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RESEARCH MEMORANDUM

PRELIMINARY COMPARISON OF 17- AND 75-MILLIMETER-BORE CAGELESS CYLINDRICAL ROLLER BEARINGS WITH CONVENTIONAL CYLINDRICAL ROLLER BEARINGS AT HIGH SPEEDS

By E. Fred Macks, W. J. Anderson, and Zolton N. Nemeth

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NATIONAL ADVISORY COMMITTEE
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SUMMARY

Conventional and special designs of 17- and 75-millimeter-bore cageless roller bearings were compared by means of a single test of each bearing type, over a DN range of 0.035×10^6 to 1.2×10^6 (DN is the product of the bearing bore in mm times the shaft speed in rpm), loads from 2 to 1120 pounds, and lubricant-flow rates from 0 to 12.7 pounds per minute. Preliminary results at high speeds indicate lower bearing temperatures, less internal bearing wear, and greater reliability of the conventional, cage-type cylindrical roller bearings than of either full-complement cageless cylindrical roller bearings or special cageless roller bearings of the types investigated. However, full-complement cageless roller bearings as well as special cageless roller bearings of the type investigated have been designed to operate successfully at DN values to 1.0×10^6. A special 1.112-inch-pitch-diameter cageless roller bearing ran without oil supply for 2 hours 3 minutes at 16,600 rpm and a load of 600 pounds per square inch without showing distress or visible damage. Investigation of a special cageless roller bearing made of bonded tungsten carbide containing 11 percent cobalt indicated that this material may be too brittle to be used for cylindrical roller bearings.

Under particular conditions of operation, both the full-complement cageless cylindrical roller bearing and the special cageless cylindrical roller bearing have favorable operating characteristics. Before definite conclusions can be drawn regarding these particular operating conditions, more extensive experimental data must be obtained.

INTRODUCTION

A high-speed, high-temperature, long-life bearing to support radial load and to operate under conditions of oil interruption is desired for applications such as the turbine support bearing in gas-turbine units for aircraft propulsion.

A large number of present bearing failures of gas-turbine units and other high-speed bearing applications have been attributed to lubrication or cage failures (refs. 1 to 3). Newer engines in the development stage encounter excessive bearing-cage and roller wear in addition to cage failures. It was felt that a roller bearing without a cage might eliminate the source of trouble and therefore offer a solution to the cage problem. The possibility of a bearing without a cage therefore invites interest and is presently available in the needle or full-complement roller bearing. Of added interest is the possibility of a special cageless roller bearing without the roller-to-roller sliding encountered in conventional needle bearings or conventional full-complement roller bearings.

As is true in most rolling contact bearing research, theoretical analysis does not give quantitative results of bearing operating characteristics, such as running temperature, friction torque, and end thrust, nor does it indicate the effect of oil flow and geometric or dimensional differences. Therefore, theory must be correlated with test results, and only then may it be used to predict bearing performance. The influence of geometry upon the operating characteristics of needle bearings is reported in references 4 and 5, and that of roller bearings with cages in references 6, 7, and 8. This investigation was conducted at the NACA Lewis laboratory in order to determine the applicability of cageless roller bearings for high-speed operation. The design parameters and operating characteristics of such special cageless roller bearings which were evolved during the present investigation are included herein.

Two sizes of bearing were employed in this experimental investigation: (a) $1\frac{1}{8}$-inch pitch diameter, and (b) approximately 4-inch pitch diameter. Conventional cylindrical roller bearings, needle bearings, and a special cylindrical cageless roller bearing of NACA design with "theoretically pure rolling motion" are compared at speeds to 16,600 rpm (DN values to 0.316×10^6) under various loads and oil flows for the $1\frac{1}{8}$-inch-pitch-diameter bearings. In the larger size, conventional cage-type cylindrical roller bearings, full-complement cylindrical roller bearings, and special cageless cylindrical roller bearings having "theoretically pure rolling motion" are compared at speeds to 16,000 rpm (DN values to 1.2×10^6) under various loads and oil flows.

APPARATUS

The apparatus section is divided into two parts for convenience: (a) small apparatus, used for small-bearing investigation ($1\frac{1}{8}$-in. pitch diameter), and (b) large apparatus, used for the large-bearing investigation (approximately 4-in. pitch diameter).

