. 11, p. 30), and each group is independent, because it contains a variable not present in any other group (hence no one particular group can be expressed as a function of any combination of the other groups). As an approximation, the function

Ω in equation (1) is assumed to be the product of the independent dimensionless groups each raised to an empirically determined power. When written as a power function, equation (1) takes the following form:

$$\frac{\Delta TC}{(DN)^2} = A\left[\frac{\rho D(DN)}{\mu}\right]^q \left[\frac{k}{\mu C}\right]^r \left[\frac{d}{D}\right]^s \left[\frac{w}{\mu D}\right]^t \left[\frac{W}{\mu D(DN)}\right]^u \tag{2}$$

where A, q, r, s, t, and u are constants. The term containing W has been retained in equation (2) for ball bearings (it was deleted for roller bearings in ref. 7) although experimental results showed only a small effect of W on ΔT in the load range used in this investigation (approximately 7 to 1113 lb).

For the oils used and for the range of oil inlet temperatures investigated herein, the effects of specific heat, density, and thermal conductivity of the oil are small with respect to the effect of change in viscosity. In addition, for a given bearing, the quantity D is fixed. Since jet size was not varied, d is also fixed. Equation (2) may thus be simplified to the following expression:

$$\Delta T = B\left[(DN)^a W^x w^y \mu^z\right] \tag{3}$$

The following inner-race equation best fits the data of this report; the equation was obtained by use of equation (3) and the method of determining the exponents given in reference 7:

$$T_{IR} - T_{OI} = B_1\left[\frac{(DN)^{1.75} W^{0.10} \mu^{0.35}}{w^{0.42}}\right] \tag{4}$$

where

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

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

B_1 constant

W bearing load, lb

μ oil viscosity at oil inlet temperature, lb-sec/sq in., reyns

w oil flow, lb/min

similarly, the outer-race equation is

$$T_{OR} - T_{OI} = B_2\left[\frac{(DN)^{1.50}W^{0.07}\mu^{0.25}}{w^{0.42}}\right] \tag{5}$$

where

T_{OR} outer-race temperature, $^{\circ}F$ 2730

B_2 constant

The horsepower rejected to the oil can also be correlated with the independent variables of bearing operation; the resulting relation is similar to that obtained for both the inner-race and outer-race temperature rise above oil inlet temperature (eq. (3)). The equation developed for horsepower rejected to the oil is

$$q = B_3\left[(DN)^{1.50}W^{0.07}\mu^{0.25}w^{0.42}\right] \tag{6}$$

where

B_3 constant

q rate of heat rejection, horsepower

Combining equations (5) and (6) yields the following equation for horsepower rejected to oil:

$$q = B_4\left[w^{0.84}\,(T_{OR} - T_{OI})\right] \tag{7}$$

where

B_4 constant

Under conditions investigated for bearing 19, the power rejected to the oil is a function only of the mass flow of oil and bearing outer-race-maximum temperature rise above oil inlet temperature. The power rejected to the oil increases with increasing oil flow and increasing outer-race temperature and decreases with increasing oil inlet temperature.

Cooling-correlation application. - The final cooling correlation curves for the inner- and outer-race temperatures and the power rejected to the oil are given in figures 11 to 14 for the following range of variables: DN, 0.3×10^6 to 1.2×10^6; load, 113 to 1113 pounds; oil viscosity at inlet temperatures, 2.18×10^{-7} to 42.6×10^{-7} reyns (kinematic viscosity of 1.77 to 34.5 centistokes); oil flow, 1.6 to 8 pounds per minute; and oil inlet temperature, 100° to 205° F.

The plots represent a useful method of obtaining a first approximation of the bearing inner- and outer-race temperatures and the power rejected to the oil, and of determining the effects of DN, load, oil viscosity, oil inlet temperature, and oil flow on these variables.

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Comparison of Cooling-Correlation Results

The constants of the correlation curves of equation (3) for the inner- and outer-race temperatures and equations (6) and (7) for the power rejected to the oil are as listed in the following table:

Constant	Inner race	Outer race	Power rejected to oil	
	$\Delta T = B\left[(DN)^a W^x w^y \mu^z\right]$		$q = B\left[(DN)^a W^x w^y \mu^z\right]$	$q = B w^y \Delta T$
B	1.333×10^{-7}	16.36×10^{-7}	10.53×10^{-9}	6.15×10^{-3}
a	1.75	1.50	1.50	---------
x	.10	.07	.07	---------
y	-.42	-.42	.42	.84
z	.35	.25	.25	---------

For the outer-race and power-rejection correlations (equations (5) and (6)), all constants are identical except for the slope B and the difference in sign of y. For outer-race temperature, y is negative, indicating an inverse relation between oil flow and temperature. For the power rejected, the sign is positive, indicating a direct relation between oil flow and the power rejected.

The correlation curves presented herein cannot be used directly for jet engine or other design applications, but the curves may be used qualitatively. The exponents as determined in the laboratory test rig for roller bearings remained the same when applied to an engine bearing (ref. 12), and a similar phenomenon should be expected to occur for ball bearings. The difference in heat flow in test rig and in engine operation, because of the external heat associated with engine operation, would change the constant B and add an intercept to the curves. The intended value of the present work lies in the possibility that the method of correlation may be applied to jet-engine bearings. Application of the method is demonstrated in detail in reference 8.

Additional Observations

From the experimental data, its analysis, and interpretation of the cooling correlation figures 11, 12, 13, and 14, the following additional observations may be added:

Effect of viscosity on bearing operating temperature. - Lubrication with an oil of higher viscosity has the effect of increasing the operating temperature of a deep-groove ball bearing of the type investigated at any DN, load, oil inlet temperature, and oil flow (figs. 11 and 12). This observation is shown to hold for cylindrical-roller bearings in reference 6 where five oils of different viscosities were used. The two basic causes for higher operating temperatures resulting from the use of high viscosity oils, as explained in reference 6 for cylindrical-roller bearings, apply equally well to deep-groove ball bearings; these are: (1) a decrease in the film coefficient of heat transfer with increasing oil viscosity, which reduces the transfer of heat from the bearing surfaces to the oil, and (2) an increase in heat developed in shearing an oil of high viscosity.

Effect of viscosity on power rejected to oil. - The power rejected to the lubricating oil increased with increasing oil viscosity (fig. 13). As the power rejected to the oil varies directly with increasing oil temperature, it is obvious that the oil outlet temperatures should also increase with increasing viscosity.

Effect of oil inlet temperature on bearing operating temperature. - Bearing operating temperatures are influenced to a large extent by the oil inlet temperature. Figure 12 shows the difference between outer-race bearing temperature and oil inlet temperature plotted against other variables. A change in oil inlet temperature produces an equal change in the bearing temperature over the range of oil inlet temperatures investigated, the other variables including oil inlet viscosity being held constant. In a practical application, oil viscosity would increase with a decrease in oil temperature, resulting in higher churning losses and less heat transfer to the oil. Therefore, the decrease in the race temperatures would be less than the decrease in oil inlet temperature. The decrease in both inner- and outer-race bearing temperatures is between 60 and 100 percent of the decrease in oil inlet temperature for the conditions investigated.

Effect of oil inlet temperature on power rejected to oil. - The power rejected to the oil would be constant at various oil inlet temperatures for constant operating conditions if oil inlet viscosity were held constant over the temperature range by use of various grades of oil (fig. 13). In a practical application, oil viscosity would decrease with increasing oil inlet temperature and the power rejected to the oil would also decrease in accordance with the change in viscosity (fig. 13).

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SUMMARY OF RESULTS

The following results were obtained in an experimental investigation of the effects of oil inlet temperature and viscosity on the operating characteristics of two 75-millimeter-bore (size 215) inner-race-riding cage-type ball bearings, which were operated over a range of DN values (bearing bore in mm times shaft speed in rpm) from 0.3×10^6 to 2.4×10^6, oil flows from 1.6 to 8 pounds per minute, and loads from 7 to 1113 pounds:

1. A previously developed cooling-correlation analysis for cylindrical-roller-bearing temperatures was found to be applicable to ball bearings. The effect of load, although small, has been included. A similar cooling correlation was developed for the power rejected to the oil. For a bearing temperature rise above oil inlet temperature, the form of the equation used was

$$\Delta T = B\left[(DN)^a W^x w^y \mu^z\right]$$

For power rejected to the oil, the form of the equation was

$$q = B\left[(DN)^a W^x w^y \mu^z\right]$$

or

$$q = B(W^y \Delta T)$$

where ΔT is the difference between outer-race bearing temperature and the oil inlet temperature; B, a constant; DN, the product of the bearing bore in millimeters and shaft speed in rpm; q, the power rejected to the oil; W, the bearing load in pounds; w, the oil flow in pounds per minute; and μ, the oil inlet viscosity in pound-seconds per square inch.

