NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 2128

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

I - INITIAL STUDIES

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SUMMARY

An experimental investigation of 75-millimeter-bore (size 215) roller bearings of three different cage types was conducted over a range of DN values (product of bearing bore in mm and shaft speed in rpm) from 0.3×10^6 to 1.65×10^6 and static radial loads from 7 to 1613 pounds with circulatory oil feed.

Of the seven bearings investigated three incipient failures occurred. One failure was of an inner-race riding-cage type bearing and the other two were of roller-riding cage-type bearings. The maximum DN value reached was 1.65×10^6 for the inner-race-riding cage-type bearing failure and 1.4×10^6 for each of the other failures.

The operating temperatures of the three types of bearing were found to differ most appreciably in the low-load, high-speed range where the roller-riding cage-type bearing exhibited significantly lower operating temperatures than the one- and two-piece inner-race-riding cage-type bearings. The operation of the roller-riding cage-type bearings, however, was considerably rougher than that of inner-race-riding cage-type bearings, and the bearings showed prohibitive roller and cage wear after relatively short high-speed operation (DN values over 1×10^6) as compared to the inner-race-riding cage-type bearings.

In general, percentage of slip within the bearings increased with increase in DN value and decreased with increase in load, reaching average values greater than 50-percent slip at light loads for DN values over 1×10^6.

Under certain operating conditions, namely moderate speeds and loads, the inner-race-riding cage-type bearing operated with a cage speed greater than the theoretical value indicating that the cage and

rollers were driven by the cage-locating surface rather than the cage being driven by the rollers. This condition was not observed for an equivalent bearing having a roller-riding cage.

A circumferential temperature gradient existed around the outer race of the turbine roller bearing of a commercial gas-turbine engine; this gradient was qualitatively similar to that obtained in the bearing rig in that the maximum temperature occurred in the region 270° to 300° after the oil-jet location in the direction of shaft rotation, whereas the minimum temperature occurred in the region 60° to 90° after the oil-jet location in the direction of shaft rotation.

The actual life of cylindrical roller bearings operating at DN values over about 0.7×10^6 and light loads may be appreciably greater than the predicted fatigue life based upon the sum of the external load and the theoretical value of centrifugal load. This apparent increase in life is due to slippage within the bearing, which effect has not heretofore been considered in such calculations.

Although considerable slippage occurred at high speeds and light loads for the cylindrical roller bearings investigated, there was negligible roller wear in the bearings that did not fail. It is therefore postulated that there may exist a hydrodynamic film of oil between rollers and raceways under such operating conditions.

INTRODUCTION

A dependable bearing to carry radial load at extreme speed, and preferably to operate at high ambient temperatures, is desired for use as the turbine-support bearing in gas-turbine-type aircraft-propulsion units, where gravity loads generally under 1000 pounds and DN values (product of bearing bore in mm and shaft speed in rpm) to 1×10^6 are presently encountered, as well as for other high-speed applications. It is desirable to know the operating characteristics and limitations of conventional rolling-contact bearings at high speeds and how these characteristics and limitations may be improved and extended by such means as improved lubrication methods and design modifications.

The literature contains only a little information on roller bearings near a DN value of 1×10^6; except for reference 1, this information is unavailable. Cage failures of the rear turbine bearing of aircraft gas-turbine engines due to faulty lubrication upon starting has been reported by Wilcock (reference 2).

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A preliminary investigation was conducted at the NACA Lewis laboratory to determine experimentally the operating characteristics of conventional cylindrical-roller bearings at high speeds. Three types of bearing were studied, the main difference being in the cage construction. The three types are used interchangeably in aircraft gas-turbine engines and are of 75-millimeter bore, 130-millimeter outside diameter, and 25-millimeter width. Seven bearings in all were tested. The ranges of controlled variables in this investigation were as follows: DN values from 0.3×10^6 to 1.65×10^6 (4000 to 22,000 rpm), and static radial loads from 7 to 1613 pounds. Oil at 100° F was supplied to the bearing under investigation by means of a single jet of 0.089- or 0.180-inch diameter directed at the space between the cage and the inner-race flange except in two cases where twin jets, each of 0.100-inch diameter, were used on either side of the bearing. External heat was not applied to the bearing housing or to the shaft. Data from the rear turbine bearing of a commercial gas-turbine engine operating to maximum speed (DN value, 0.86×10^6) is included for comparison with the bearing data from the test rig.

APPARATUS

Bearing rig. - The radial-load rig used in this investigation is shown in figure 1.

The bearing under investigation was mounted on one end of the test shaft, which is supported in a cantilever fashion for purposes of observing the component parts and the flow of the lubricant during operation. This arrangement also facilitates assembly and disassembly of the test bearing for inspection at frequent intervals. The outer race of the test bearing was mounted in the turbine bearing housing of an aircraft gas-turbine engine and for rigidity this assembly was mounted in a steel housing of 21/32-inch radial cross section. The complete outer-race assembly was axially positioned between two uprights with 0.002-inch axial clearance. In order to prevent rotation of the housing assembly during no-load operation, a stop pin located in one upright engaged a clearance hole in the outer-race housing.

A radial load up to 1613 pounds was applied to the bearing under investigation by means of a lever and dead-weight system. The end of the load arm, which contacts the bottom of the bearing housing, was provided with a locating pilot pin and convex upper surface. The curvature of the surface serves to compensate for effects of shaft deflection. Load on the bearings was applied vertically upward.

With these methods of mounting and loading, the operating characteristics of the bearing under investigation are essentially unaffected by small shaft deflection as well as small shaft and load-arm misalinements.

The test-shaft assembly is shown in figure 2. The shaft is supported on two deep-groove Conrad type ball bearings of sizes 207 and 209. The size 207 bearing, mounted at the drive end of the shaft, locates the shaft in the axial direction. The size 209 bearing, its outer race free to move axially, is mounted midway between the 207 bearing and the bearing under investigation; in consequence, the size 209 bearing carries twice the effective applied load and the size 207 bearing supports the same load as does the bearing under investigation. The inner race of each support bearing was mounted with an interference fit on the shaft and held in place by means of a star-lock washer and lock nut. A radial set screw through the lock nut and seated in a keyway of the shaft was required at the extreme operating conditions to insure positive location of the bearings on the shaft.

<u>Drive equipment</u>. - The high-speed drive equipment consists of a shunt-wound 225-horsepower direct-current motor connected to a 10:1 speed increaser. The high-speed shaft of the speed increaser is connected to the test shaft by means of a floating spline coupling. The speed range of the test shaft is 800 to 50,000 rpm controllable to within ±1 percent.

<u>Test bearings</u>. - The seven test bearings (table I) were standard cylindrical roller bearings of a conventional aircraft gas-turbine engine. Three bearing designs, each having double flanges on the inner race and a removable outer race are used interchangeably as the turbine rotor bearing in this engine, the bearing dimensions being 75-millimeter bore, 130-millimeter outside diameter, and 25-millimeter width. A drawing emphasizing the differences of the three types of bearing is shown in figure 3. The designs differ principally in the construction of the cage and the radial location of the cage, one being a one-piece inner-race-riding cage, another a two-piece riveted roller-riding cage, and the third a two-piece riveted inner-race-riding cage.

<u>Temperature measurement</u>. - Six iron-constantan thermocouples were located at 60° intervals around the outer-race periphery of the bearing under investigation at the axial center line. The thermocouples were embedded in the outer-race housing, flush with the housing bore, and made contact with the outer-race outside surface.

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A copper-constantan thermocouple was pressed against the inner-race inside surface at the axial center line of the test bearing. The signal generated was transmitted through copper and constantan slip rings at the end of the shaft. The brushes, through operation of a solenoid, contacted the slip rings only at the time a reading was made.

Thermocouples contacting the outer-race outside surface were installed on the top and the bottom of each support bearing.

Thermocouples were located in the oil lines immediately upstream of the points at which oil entered the bearings. Oil-out, experimental-unit ambient, and sump temperatures were also measured.

Lubrication system. - The lubrication system used was of the circulating type. Oil inlet temperature was controlled to within $\pm 1^\circ$ F and oil inlet pressure to within ± 0.2 pound per square inch. Oil flow was determined by calibrated rotameters. Oil entering and leaving the support bearings was kept from the vicinity of the test bearing by means of a shaft slinger and an oil shield. The oil was drained by gravity from the base of the rig to a sump and then recirculated. A full-flow filter was provided after the oil-supply pump.

Cage-speed determination. - Cage speed was measured by a magnetic pickup (fig. 4), which counted the rollers as they passed a given station. The signal from the pickup was amplified, read on a frequency meter, and recorded on a recording ammeter.

PROCEDURE

Lubrication of test bearing. - Lubricant at 100° F was supplied to the bearing under investigation through a single jet of 0.089-inch (bearings 4 to 6) or 0.180-inch (bearing 1) diameter. Bearings 2 and 3 were lubricated by single-opposed jets, each 0.100-inch in diameter, that supplied 5.2 pounds per minute total flow at 10 pounds per square inch. (The lubrication of bearing 7 will be discussed later.) The oil was directed at the space between the cage and the inner-race flange on the unloaded side of the bearing and perpendicular to the bearing face in all cases.

The properties of the oil used are given in figure 5. This oil was a commercially prepared blend of a highly refined paraffin base with a small percentage of a polymer added to improve the viscosity index.

Lubrication of support bearings. - Inasmuch as temperature gradients along a shaft axis cause a flow of heat to or from the bearings, it is important in comparative bearing research to maintain the support bearings at nearly constant temperatures for a given operating condition. A large quantity of oil was supplied to each of the support bearings; the flow and the temperature of the oil was held constant for all the investigations reported. For all runs, the oil to each support bearing was supplied at 100° F and a pressure of 10 pounds per square inch through 0.180-inch-diameter jets directed at the cage-locating surface (flow of 8 lb/min). 1275

Test-bearing measurements. - The test bearings were measured on a standard fixture to determine the unmounted eccentricity and internal clearance (table I). A 0.0001-inch dial indicator was used in conjunction with the fixture. The measurements of clearance and eccentricity were accurate to within ±.0001 inch.

Inasmuch as the interference between the inner-race bore and the shaft diameter was of the order of 0.001 inch, the inner-race and cage assembly of the test bearing were heated in an oil bath to a temperature of approximately 225° F in order to facilitate assembly of the inner race on the shaft.

After assembly, the bearing was again measured for eccentricity and internal clearance. At intervals during long runs, the bearing was visually inspected and checked for internal and cage clearance.

The surface finish (obtained using a Profilometer) is given in table II and the hardness of the component parts of each cage-type test bearing, before and after running, is given in table III. Contact-surface crowning data for new bearings was obtained using a Pratt and Whitney Electrolimit Gage and is given for each bearing type in table IV.

Reference conditions. - In order to determine the influence of running time on bearing performance, frequent checks of the bearing operating characteristics were made with bearings 5 and 6 at a predetermined set of operating conditions, that is, DN value of 1.2×10^6 (16,000 rpm), 368 pound load, and lubrication of 2.75 pounds per minute through a 0.089-inch-diameter jet. Oil samples were taken on many of these occasions. In this manner, it was possible to determine the changes in bearing characteristics as well as changes in the oil over long running periods.

Engine data. - An attempt was made to correlate test-bearing data with actual engine operation data by instrumentation of the rear-turbine bearing (bearing 7, table I) of a commercial gas-turbine engine. This bearing was lubricated by a single jet of 0.052-inch diameter. The engine oil used was Navy 3042 (6.1×10^{-6} reyns at 100° F, 0.72×10^{-6} reyns at 210° F, viscosity index of 100, and a flash point of 390° F). Six outer-race thermocouples were located at 60° intervals around the bearing circumference. The thermocouples were mounted radially through the outer-race housing so that the sensitive element made contact with the outer race (the method used in the bearing-test machine). All the thermocouples were mounted at the axial center line of the bearing. A special oil-inlet thermocouple and pressure take-off was installed immediately before the oil-jet nozzle of the engine to obtain more accurate values of oil-inlet conditions than are given by engine-sump temperature and engine pressure. The estimated static load on the bearing was 375 pounds.

RESULTS AND DISCUSSION

The results of the experimental investigation of bearings 1 to 7 (table I) are given in figures 6 to 17. Bearing temperature was chosen as the principle criterion of operation inasmuch as, in the final analysis, temperature is an over-all indication of the effects of all the operating conditions. In addition, bearing temperature is a direct indication of whether operation is at an equilibrium condition and to what degree of severity the bearing is being subjected in relation to its maximum permissible operating temperature.

Rig bearing temperatures. - The effect of speed for DN values to 1.65×10^6 on the temperature at the loaded and unloaded sides of bearing 1 and the representative support bearings is shown in figure 6(a) for a 7 pound load on test bearing 1. The effect of speed for DN values up to 1.43×10^6 and a 1613-pound load is shown in figure 6(b).

The temperature data shown in figure 6 illustrate two facts:

(a) At very light loads and for the speed range covered, the increase in bearing temperatures with speed is greater than linear (fig. 6(a)); whereas with an appreciable load on the bearings, the increase in bearing temperature with speed is approximately linear (fig 6(b)).

(b) Although the test bearing is of the cylindrical-roller type, the two support bearings of the ball type, and all of different sizes, the foregoing facts hold qualitatively for all three bearings.

Bearing failures. - During the investigations reported, three incipient failures of test bearings occurred. An incipient failure is said to occur when the bearing, operating at a given set of conditions, does not reach a state of temperature equilibrium below 390° F or when visual examination indicates excessive wear or damage to any of the component parts.

One of the failures occurred in a run to determine the maximum DN value over a range of loads at which a 75-millimeter bore (size 215) roller bearing would operate at a stable temperature. Experimental bearing 2 (one-piece inner-race-riding cage) was operated at a DN value of 1.65×10^6 with loads of 7, 113, 613, and 1113 pounds. The bearing failed as the 1113 pound load was removed in preparation to going to a higher speed. The inner-race temperature increased rapidly from 308° to 390° F and showed no signs of reaching equilibrium. The rig was shut down and the test bearing inspected. The cage was cut to remove it from the inner race and appearances (fig. 7) indicated an incipient lubrication failure had occurred at the cage-locating surface. A possible explanation of this failure is that although the bearing had previously operated satisfactorily at the conditions of 7-pound load and DN value of 1.65×10^6, upon return to these conditons the bearing operating temperature was greater because of the intervening operation at a 1113-pound load. As the load was removed, the relative speed between the cage and the inner race increased due to slippage (although shaft speed remained constant). Even though the cage load decreased with the increase in slippage, the lower oil-film viscosity resulting from the higher bearing operating temperatures produced operation in the higher friction region of boundry lubrication. The additional heat thus generated was sufficient to cause an incipient failure at this critical surface. The diametral clearance of this bearing, which failed after 17 hours, increased from 0.0018 to 0.0025 inch, and the cage diametral clearance increased from 0.019 to 0.043 inch.

The second incipient failure occurred with bearing 3. The maximum operating conditions that this bearing was subjected to were: DN value, 1.4×10^6; load 1113 pounds; and temperature, 356° F. The bearing operated with considerable vibration at DN values above 1×10^6. Although equilibrium of the test-bearing temperature existed at all conditions, the rig was shut down after the oil had darkened noticeably. Upon examination of the test bearing, it was found that the cage pockets and rollers had worn appreciably (see table I). This bearing probably would have run for some time at DN values below 1×10^6.

The third incipient failure was of a somewhat different nature and occurred with bearing 4. The bearing exhibited rough operation at DN values above 1.2×10^6. After the shaft speed was increased to change the DN value from 1.2×10^6 to 1.4×10^6, at a bearing load of 7 pounds, the temperature increased from 125° to 245° F and then decreased to an equilibrium temperature of 120° F. With the application of a 113-pound load, the bearing temperature rose to 330° F and then decreased to an equilibrium temperature of 188° F. On the application of a 368-pound load, the bearing temperature rose to 390° F. The rig was shut down shortly and the bearing inspected. The diametral clearance of this bearing, which had run a total of 16.7 hours, increased from 0.0020 to 0.026 inch, whereas the rollers decreased in diameter from 0.5255 to 0.5136 inch. The cage-pocket diameter, however, increased but a small amount. The exact increase was unknown inasmuch as the pocket diameter could not be measured before the investigation. (It can be recalled from table I that a sample bearing was taken apart and measured for dimensions that could not be obtained from the assembled bearing; the tolerances on such dimensions as roller diameters are maintained much closer than are the tolerances on cage-pocket diameters.) With the large radial clearance between the cage pockets and the rollers (table I), this bearing type is subject to vibration at high speeds inasmuch as the cage so locates itself that its center is displaced appreciably from the bearing center, thereby causing unbalance. This condition became more severe as operation progressed; and eventually, the clearance between the rollers and the cage pockets had become so great (fig. 8) that the cage was located at the inner race. The diametral clearance between the cage and the inner race had increased from 0.041 to 0.070 inch. It is probable that some of the worn cage material mixed with the oil and acted as a lapping compound that promoted wearing of the rollers and races.