Small Apparatus

Test rig. - The experimental unit used for the present investigation was essentially the same as that described in references 4 and 5. The basic elements of the test rig are shown in figure 1(a). Radial load was applied to the test bearing by means of a lever and dead weight. The speed of the test shaft was variable over the range from 1600 to 17,000 rpm. The test bearing was mounted in a housing which was isolated from the rest of the rig by a film of externally pressurized oil. This permitted the friction torque and the end thrust of the test bearing to be measured. Other variables measured included speed, load, lubricant flow, bearing outer-race temperature, and lubricant inlet and outlet temperatures. In addition, the inner-race bearing temperature was measured in certain of the tests.

The two shafts used, A and B, are shown in figures 1(b) and 1(c). Shaft A was used for the needle-bearing tests, while shaft B was used for the conventional roller and special cageless roller-bearing tests. Lubricating oil was supplied under pressure to the test bearing through a hole in the inner race as shown in figure 1(a). Lubricating oil was supplied to the test bearings of shaft B by means of a single jet of 0.015-inch orifice diameter directed between the inner race and the cage in the case of the conventional roller bearing. In the case of the special cageless roller bearing, the oil jet was directed through one of the holes in the outer-race flange in the plane of the load vector in the loaded region.

Test bearings. - The needle bearing, conventional roller bearing, and special cageless roller bearing were designed with pitch diameters in the neighborhood of $1\frac{1}{8}$ inches. The inner races of all the bearings were a "thumb-push" fit on the shaft and were clamped axially by the spherical washer and bolt arrangement shown in figure 1(b). The needle bearing (bearing 3a of table I) was equipped with 56 cylindrical rolling elements of 0.0625-inch diameter and 0.22-inch length. The inner race of the needle bearing had a 1/8-inch radial hole at the axial center line for lubrication. Further physical characteristics of the test bearings are given in table I.

The conventional roller bearing (bearing 1a of table I) was equipped with nine cylindrical rolling elements of 0.250 diameter and 0.250 length, which were spaced by a two-piece inner-race-riding brass cage.

The special cageless roller bearing (bearing 2a of table I and fig. 2) was designed to prevent sliding contact between the rolling elements (ref. 4). The principle of operation of this bearing is as follows: The 12 small rollers are forced against the 12 larger rollers by centrifugal force and are friction driven by these larger, or load, rollers at a speed proportional to the roller diameters. However, if for any reason the small rollers are forced toward the inner race, the outer-race lips will prevent the small rollers from touching the inner race and, therefore, from sliding on the inner race. With proper design, pure rolling will also occur on the outer-race lips; and thus this bearing will have theoretically pure rolling motion except at the roller ends, that is, if $V_1 = V_5$ (see fig. 2). Therefore, since

$$V_1 = V_2\left(\frac{r_1}{r_2}\right) = V_2\left(\frac{d_1}{d_2}\right) \tag{1}$$

and

$$V_5 = V_6\left(\frac{r_5}{r_6}\right) \tag{2}$$

it follows that if the ratio $\frac{r_5}{r_6}$ is equal to $\frac{r_1}{r_2}$, pure rolling is theoretically possible even though the small rollers are forced against the outer-race lips. (V = surface velocity, r = radius, and d = diameter.) Other variations of this bearing may be designed, depending on the choice of roller and race diameter ratios. However, the only basic variation investigated is that shown in figure 2.

Large Apparatus

Test rig. - The bearing rig (fig. 3) used for evaluating the 75-millimeter-bore bearings (approximately 4-in. pitch diameter) is the same as that used in reference 8. The bearing under investigation was located on one end of the test shaft, which was mounted in cantilever fashion, in order to observe bearing component parts and lubricant flow during operation. A radial load was applied to the test bearing by means of a lever and dead-weight system. The variables measured included speed, load, oil flow, and inner- and outer-race bearing temperatures.

The method of temperature measurement is described in reference 8. Iron-constantan thermocouples were located at 60° intervals around the outer-race periphery at the axial center line of the bearing under investigation. A copper-constantan thermocouple was pressed against the bore of the inner race at the axial midpoint of the test bearing; the voltage was transferred from the rotating shaft by means of slip rings.