2. For the inner- and outer-race temperatures and the power rejected to the oil, the constants found to give the best degree of correlation were as follows:

Constant	Inner race	Outer race	Power rejected to oil	
	$\Delta T = B\left[(DN)^a W^x w^y \mu^z\right]$		$q = B\left[(DN)^a W^x w^y \mu^z\right]$	$q = Bw^y \Delta T$
B	1.333×10^{-7}	16.36×10^{-7}	10.53×10^{-9}	6.15×10^{-3}
a	1.75	1.50	1.50	---------
x	.10	.07	.07	---------
y	-.42	-.42	.42	.84
z	.35	.25	.25	---------

3. Bearing inner- and outer-race temperature varied inversely to the 0.42 power, and power rejected to the oil varied directly to the 0.42 power with the mass flow of oil.

4. A more viscous oil produced higher bearing temperatures and caused the power rejected to the oil to be greater at a given operating condition (DN, load, oil flow, and oil inlet temperature constant).

5. Bearing temperatures rose very nearly 1° F for each increase of 1° F in oil inlet temperatures over the range of oil inlet temperatures investigated at a constant DN, load, oil flow, and inlet viscosity. For a specific oil, the bearing temperature rise was from 0.6° to 1° F for each increase of 1° F in oil inlet temperature, because of the change in viscosity with change in oil temperature, the exact increase depending on the particular operating condition.

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6. The power rejected to the oil remained constant over the range of oil inlet temperatures investigated at a constant DN, load, oil flow, and oil inlet viscosity. For a specific oil, the power rejected to the oil decreased with increasing oil inlet temperatures because of decreasing oil inlet viscosity.

7. In a speed run, one of the bearings ran to a DN of 2.4×10^{6} (32,000 rpm) at a load of 368 pounds, an oil flow of 2.75 pounds per minute, and an oil inlet temperature of 100° F before failure. Failure occurred $3\frac{1}{2}$ minutes after bearing equilibrium temperature had been established. The time to establish equilibrium temperature was 6 minutes. Failure was caused by the breakage of the phenolic cage. The races and balls showed no evidence of wear after running for 13 hours at DN values from 1.2×10^{6} to 2.4×10^{6}.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, July 17, 1953

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TABLE I. - SURFACE FINISH[a] OF TEST BEARING COMPONENT PARTS

Bearing			19		20	
Construction			Two-piece inner-race-riding cage		Two-piece inner-race-riding cage	
			Before (b)	After	Before (b)	After
Total running time, hr			0	59.6	0	78.4
Severity factor[c]			0	200,688	0	302,112
Surface finish of outer-race track		Axial	-	---	-	---
		Circumferential	4-5	4-5	4-5	4-5
Surface finish of inner race	Track	Axial	---	---	---	---
		Circumferential	1.5-2	2-3	1.5-2	2-4
	Lands	Axial	15-17	5-8	15-17	6-9
		Circumferential	7-8	5-7	7-8	6-8
Surface finish of balls			1	1.5-2	1	1.5-2.5

[a]Surface finish measured in microin. rms by means of a profilometer.

[b]Measurements obtained from sample bearing.

[c]Summation of products of difference between outer-race-maximum temperature and lubricant inlet temperature for each operating condition and corresponding operating time in min at that particular condition.

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TABLE II. - HARDNESS OF TEST BEARING COMPONENT PARTS

Bearing		19		20	
Construction		Two-piece inner-race-riding cage		Two-piece inner-race-riding cage	
		Before	After	Before	After
Total running time, hr		0	59.6	0	78.4
Severity factor[a]		0	200,688	0	302,112
Hardness, Rockwell-C	Outer race	[b]63-64	60-63	[b]63-64	60.63
	Inner race	[b]61-63	62-63	[b]61-63	60.5-62
	Balls	[b,c]69	[c]66.5	[b,c]69	[c]60-63

[a]Summation of products of difference between outer-race-maximum temperature and lubricant inlet temperature for each operating condition and corresponding operating time in min at that particular condition.

[b]Measurement obtained from sample bearing.

[c]Measurement obtained on ground flat 0.005 in. from surface of ball.

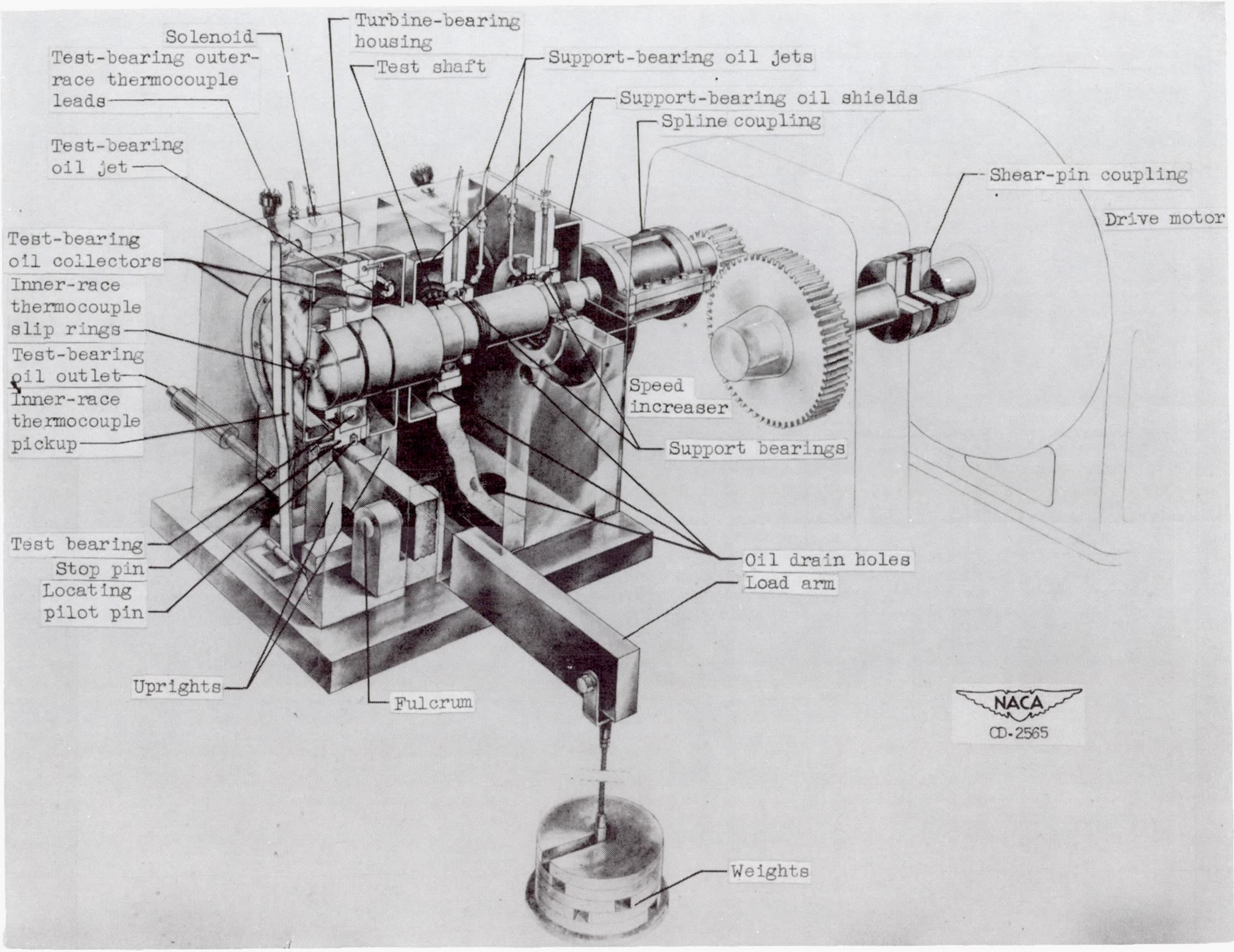

Figure 1. - Cutaway view of radial-load rig.