Effect of running time. - At extreme operating conditions, where usually small effects are magnified, it has been found that the operating characteristics of a roller bearing change with time over long running periods. Thorough evaluation of these characteristics of roller bearings requires much more data than are available at present. Two examples are presented, however, to illustrate the nature of the problem and to emphasize how running time affects the results of comparative investigations that must necessarily be run for long periods at extreme operating conditions. The data obtained at the reference condition (DN value, 1.2×10^6 (16,000 rpm); load, 368 lb; oil-jet diameter, 0.089 in.; oil flow, 2.75 lb/min; and oil inlet temperature, 100° F) are given in figure 9 for bearing 5 over a running time of 133.5 hours and for bearing 6 over a running time of 195.9 hours. The abscissa was arbitrarily selected to represent, to a first approximation, severity of bearing operation. This severity factor is the summation of the products of the difference between equilibrium bearing temperature and oil inlet temperature for each operating condition and the corresponding operating time in minutes at that particular condition. Temperature was chosen rather than any combination of the operating variables such as load, speed, oil flow, and so forth inasmuch as, in the final analysis, temperature combines the effects of all the operating variables. The following data are presented in figure 9 for each bearing; maximum and mimimum outer-race temperatures, inner-race temperature, cage slip, oil viscosity at 100° F, and oil viscosity index.

It is evident from figure 9 that the bearing operating characteristics vary in a somewhat erratic manner. It is significant that the three bearing temperatures followed the same general trend indicating similarity of oil distribution for the various reference points. It may be seen that bearing 5 operated at lower temperatures when it was relatively new. During operation, in the period represented by change in severity factor from 1.35×10^5 to 2.7×10^5, however, the inner-race temperature increased approximately 100° F, whereas the outer-race temperature increased by a slightly lesser amount, and operation continued at the higher temperature levels for the remainder of the running time. After the series of runs, bearing clearance was found to have decreased approximately 0.0003 inch. This decrease in clearance indicates either a growth of the inner race or rollers or a shrinkage of the outer race. It may be seen that cage speed, and therefore precentage slip, varied in an unpredictable manner. Percentage of slip is defined as

$$\text{Percentage of slip} = \left(\frac{N_c' - N_c}{N_c'}\right) 100$$

where

N_c actual cage speed, rpm

N_c' theoretical cage speed with no slip, $N_c' = \frac{1}{2}\left(1 - \frac{D}{D_p}\right) N_s$, rpm

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D roller diameter, inches

D_p pitch diameter of bearing, inches

N_s shaft speed, rpm

It may be seen from figure 9 that there was an appreciable decrease in viscosity as well as in viscosity index of the lubricant with progression of operating severity. In contrast to bearing 5, it may be seen from figure 9 that bearing 6 operated at lower temperatures throughout the series of runs. It is of interest to note that although bearings 5 and 6 are both of the inner-race-riding cage type, the bearings are of different construction and manufacture, which may or may not be significant.

Circumferential temperature distribution. - Polar plots of temperature at six points around the outer-race circumference for bearings 4, 5, and 6 for several DN values are shown in figure 10. The temperatures were not uniform around the outer race of the bearing, being lowest just after the oil inlet in the direction of rotation and highest in the region before the oil inlet. It is also evident that the outer-race circumferential temperature gradient increases with increase of speed and increases to a lesser extent with an increase in load (fig. 11). Not only the differences in cage construction but those in the mounted clearances may account for the different temperature distributions as affected by speed and load for the three bearing types, as shown in figures 10 and 11. Getzlaff (reference 1) found that clearance considerably affects roller-bearing operating temperatures.

Effect of DN value on operating temperature. - The effects of shaft speed on operating temperatures of the three types of bearing investigated are given in figure 12 for a load of 368 pounds and an oil flow of 2.75 pounds per minute. Operating temperature essentially increased linearly with increase in speed for each of the bearings. It is also evident that the differences in operating temperatures of the various bearing types are more pronounced at the higher speeds. The three bearing types operated within $\pm 10^\circ$ F for the maximum and mean outer-race and the inner-race temperatures. (The data of fig 12 for bearing 5 was obtained before a severity factor of 2.7×10^5 had been reached.)

Effect of load on operating temperature. - The effects of load on the operating temperatures of the three types of bearing investigated are given in figure 13 for an oil flow of 2.75 pounds per minute. The data point for a DN value of 0.735×10^6 at a 7-pound load for bearing 5 has not been included inasmuch as the bearing temperatures fluctuated due to an unstable system for this particular operating combination.

As may be seen from figures 13(a) to 13(c), in general the outer-race temperatures of the three bearing types increase rather rapidly with load in the low-load range, after which the slope of temperature with load approaches zero. A significant difference in the temperature of the three bearing types, for the conditions investigated, occurs at the light loads where the two-piece roller-riding cage-type bearing operates at outer-race temperatures appreciably lower than the other bearing types, particularly at the higher DN values. The slope of outer-race temperatures with load increases with higher values of DN in the low-load range for bearing 4 (a similar effect was observed for test bearing 3); whereas the slope of outer-race temperature with load decreases slightly with higher DN values in the low-load range for bearings 5 and 6. These significant differences among bearings are, however, not apparent at the lower speeds, for example, DN value of 0.3×10^6. The marked increase in operating temperature above loads of 368 pounds at the higher DN values for bearing 5 may be explained, in part, by the small clearance values of this bearing.

The inner-race temperatures for bearing 4 show an increasing slope with speed in the very-low-load range (fig. 13(d)). As more load is applied (between 113 and 368 lb), the temperature decreases; this decrease occurs only at the higher DN values and more sharply as the DN value exceeds 0.735×10^6. For bearings 5 and 6, a somewhat different operating characteristic is noted in that the slope of temperature with load gradually decreases with increasing DN values in the low-load range and actually becomes negative at DN values of 1.2×10^6. Above loads of 368 pounds, however, the slope of temperature with load is positive or zero for all the bearing types.

The more favorable temperature characteristics of the roller-riding cage-type bearing over the inner-race-riding cage-type bearing in the low-load high-speed range may be explained in part by the large percentage slip found for both bearing types at these operating conditions. This point is discussed in the section ANALYSIS OF EXPERIMENTAL RESULTS.

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Effect of load and speed on percentage slip. - As previously stated, cage speed, or orbital roller speed, was determined by means of a magnetic pickup, which counted the rollers as they passed a given point on the outer race.

The effect of DN on percentage slip is shown in figure 14 for bearing 4, which has a roller-riding cage and in figure 15 for bearings 5 and 6, which have inner-race-riding cages. The curves of figures 14 and 15 are for relatively new bearings. It has been found that the cage speed is not readily reproducible, particularly at the higher DN values; the variations might depend upon a number of things including vibration, extent of roller skewing, and clearances within the bearing. The general trend of these curves, however, indicates an increase in slip with increase in speed and a decrease in slip with increase in load. These findings are similar to results obtained with needle bearings (reference 3). In comparing these figures, it may be observed that, under certain operating conditions (low DN values and loads of 113 lb or greater) the cage speed of a bearing having an inner-race riding-cage is greater than the theoretical cage speed (fig. 15). This result was not observed for an equivalent bearing having a roller-riding-cage (fig. 14).

The insets of figures 14 and 15 show the variation of slip with load for bearings 4 and 5, respectively. The fact that in figure 15 the slip is minus for low DN values indicates that friction at the cage-locating surface is the driving force overcoming the rolling and sliding friction of the rollers at the raceways at lower speeds. This condition, however, does not occur at higher DN values. The fact that the curve for a DN value of 0.3×10^6 approaches the zero slip line at higher loads (insert, fig. 15) is probably due to the increased roller traction at higher loads. It thus appears that as the load increases the driving force due to roller traction gradually overcomes the driving force due to sliding friction at the cage-locating surface for the inner-race-riding cage-type bearing.

Comparison with engine bearing data. - The operating characteristics of the rear bearing of an aircraft gas turbine (bearing 7 of table I) over the speed range 4000 to 11,500 rpm (DN 0.3×10^6 to 0.86×10^6) are presented in figures 16 and 17.

The maximum and mean outer-race temperatures are shown in figures 16(a) and 16(b), respectively, whereas figures 16(c) and 16(d) show the oil flow, oil pressure, oil inlet temperature, and maximum temperature difference around the outer-race circumference. Also shown in figure 16, for comparison purposes, are the corresponding data obtained in the bearing test rig for a similar bearing (bearing 6, table I). A plot of the circumferential temperature distribution is presented in figure 17. In this case, the circumferential gradient is about 35° F at 11,500 rpm (DN of 0.86×10^6). This gradient is qualitatively similar to those obtained in the bearing rig in that the maximum temperature occurred in the region 270° to 300° after the oil-jet location in the direction of shaft rotation; whereas the minimum temperature occurred in the region 60° to 90° after the oil-jet location in the direction of shaft rotation.

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It is interesting to note from figure 16 the sharp increases in maximum bearing temperature, oil inlet temperature, and circumferential temperature gradient with engine speed above approximately 8000 rpm (DN of 0.6×10^6). These increases may be accounted for in part by the fact that more heat flows to the bearing through the shaft and housing at the more severe operating conditions. (The oil-flow curve of fig. 16(c) was obtained in a setup using the equilibrium oil inlet temperatures and pressures obtained from engine operating data.) It is of interest to note that this engine is not equipped with an oil cooler.

ANALYSIS OF EXPERIMENTAL RESULTS

Explanations of all phenomena observed are not available at this time; however, the following discussion may lead to a better understanding of the results obtained.

Changes in test-bearing dimensions with running time. - The test-bearing measurements before and after running are given in table I. Inasmuch as nondestructive disassembly of bearings of the three types investigated is impossible, unused sample bearings were disassembled to obtain representative data. Comparison of the before and after running data is therefore open to question inasmuch as the data compared were not obtained from the same bearing. The manufacturing tolerances of high-speed aircraft-grade bearings, however, are so close as to make it possible to draw conclusions from certain of the data presented. Although these results are self-explanatory, a few particularly interesting facts are mentioned.

(a) The only significant change in roller diameters occurred with the roller-riding cage-type bearings.

(b) Roller lengths did not change appreciably in any of the bearings investigated, and inasmuch as the axial clearance between the rollers and inner-race flanges increased in certain of the bearings, the wearing occurred at the flange contact faces.

(c) An increase in diametral clearance between the roller and cage pocket occurred in most of the bearings; however, this increase was most marked in the roller-riding cage-type bearing.

(d) Bearing and cage diametral clearances increased appreciably in each bearing that failed. This increase was very small in bearings that gave satisfactory operation.

Changes in surface finish of test-bearing component parts with running time. - Surface-finish values of the component parts for each bearing type obtained from disassembled sample bearings, as well as from disassembled test bearings 4, 5, and 6 (which are representative of the three types of bearing used in this investigation) after running, are given in table II. Comparison of these values before and after running is questionable, just as are hardness values, because the data were not obtained using the same bearing in each case. Some facts from table II worth emphasizing are:

(a) For new bearings, values of surface finish are seen to be 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 the circumferential and the axial directions was not generally so great.

(b) Even though bearings 5 and 6 had not failed, certain of the surfaces exhibit poorer finish after running than before, which is particularly true for finish in the circumferential direction.

(c) It is evident that much can be accomplished regarding closer tolerances of surface finish at the critical locations, particularly at the cage locating-surfaces and cage pockets.

Change in hardness of test-bearing component parts with running time. - The hardness values of the component parts for each bearing type (obtained from disassembled sample bearings, as well as from the disassembled test bearings 4, 5, and 6 after running) are given in table III. The following interesting facts are apparent:

(a) The rollers are the hardest parts of each of the new bearings; however, after running there is very little difference in the hardness of any of the steel parts indicating that tempering of the rollers occurs during very high-speed operation.

(b) The hardness of the cage material varies considerably not only for cages of different manufacture but also for cages of the same manufacture.

Effect of crowning of test-bearing contact surfaces. - The measurement of roller and raceway crowning is given in table IV for each of the bearing types investigated. The primary purpose of crowning in roller bearings is to relieve the edge stress at the line of contact, which may reach values as high as one and one-half times the calculated value of the mean stress. (The mean stress is approximately 0.8 of the maximum stress for line contact.) Crowning of the contact surfaces may also influence the percentage slip within roller bearings at high speeds and light loads.

It may be seen from table IV that the two-piece inner-race-riding cage-type bearing had no crowning at any of the contact surfaces, whereas the other two bearing types had roller crowning of 0.0002 to 0.0003 inch to about 1/16 inch from the ends of each roller (not including the chamfer). In order to provide maximum roller stability in a cylindrical roller bearing, it is possible that an alternative would be to crown the inner and outer race and leave the rollers uncrowned.

Temperature-speed relation as affected by load. - The data of figure 6 illustrate that at very light loads, and for the speed range covered, the increase in bearing temperature with speed is greater than linear; whereas with an appreciable load on the bearings, the increase in bearing temperature with speed is approximately linear. Also, although the test bearing is of the cylindrical-roller type and the two support bearings are of the deep-groove ball type, and all are of different sizes, the foregoing facts hold qualitatively for all three bearings. These facts indicate that at very light loads the increase in slippage within each bearing as the speed is increased causes a greater resultant friction than without slippage.

Effect of cage type. - Significant differences exist in the operating characteristics of the three types of turbine roller bearing currently used interchangeably in some aircraft gas-turbine engines. These differences become more apparent at the very high speeds. Although all three types are apparently satisfactory at the maximum DN values currently encountered, a greater factor of safety would result under critical conditions if the rear turbine bearing of this engine were restricted to the one-piece inner-race-riding cage-type bearing because of its more reliable performance.

The main disadvantage of the roller-riding cage-type bearing is that excessive roller and cage-pocket wear occurs at high speeds, eventually causing the cage to be located on the inner race; however, inasmuch as this bearing type has inherent advantages over the inner-race riding-cage type bearing, it may be possible to refine the design and manufacturing techniques of this cage-type bearing so that wear at high speeds would be materially reduced. Improvements in this direction may be realized by closer tolerances regarding pocket spacing, pocket clearance, pocket surface finish, by balancing the assembled cage, and by decreasing the axial clearance between the rollers and the race flanges.

For applications involving very lightly loaded high-speed roller bearings (maximum load of about 100 lb), such as the idler rotor bearing in one aircraft gas-turbine engine, an improved roller-riding cage-type bearing may prove advantageous (over an inner-race riding-cage type bearing) due to its inherent characteristic of operating at lower temperatures under these conditions.

An improved roller-riding cage-type bearing may operate with less cage pickup than an inner-race-riding cage-type bearing during engine starting under faulty lubricating conditions inasmuch as the cage assembly of the roller-riding cage-type bearing is driven only by roller traction, whereas the cage speed of the inner-race-riding cage-type bearing is influenced by surface contact between the inner race and cage in addition to roller traction.

It would be advantageous to so design the cages of all high-speed bearings as to induce the flow of lubricant to the areas where it is most needed, that is, to the cage-locating surface, the roller ends, and the roller pockets. No attempt has been made to incorporate such design considerations in any of the cage types investigated herein. The addition of expediently designed chamfers, grooves, and passageways may enhance the lubricating and cooling efficiency of a given quantity of oil. Such modifications are recommended for future high-speed bearings.

Effect of slip on total roller load. - It is shown in reference 4 that the life of a rolling-contact bearing is appreciably decreased in the high-speed range due to the load imposed by the action of centrifugal force on the rolling elements. This load, which acts in addition to the applied external load, is calculated by Newton's second law with the centripetal acceleration calculated from the theoretical cage speed. A plot showing this theoretical load is given for roller bearings in figure 18. Under conditions at which the effects of centrifugal load are significant (that is, at high speeds), the experimental results reported herein indicate that a considerable percentage of slip occurs between the rollers and the raceways. The slip causes the centrifugal load to be appreciably less than that calculated from theoretical cage speed because its value is proportional to the square of cage speed. Inasmuch as many high-speed bearings are run at light loads, the effect of slippage within the bearing is quite important due to the reduction in centrifugal force, which results in a reduction in the net bearing load, as the slippage increases. As an example, it may be seen that at a DN value of 1.52×10^6 (20,250 rpm) the theoretical centrifugal force is (from fig. 18) approximately 180 pounds, whereas the actual centrifugal force with 65 percent slip (see point A, fig. 14) is determined from figure 18 by using an equivalent shaft speed N_s', where

$$N_s' = \frac{(100\text{-percent slip})}{100} \; (N_s)$$

where

N_s actual shaft speed, rpm

In this case, the actual centrifugal load is approximately 22 pounds or only 12.2 percent of the theoretical value. The difference between the theoretical and actual load is 158 pounds or 42 percent of the external load, which in this case is the gravity load of 368 pounds on this size bearing in an aircraft gas-turbine engine. It is, therefore, seen that at high speeds and light loads the slippage within the bearing is significant regarding the life of the bearing inasmuch as the life varies approximately as the cube of total load (reference 5).

Effect of slip on relative velocities. - Slippage within the bearing is also of interest because the relative surface speed at the cage-locating surface is a function of the cage speed.

