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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 3002

EFFECT OF BRONZE AND NODULAR IRON CAGE MATERIALS ON CAGE SLIP AND OTHER PERFORMANCE CHARACTERISTICS OF 75-MILLIMETER-BORE CYLINDRICAL-ROLLER BEARINGS AT DN VALUES TO 2×10^6

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Lewis Flight Propulsion Laboratory
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SUMMARY

An experimental investigation to determine the relative merits of bronze and nodular iron as cage materials for cylindrical-roller bearings was conducted using four identical 75-millimeter-bore outer-race-riding cage-type cylindrical-roller bearings (two with bronze cages and two with nodular iron cages). The test bearings were operated over a range of DN values (product of bearing bore in mm and shaft speed in rpm) from 0.3×10^6 to 2.1×10^6, oil flows from 2 to 8 pounds per minute, and loads from 7 to 1613 pounds.

The results, which are applicable with certainty only to the type of bearing tested, indicate that wear rates vary with cage slip and appear to be a function of cage material and also of the sliding velocity and load between the rollers and inner race. Cage slip occurs more readily at DN values above 1.2×10^6 if the cage material is nodular iron rather than the bronze investigated. Consequently, bearings with nodular iron cages showed more wear of rollers and cages (especially in cage-roller pockets) than did bearings with bronze cages.

The severe roller and cage wear obtained in the bearings with nodular iron cages suggests that roller-pocket and cage-locating-surface friction may, under certain operating conditions, exceed the tractive force between the rollers and inner race and may, therefore, be responsible for the onset of cage slip. Cage slip in bearings with bronze cages was accompanied by severe roller wear and light cage wear in the cage-roller pockets. This fact, together with the severe cage wear associated with nodular iron cage-type-bearing slip, suggests that cage material may be a factor in reducing cage wear in a bearing operating under slip conditions.

There was little difference in performance between the bronze and nodular iron cages as measured by bearing temperature, heat dissipation

to the oil, and bearing wear at DN values to 1.2×10^6 (16,000 rpm). The wear in all test bearings at speeds to 16,000 rpm was light and was not of a magnitude that would indicate a very limited life.

INTRODUCTION

The importance of the cage or retainer as a component part of rolling-contact bearings has been brought to light by the fact that cage failures have ranked high among the causes of failure in high-speed roller bearings (refs. 1 and 2). In reference 1 it is reported that the greatest number of bearing failures occurs as a result of an inner-race-to-cage seizure in roller bearings or in a ball-to-cage seizure in ball bearings. In reference 2 the author states that, to his knowledge, there has never been a roller-bearing failure in aircraft bearings that could be attributed to fatigue and that all the failures had been caused by the cages. In references 3 and 4 it is reported that bearing life is being limited by wear (cage, ball or roller, and race) rather than fatigue. In reference 5 one of the discussers states that practically all investigators have concluded that the critical factors at higher speeds are cage failure, heat dissipation, and lubrication. Such statements are rather strong, and they must not be interpreted to mean that the problem of fatigue does not exist in high-speed ball and roller bearings. They do, however, point out the critical nature of the cage problem. Published service experience on turbojet engines (ref. 6) indicates that failure of main rotor bearings has been one of the principal causes of engine removal and that the bearing failures have indicated the need for, among other things, more research on cage problems.

Unfortunately, very little information on cage design and material research is available in the literature. In reference 7 the comparative performance of several experimental roller-bearing-cage designs is reported. Wear and friction studies on cage materials are reported in references 8 to 10.

It would be ideal to design a cage with hydrodynamic lubrication at all points of contact between the cage and races and rollers, for then failures due to wear, galling, and cage pickup would be minimized. Unfortunately, since success has not yet been achieved in this regard, the cage-material problem remains to be solved. Until hydrodynamic lubrication at all cage-contacting surfaces can be achieved, boundary lubrication will exist at points of sliding contact, and the wear and frictional properties of the cage material will be of extreme importance.

Much work remains to be done on the evaluation of the merits of the more promising cage materials in actual bearings. Data obtained with friction and wear machines act as a guide in the choice of materials from which cages should be made, but results cannot be considered conclusive until bearing performance data are at hand.

The experimental results reported herein, obtained from an investigation conducted at the NACA Lewis laboratory, consist of comparisons of the operating characteristics of four 75-millimeter-bore cylindrical-roller bearings with identical races and rollers at DN values of 0.3×10^6 to 2.1×10^6, loads of 7 to 1613 pounds, and oil flows of 2 to 8 pounds per minute. Test-bearing operating temperatures, limiting DN values, wear, and heat dissipation to the oil are compared. The test bearings were equipped with double-flanged outer races, straight inner races, and outer-race-riding cages. Two test bearings had nodular iron cages, and two had bronze cages. Great care was taken to keep all the significant variables except cage material constant in the four test bearings.

APPARATUS

Bearing rig. - The bearing rig (fig. 1) used for this investigation is the same as that used in reference 11, modified according to reference 7. The bearing under investigation was mounted on one end of the test shaft, which was supported in cantilever fashion in order that bearing component parts and lubricant flow could be observed 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 loading 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 11. Oil was supplied to the support bearings at a pressure of 10 pounds per square inch through 0.180-inch-diameter nozzles and at a temperature equal to that of the test-bearing oil (100° F).

Drive equipment. - The high-speed drive equipment consisted of a shunt-wound 30-horsepower direct-current motor connected to a 14:1 speed increaser. The high-speed shaft of the speed increaser was connected to the test shaft by means of a floating spline coupling. The speed range of the test shaft was 1100 to 36,000 rpm, controllable to within ±50 rpm at all speeds.

Temperature measurement. - The method of temperature measurement is described in reference 11. Iron-constantan thermocouples were located in the outer-race housing at 60° intervals around the outer-race periphery at the axial midpoint 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 thermocouple electromotive force was transmitted from the rotating shaft by means of small slip rings located on the end of the test shaft (ref. 12).

Lubrication system. - The lubrication system used was of the circulating type. Separate pumps were used to supply oil to the support bearings and the test bearing, and full-flow filters were provided after the oil-supply pumps. Oil inlet temperature was controlled to within $\pm 1^\circ$ F and oil inlet pressure to within ± 0.5 pound per square inch. Support-bearing oil flow was drained by gravity from the base of the rig. Test-bearing oil was collected in cans and pumped either to weighing buckets or back to the sump.

TEST BEARINGS

Four test bearings (size 215) were used for this investigation. The over-all dimensions of all four bearings were: bore, 75 millimeters; outside diameter, 130 millimeters; and width, 25 millimeters. Test bearings were of the double-flanged outer race, straight inner-race type. The cages of all four test bearings were of the outer-race-riding type with broached roller pockets and were dimensionally alike. The cages of bearings A and B were made of bronze and those of bearings C and D of pearlitic nodular iron. Physical properties and chemical composition of the cage materials are given in table I. Nodular iron was chosen as the first material to be used in this experimental cage-material program because it affords a good combination of anti-weld, wear, expansion-coefficient, and strength properties (ref. 8).

Great care was taken to keep all the significant variables as nearly constant as possible. The component parts of the four test bearings were carefully inspected and all critical dimensions were tabulated. The test bearings were then selectively assembled to keep the following variables as nearly constant as possible:

1. Bearing diametral clearance

2. Cage diametral clearance

3. Roller axial clearance

4. Roller diametral variation

5. Roller length variation

6. Roller clearance in cage pocket

The dimensions of all four test bearings are given in table II, and a schematic drawing of a test bearing is shown in figure 2.