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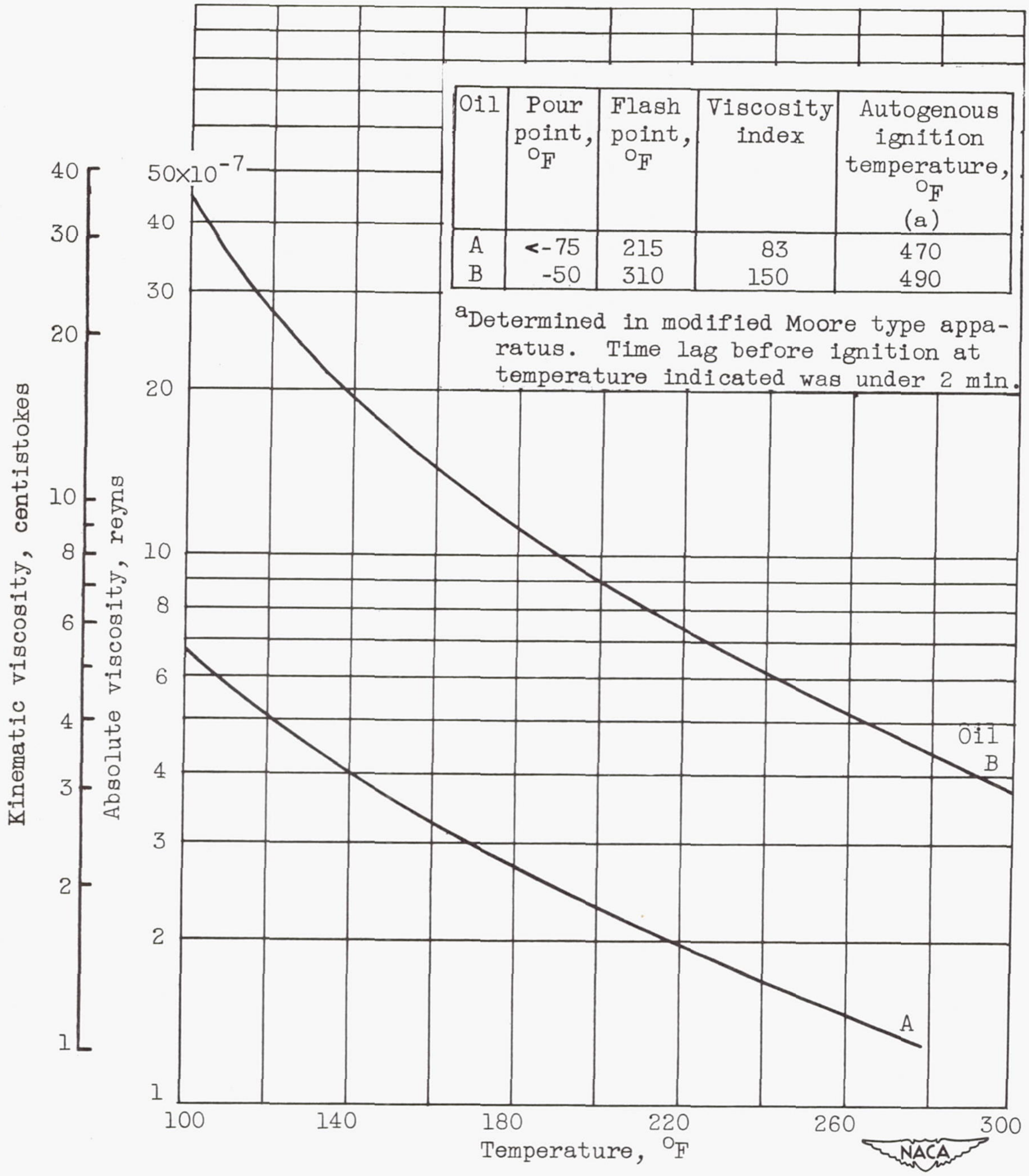
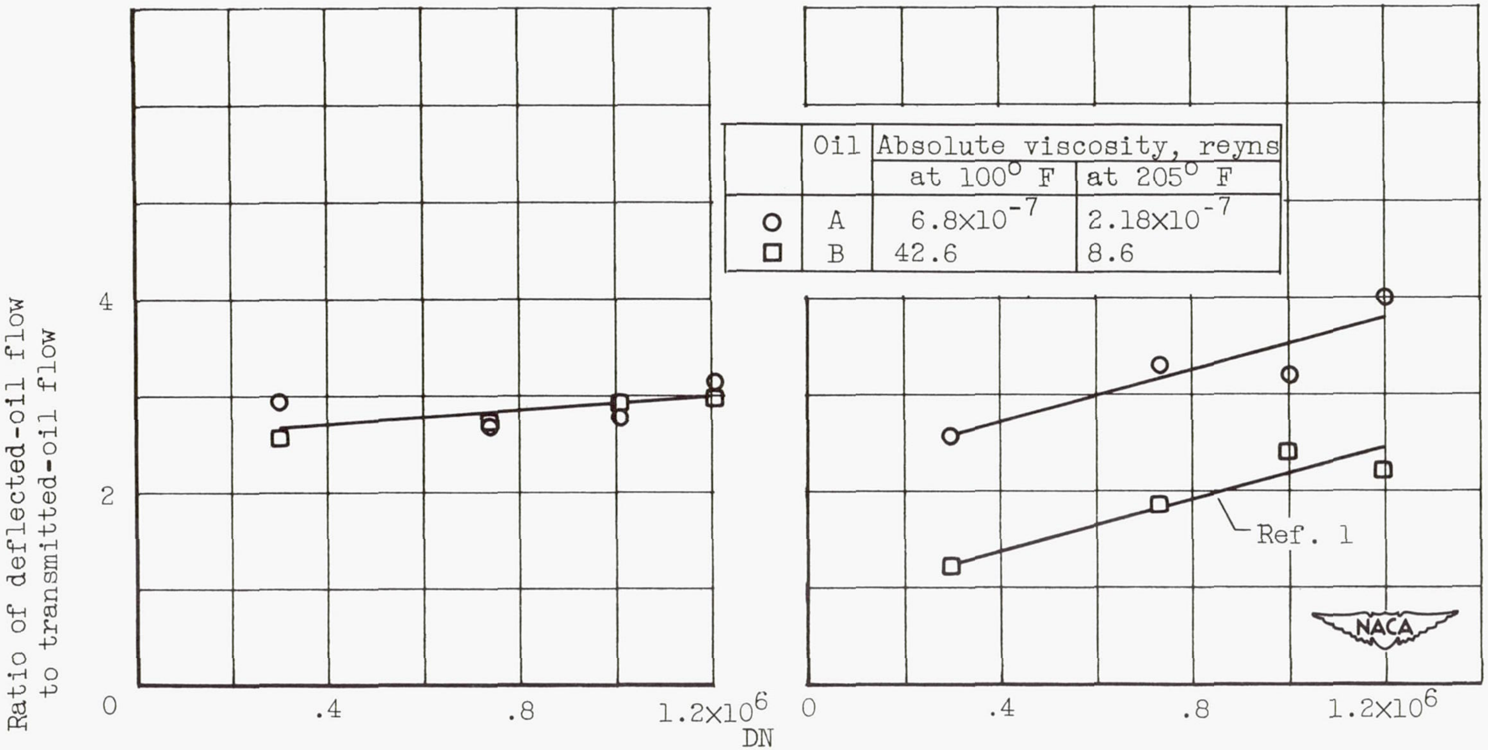


Oil	Pour point, °F	Flash point, °F	Viscosity index	Autogenous ignition temperature, °F (a)
A	<-75	215	83	470
B	-50	310	150	490

[a]Determined in modified Moore type apparatus. Time lag before ignition at temperature indicated was under 2 min.