Lubricant was supplied to the conventional and full-complement roller bearings by means of a single jet of 0.089-inch diameter directed at the space between the cage and the inner race and perpendicular to the bearing face. Lubricant was supplied to the special cageless roller bearings through one of the holes in the outer-race flange in the plane of the load vector in the unloaded region.

<u>Test bearings</u>. - The conventional roller bearing, full-complement roller bearing, and special cageless roller bearing all had pitch diameters in the neighborhood of 4 inches. All the bearings of this size (except the special cageless roller bearing made of tungsten carbide) had an approximately 0.001-inch shrink fit on the shaft to eliminate relative motion and therefore fretting and galling between the inner race and shaft surfaces at the extreme DN values encountered.

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The conventional cylindrical roller bearing (bearing 6 of table I) was equipped with a one-piece inner-race-riding brass cage which spaced 18 cylindrical rolling elements of 0.551-inch diameter by 0.551-inch length.

Two large special cageless roller bearings (bearings 4a and 5a of table I) of the same basic design (fig. 4) as the small special cageless roller bearing were designed and tested. Test bearing 4a was made entirely from tungsten carbide while test bearing 5a was made from SAE 52100 chrome steel. There were also slight design differences of the two bearings which may be noted by comparing figures 4(a) and 4(b). The roller crowning and roller end design are different for the two bearings. Each bearing was equipped with 20 cylindrical load rollers and 20 idler rollers.

Three full-complement cageless roller bearings (bearings 17, 19, and 20 of table I) were also investigated. Two of the bearings of this type were equipped with 22 cylindrical rollers, while the third was equipped with 23 cylindrical rolling elements. The circumferential clearance per roller was the primary variable in these bearings, being 0.011 inch for bearing 17, 0.0019 inch for bearing 19, and 0.023 inch for bearing 20.

PROCEDURE

The oil used during the entire investigation reported herein was a commercial, highly refined paraffin-base blend including a small percentage of polymer to improve the viscosity index. Viscosity and specific gravity of the oil are given in figure 5.

The oil was supplied to the test bearings at an inlet temperature of approximately 100° F and at flows ranging from 0 to 5 pounds per hour in the case of the small bearings and from 0 to 12.7 pounds per minute in the case of the larger bearings. Circulating oil feed was used in all the tests reported herein.

During the tests to determine the effect of speed upon the bearing operating characteristics, the load and the oil flow were held constant. After the test bearing temperature had essentially reached equilibrium, data were recorded and the speed was increased to the next predetermined value. During a run to determine the effect of oil flow upon the operating characteristics, the speed and the load were held constant; after equilibrium was essentially established, data were recorded and the oil flow was increased to the next predetermined value.

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The total bearing load is given in pounds, and the unit bearing load in pounds per square inch, in order to facilitate comparison of the small and large bearings. In the latter case the unit load is based upon the bearing projected area (product of the load roller effective length and the bearing pitch diameter).

RESULTS AND DISCUSSION

Small-Bearing Investigation

Three tests were made for the small-bearing investigation. The operating data together with the physical characteristics of the test bearings are summarized in table I. These results are presented mainly for comparative purposes to facilitate evaluation of the different bearing designs.

Test 1. - Test 1 was conducted with bearing 1a (fig. 1(c)), which was a conventional, size 203 cylindrical roller bearing with a two-piece inner-race-riding brass cage having a pitch diameter of 1.112 inches. The bearing was operated over a speed range from 2000 to 17,000 rpm (maximum DN value 0.282×10^6). The load was varied from 2 to 169.5 pounds (7 to 600 lb/sq in.). The lubricant-flow rate was from 3 to 5 pounds per hour.

The effect of speed and load on bearing outer-race and inner-race temperatures with load as parameter is shown in figure 6(a). It may be observed that both outer-race and inner-race bearing temperatures increase approximately linearly with an increase in speed. Bearing load increases bearing operating temperature of both races over the load range investigated. The inner-race bearing temperature is seen to be significantly higher than the outer-race bearing temperature except at low speeds.

Test 2. - Test 2 was conducted with bearing 2a, which was a special cageless roller bearing of 1.112-inch pitch diameter. The bearing was operated over a speed range from 2000 to 17,000 rpm (maximum DN value 0.282×10^6), a load range from 2 to 169.5 pounds per square inch, and a lubricant-flow rate from 0 to 5 pounds per hour.