PROCEDURE

Order of test. - Each test bearing was first subjected to a number of tests at DN values of 0.3×10^6 to 1.2×10^6. These tests were collectively termed the low-speed tests, and at their conclusion each bearing was removed from the rig and checked for wear. After inspection, the bearing was reinstalled in the rig and run to its limiting DN values at oil flows of 2 to 8 pounds per minute. These tests were collectively termed the high-speed tests, and at their conclusion each bearing was again checked for wear. All test bearings were subjected to an identical series of tests so that the total running time on each bearing was approximately equal.

Lubrication of test bearings. - Lubricant was supplied to the test bearings through a single jet of 0.050-inch-diameter orifice. The oil was directed normal to the bearing face opposite the load zone. The optimum radial position for the oil jet was determined for each of the test bearings and was found to be between the cage and inner race.

A highly refined nonpolymer-containing petroleum-base lubricating oil used to lubricate the bearings in several current turbojet engines was used to lubricate the test bearings. The viscometric properties of the test oil are shown in figure 3. Data for figure 3 were obtained by taking daily samples of test oil. Viscosities were obtained by standard laboratory procedures, and the data plotted in figure 3 represent the mean for all the samples of oil.

Oil was supplied to the test bearings at a temperature of 100° F and at pressures of 30 to 405 pounds per square inch, which corresponded to oil flows of 2 to 8 pounds per minute.

Test-bearing measurements. - Measurements of test-bearing component parts were obtained in a constant-temperature gage room. Standard, precision inspection instruments measuring to 0.0001 inch per division were used to obtain dimensions of the races and cages. A comparator-type gage measuring to 2×10^{-6} inch was used to obtain roller measurements. Test-bearing-hardness measurements are given in table III, and test-bearing surface-finish measurements are given in table IV.

RESULTS AND DISCUSSION

The results of the experimental investigation are presented in figures 4 to 8 and in tables V and VI. Bearing temperature, heat dissipation to the oil, bearing wear, and bearing limiting speeds were chosen as the criteria of operation. Each of these is discussed with respect to the two cage materials under investigation.

Low-Speed Tests

Effect of DN on operating temperatures. - The effect of DN on the outer-race maximum temperatures of the four test bearings is shown in figure 4. Curves for oil flows of 2.0, 2.75, 5.5, and 8.0 pounds per minute are shown. The curves for all four test bearings at all four oil flows show that bearing temperature is an approximately linear function of DN throughout the speed range shown. Differences in operating temperatures for the four test bearings were small, with the maximum difference in bearing temperatures occurring at an oil flow of 2.75 pounds per minute and a DN of 1.2×10^6.

Effect of DN on power rejected to oil. - The effect of DN on the power rejected to the oil in the four test bearings is shown in figure 5. Curves for oil flows of 2.0, 2.75, 5.5, and 8.0 pounds per minute are shown. Differences in the power rejected to the oil in the test bearings became greater at the higher oil flows. The power rejected in bearing A is seen to be greater at the two highest oil flows than that in bearings B, C, and D. Since only the power rejected in bearing A and not that in both bearings A and B seems to be high, the differences are probably due to test conditions, and not to variations in frictional characteristics of the two cage materials. At the conclusion of the tests of bearing A, it was discovered that a slight load misalinement had existed during the tests. This condition manifested itself in the form of slightly greater roller wear on one end than on the other and in slightly uneven wear patterns in the cage pockets. This condition could have been responsible for the discrepancy in results between bearings A and B.

Effect of load on operating temperatures. - Figure 6 shows the effect of load on both the outer-race maximum and the inner-race temperatures of all four test bearings. Inner-race data for bearing A are not shown, because the inner-race thermocouple system was inoperative during that test. The curves for all bearings are similar in shape with the greatest discrepancies between bearings occurring at the two lowest loads, where slip occurred between the rollers and the inner race, causing the cage to slip or to rotate at a rotational speed somewhat less than its theoretical rotational speed. Percentage cage slip is defined as follows:

$$\text{Percentage cage slip} = \left(\frac{N_{CT} - N_C}{N_{CT}}\right) 100$$

where

N_{CT} theoretical cage rotational speed, rpm

N_C actual cage rotational speed, rpm

Reproducibility of bearing temperature cannot be obtained at loads where any appreciable amount of cage slip occurs. Although care was taken to assemble test bearings with nearly identical diametrical clearances, cage slip varied from 27 to 86 percent in the four test bearings when running at a load of 7 pounds, an oil flow of 2.75 pounds per minute, and a DN value of 1.2×10^6. Values of cage slip for the four test bearings operating at this combination of conditions are shown by points a to d in figure 6. It has been found impossible to duplicate cage slip in the same bearing from day to day. The data of figure 6 show that, at a load of 7 pounds, the two bearings with greatest slip had the lowest temperatures.

It is evident from the curves of figure 6 that outer-race maximum temperatures increase sharply in going from the low-load to the high-load range while inner-race temperatures do not. In a bearing of the type being investigated, outer-race temperatures decrease as cage slip increases, because the relative sliding velocity between the cage and the outer race and the relative rolling and sliding velocities between the rollers and the outer race all decrease. Inner-race temperatures do not increase as sharply in going from the low-load to the high-load range as outer-race temperatures, because the combination of sliding and rolling that exists between the rollers and inner race when the cage is not rotating at theoretical speed apparently generates almost as much heat as is generated at zero cage slip. This result is predicted in reference 13.

At the conclusion of the load tests, the characteristic inner-race frosting, which accompanies roller sliding and cage slip, was observed on bearings A, C, and D.

Test-bearing wear. - The low-speed-test wear data (wear indicated by weight change) for the test-bearing component parts are tabulated in table V(a). The data show that there is little difference in wear characteristics between the bronze and nodular iron investigated at DN values to 1.2×10^6 (16,000 rpm), although the nodular iron cages did show slightly greater wear. At 16,000 rpm and zero cage slip, the sliding velocity is 7970 feet per minute at the cage-locating surface and 7310 feet per minute in the cage pockets. The wear of all test-bearing component parts was low and not indicative of any severe bearing-life limitations at speeds to 16,000 rpm.

High-Speed Tests

Effect of oil flow on limiting DN values. - The effect of oil flow on the limiting DN values for all four test bearings is shown in figure 7. At each oil flow, each test bearing was run until an equilibrium operating temperature could no longer be obtained. The highest DN value at which equilibrium operation could be obtained has been termed the limiting DN value. Limiting DN values, in general, increased with increasing oil flow, and variations in limiting DN values between bearings were relatively small.

Bearing operating temperatures at very high speeds. - Figure 8 shows plots of bearing outer-race maximum temperature as functions of DN values from 1.2×10^6 to the limiting values for all four test bearings. Curves for oil flows of 2.0, 2.75, 5.5, and 8.0 pounds per minute are shown. Unlike the DN curves at low speeds, the curves become rather erratic at high speeds, especially at the two highest oil flows. During the course of the investigation, it was found that the sharp changes in slope occur at the onset of cage slip. The tests seem to indicate that a roller bearing is capable of absorbing only a fixed amount of torque when operating at a specific load and oil flow. When this limiting bearing torque is reached, the maximum friction force that can be developed between the rollers and inner race is being utilized. An increase in shaft speed beyond that at which the maximum bearing torque is developed will result in cage slip. Several significant points on figure 8 are identified, and the corresponding values of cage slip are tabulated on the figure. At an oil flow of 5.5 pounds per minute, the slope of the curve for bearing C decreased in going from point "a" to point "b," and at the same time cage slip increased from 2 to 30 percent. At the same oil flow, the curve for bearing D exhibited a marked increase of slope in going from point "c" to point "d"; this change was accompanied by a decrease in cage slip from 23 to 5 percent. Sharp changes in slope apparently indicate changes in cage slip.