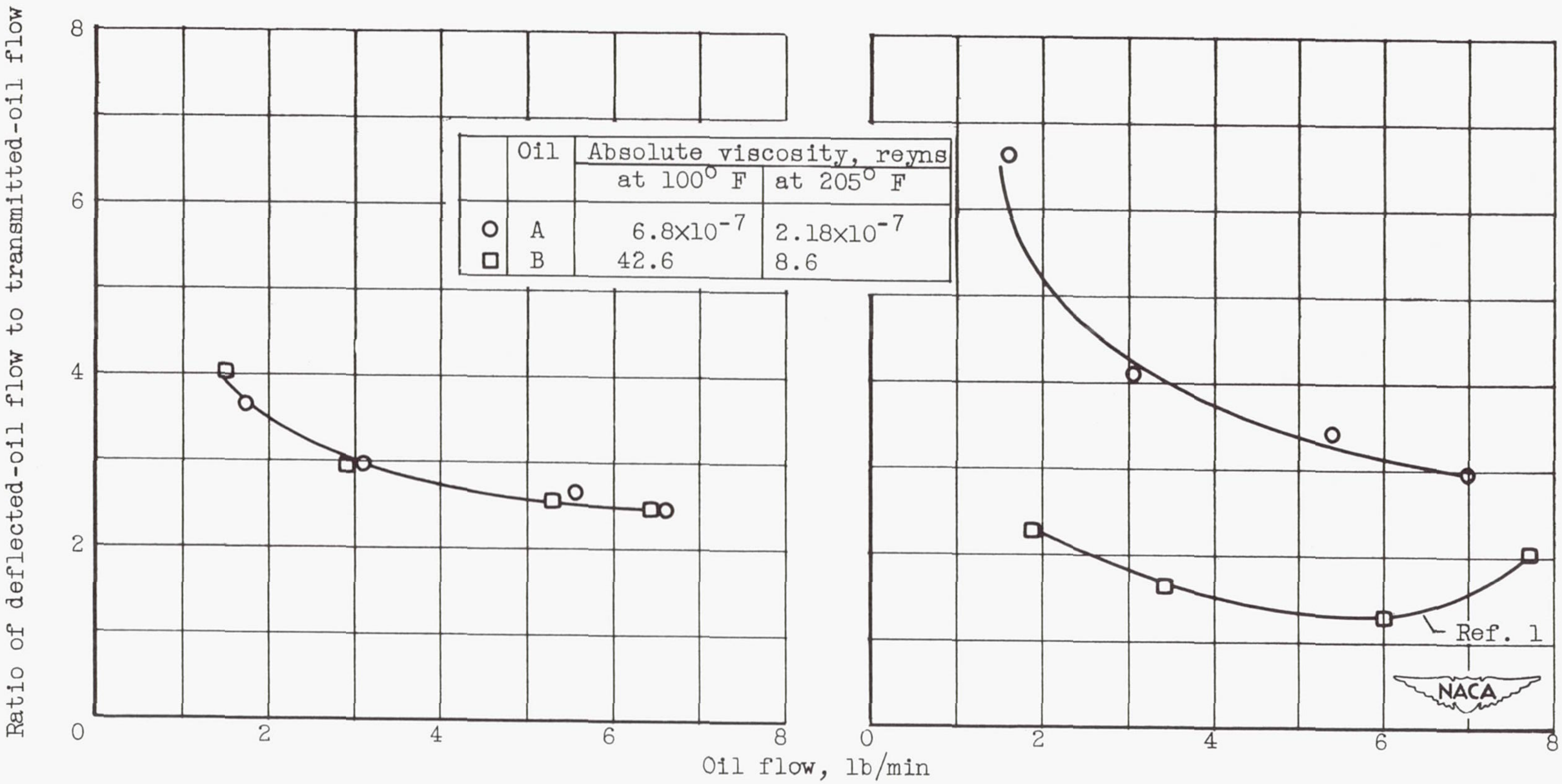
Figure 2. - Effect of temperature on kinematic and absolute viscosities of two test oils.



(a) Oil inlet temperature, 205° F.

(b) Oil inlet temperature, 100° F.

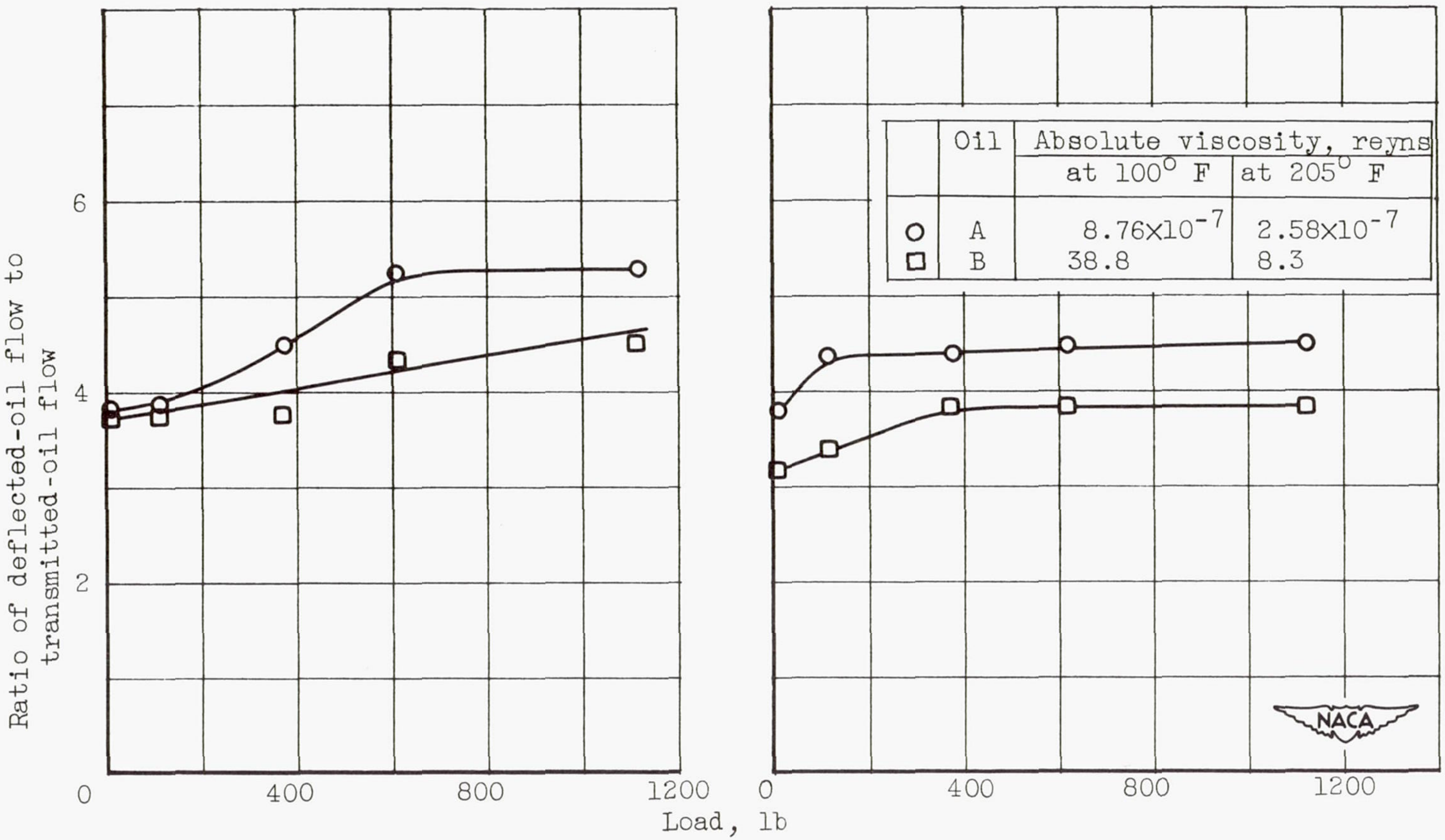
Figure 3. - Effect of DN on ratio of deflected- to transmitted-oil flow for bearing 19 for oils A and B. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperatures, 100° and 205° F; oil-jet diameter, 0.050 inch.



(a) Oil inlet temperature, 205° F.

(b) Oil inlet temperature, 100° F.

Figure 4. - Effect of oil flow on ratio of deflected- to transmitted-oil flow for bearing 19 for oils A and B. DN, 1.2×10^6; load, 368 pounds; oil inlet temperatures, 100° and 205° F; oil-jet diameter, 0.050 inch.



(a) Oil inlet temperature, 205° F.

(b) Oil inlet temperature, 100° F.

Figure 5. - Effect of load on ratio of deflected- to transmitted-oil flow for bearing 19 for oils A and B. DN, 1.2×10^{6}; oil flow, 2.75 pounds per minute; oil inlet temperatures, 100° and 205° F; oil-jet diameter, 0.050 inch.