In a bearing having a roller-riding cage, only one source of sliding friction exists at the cage surfaces and that is at the cage pockets. In a bearing having an inner-race-riding cage, however, the roller pockets and the surfaces between the cage and the inner race are the sources of cage friction. For the case of the roller-riding cage, the relative velocity at the cage-locating surface (that is, at the cage pockets) decreases as the percentage slip increases, inasmuch as the rollers do not rotate about their respective axes as fast with slippage as when no slippage occurs. For the inner-race-riding cage, a similar decrease in relative velocity occurs at the cage pockets with an increase in slip in addition however, the relative velocity at the cage-locating surface (that is, between the cage and the inner race) increases as the slip increases inasmuch as the inner-race surface speed remains constant, whereas the cage-surface speed decreases. This fundamental difference of the two cage designs may account, in part, for the large difference in operating temperatures in the high speed, low-load range (fig. 13). It is seen, however, that for loads above 400 pounds the bearings operate at more nearly the same temperatures, which indicates that as load is applied the effects of the different cage types are not as significant.

Inasmuch as the cage-locating surface of a bearing having an inner-race-riding cage is a plain bearing of very small length-diameter ratio (about 0.03 to 0.04 as compared to approximately 1 in a sleeve bearing), little oil-film-load capacity is available and it is believed that sliding in the boundary region of lubrication occurs at this surface. Design changes resulting in hydrodynamic lubrication at this locating surface at high speeds should give increased cage life and operating reliability.

<u>Hydrodynamic lubrication of roller bearings</u>. - That a hydrodynamic film may exist between rollers and races in a roller bearing is shown by Büche (reference 6), who applied the hydrodynamic theory of lubrication to roller bearings. Gatcombe (reference 7) applied the hydrodynamic theory of lubrication to spur gears and found that high hydrodynamic pressures are developed between two disks rolled together in oil. The results included herein with roller bearings that operated at less than the theoretical cage speed indicate that the rollers may be separated from the races by a film of oil inasmuch as negligible roller wear was in evidence in the bearings that did not fail.

Effect of circumferential temperature distribution. - The exact effect of the circumferential temperature gradient of the outer race, as evidenced in figures 10, 11, and 17, is unknown; however, it either causes the outer race to become out of round or causes thermal stresses in the outer race and housing. The magnitude of the noncircularity may be estimated by calculating the expansion between diametrically opposed points on the circumference of the outer race assuming that the housing and the outer race both expand according to the circumferential temperature distribution at the interface between the two parts. A qualitative picture of the distortion occurring under such circumstances, is evident in figures 10, 11, and 17.

Effect of differential expansion upon running clearance. - The difference between running clearance and mounted clearance depends upon the relative differential expansion between the inner race, the rollers, and the outer race. If it is assumed; that the mean roller temperature is the average of the mean outer-race temperature and the inner-race temperature, that the outer race and its housing both expand according to the circumferential temperature distribution at the interface between the two parts, and that the circumferential temperature gradient about the inner race is slight and therefore that the single thermocouple gives the inner-race temperature, then the running clearance may be estimated from the data given herein. Inasmuch as the first two assumptions are open to considerable conjecture, calculated running clearances are not included herein.

The data obtained from an aircraft gas-turbine engine or other similar applications may give results far different than the foregoing results regarding estimated running clearance because the rotor of a jet engine is considerably warmer than the outer-race housing.

SUMMARY OF RESULTS

From the experimental investigation and analysis of the results of 75-millimeter-bore (size 215) cylindrical roller bearings operated over a range of DN values (product of bearing bore in mm multiplied by shaft speed in rpm) from 0.3×10^6 to 1.65×10^6 under loads, from 7 to 1613 pounds, and with a jet-type circulatory oil feed, the following results were obtained:

1. Bearing operating temperatures were much more sensitive to changes in speed than to changes in load except in the low-load range. The temperature of a loaded roller bearing was found to increase

approximately linearly with an increase in speed; however, the temperature of an unloaded roller bearing increased at a rate greater than linear presumably due to slippage within the bearing at the higher speeds.

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2. Of the seven bearings investigated, three incipient failures occurred. Two failures were of roller-riding cage-type bearings and the other of an inner-race-riding cage-type bearing. The maximum DN value reached was 1.4×10^6 for the first failures and 1.65×10^6 for the other failure.

3. The test bearings were found to undergo changes in operating characteristics with running time in that at a given operating condition the inner- and outer-race temperatures and the cage speed varied as the operation progressed.

4. The operating temperatures of the three types of bearing were found to differ most in the low-load, high-speed range where the roller-riding cage-type bearing exhibited significantly lower operating temperatures than the one- and two-piece, inner-race-riding cage-type bearings. The operation of the roller-riding cage-type bearing was considerably rougher, and the bearing showed prohibitive roller and cage wear after relatively short high-speed operation (DN values over 1×10^6), as compared to the inner-race-riding cage-type bearings.

5. In general, the percentage of slip within the bearing increased with an increase in DN value and decreased with increase in load, reaching average values greater than 60-percent slip at DN values from 1.35×10^6 to 1.65×10^6.

6. Under certain operating conditions, namely moderate speeds and loads, (that is, DN values in the range 0.3×10^6 to 0.7×10^6 and loads in the range 100 to 1100 lb), the inner-race-riding cage-type bearing operated with a cage speed greater than the theoretical value, which indicated that the cage and rollers were driven by the cage-locating surface rather than the cage being driven by the rollers. This condition was not observed for an equivalent bearing having a roller-riding cage.

7. A circumferential temperature gradient existed around the outer race of the turbine roller bearing of an aircraft turbine engine; this gradient was qualitatively similar to that obtained in the bearing rig in that the maximum temperature occurred in the region 270° to 300° after the oil-jet location; whereas the minimum temperature occurred in the region 60° to 90° after the oil-jet location in the direction of shaft rotation.

8. Although considerable slippage occurs at high speeds and light loads for the cylindrical roller bearings investigated, there was little evidence of roller wear in the bearings that did not fail. It is therefore postulated that there may exist a hydrodynamic film of oil between rollers and raceways under such operating conditions. 1273

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, October 24, 1949.

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TABLE I - PHYSICAL CHARACTERISTICS OF TEST BEARINGS

Bearing number	1		2		3		4		5		6		7(a)	
Construction	Two-piece inner-race-riding cage		One-piece inner-race-riding cage		Two-piece roller-riding cage		Two-piece roller-riding cage		Two-piece inner-race-riding cage		One-piece inner-race-riding cage		One-piece inner-race-riding cage	
Number of rollers	17		18		17		17		17		18		18	
Roller $\frac{l}{d}$ ratio	1		1		1.237		1.237		1		1		1	
Pitch diameter of bearing (in.)	4.031		4.036		4.0295		4.0295		4.031		4.036		4.036	
	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After
Total running time (hr)	0	8.4	0	17	0	17.3	0	16.7	0	133.5	0	195.9	0	15.8
$\Sigma\Delta$ temp. (°F) x time (min)	0	0.59×10^5	0	0.63×10^5	0	0.92×10^5	0	0.68×10^5	0	11.79×10^5	0	11.64×10^5	0	-----
Roller diameter (in.)	[b]0.5625	-----	[b]0.5613	0.5511	[b]0.5255	0.5210	[b]0.5255	0.5136	[b]0.5625	0.5624	[b]0.5513	0.5510	[b]0.5513	-----
Roller length (in.)	[b]0.5625	-----	[b]0.5510	0.5505	[b]0.6500	0.6500	[b]0.6500	0.6496	[b]0.5625	0.5625	[b]0.5510	0.5507	[b]0.5510	-----
Diametral clearance between cage and roller (in.)	[b]0.0097	-----	[b]0.0087	0.0159	[b]0.0099	0.0170	[b]0.0099	0.0204	[b]0.0097	0.0082	[b]0.0087	0.010	[b]0.0087	-----
Axial clearance between roller and inner-race flange (in.)	[b]0.003	-----	[b]0.002	0.002	[b]0.0014	0.010	[b]0.0014	0.004	[b]0.003	0.005	[b]0.002	0.002	[b]0.002	-----
Axial clearance between roller and cage (in.)	[b]0.011	-----	[b]0.007	0.015	[b]0.017	0.013	[b]0.017	0.017	[b]0.011	0.011	[b]0.007	0.009	-----	-----
Unmounted bearing: [c]Diametral clearance (in.) Bearing	0.0016	0.0017	0.0018	0.0025	0.0019	0.011	0.0020	0.026	0.0018	[d]0.0015	0.0020	0.0022	0.0017	0.0018
Cage	.013	.014	.019	.043	[b].024 to .048	-----	.024 to .048	.070	.011	.012	.015	.018	.012 to .016	.015 to .016
[e]Eccentricity (in.)	.0000	.0002	.0002	-----	.0001	-----	.0001	-----	.0001	.0001	.0000	.0002	.0001	.0001
Mounted bearing: [f]Diametral clearance (in.) Bearing	0.0002	0.0008	0.0005 to .0006	-----	0.0007	0.012	0.0010	0.0250	0.0005	0.0002	0.0009	0.0009	-----	-----
Cage	-----	-----	.018	.042	[b].023 to .041	-----	.023 to .041	.070	.010	.011	.014	.017	-----	-----
[g]Eccentricity (in.)	-----	-----	.0003	.0003	.0004	-----	.0004	-----	.0005	-----	.0005	.0004	-----	-----
Remarks	Satisfactory operation		Incipient failure at surfaces between cage and inner race		Incipient failure due to excessive roller and cage pocket wear		Incipient failure due to roller and cage pocket wear		Unusually hot operation but no failure		Satisfactory operation		Reinstalled in engine	

[a]Bearing used in engine.
[b]Measurements obtained from sample bearing.
[c]Measurement obtained in fixture with dial gage.
[d]Diametral clearance actually decreased due to apparent growth of inner race.
[e]Measurement obtained in fixture with dial gage, inner race rotating and outer race stationary.
[f]Measurements obtained as mounted in test rig with dial gage.
[g]Measurements obtained as mounted in test rig with dial gage, inner race rotating, and outer race stationary.

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

Bearing number			4		5		6	
Construction			Two-piece roller-riding cage		Two-piece inner-race-riding cage		One-piece inner-race-riding cage	
			[b]Before	After	[b]Before	After	[b]Before	After
Total running time (hr)			0	16.7	0	133.5	0	195.9
$\Sigma\Delta$ Temp. ($^{\circ}$F) $\times$ time (min)			0	0.68×10^{5}	0	11.79×10^{5}	0	11.64×10^{5}
Outer-race track		Axial	5-6	7-8	3-3.5	7-8	10-11	6-7
		Circumferential	3.5-4.0	8-9	2-2.5	4-5	3-4	4-5
Inner-race	Track	Axial	5-6	6-8	10-11	4-5	6.5-7.5	6-7
		Circumferential	3-5	[c]1.5-2 5-7 and 110-130	3-5	3-3.5	3-4	3-4
	Lands	Axial	11-14	10-12	[d]10-11 13-17	[d]10-12 25-30	[d]10-12 8-9	[d]50-70 13-16
		Circumferential	4-6	3.5-5.5	14-16 7-8	2-2.5 13-18	3-5 3-4	3-4 3-4
Rollers		Axial	7	8-15	2	3.5-4	3-3.5	3
		Circumferential	3-3.5	12-14	1.5	2	3	2.5
		Ends	8-10	10-12	10-16	10-15	4-5	2-4
Cage	Locating surface	Axial	-------	-------	25-30	40-50	30-40	[d]40-60 70-80
		Circumferential	-------	-------	15-20	10-15	20-25	15-20 60-70
	Pocket	Axial	30-35	20-25	19-21	[e]80-90 25-35	18-22	15-20
		Circumferential	15-20	18-22	13-15	35-40 20-25	18-22	10-15

[a]Surface finish measured in microinches, rms.
[b]Measurement obtained from sample bearings.
[c]Surface finish at three representative surfaces.
[d]Surface finish for the two cage-locating surfaces.
[e]Surface finish from same cage pocket for trailing and for leading sides.

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TABLE III - HARDNESS OF TEST-BEARING COMPONENT PARTS

Bearing number		4		5		6	
Construction		Two-piece roller-riding cage		Two-piece inner-race riding cage		One-piece inner-race-riding cage	
		[a]Before	After	[a]Before	After	[a]Before	After
Total running time (hr)		0	16.7	0	133.5	0	195.9
$\Sigma\Delta$ temp. (°F) × time (min)		0	0.68×10^5	0	11.79×10^5	0	11.64×10^5
Hardness of	Outer race (Rockwell C-scale)	59-60	58-60	61-62	60-61	59-60	59-60
	Inner race (Rockwell C-scale)	58-59	58-59	59-60	58-59	58-59	57-60
	Rollers (Rockwell C-scale)	60-61	57-58	60-63	57-59	62-63	59-60
	Cage (Rockwell B-scale)	51-54	[b]58-62	65-67	[b]49-52	50-53	[b]47-50

[a]Measurement obtained from sample bearing.
[b]Hardness at several cage locations was found to be within this range.

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TABLE IV - CROWNING OF TEST-BEARING CONTACT SURFACES

Construction	Two-piece roller-riding cage	Two-piece inner-race-riding cage	One-piece inner-race-riding cage
Outer-race crowning	None	None	None
Inner-race crowning	Edge of race track 0.00005 - 0.0001 inch high	None	None
Roller crowning	0.0002 - 0.0003 inch (to about 1/16 in. from each end)	None	0.0002 - 0.0003 inch (to about 1/16 in. from each end)

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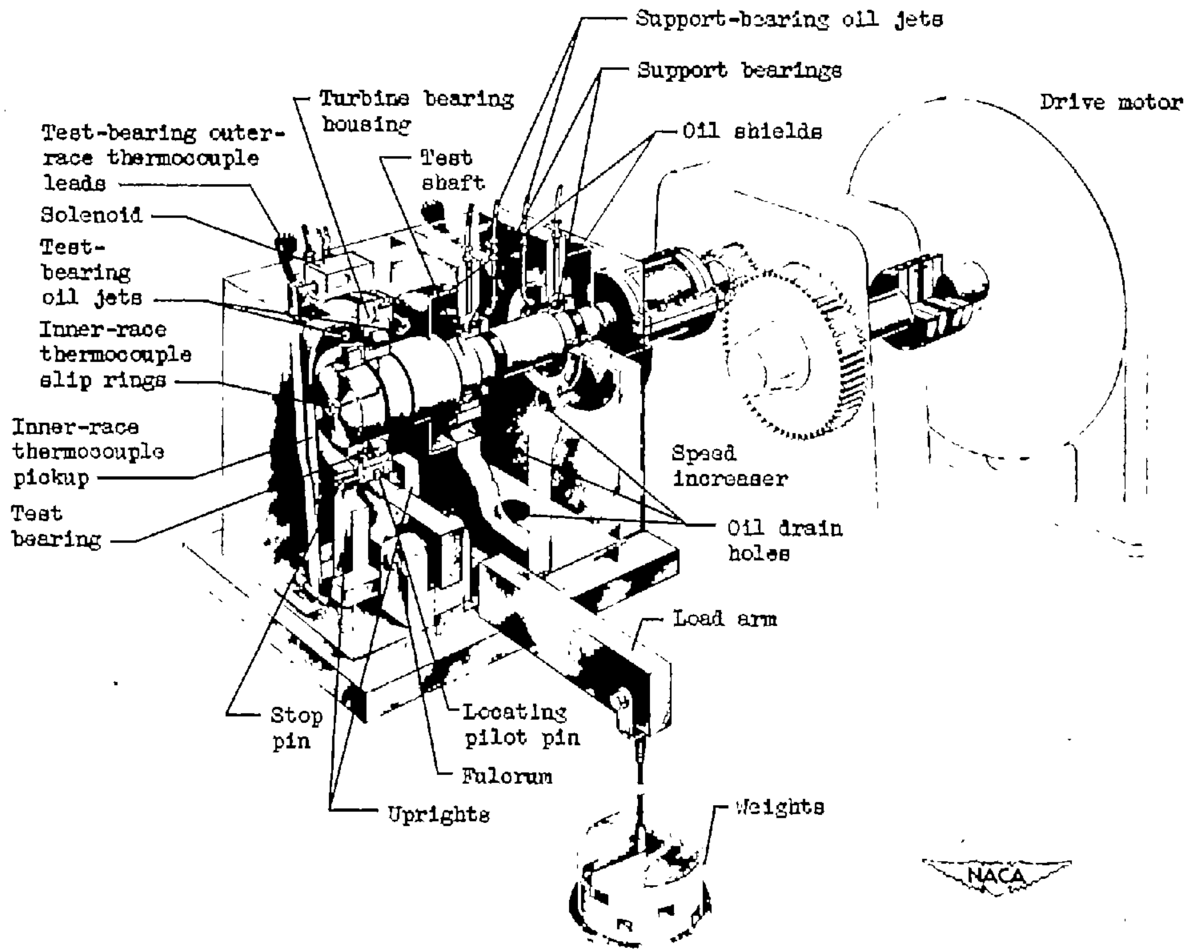


Figure 1. - Cutaway view of radial-load rig.