Test-bearing wear. - The high-speed-test wear data for the test-bearing component parts are tabulated in table V(b). The data show that the bearings with bronze cages incurred much less wear than did the bearings with nodular iron cages at bearing DN values above 1.2×10^6 (16,000 rpm). Maximum sliding velocities (assuming no cage slip) in the test bearings at DN values of 1.2×10^6 to 2.1×10^6 are 7970 to 13,950 feet per minute. Cage, roller, and outer-race wear were severe in both bearings C and D. Roller wear was heavy in bearing A, presumably, because of the slight load misalinement discussed in the low-speed test results. The most significant indication of the high-speed wear data is the relative wear of the four test-bearing cages. The results of basic friction and wear tests (ref. 8) and of the low-speed tests reported herein (which show negligible wear) indicate strongly that the excessive wear incurred by the nodular iron cages results not only from differences in material properties but also from other factors. Comparison of the high-speed DN curves (fig. 8) and the high-speed-test wear data (table V(b)) suggests strongly that cage slip is responsible, at least in part, for high wear rates and is therefore highly detrimental to bearing life.

Cage-Slip Tests

The exact effects of increased cage slip, other than the tendency to reduce bearing torque and bearing operating temperatures because of reduced sliding velocities within the bearing, were unknown at the time the high-speed tests were run, so special tests were run to determine the effect of cage slip on the wear rates of the bearing component parts. These results are discussed in the section following.

Effect of cage slip on wear rates of test-bearing component parts. - To determine the effect of cage slip on wear rates, three tests were run using bearing B. Bearing B was chosen, because it had sustained the least wear of all four test bearings and was not worn to any damaging extent. Three tests, each of 4 hours duration, were run at a DN value of 1.2×10^6 and at an oil flow of 2.75 pounds per minute. The load during each test was different in order to vary the cage slip. The results of these tests are tabulated in table VI. The data of table VI show that test bearing B incurred no appreciable wear when cage slip was as high as 30 percent but that very severe wear occurred at 95-percent cage slip. At the start of the run at a load of 7 pounds, cage slip was about 40 percent and did not reach 95 percent until about 30 minutes had elapsed. The gradual increase in cage slip was accompanied by a steady decline in bearing temperatures indicative of the decreased rate of heat generation within the bearing.

Comparison of these results with those of the low-speed tests, where values of cage slip of 80 and 86 percent were obtained in bearings A and C at a load of 7 pounds and a DN of 1.2×10^6 (fig. 6), shows a marked difference in wear. The wear during the low-speed tests may have been small because of the short time (approx. 15 min) the bearings were run under these conditions. At DN values of 1.5×10^6 and higher and at a load of 368 pounds, very severe wear occurred at values of cage slip on the order of 30 percent in bearings C and D (see fig. 8 and table V(b)).

Cage material undoubtedly has an effect on wear rate, but for both cage materials investigated the results seem to indicate that additional important factors affecting wear are the sliding velocities and the loads between the component parts. The sliding velocity between the rollers and inner race is, in turn, a function of the percentage cage slip and the DN value.

It can be concluded then that cage slip, despite the fact that it is accompanied by a decrease in heat generation, is extremely harmful, from a wear standpoint, to a cylindrical-roller bearing with an outer-race-riding cage. For this reason, nodular iron cannot be recommended as a cage material for this type of high-speed roller bearing, because it apparently tends to promote cage slip. In reference 8 it is shown that nodular iron, although exhibiting low wear rates in sliding against SAE 52100 steel, showed higher coefficients of friction in many cases than did bronze. It seems most probable that, with nodular iron cages under poor lubrication conditions introduced by high sliding speeds, the friction forces between the cage and rollers and between the cage and outer race become great enough to exceed the tractive force between the rollers and inner race. When this happens, cage slip occurs, causing surface welding between the rollers and inner race. Transferred and loose particles of steel that result from failure in sliding serve to abrade and lap, producing the characteristic frosted appearance on the inner race. The loose particles of steel would accelerate wear of all component parts, and the transferred metal could accelerate wear of all surfaces contacting the periphery of the rollers.

Effect of cage slip on limiting DN values. - Limiting DN value was previously defined as the highest DN at which the bearing under investigation would operate at an equilibrium temperature. In the light of the cage-slip and high-speed test results, however, it might be advisable to modify that definition to include the condition that no damaging cage slip occur. The limiting DN value for a bearing operating at a specific load and lubricated in a specific manner would then be the highest DN value at which the bearing would run at an equilibrium temperature without damaging cage slip; limiting DN value would thus be a function of load, oil flow, and operating diametrical clearance, as well as other factors such as surface finish.

General Observations

Several critical clearances which reflect the magnitude of the wear at different points in the bearings are tabulated in table II. It was observed that inner-race frosting accompanied cage slip in all test bearings and that in bearings with nodular iron cages the races and rollers appeared highly polished where not frosted.

As shown in table III, test-bearing-hardness measurements changed little during the tests. Roller and cage hardnesses show a consistent small decrease, while race hardnesses decreased slightly in some instances and remained constant in other cases.

Test-bearing surface finishes (table IV) became generally rougher during the tests. Frosting considerably roughened the inner races, and heavy roller wear resulted in rougher surface finishes on both roller periphery and ends.

SUMMARY OF RESULTS

An experimental investigation to determine the relative merits of bronze and nodular iron as cage materials for cylindrical-roller bearings was conducted using four identical 75-millimeter-bore outer-race-riding cage-type cylindrical-roller bearings (two with bronze cages and two with nodular iron cages). The test bearings were operated over a range of DN values (bearing bore in mm times shaft speed in rpm) from 0.3×10^6 to 2.1×10^6, oil flows from 2 to 8 pounds per minute, and loads from 7 to 1613 pounds. Test bearings were lubricated by a single jet of oil at an inlet temperature of 100° F. The following results, applicable with certainty only to the type of bearing tested, were obtained:

1. Wear rates varied with cage slip and appeared to be a function of cage material and also of the sliding velocity and load between the rollers and inner race.

2. Cage slip occurred more readily at DN values above 1.2×10^6 with nodular iron cages than with the bronze cages investigated. Consequently, bearings with nodular iron cages showed more wear of rollers and cages (especially in cage-roller pockets) than did bearings with bronze cages.

3. The severe roller and cage wear obtained in the bearings with nodular iron cages suggests that roller-pocket and cage-locating-surface friction may, under certain operating conditions, exceed the tractive force between the rollers and inner race and may therefore be responsible for the onset of cage slip.

4. Cage slip in bearings with bronze cages was accompanied by severe roller wear and light cage wear in the cage-roller pockets. This fact together with the severe cage wear associated with nodular iron cage-type-bearing slip suggests that cage material may be a factor in reducing cage wear in a bearing operating under slip conditions.

5. There was little difference in performance between the bronze and nodular iron cages as measured by bearing temperature, heat dissipation to the oil, and bearing wear at DN values to 1.2×10^6 (16,000 rpm). The wear in all test bearings at speeds to 16,000 rpm was light and was not of a magnitude that would indicate a very limited life.

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TABLE I. - PROPERTIES AND COMPOSITION OF CAGE MATERIALS

Material	Bronze	Nodular iron[a]
Yield point, lb/sq in.	32,000-37,000	56,800[b]
Tensile strength, lb/sq in.	65,000-73,000	80,500
Elongation, percent	25-35	2
Reduction of area, percent	20-30	1.5
Modulus of elasticity, lb/sq in.	15×10^6	20.9×10^6
Chemical composition, percent	Cu 61.4 Zn 36.5 Pb 1.0 Al 1.1	C 3.70 Mn .33 Si 2.33 Cr .02 Cu .47 P .03 S .011 Mg .061

[a]Obtained from the Ford Motor Company foundry.