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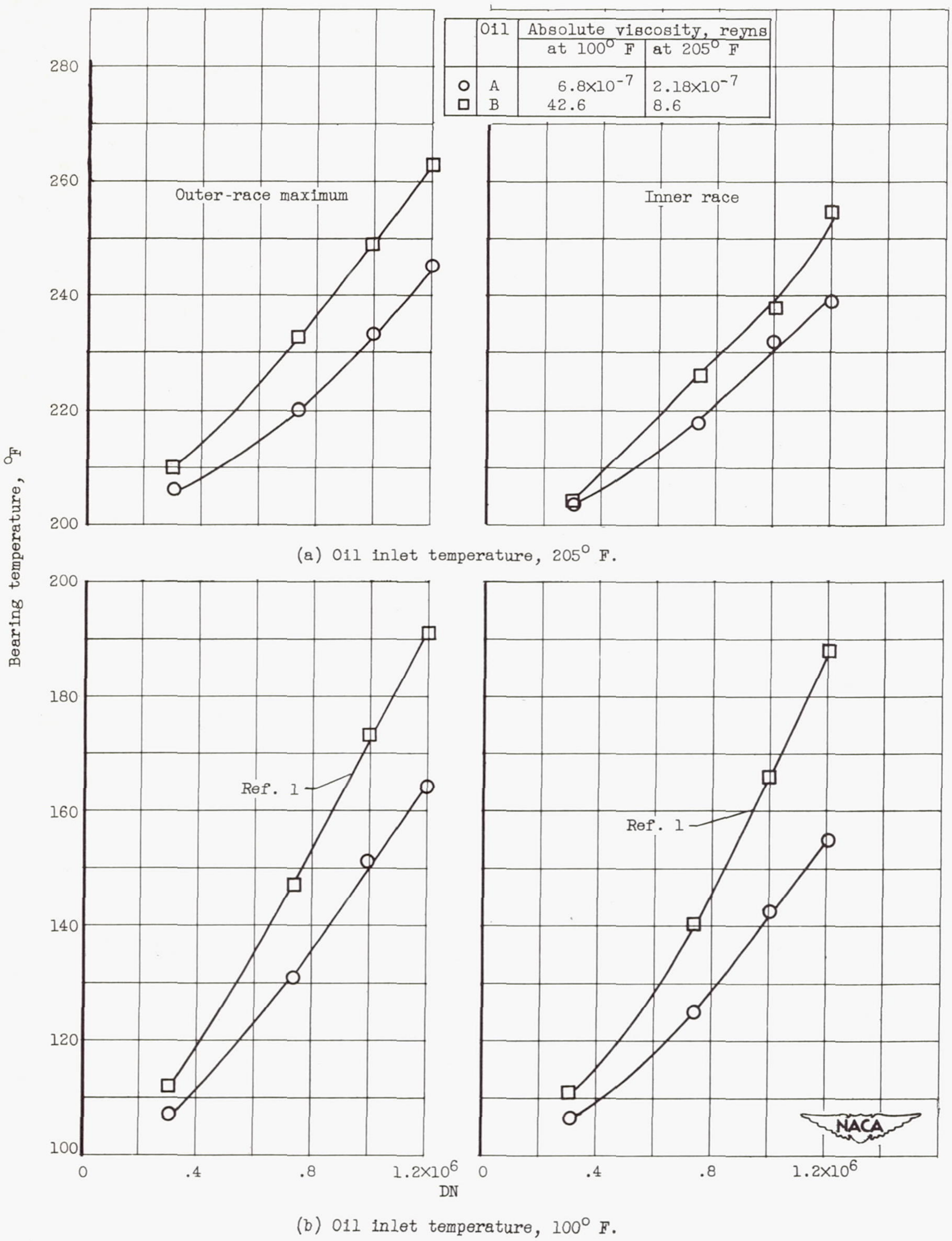

Figure 6. - Effect of DN on outer-race-maximum and inner-race temperatures of bearing 19 for oils A and B. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperatures, 100° and 205° F; oil-jet diameter, 0.050 inch.

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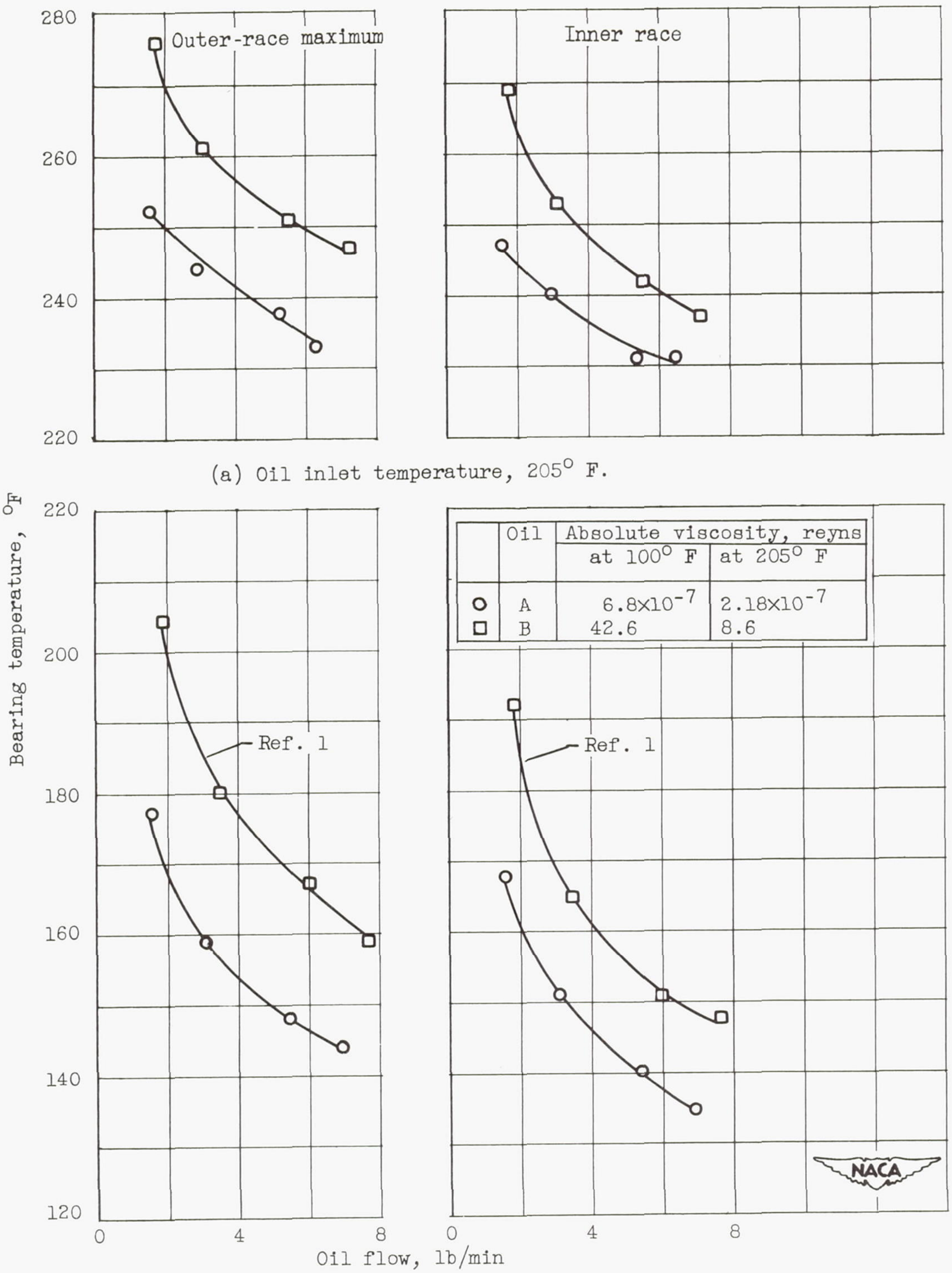
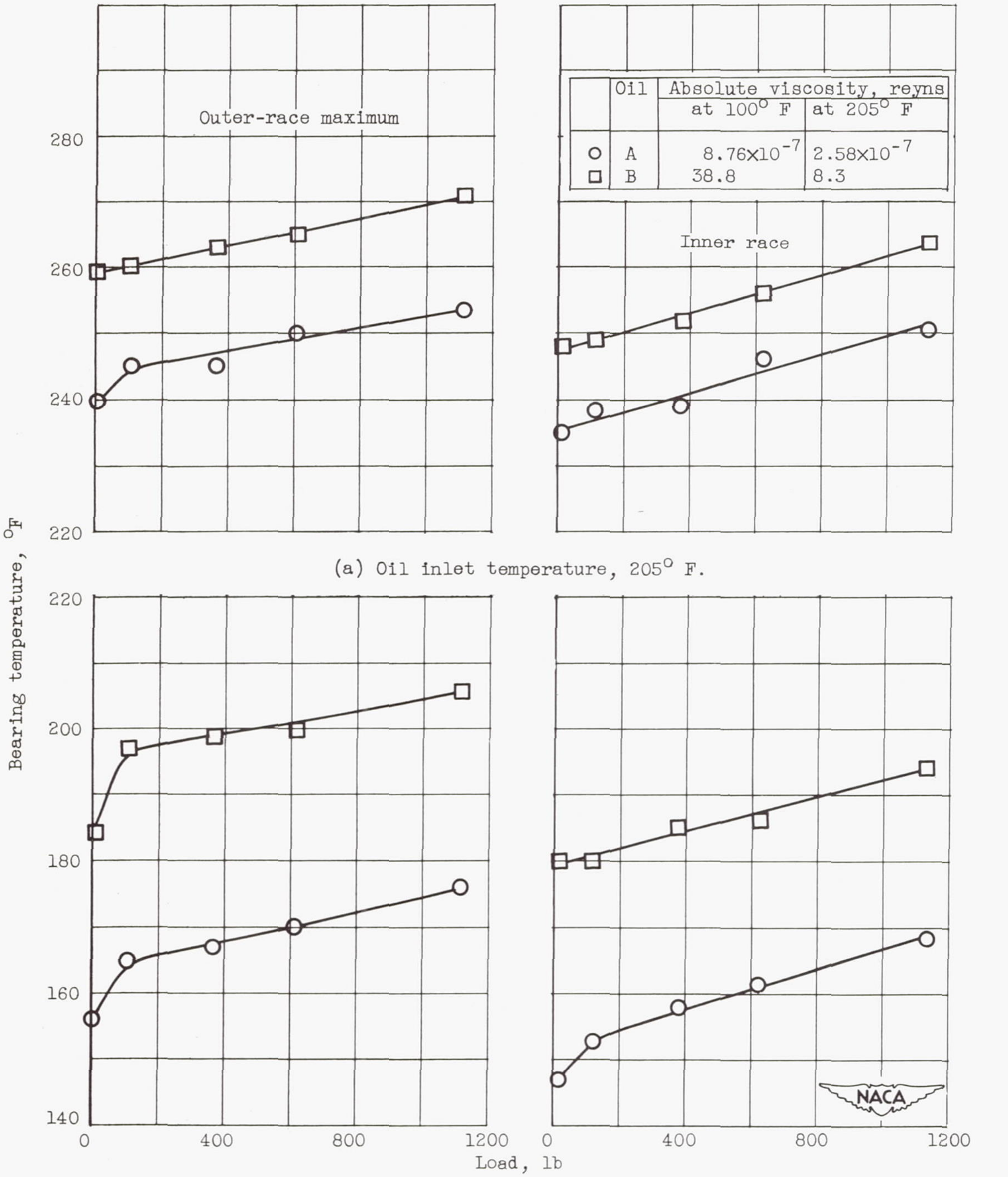


(a) Oil inlet temperature, 205° F.