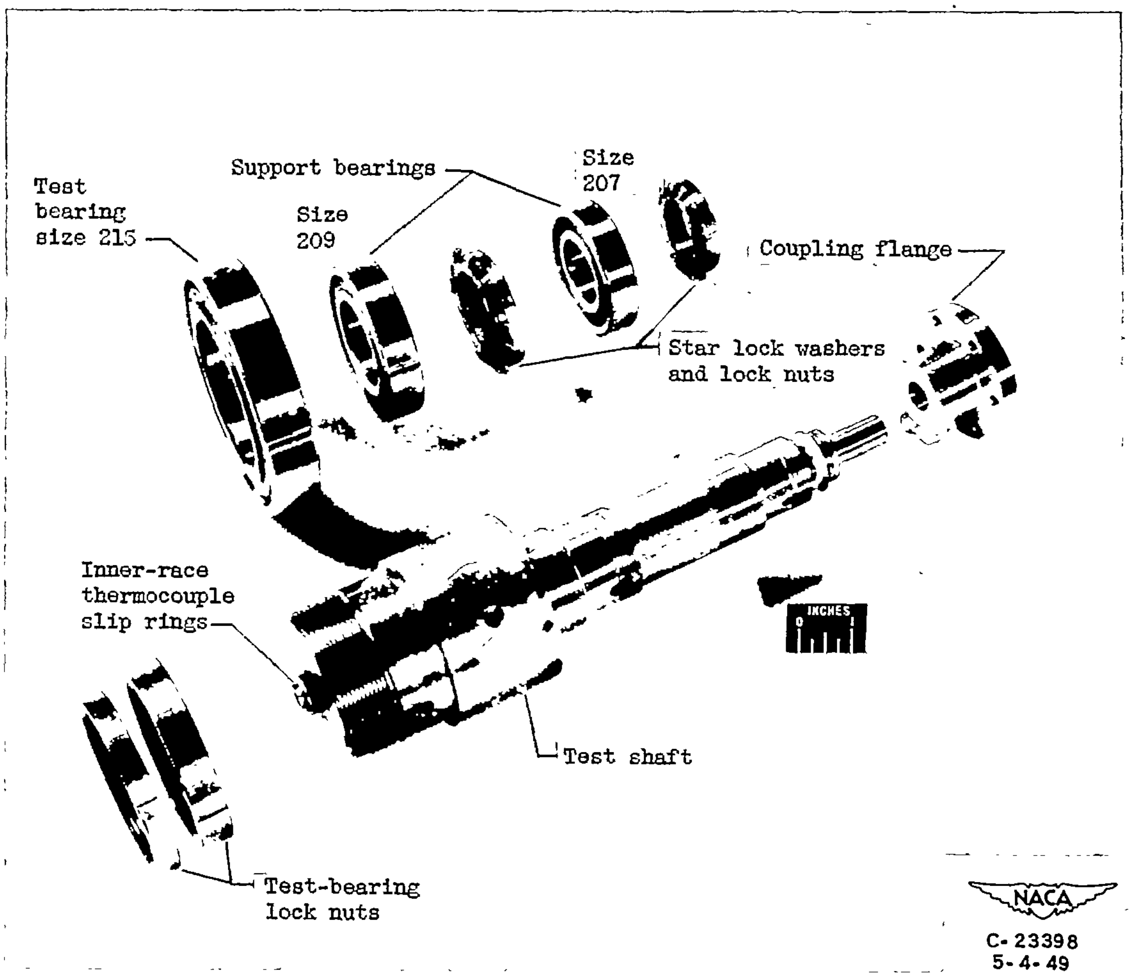
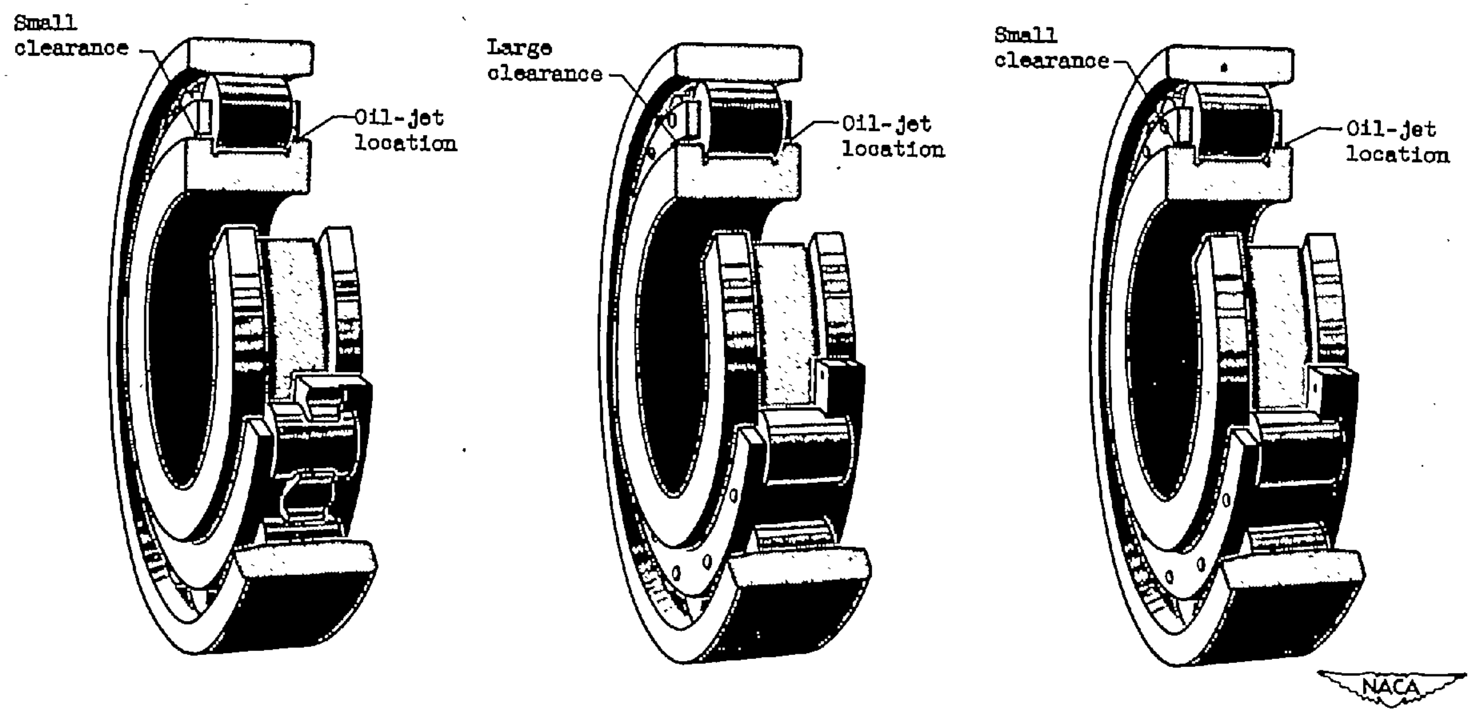


Figure 2. - Test-shaft assembly.



(a) One-piece inner-race-riding cage.

(b) Two-piece riveted roller-riding cage.

(c) Two-piece riveted inner-race-riding cage.

Figure 3. - Schematic diagram of test bearings, emphasizing differences in three types of cage construction.

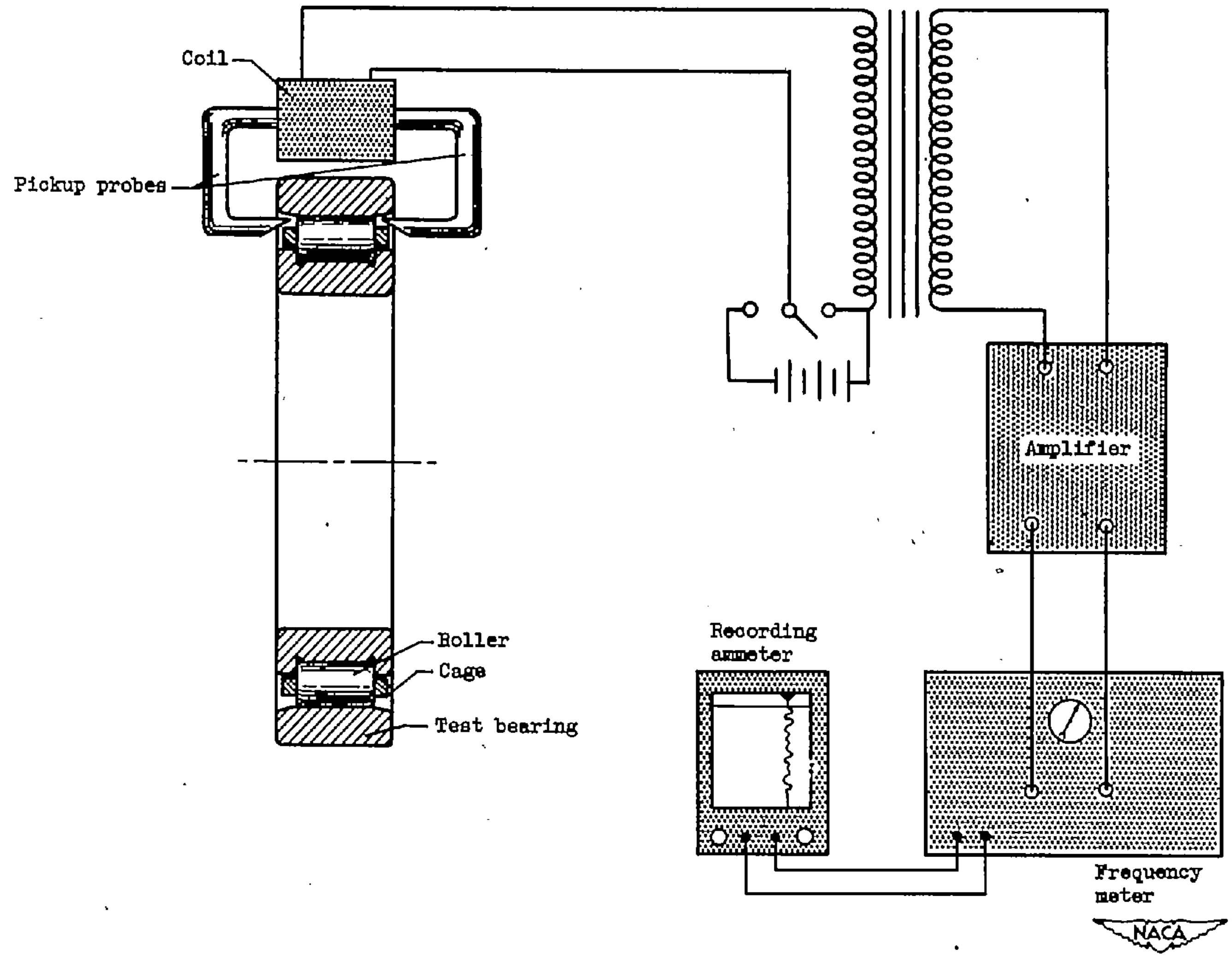


Figure 4. - Magnetic pickup circuit for measuring cage speed.

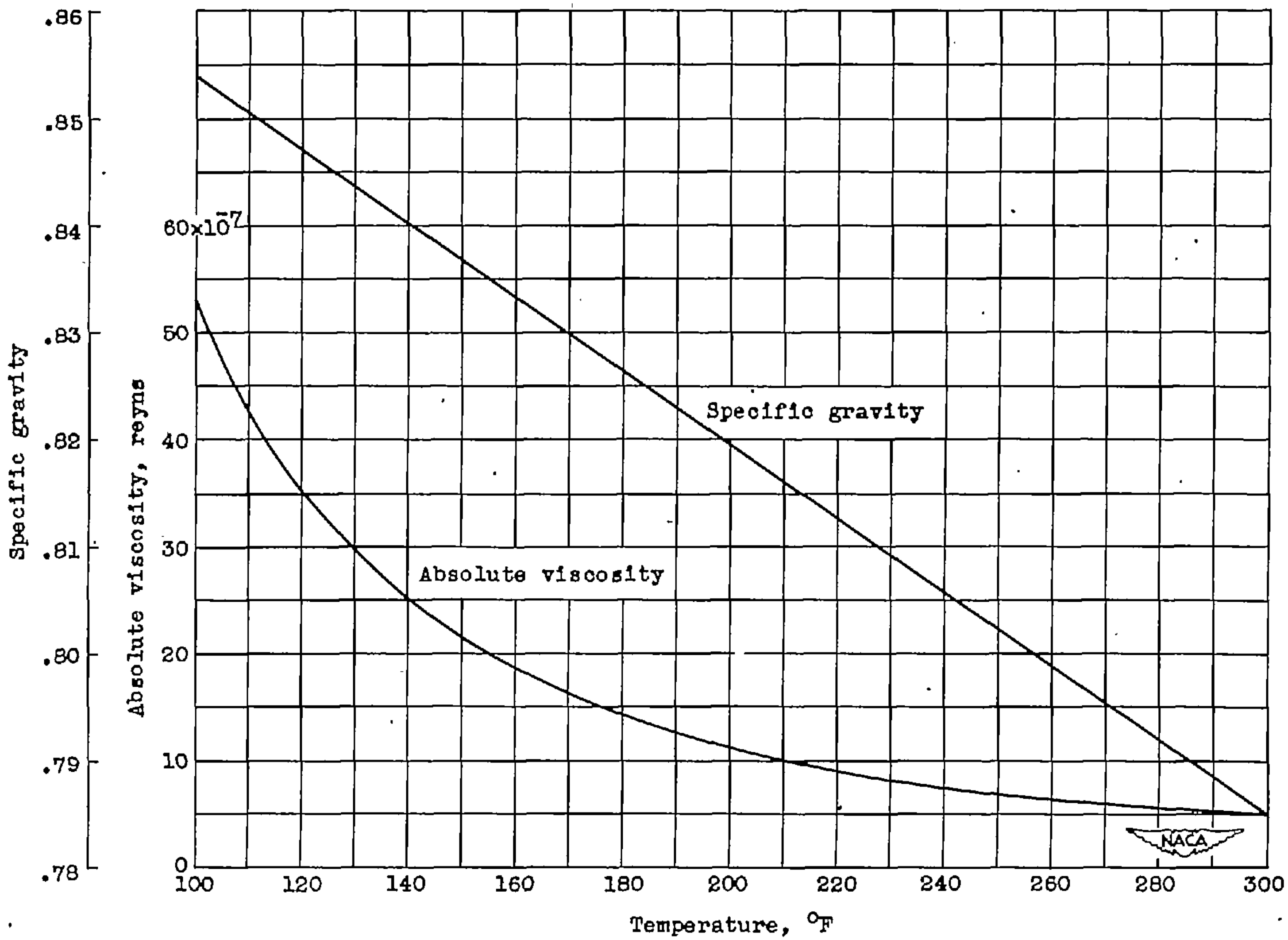
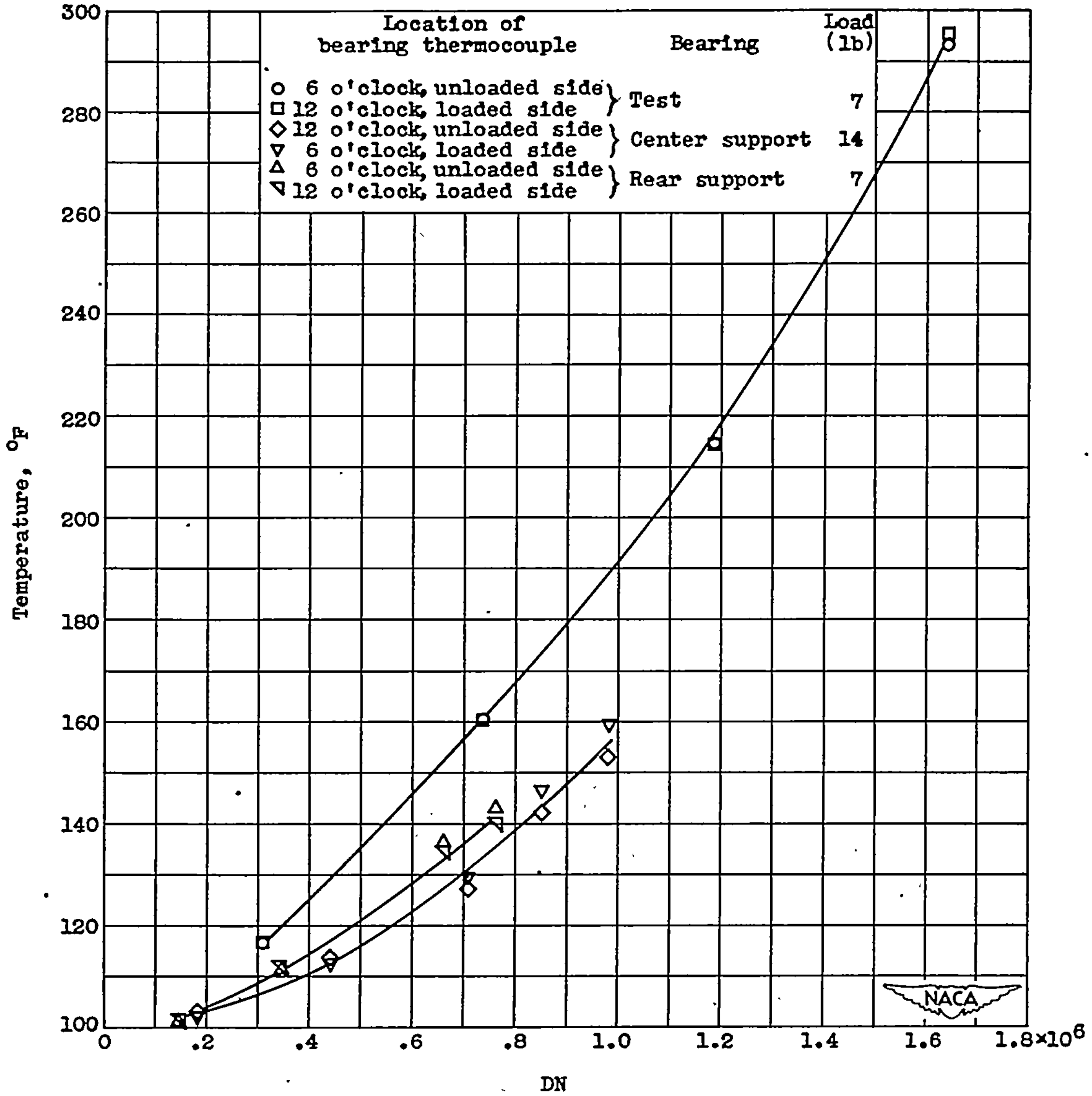


Figure 5. - Absolute viscosity and specific gravity of oil. Pour point, -50° F; flash point, 310° F; viscosity index, 150.