[b]Specimen taken from $2\frac{1}{4}$-in. radius of 6-in.-diameter by 12 in.-long ingot.

TABLE II. - DIMENSIONS OF TEST BEARINGS

Bearing	A		B		C		D	
Number of rollers	20		20		20		20	
Pitch diameter of bearing, in.	4.036		4.036		4.036		4.036	
Length-diameter ratio of roller	1		1		1		1	
Cage material	Bronze		Bronze		Nodular iron		Nodular iron	
Cage type	One-piece outer-race riding		One-piece outer-race riding		One-piece outer-race riding		One-piece outer-race riding	
	Before	After	Before	After (a)	Before	After	Before	After
Total running time, hr	0	26.1	0	22.8	0	23.2	0	22.5
Average roller diameter, in.	0.5511	0.5505	0.5511	0.5510	0.5511	0.5501	0.5511	0.5500
Average roller length, in.	0.5507	0.5502	0.5508	0.5506	0.5507	0.5500	0.5508	0.5500
Total roller diametral variation, in.	5×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------
Total roller length variation, in.	0.0004	------	0.0004	------	0.0004	------	0.0004	------
Axial clearance between roller and outer-race flanges, in.	0.0019	0.0026	0.0018	0.0021	0.0022	0.0038	0.0020	0.0039
Axial clearance between roller and cage, in.	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012
Circumferential clearance between roller and cage, in.	0.010	0.0115	0.011	0.012	0.0115	0.051	0.0115	0.051
Mounted bearing: Diametral clearance, in.								
Bearing	0.0008	0.0027	0.0011	0.0015	0.0010	0.0030	0.0009	0.0033
Cage	.019	.019	.0205	.0205	.0205	.0225	.0205	.0225
Eccentricity, in.	0.0002	0.0003	0.0001	0.0003	0.0001	0.0003	0.0001	0.0006
Remarks	Inner-race roller track frosted because of cage slip; outer race unmarked. Cage wear light. Roller wear fairly heavy on periphery and ends.		No inner-race frosting after low- and high-speed tests. Outer race unmarked. Cage wear very light. Roller wear light. Severe inner-race frosting and slight surface galling after slip tests.		Inner-race roller track frosted because of cage slip and polished in nonfrosted areas. Outer-race roller track and cage-locating surface polished. Roller and cage wear very severe.		Inner-race roller track frosted because of cage slip and polished in nonfrosted areas. Outer-race roller track appeared dull. Roller and cage wear very severe.	

[a]Values obtained after high-speed tests and before cage-slip tests.

TABLE III. - TEST-BEARING-HARDNESS MEASUREMENTS[a]

Bearing	A		B		C		D	
Cage material	Bronze		Bronze		Nodular iron		Nodular iron	
	Before	After	Before	After (b)	Before	After	Before	After
Outer-race face (Rockwell C-scale)	60.5-62	58-60	58.5-60	58-60	61-62	59-61	60-61	60-61
Inner-race face (Rockwell C-scale)	59-59.5	58.5-59.5	61-62.5	60-61	62	60-61	60-62	60-62
Roller end (Rockwell C-scale)	63-66	62-63.5	63-65.5	62-63	63-65	61-63	63-66	62-64
Cage face (Rockwell B-scale)	60-65	58-62	61-65	59-62	97	92-95	98	93-96

[a]Obtained with Rockwell superficial hardness tester.
[b]Obtained after high-speed tests but before cage-slip tests.

TABLE IV. - TEST-BEARING SURFACE-FINISH MEASUREMENTS[a]

Bearing	A		B		C		D	
Cage material	Bronze		Bronze		Nodular iron		Nodular iron	
	Before	After	Before	After (b)	Before	After	Before	After
Inner-race roller riding diameter, circumferential	2.5-4	8-20	2-3	5-20	2-3.5	2.5-20	2-3	2.5-20
Roller diameter, axial	3-4	1.5-4	1.5-4.5	1.5-4.5	3-4.5	2-6	3-4.5	2-8
Roller end	2-3.5	1.5-3	3-4	1.5-4	3-4	1.5-10	3-4	1.5-12
Outer-race cage-locating surface, circumferential	3-5	1.5-4	2-4	2-3.5	3-5	2-5	3-5	1.5-4
Cage outside diameter, circumferential	20-25	20-30	20-25	20-30	10-16	15-20	10-15	15-20
Outer-race roller riding diameter, circumferential	3-4.5	2-4	2-4	2-3.5	3-5	1.5-4	3-4	2-6.5

NACA

[a]Measured in microinches, rms with a profilometer.
[b]Obtained after high-speed tests but before cage-slip tests.

TABLE V. - TEST-BEARING-WEAR DATA[a]

(a) Low-speed tests.

Bearing	A	B	C	D
Cage material	Bronze	Bronze	Nodular iron	Nodular iron
Outer-race wear, mg	16	5	17	23
Inner-race wear, mg	7	2	5	1
Roller wear, mg	68	22	28	67
Cage wear, mg	18	13	37	51

(b) High-speed tests.

Bearing	A	B	C	D
Cage material	Bronze	Bronze	Nodular iron	Nodular iron
Outer-race wear, mg	69	4	248	515
Inner-race wear, mg	23	10	25	34
Roller wear, mg	883	48	1744	1882
Cage wear, mg	30	18	6233	8913

[a]Measured by weight loss.

TABLE VI. - TEST-BEARING-B WEAR DATA FOR CAGE-SLIP INVESTIGATION[a]

[Conditions for each test: time, four hr; DN, 1.2×10^6 (16,000 rpm); oil flow, 2.75 lb/min; oil inlet temperature, 100° F.]

Test	Load, lb	Average cage slip, percent	Wear, mg			
			Outer race	Inner race	Rollers	Cage
1	368	2	3	-1	3	6
2	113	30	4	10	3	6
3	7	95	7	132	1762	49

[a]Measured by weight loss.