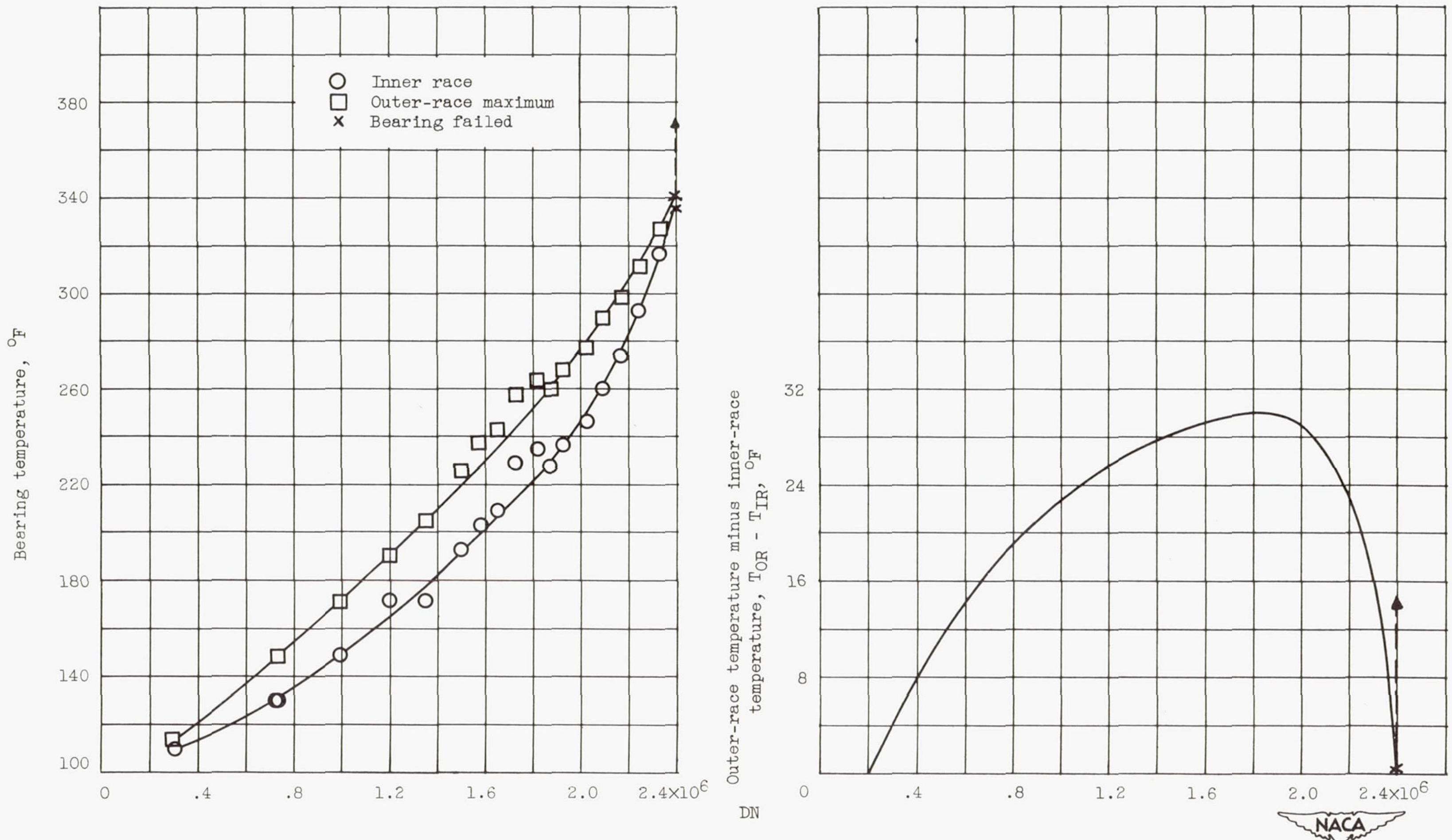
(b) Oil inlet temperature, 100° F.

Figure 7. - Effect of oil flow on outer-race-maximum and inner-race temperatures of bearing 19 for oils A and B. DN, 1.2×10^6; load, 368 pounds; oil flow, 1.7 to 7.65 pounds per minute; oil inlet temperatures, 100° and 205° F, and oil-jet diameter, 0.050 inch.



(b) Oil inlet temperature, 100° F.

Figure 8. - Effect of load on outer-race-maximum and inner-race temperatures of bearing 19 for oils A and B. DN, 1.2×10^{6}; oil flow, 2.75 pounds per minute; oil inlet temperatures, 100° and 205° F; oil-jet diameter, 0.050 inch.



(a) Race temperature.

(b) Temperature difference.

Figure 9. - Effect of DN on outer-race-maximum and inner-race temperatures and on temperature difference of bearing 20. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch; oil B.

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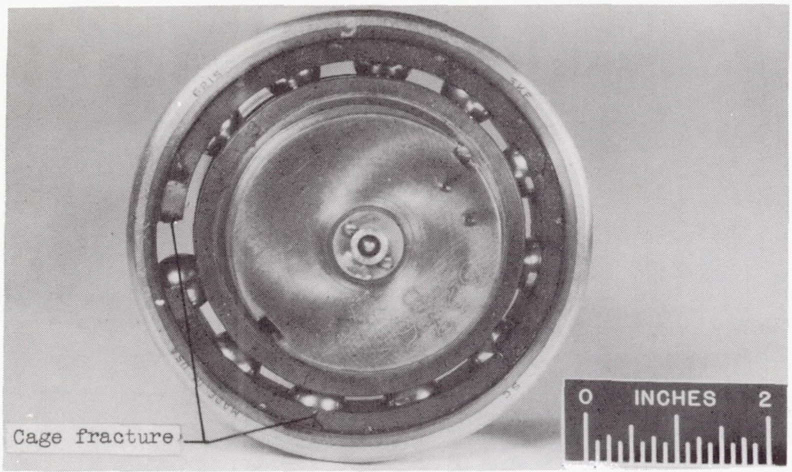

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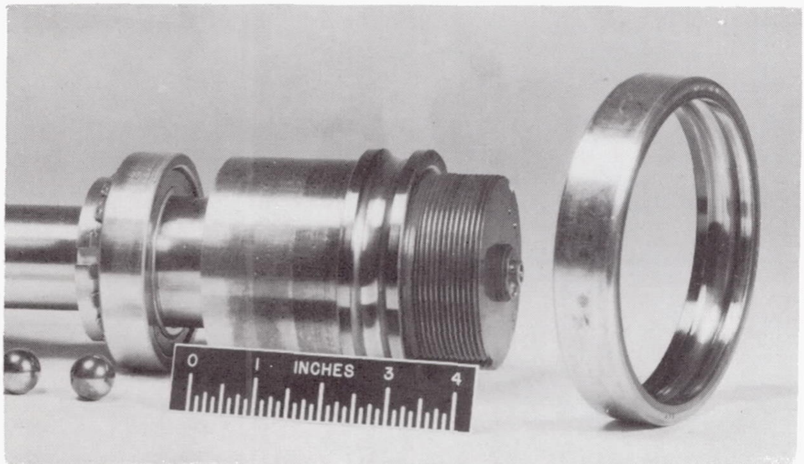
(a) Assembled bearing.