(a) Test-bearing load, 7 pounds.

Figure 6. - Effect of speed and load on operating temperatures of bearing 1 and representative support bearings. Oil flow for each bearing, 8 pounds per minute; oil-jet diameter, 0.180 inch; oil inlet pressure, 10 pounds per square inch; oil inlet temperature, 100° F. Oil jet located at 12 o'clock.

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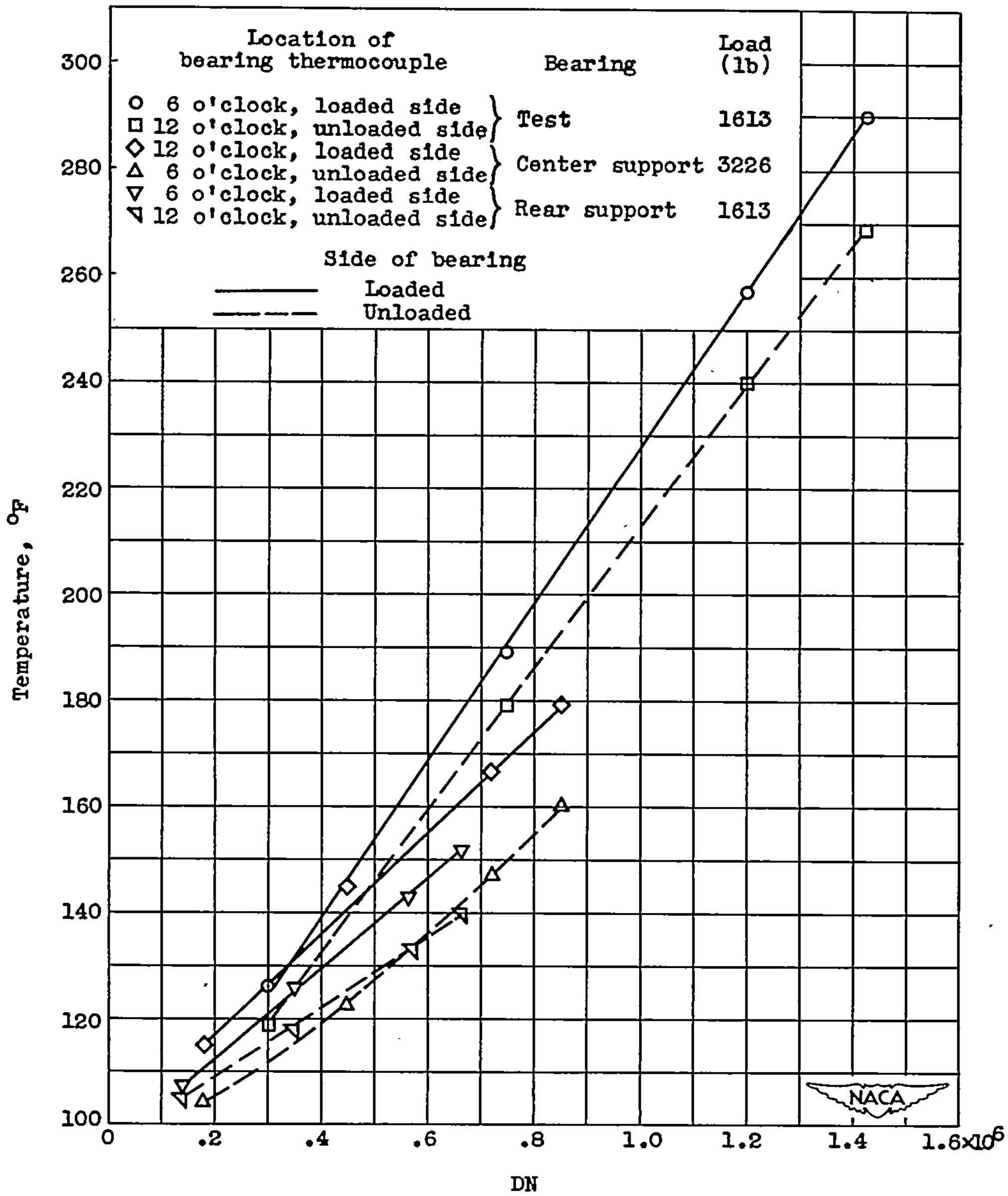


(b) Test-bearing load, 1613 pounds.

Figure 6. - Concluded. Effect of speed and load on operating temperatures of bearing 1 and representative support bearings. Oil flow for each bearing, 8 pounds per minute; oil-jet diameter, 0.180 inch; oil inlet pressure, 10 pounds per square inch; oil inlet temperature, 100° F. Oil jet located at 12 o'clock.

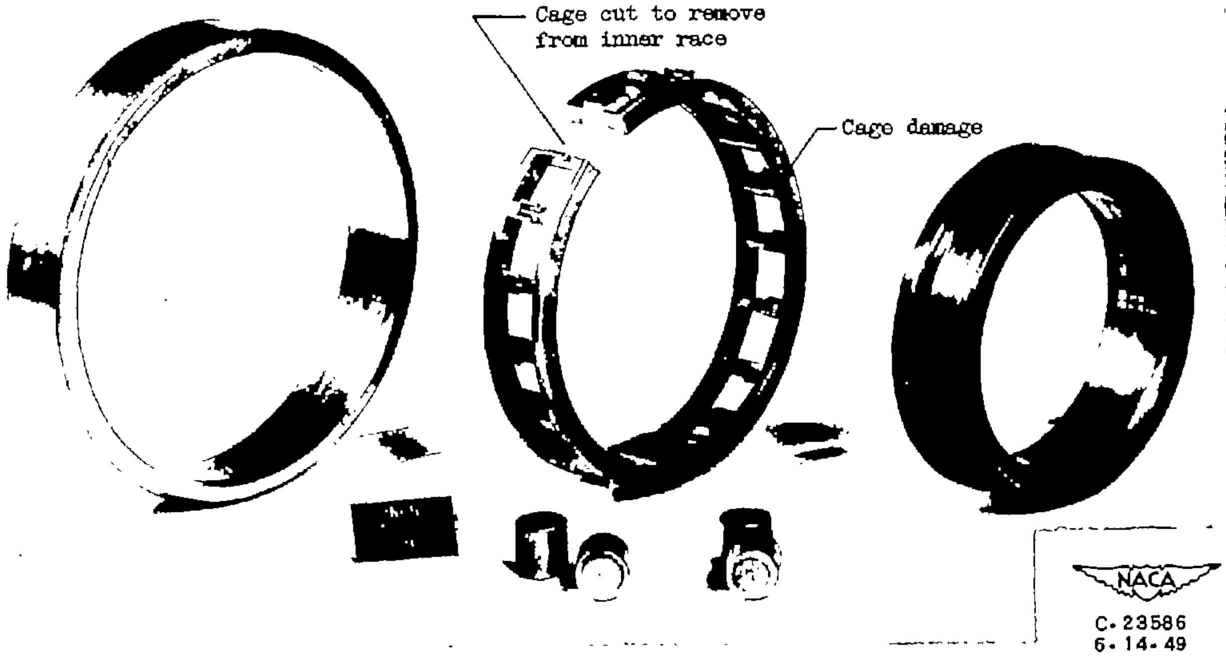


Figure 7. - Roller bearing 2 after investigation.

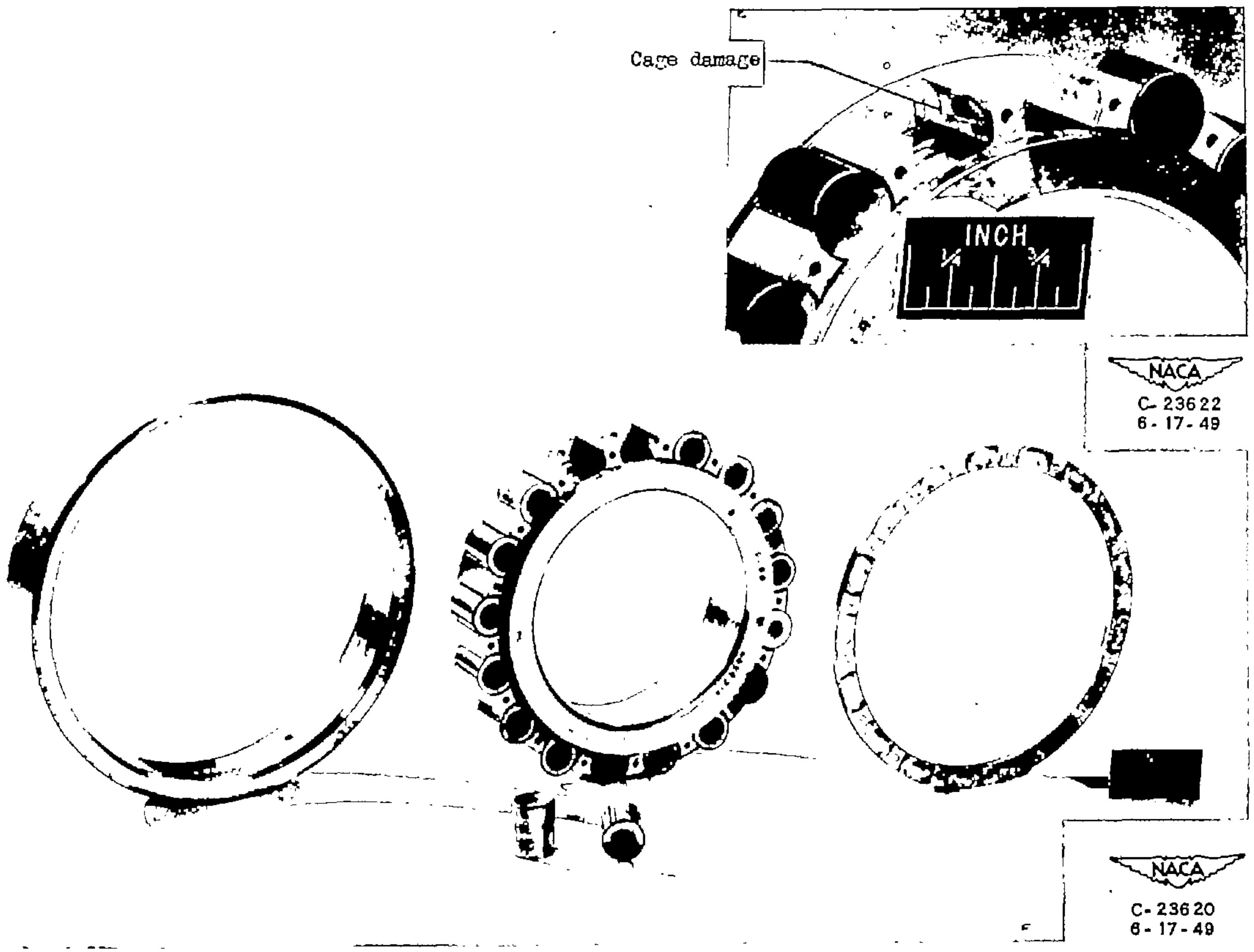


Figure 8. - Roller bearing 4 after investigation.

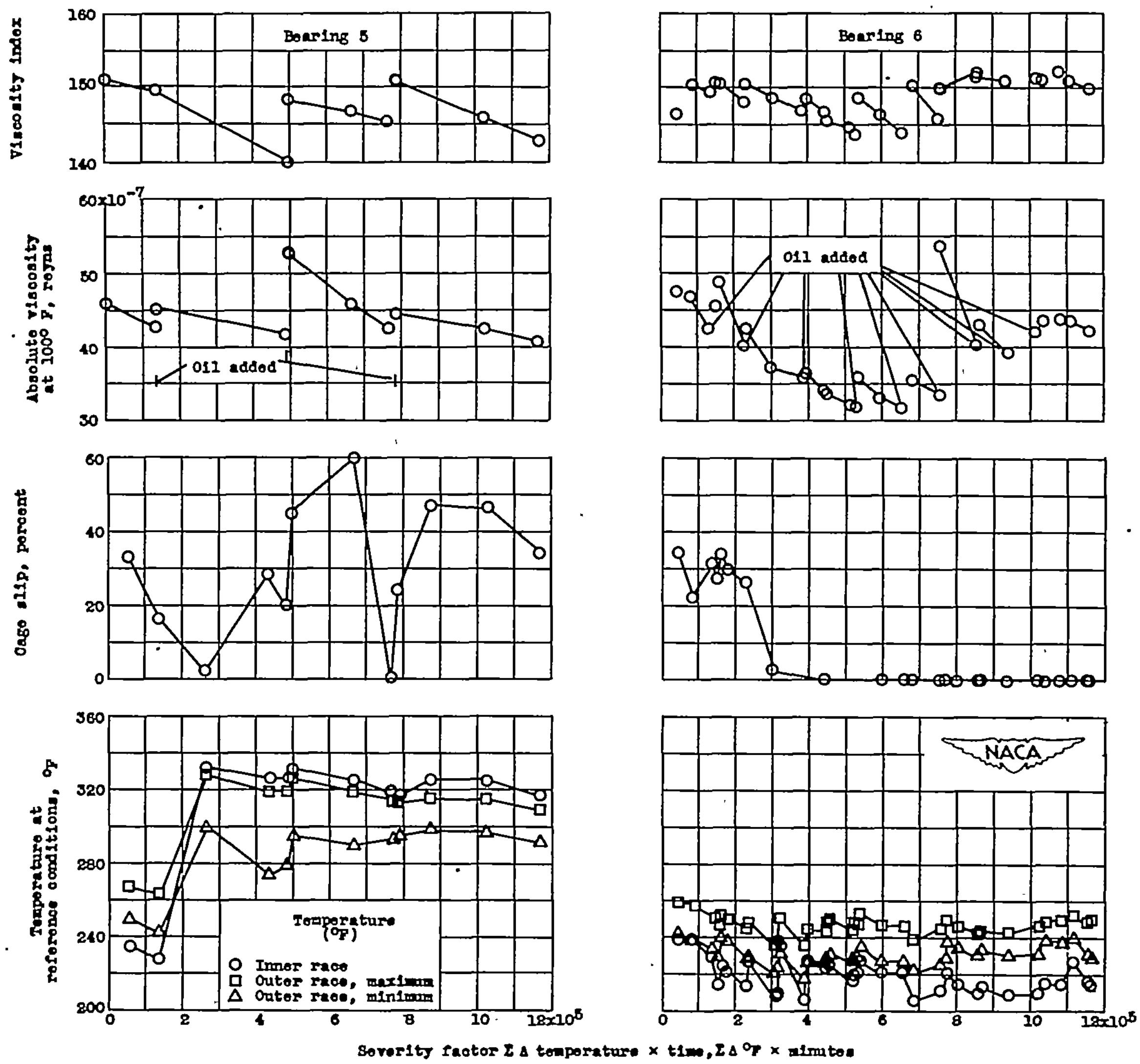
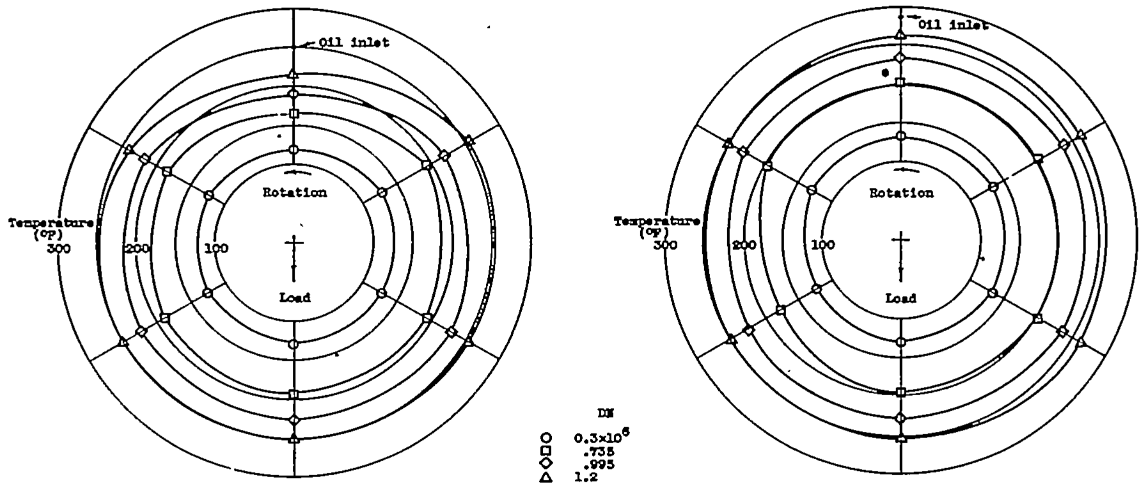
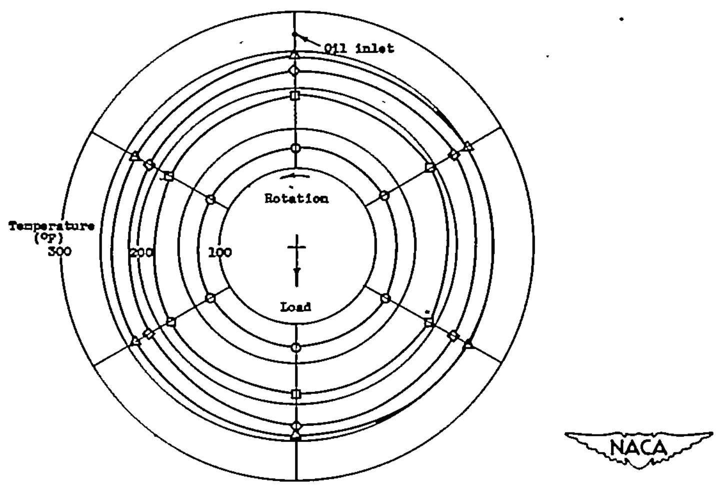