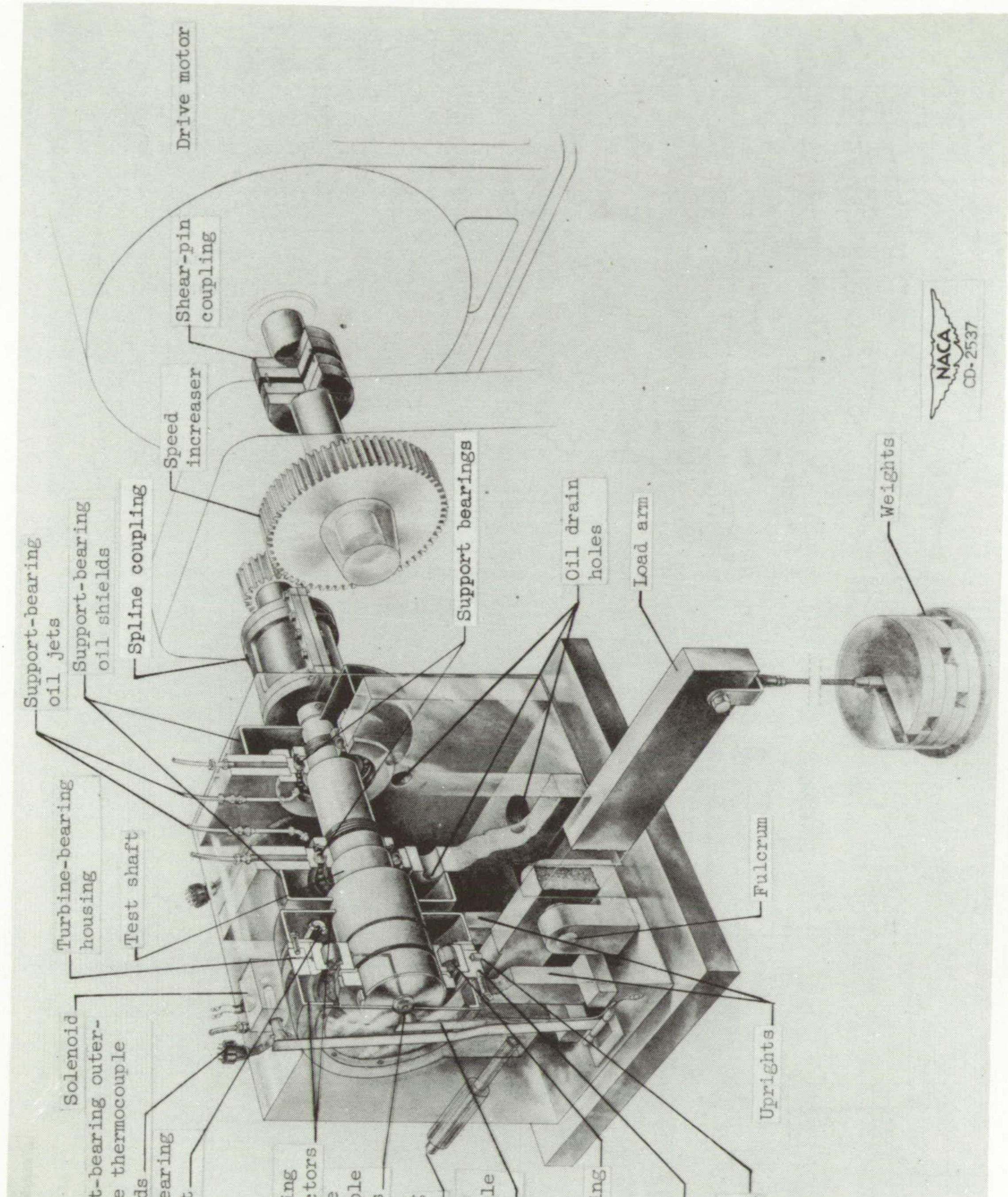


Figure 1. - Cutaway view of radial-load rig.

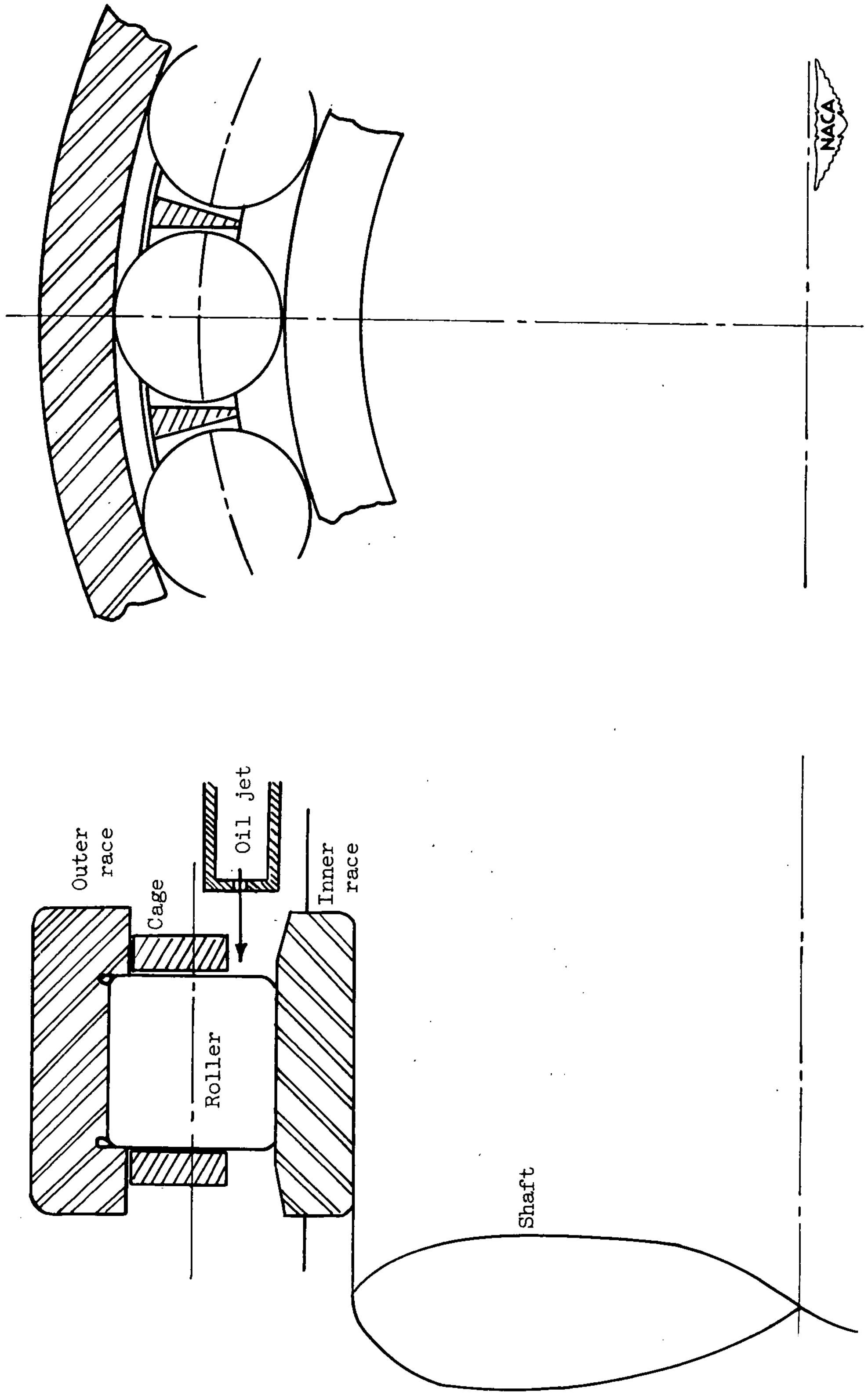


Figure 2. - Schematic drawing of test bearings showing location of oil jet.