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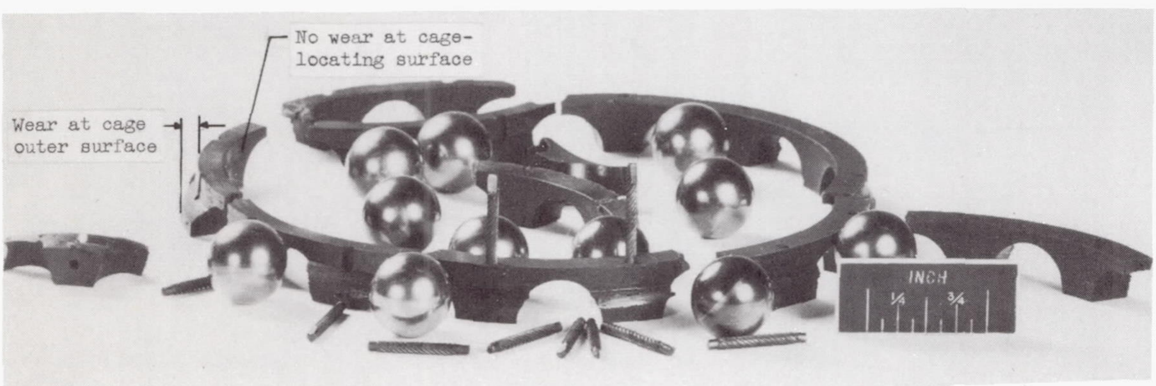
(b) Inner race and outer race.



(c) Balls and cage.

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Figure 10. - Bearing 20 after failure.

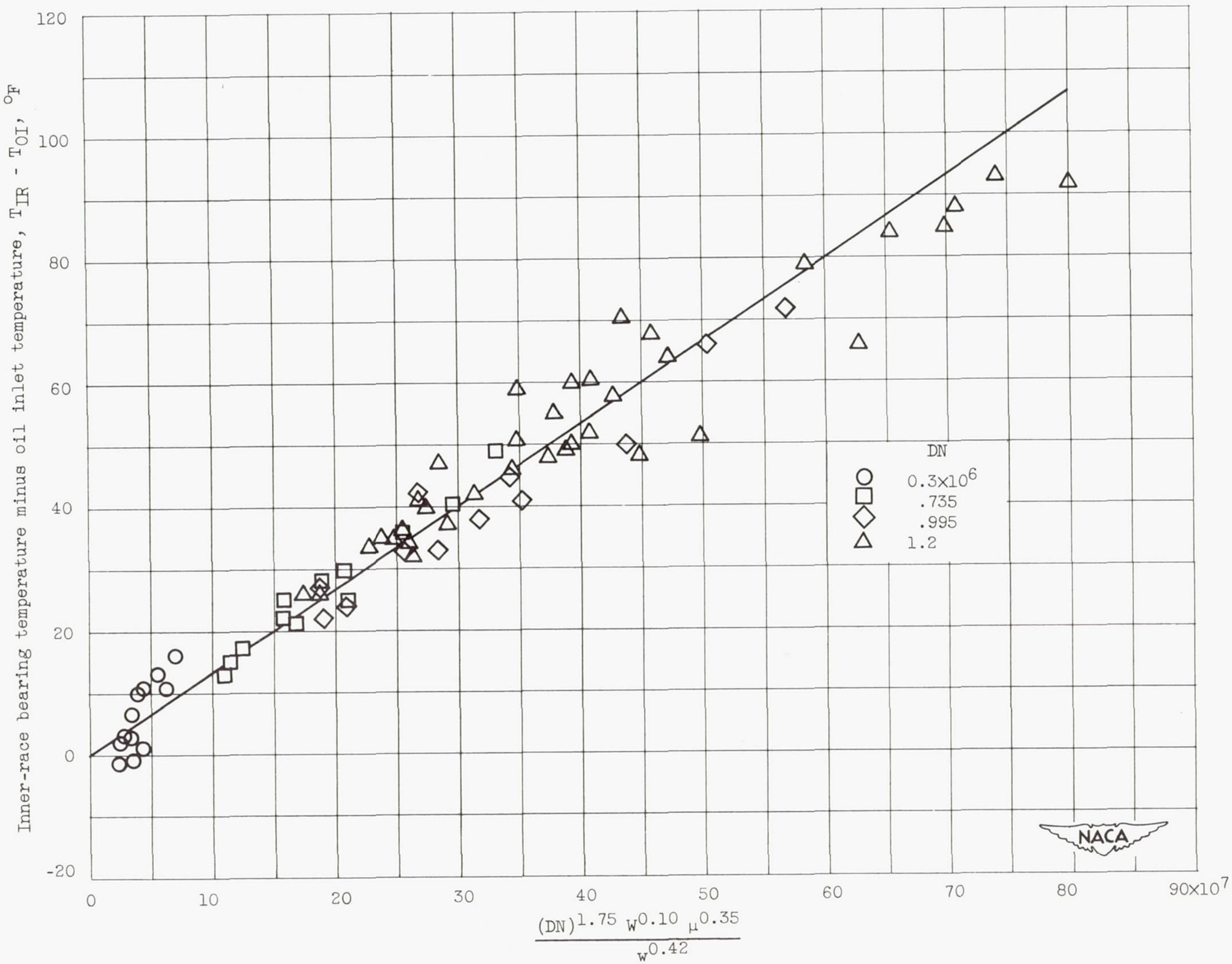

Figure 11. - Cooling-correlation curve for inner-race temperature of bearing 19. (Temperature T, °F; DN, bearing bore in mm times shaft speed in rpm; load W, lb; viscosity μ, lb-sec/sq in.; oil flow w, lb/min.)

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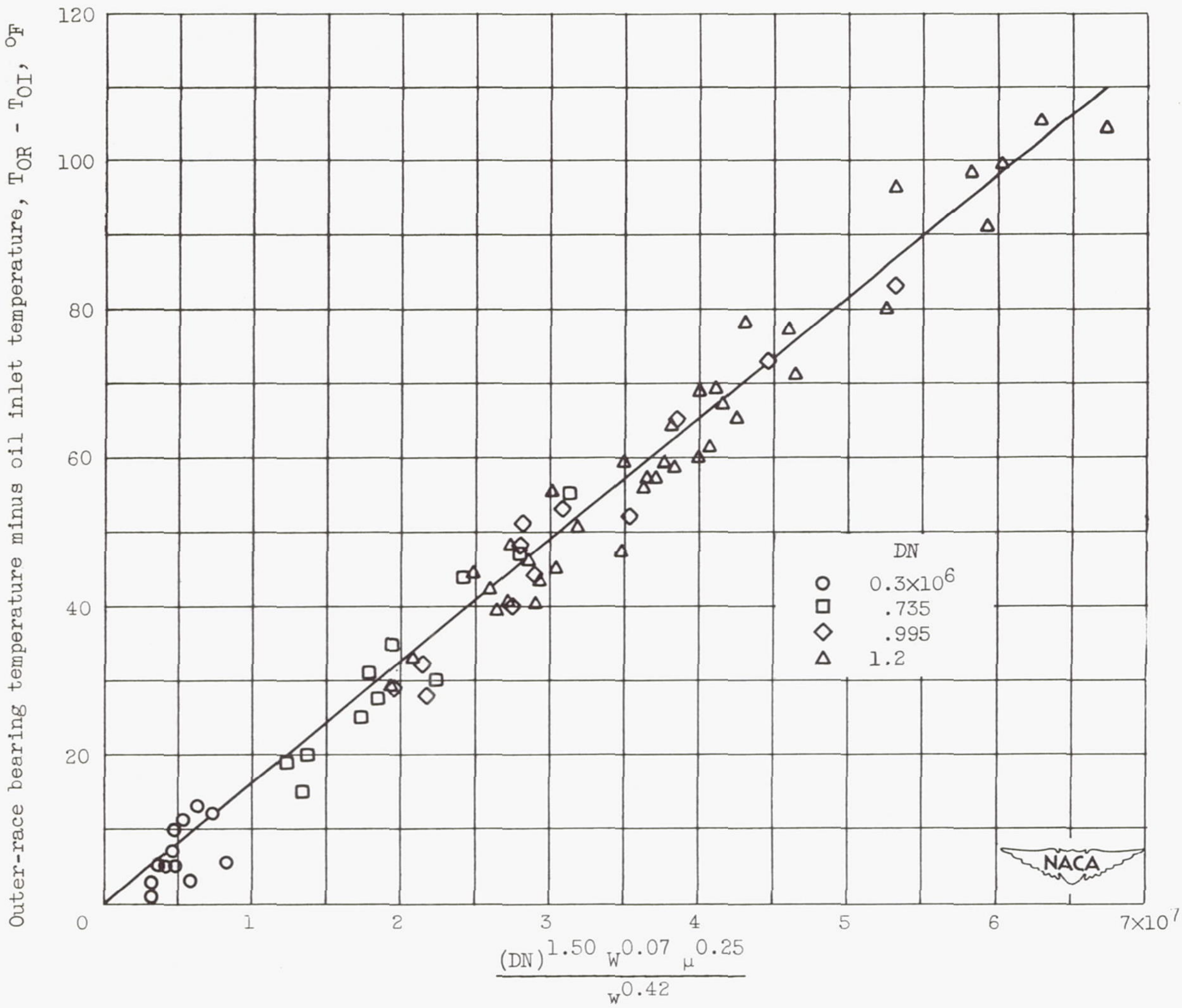

Figure 12. - Cooling-correlation curve for outer-race-maximum temperature of bearing 19. (Temperature T, °F; DN, bearing bore in mm times shaft speed in rpm; load W, lb; viscosity μ, lb-sec/sq in.; oil flow w, lb/min.)