Figure 9. - Effect of severity factor on bearing operating characteristics and lubricant properties for bearings 5 and 6. DN, 1.2×10^6; load, 368 pounds; oil-jet diameter, 0.089 inch; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F.



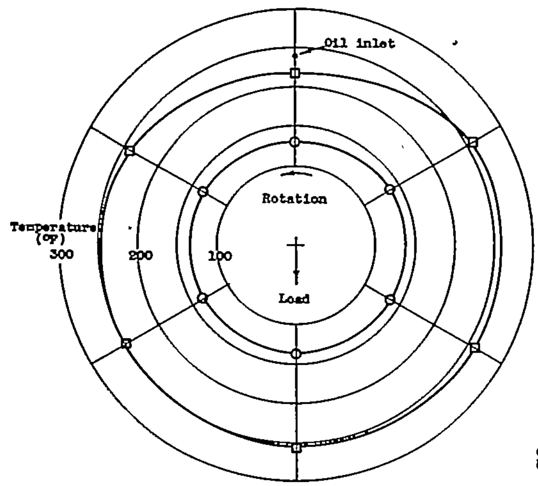
(a) Bearing 4; two-piece roller-riding cage.

(b) Bearing 5; two-piece inner-race-riding cage.

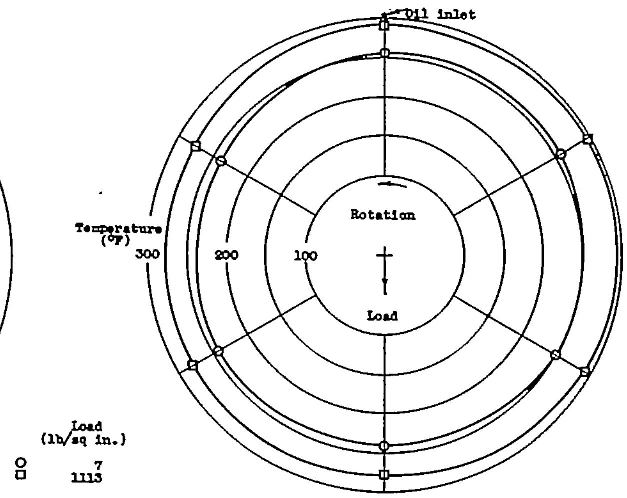


(c) Bearing 6; one-piece inner-race-riding cage.

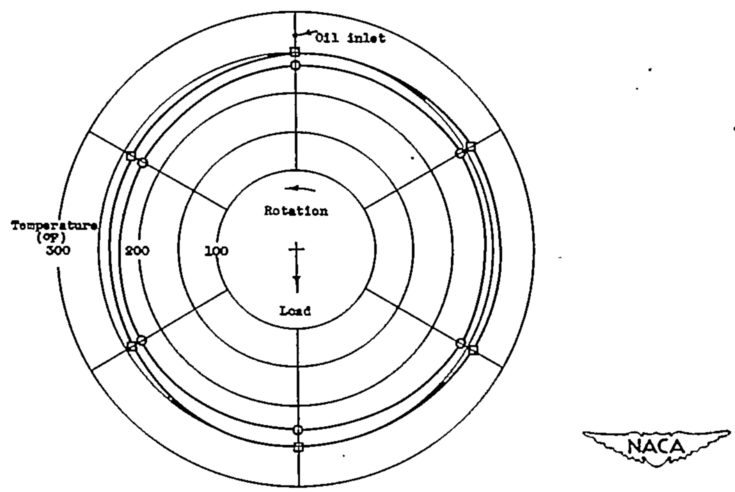
Figure 10. - Outer-race circumferential temperature distribution for bearings 4, 5, and 6 for DN values from 0.3×10^6 to 1.2×10^6. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.



(a) Bearing 4; two-piece roller-riding cage.



(b) Bearing 5; two-piece inner-race-riding cage.



(c) Bearing 6; one-piece inner-race-riding cage.

Figure 11. - Outer-race circumferential temperature distribution for bearings 4, 5, and 6 for loads of 7 and 1113 pounds. DN value, 1.2×10^6; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.

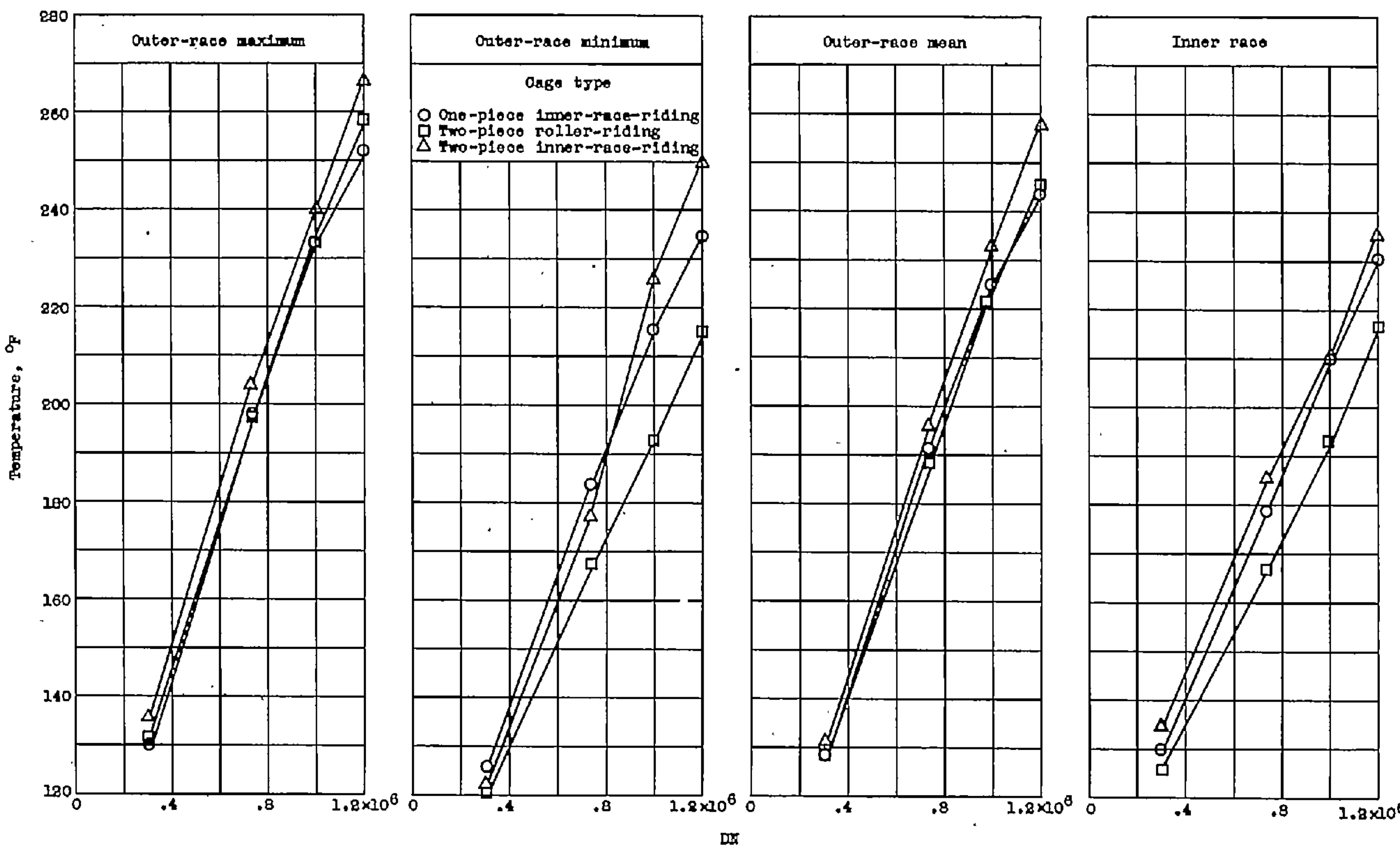


Figure 12. - Effect of DN on operating temperatures for bearings 4, 5, and 6 with two-piece roller-riding cage, two-piece inner-race riding cage, and one-piece inner-race riding cage, respectively. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet temperature, 100° F.

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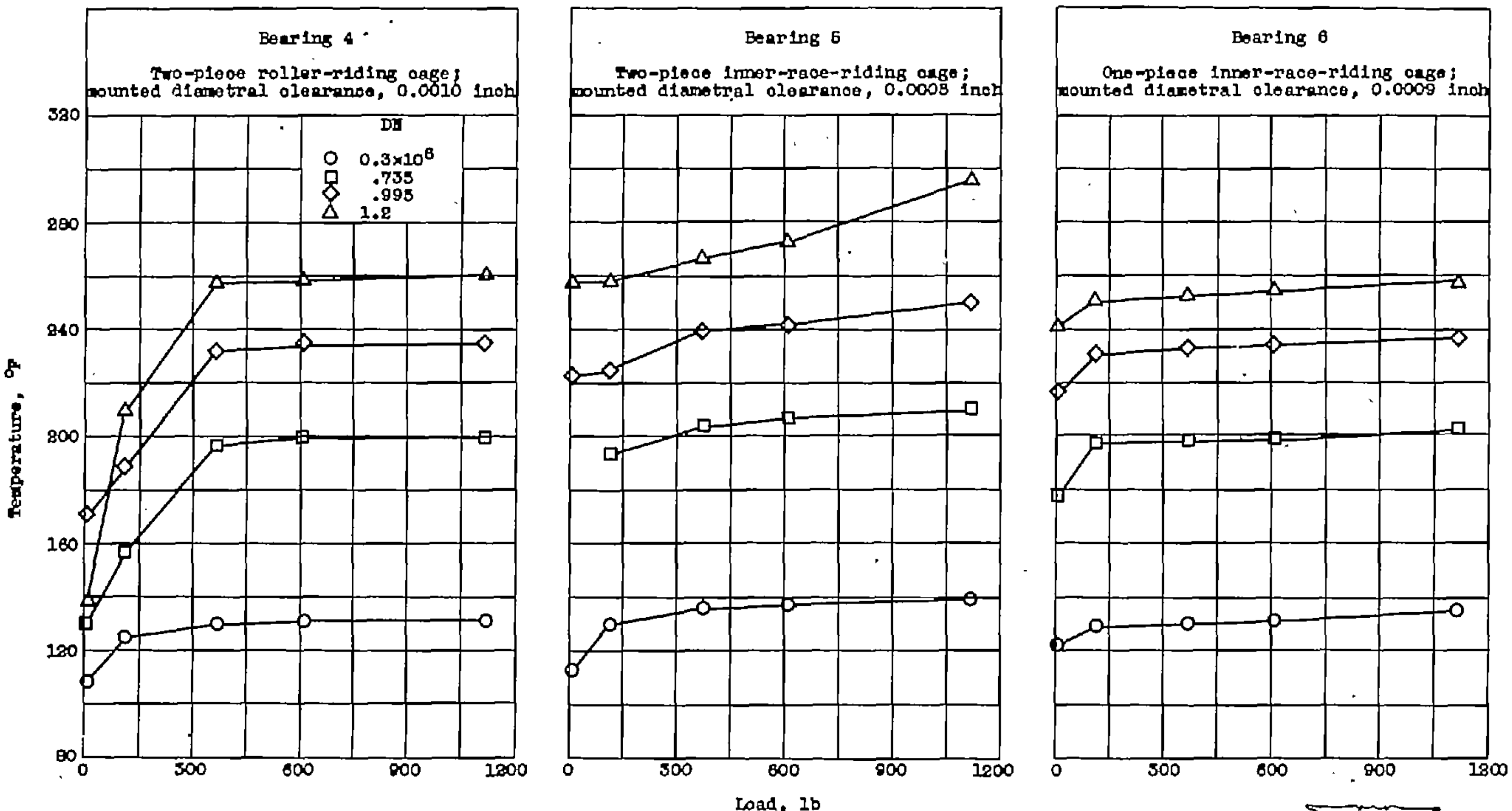
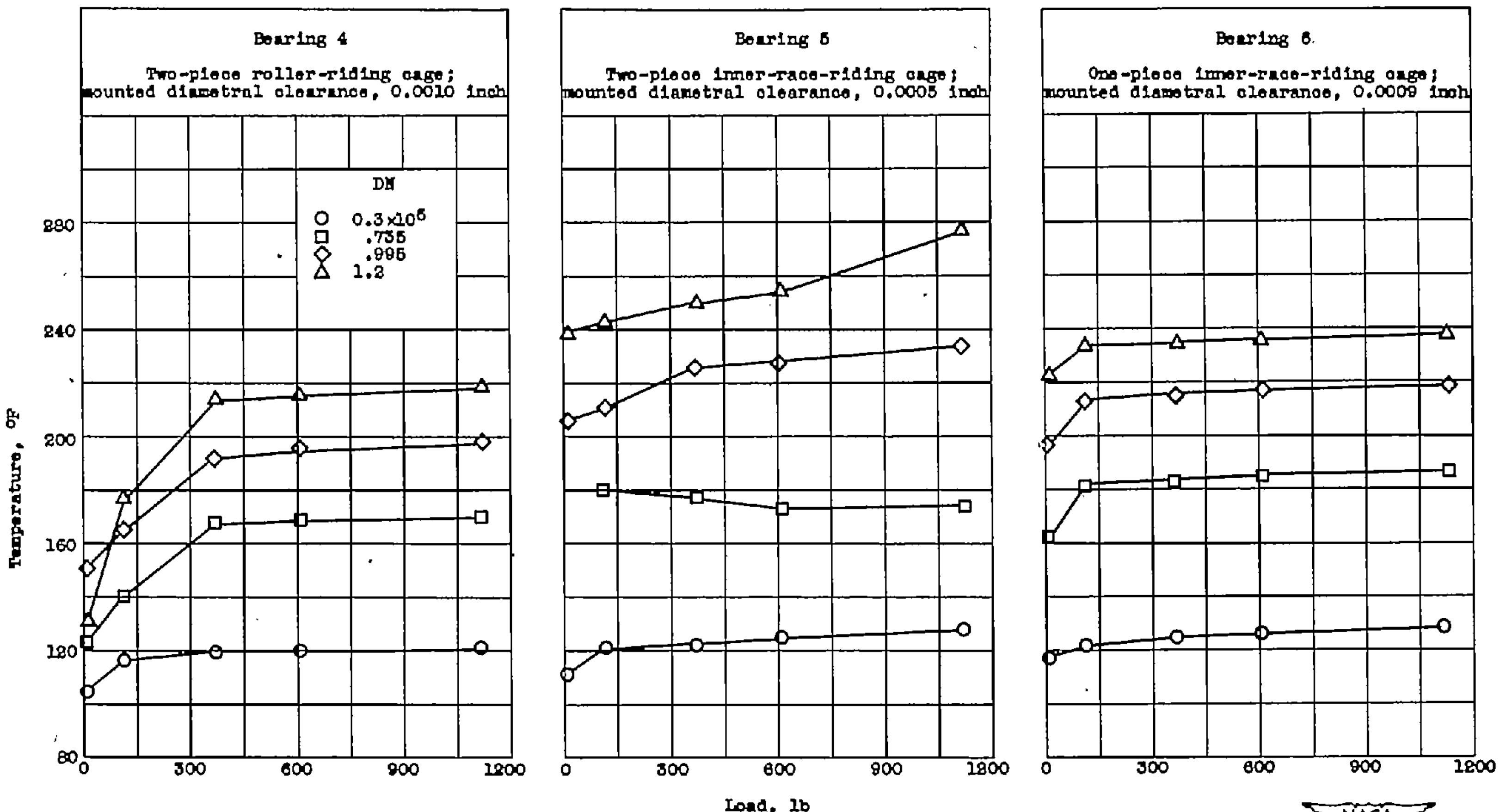