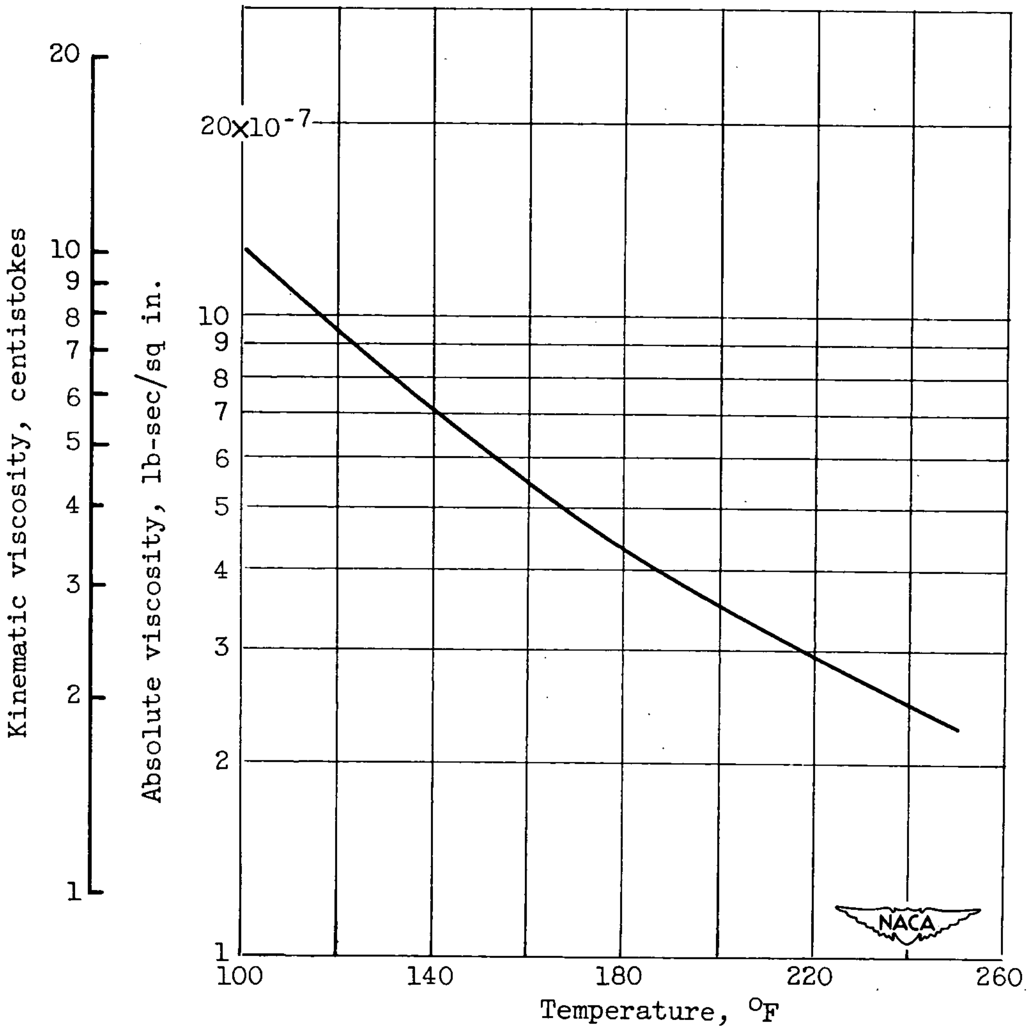
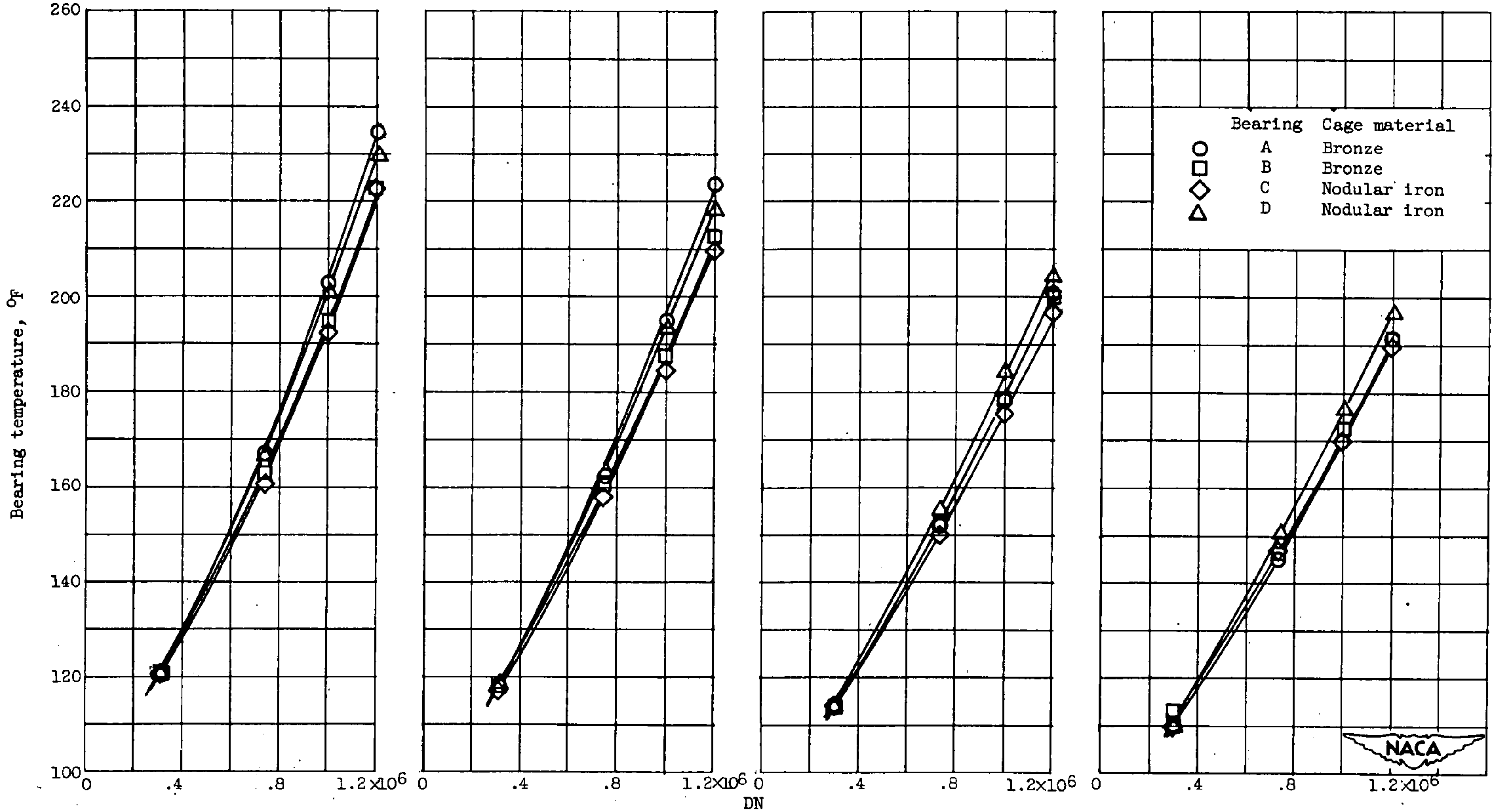


Figure 3. - Effect of temperature on kinematic and absolute viscosities of test oil. Pour point, less than -75° F; flash point, 300° F; viscosity index, 75; autogenous ignition temperature, [a]500° F.

[a]Time lag before ignition at temperature indicated was under 2 min.