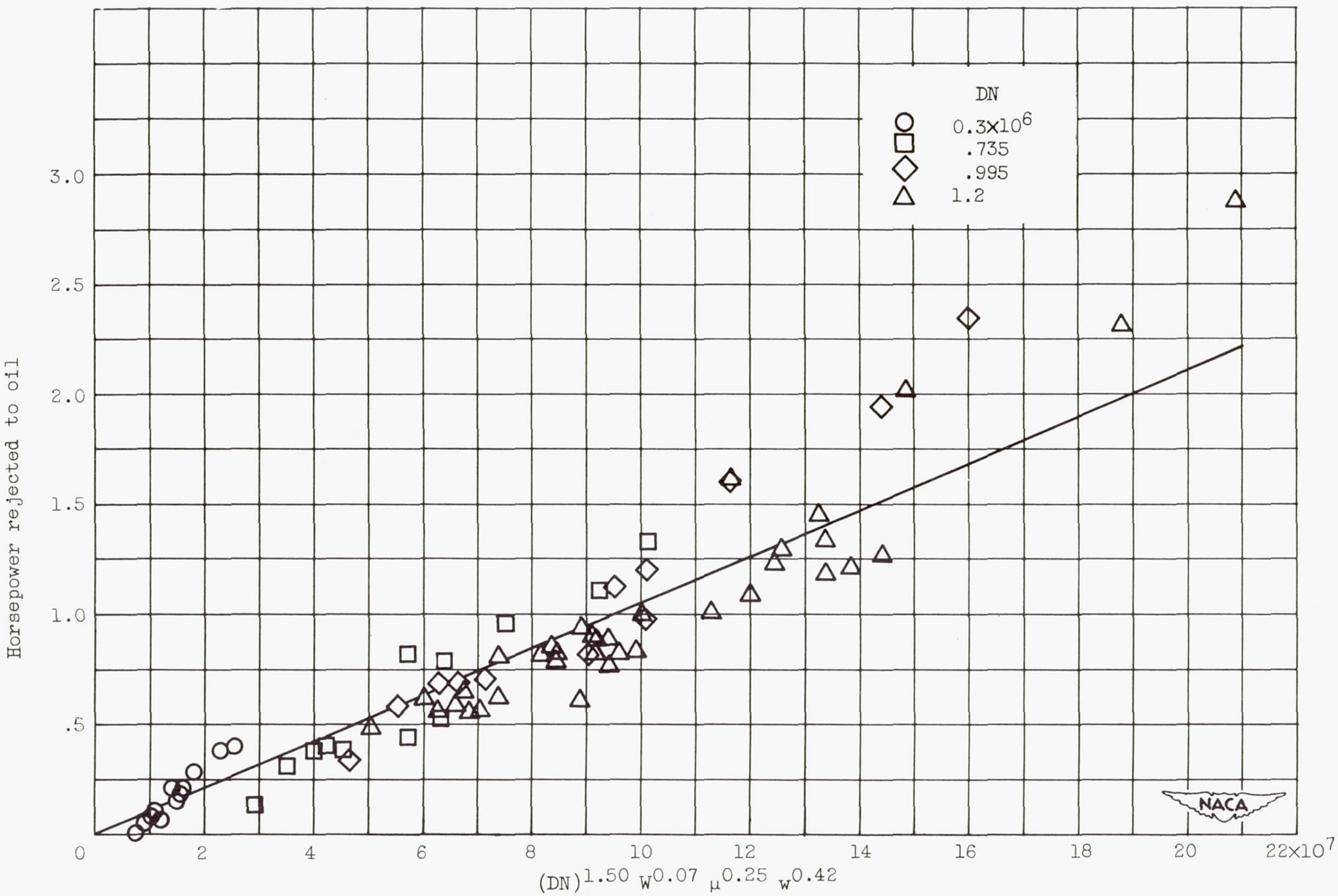

Figure 13. - Correlation curve for horsepower rejected to oil of bearing 19. (DN, bearing bore in mm times shaft speed in rpm; load W, lb; viscosity μ, lb-sec/sq in.; oil flow w, lb/min.)

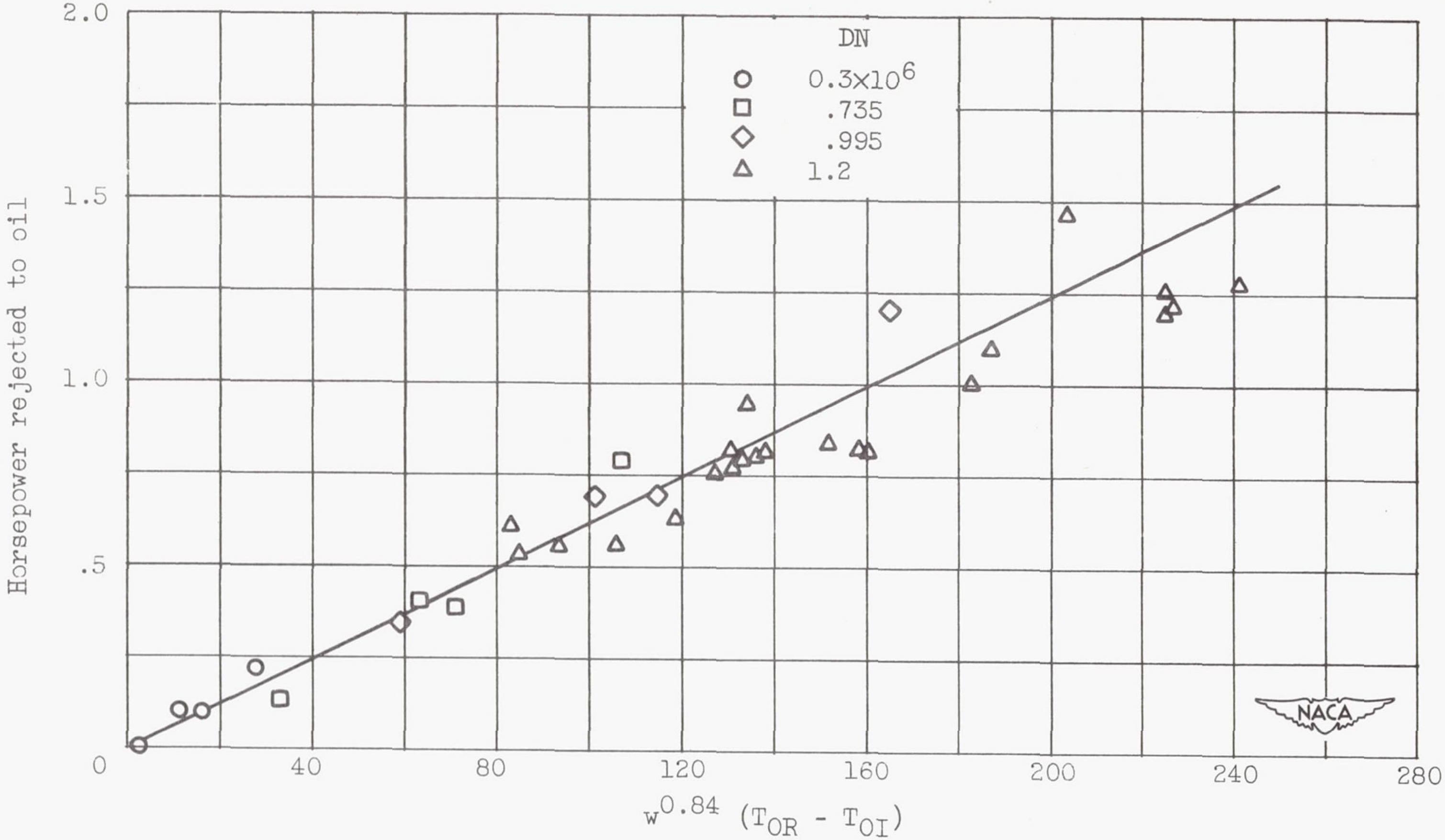

Figure 14. - Correlation curve for horsepower rejected to oil of bearing 19. (Oil flow w, lb/min; temperature T, $^{\circ}$F.)

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