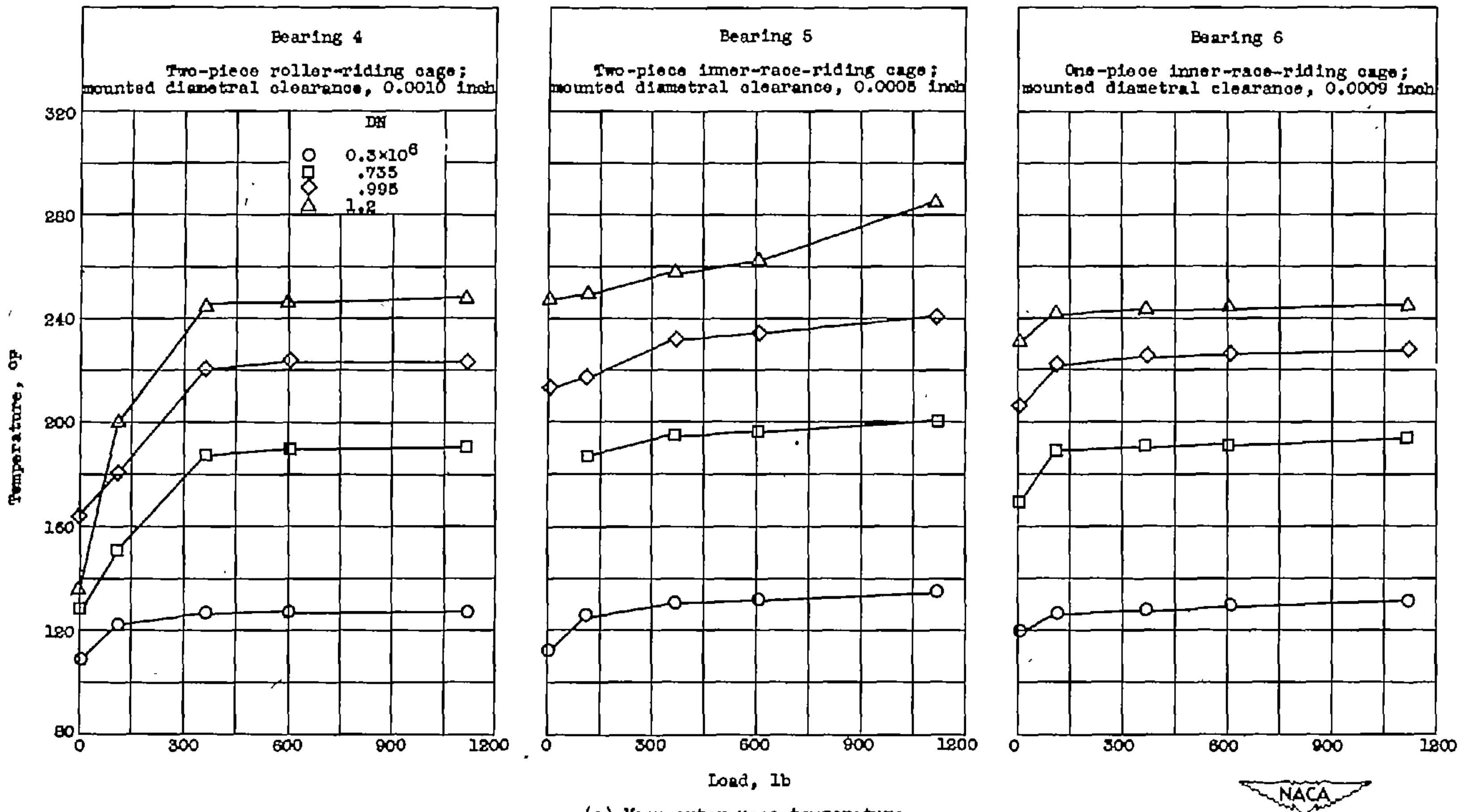
(a) Maximum outer-race temperature.

Figure 13. - Effect of load on operating temperatures of bearings 4, 5, and 6. DN values, 0.3, 0.735, 0.995, and 1.2×10^6; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.



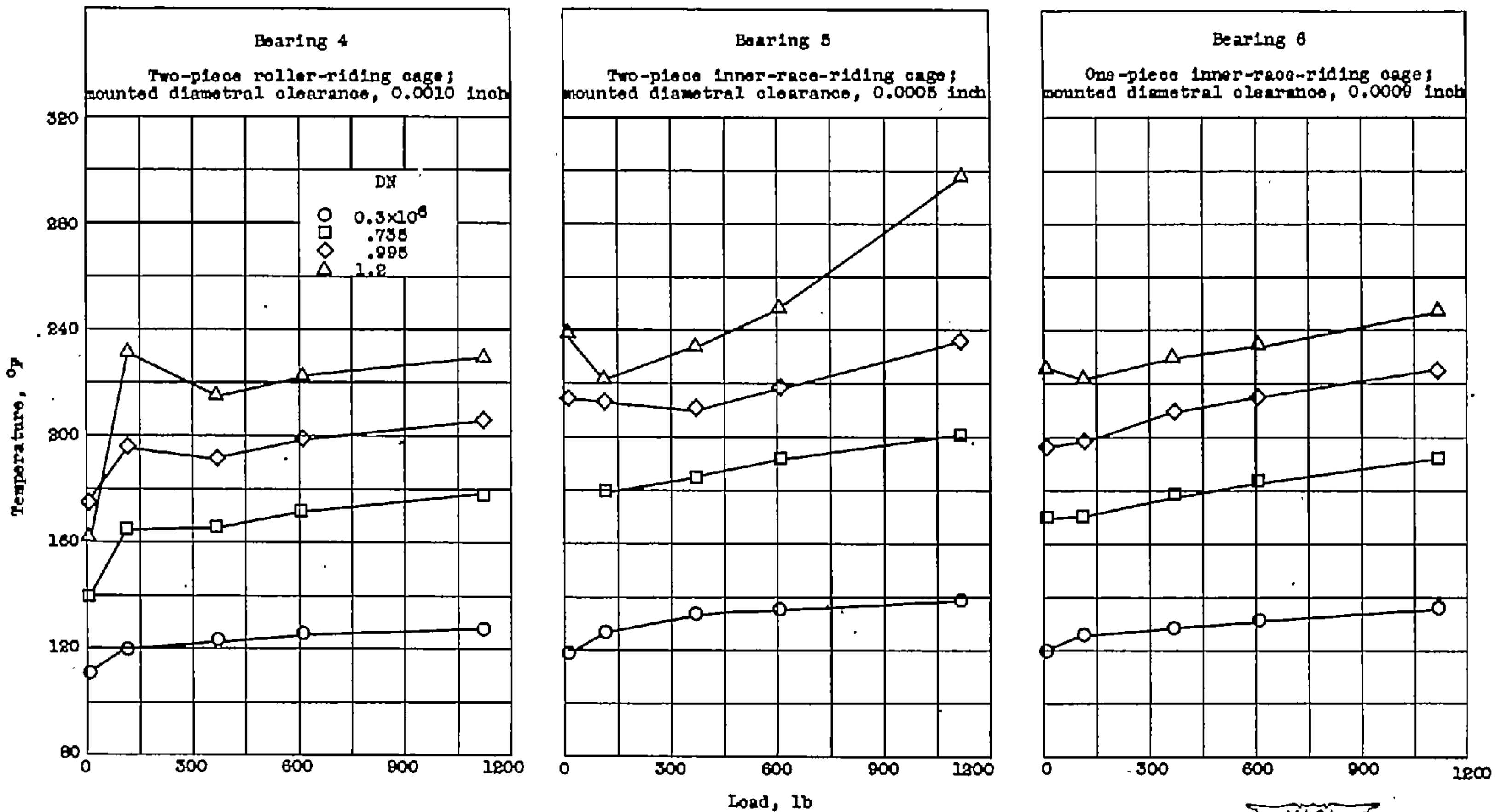
(b) Minimum outer-race temperature.

Figure 13. - Continued. Effect of load on operating temperatures of bearings 4, 5, and 6. DN values, 0.3, 0.735, 0.995, and 1.2×10^6; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.



(c) Mean outer-race temperature.

Figure 13. - Continued. Effect of load on operating temperatures of bearings 4, 5, and 6. DN values, 0.3, 0.735, 0.995, and 1.2×10^6; oil flow, 3.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.



(d) Inner-race temperature.

Figure 13. - Concluded. Effect of load on operating temperatures of bearings 4, 5, and 6. DN values, 0.3, 0.735, 0.995, and 1.2×10^6; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.

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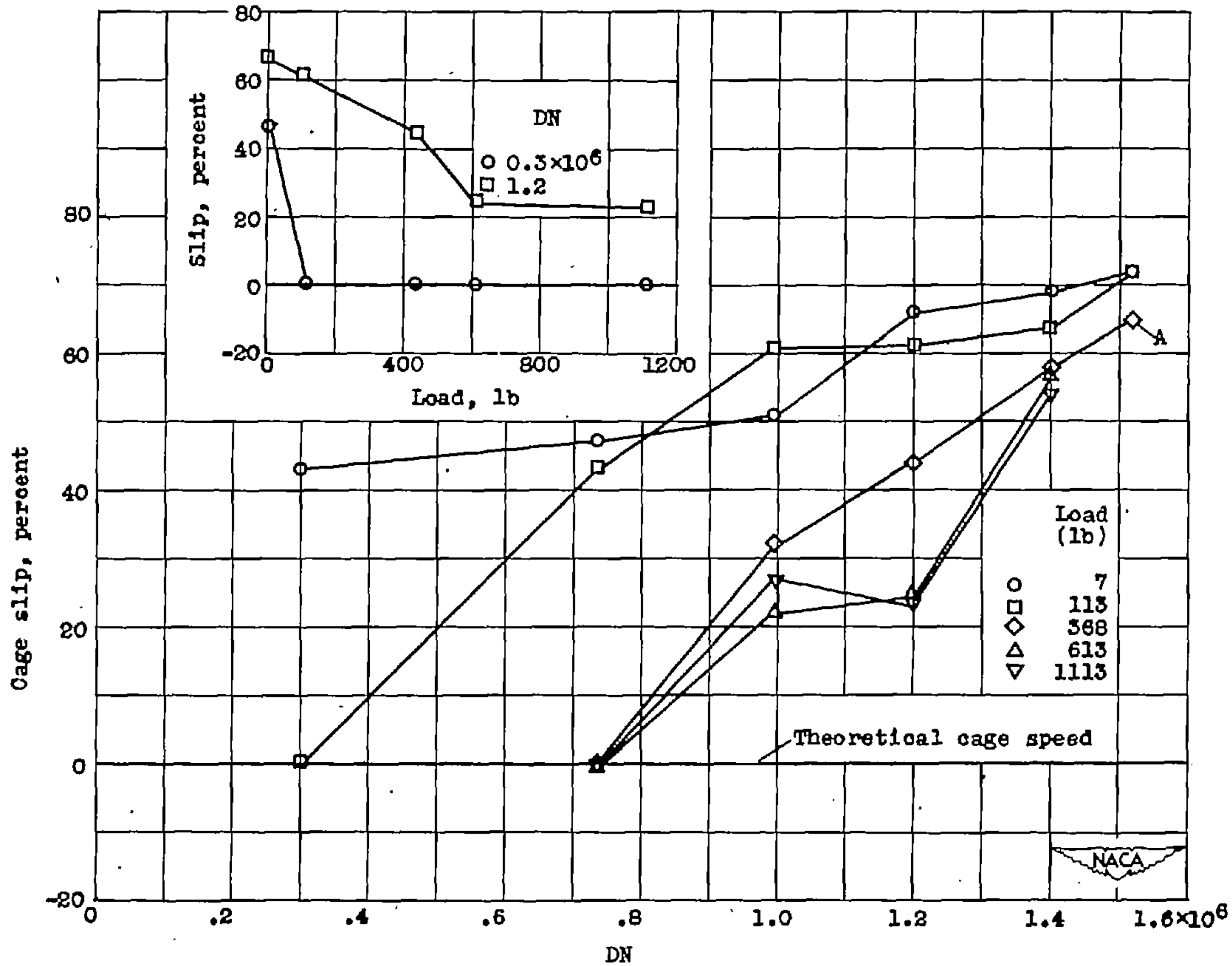


Figure 14. - Effect of speed on percentage slip for bearing 4. Load, 7, 113, 368, 613, and 1113 pounds; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.