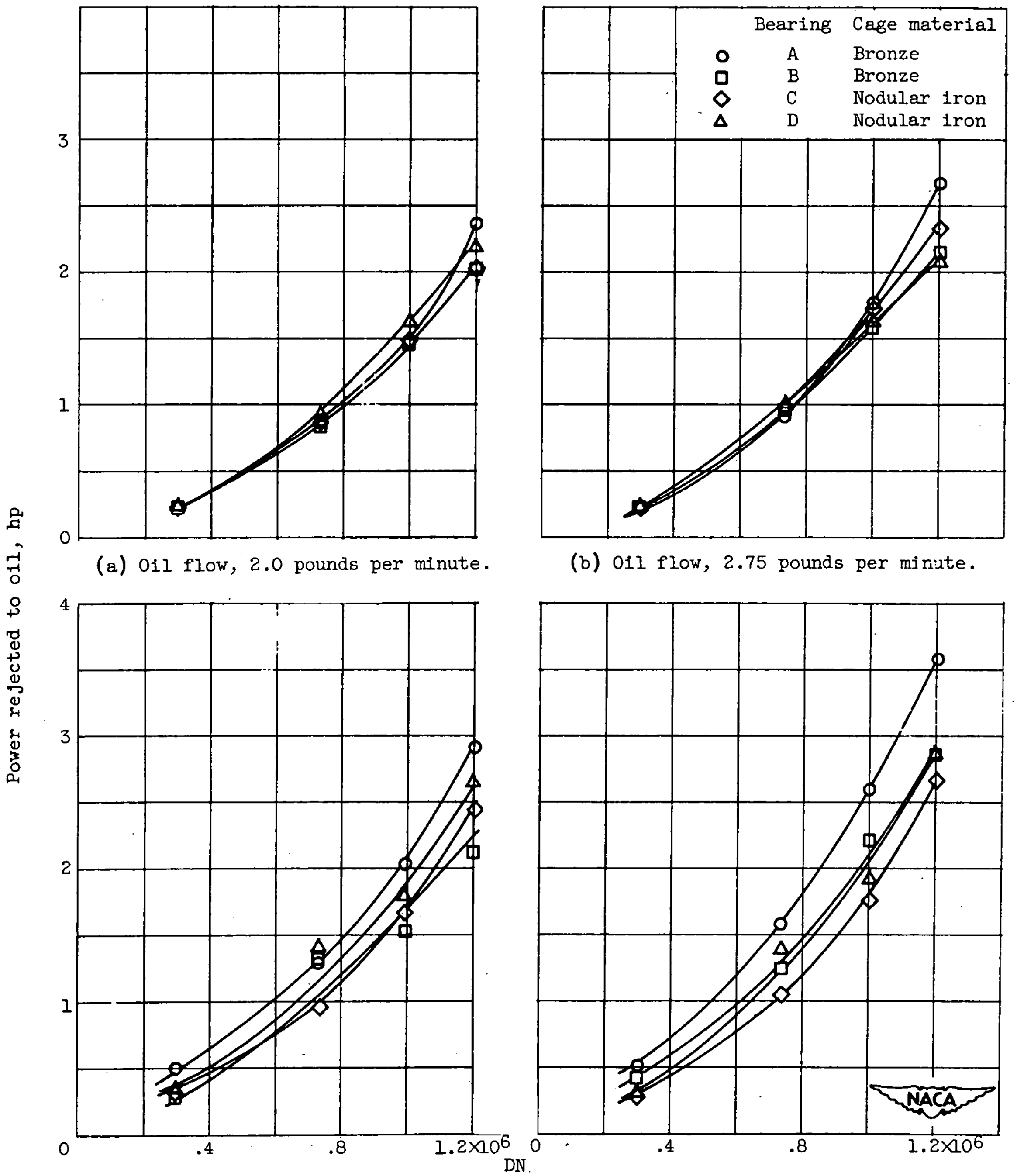
(a) Oil flow, 2.0 pounds per minute.

(b) Oil flow, 2.75 pounds per minute.

(c) Oil flow, 5.5 pounds per minute.

(d) Oil flow, 8.0 pounds per minute.

Figure 4. - Effect of DN on outer-race maximum temperatures of bearings A, B, C, and D for four oil flows. Load, 368 pounds; oil inlet temperature, 100° F.



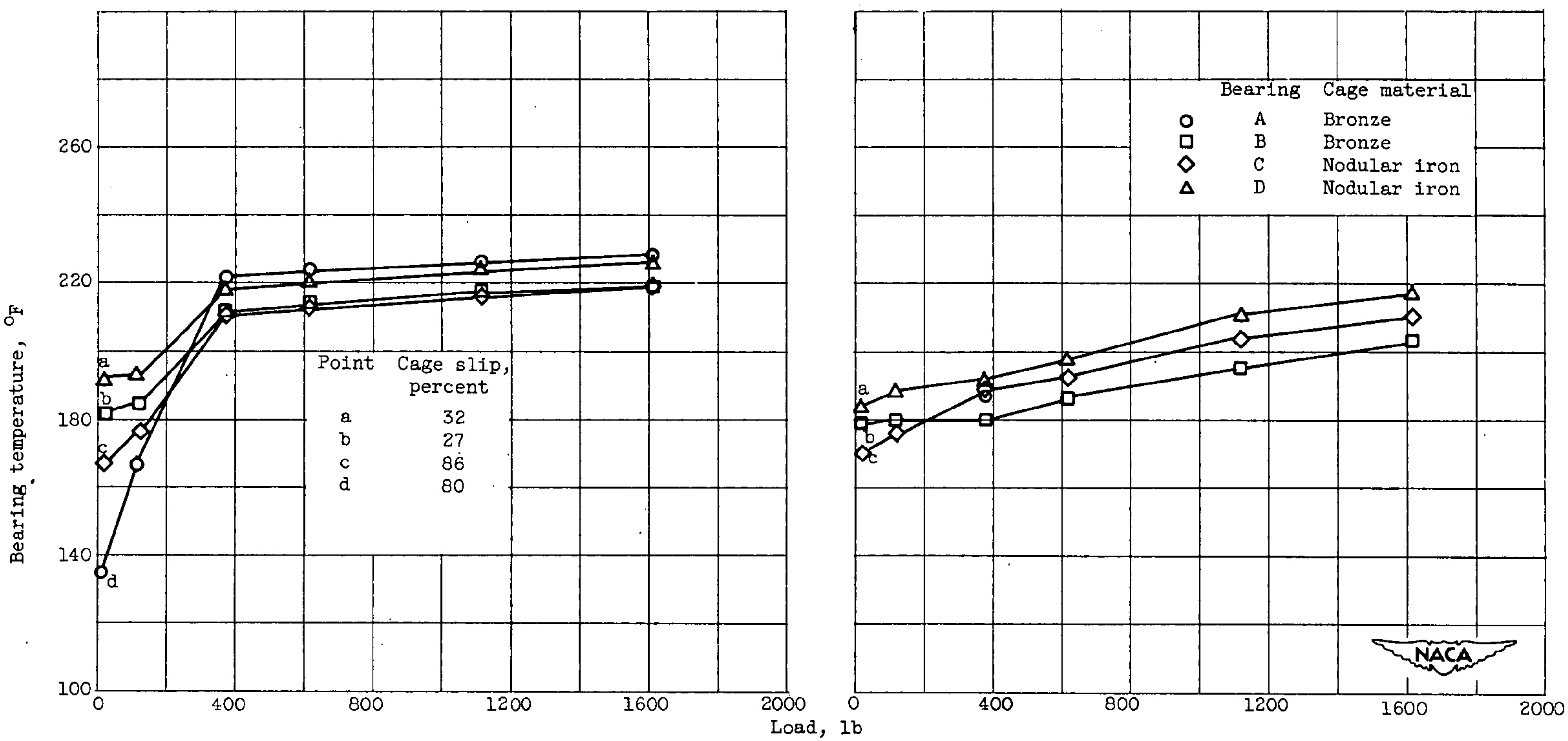
(a) Oil flow, 2.0 pounds per minute.

(b) Oil flow, 2.75 pounds per minute.

(c) Oil flow, 5.5 pounds per minute.

(d) Oil flow, 8.0 pounds per minute.

Figure 5. - Effect of DN on power rejected to oil in bearings A, B, C, and D at four oil flows. Load, 368 pounds; oil inlet temperature, 100° F.



(a) Outer-race maximum temperature.

(b) Inner-race temperature.

Figure 6. - Effect of load on outer-race maximum and inner-race temperatures of bearings A, B, C, and D. DN, 1.2×10^6; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F.

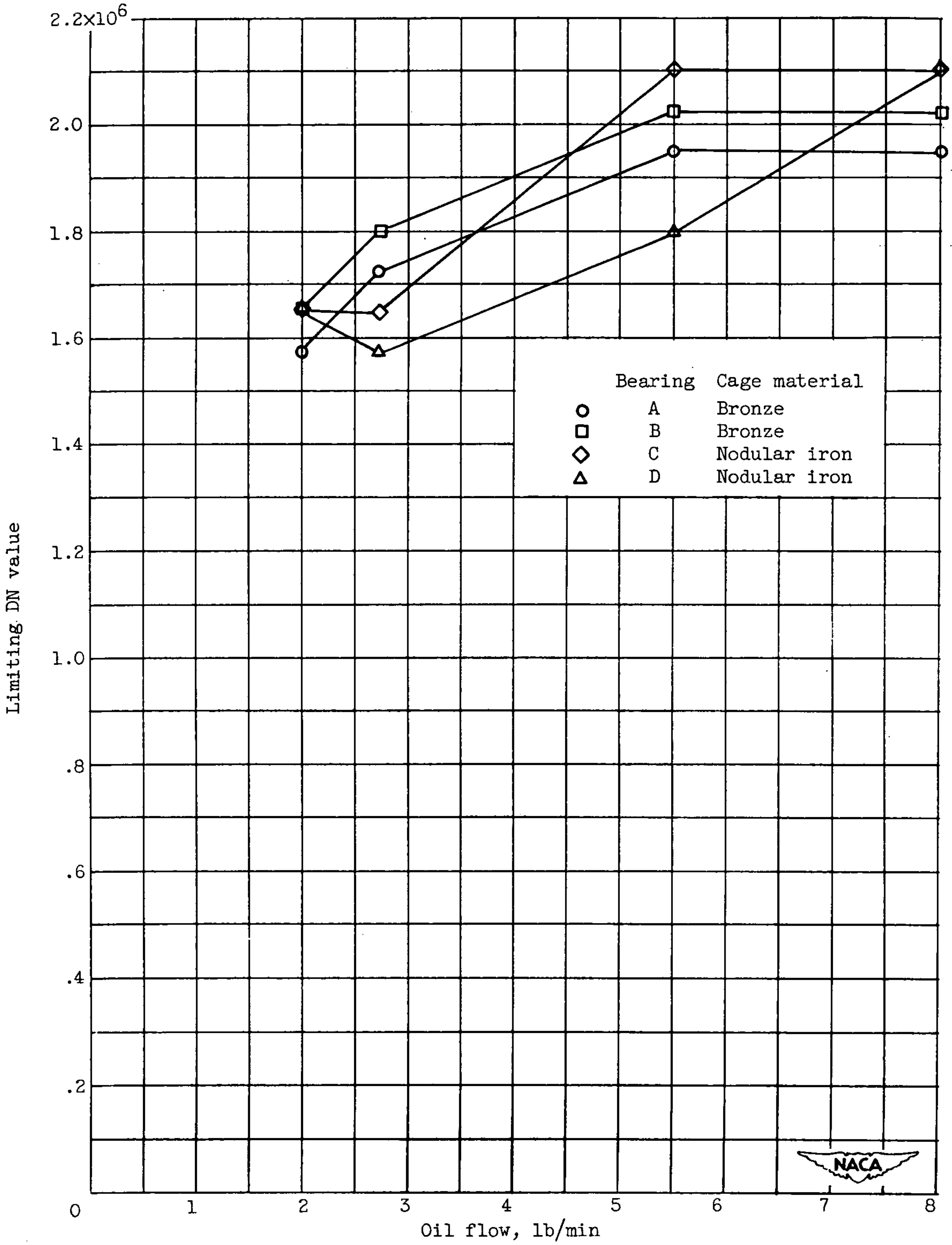


Figure 7. - Effect of oil flow on limiting DN values for bearings A, B, C, and D. Load, 368 pounds; oil inlet temperature, 100° F.

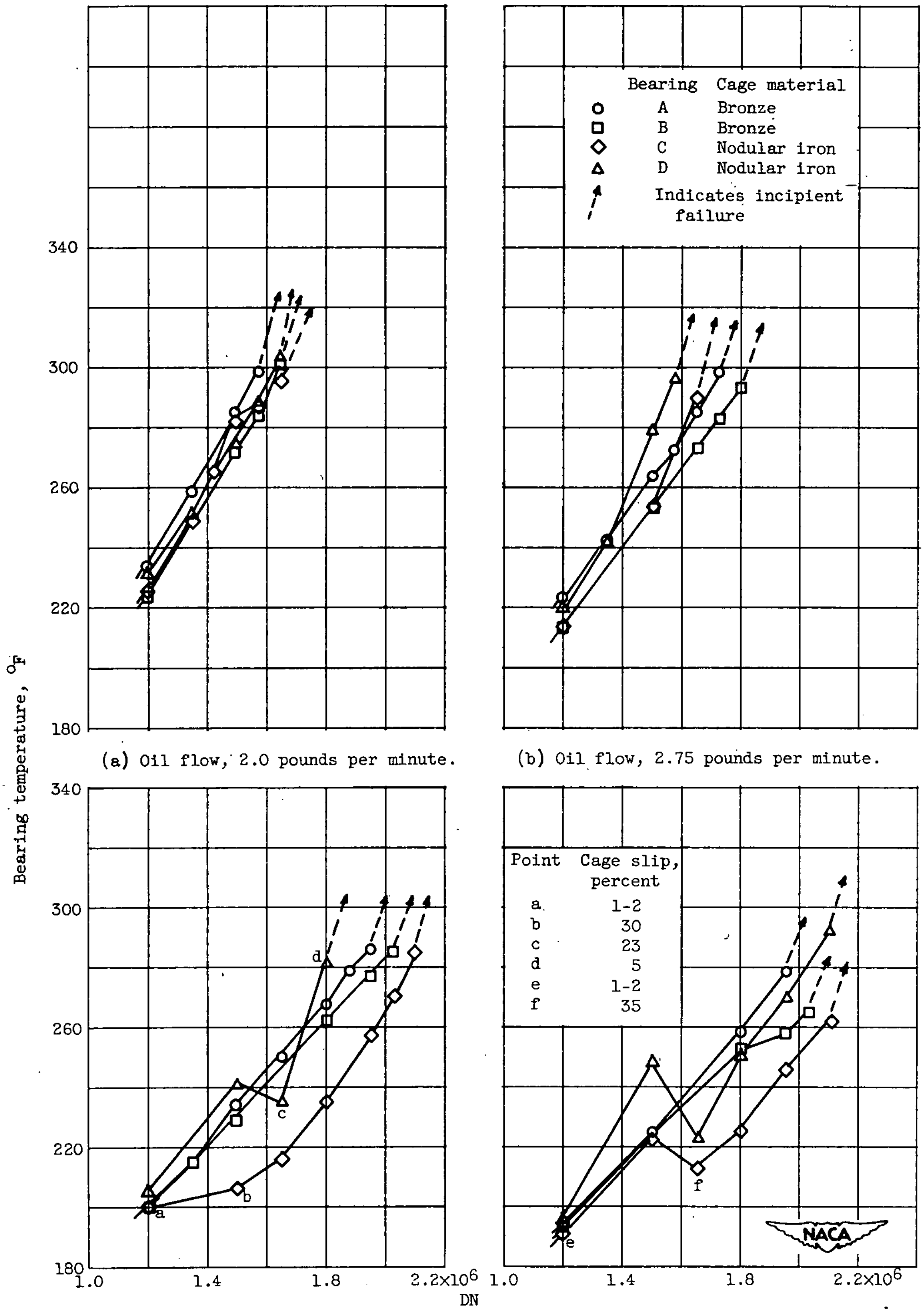


Figure 8. - Bearing outer-race-maximum temperature as affected by DN values from 1.2×10^6 to the limiting value at four oil flows for bearings A, B, C, and D. Load, 368 pounds; oil inlet temperature, 100° F.