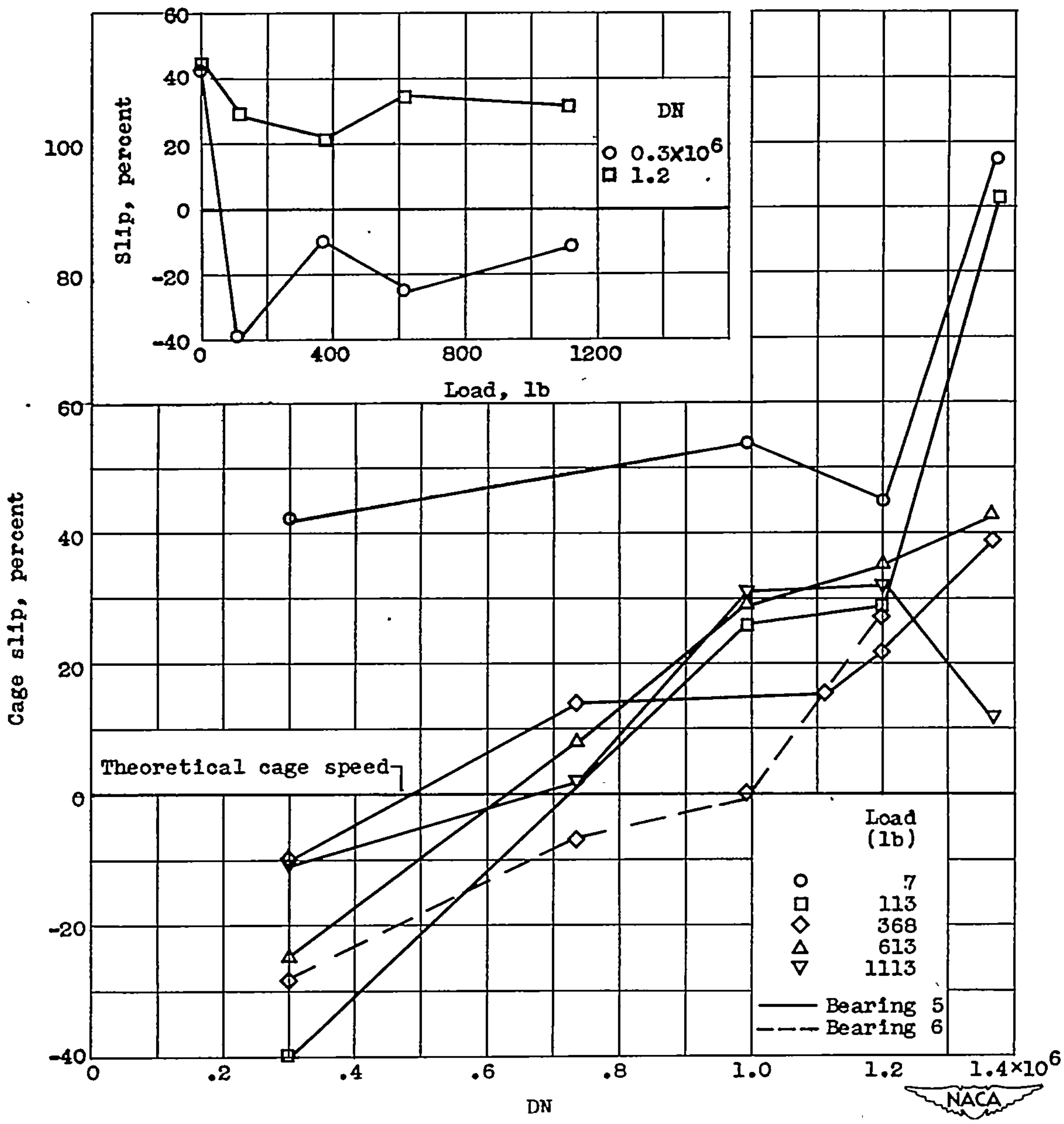
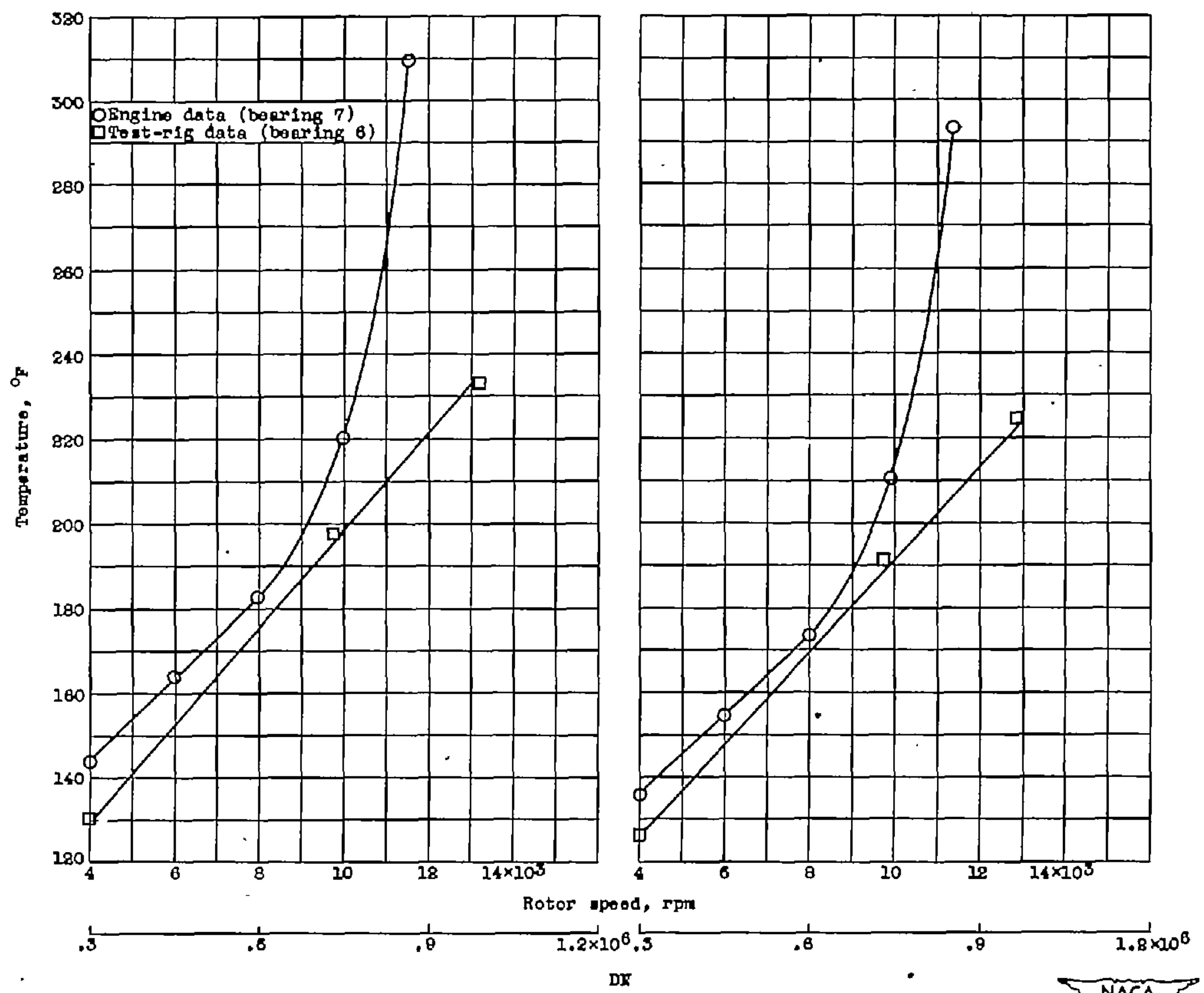


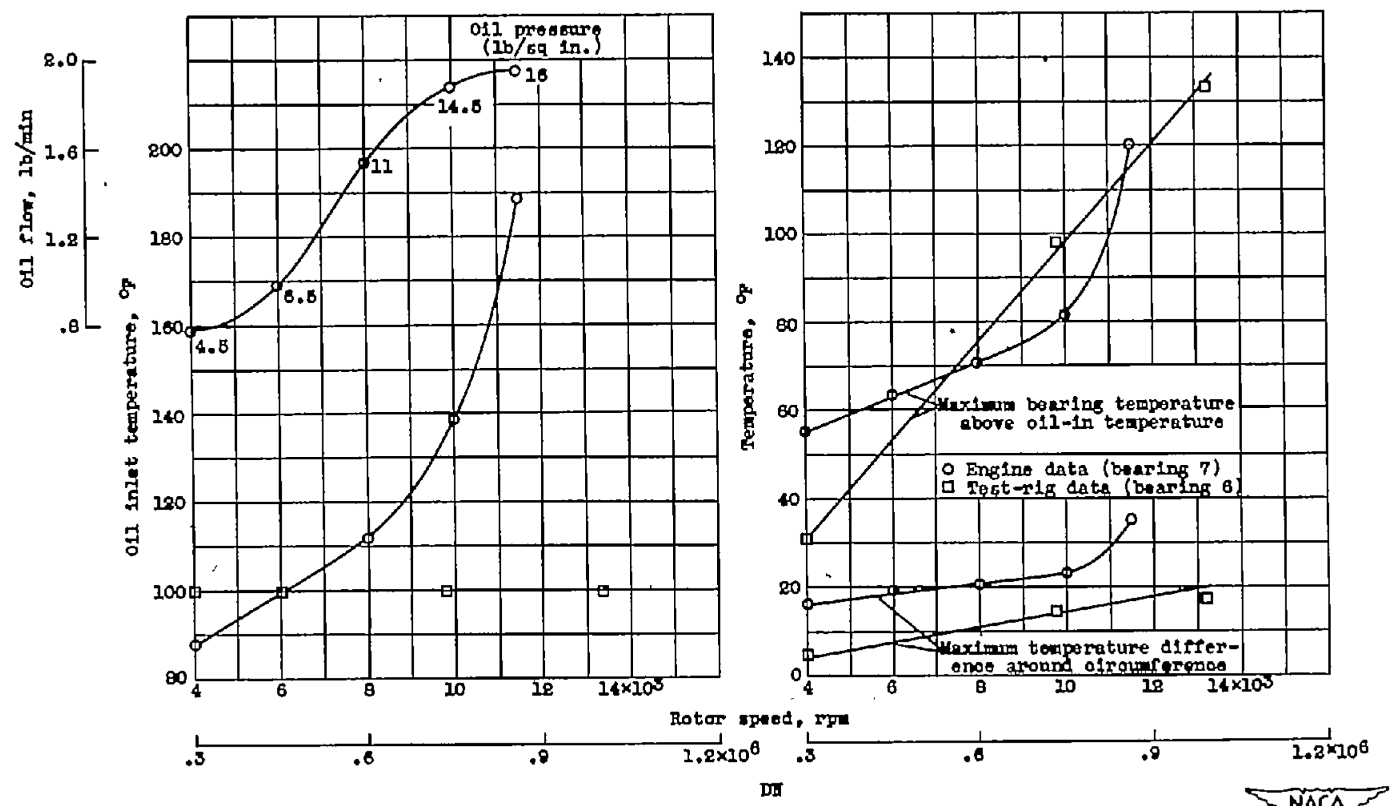
Figure 15. - Effect of speed on percentage slip for bearings 5 and 6. Load on bearing 5: 7, 113, 368, 613, and 1113 pounds; load on bearing 6: 368 pounds; oil flow to each bearing, 2.75 pounds per minute; oil-jet diameter, 0.089 inch; oil inlet pressure, 9.5 pounds per square inch; oil inlet temperature, 100° F.



(a) Maximum outer-race temperature.

(b) Mean outer-race temperature.

Figure 16. - Operating characteristics of commercial gas-turbine rear turbine roller bearing (bearing 7, table I) compared with test-rig operation, (bearing 6, table I). Load on engine bearing, 375 pounds; jet diameter, 0.052 inch; load on test-rig bearing, 388 pounds, jet diameter, 0.089 inch; oil flow, 2.76 pounds per minute; oil inlet pressure, 9.5 pounds per square inch.



(c) Oil data.

(d) Temperature data.

Figure 16. - Concluded. Operating characteristics of commercial gas-turbine rear turbine roller bearing (bearing 7, table I) compared with test-rig operation (bearing 6, table I). Load on engine bearing, 375 pounds; jet diameter, 0.052 inch; load on test-rig bearing, 368 pounds, jet diameter, 0.059 inch; oil flow, 8.75 pounds per minute; oil inlet pressure, 9.5 pounds per square inch.

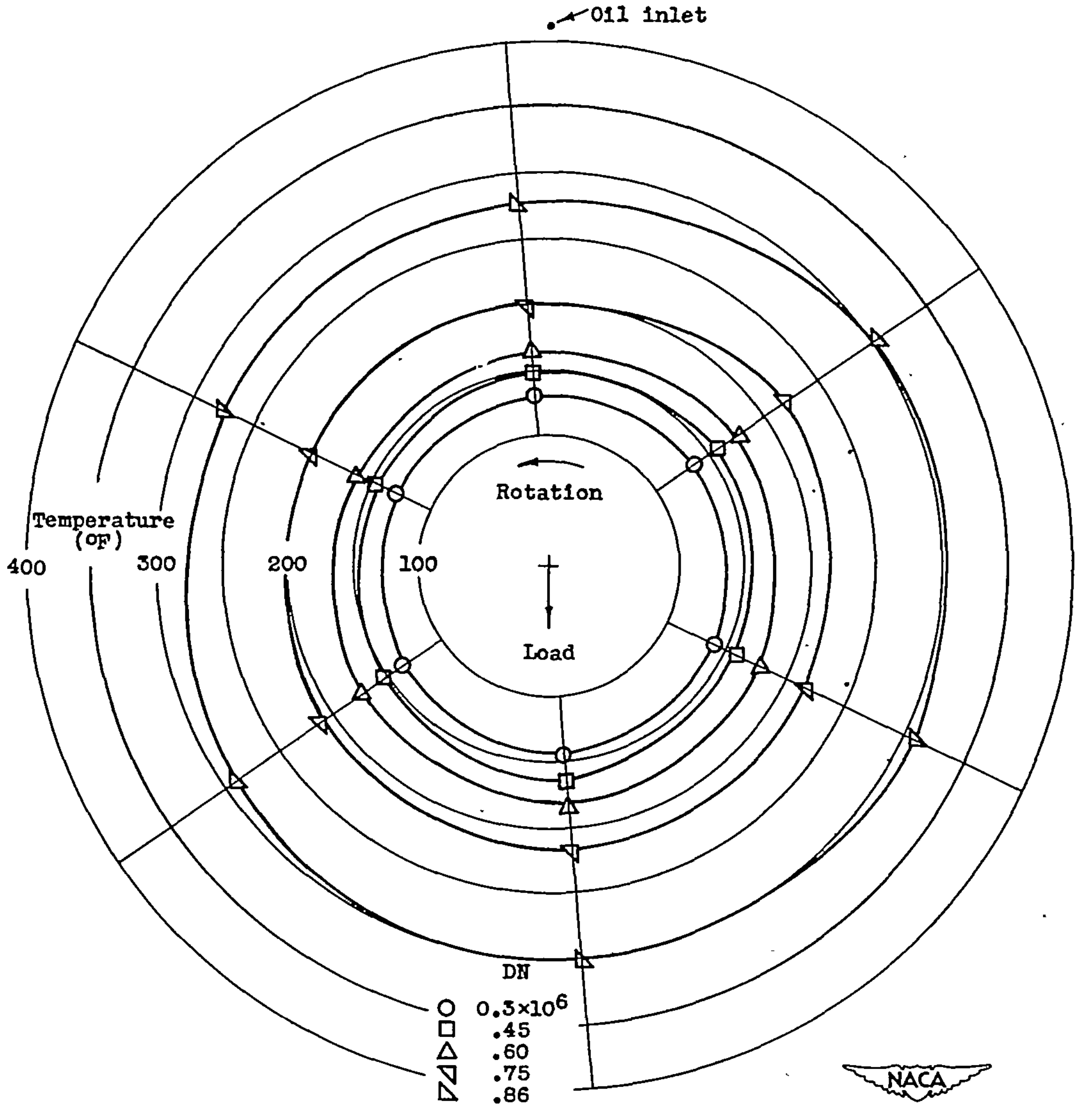


Figure 17. - Circumferential temperature distribution around outer race of aircraft gas-turbine-engine roller bearing (bearing 7 from table I). Load, 375 pounds. Oil-jet diameter, 0.052 inch.

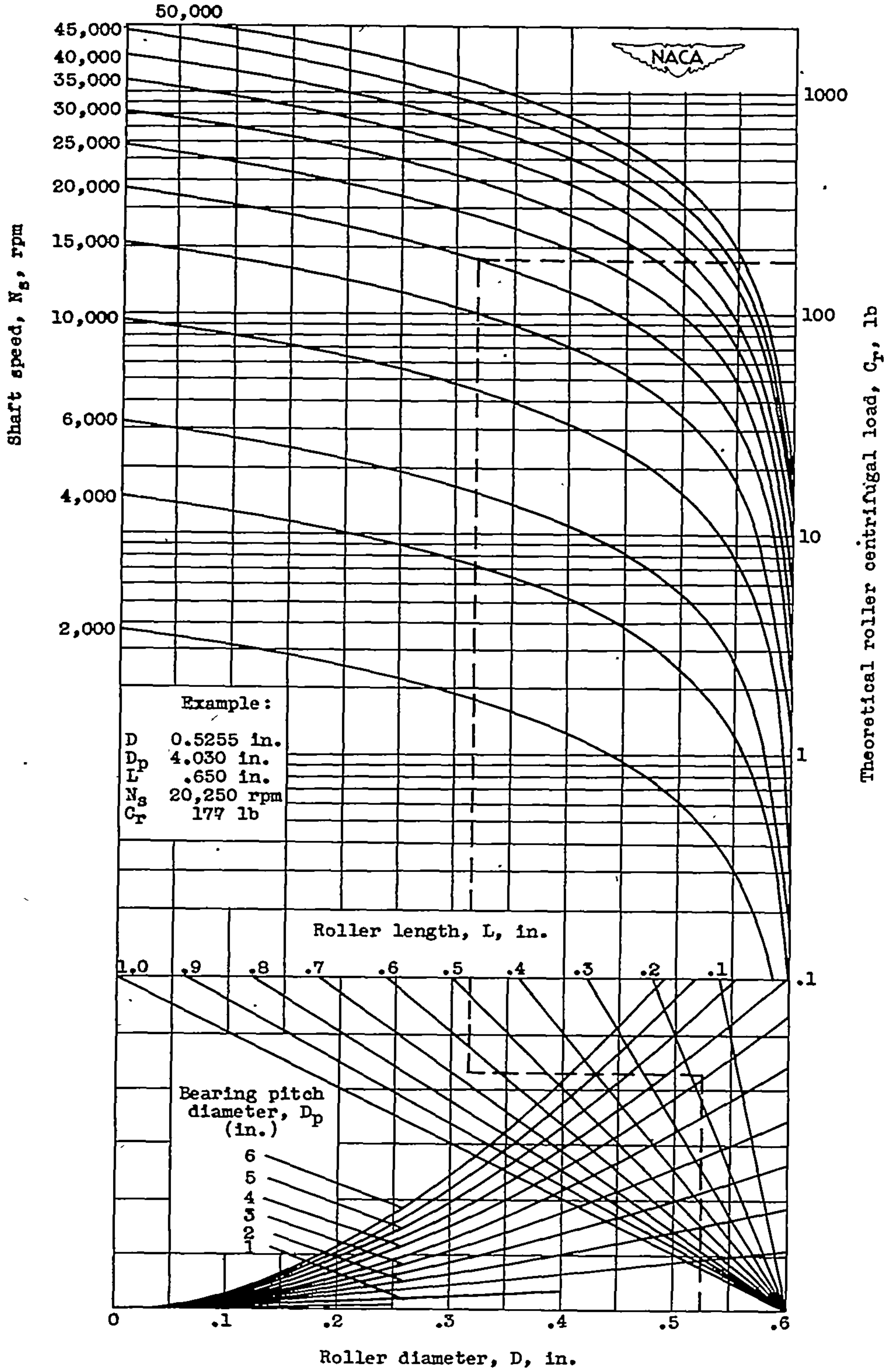


Figure 18. - Variation of theoretical roller centrifugal load with bearing dimensions and shaft speed.

$$\left(C_r = 0.792 \times 10^{-6}\, D^2 L (N_s)^2\, D_p \left(1 - \frac{D}{D_p}\right)^2\right)$$