The effect of bearing speed on outer- and inner-race bearing temperatures with load as parameter is shown in figure 6(b). It is seen that both outer- and inner-race bearing operating temperatures increase approximately linearly with an increase in speed and that the inner race operates hotter than does the outer race. It is also seen that an increase in load increases both race temperatures. The bearing after test is shown in figure 7.

Test 3. - Test 3 was conducted with bearing 3a, which was a needle-type bearing having a pitch diameter of 1.117 inches.

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The bearing was operated over a speed range from 2000 to 16,600 rpm (maximum DN value 0.316×10^6), a radial load range of 2.5 to 147 pounds (10 to 600 lb/sq in.), and a lubricant-flow rate from 0.5 to 5 pounds per hour. The effect of speed on outer-race bearing temperature, friction torque, and end thrust with load as parameter is shown in figure 8. It is seen that bearing temperature and friction torque are influenced by bearing load. End thrust increases appreciably with bearing load even though the end thrust changes direction. More extensive data on the operating characteristics of needle-type bearings are given in references 4 and 5 and 9 to 13.

Comparison of Small Bearings

Effect of load and speed. - Test bearings 1a and 2a are compared in figure 9 over a speed range from 2000 to 16,600 rpm (DN range to 0.282×10^6) for loads of 2 and 169.5 pounds. Outer-race and inner-race bearing temperatures, friction torque, and end thrust are shown. It is seen that test bearing 1a (conventional roller bearing) operated at lower inner- and outer-race bearing temperatures over the speed range investigated, the difference in raceway temperatures decreasing at the higher load. It may be observed that the friction torque of the conventional roller bearing (1a) was appreciably less over the entire speed range. There is little difference in end thrust of both bearing types at the low load; however, at the higher load the end thrust of the conventional roller bearing is appreciably less than that of the special cageless roller bearing.

Effect of oil flow. - A comparison of the operating temperatures of bearings 1a, 2a, and 3a as affected by oil flow is given in figure 10. Inner-race bearing temperature was not measured in the case of bearing 3a. Here again it is seen that the conventional roller bearing operated at lower bearing temperatures over the flow range investigated than did either the special cageless roller bearing or the needle-type bearing.

After a run at an oil flow of 3 pounds per hour, the oil was shut off and bearing 2a was allowed to operate at 16,600 rpm and 600-pounds-per-square-inch load to determine whether the bearing would come to equilibrium temperature. It is seen from figure 10 that at zero oil flow the bearing equilibrium operating temperature was very nearly the same as it was at an oil flow of 5 pounds per hour and somewhat less than it was at oil flows of both 4 and 3 pounds. This result may be explained in part by the reduced oil churning at zero oil flow. The bearing was operated for a period of 2 hours 3 minutes at zero oil flow prior to a support bearing failure. Post-test examination of the special cageless roller bearing (bearing 2a) showed no visible damage or distress of any nature (see fig. 7). 2679

Many significant variables, such as surface finish, internal bearing clearances, number of rolling elements, and so forth, may alter a comparison of the subject bearing types considerably. However, it may be stated from the available investigations that the conventional roller bearing operates at lower temperatures than does the needle-type bearing and has a slightly improved performance over the special cageless roller bearing when provided with a copious supply of lubricant. It is noteworthy, however, that the special cageless roller bearing operated satisfactorily over the range of conditions investigated and particularly that it operated without lubricant for an extended period at a speed of 16,600 rpm and a load of 600 pounds per square inch.

Large-Bearing Investigation

Six tests were made in the large-bearing investigation, which included a comparison of special cageless roller bearings and full-complement cageless roller bearings with a conventional roller bearing having a one-piece inner-race-riding cage. The operating data and the physical characteristics of the test bearings are summarized in table I. Extensive data on the high-speed operating characteristics of conventional roller bearings are given in references 6 and 14 to 16.

Test 4. - Test 4 was conducted with bearing 4a (fig. 11), which was a special cageless roller bearing made entirely of bonded tungsten carbide (11 percent cobalt). This bearing had a pitch diameter of 4.223 inches. The bearing was operated over a speed range from 4000 to 16,000 rpm (DN value of 0.3×10^6 to 1.2×10^6), a load of 10 pounds, and a lubricant-flow rate from 0 to 0.42 pounds per minute. (The bearing did not reach an equilibrium temperature at 16,000 rpm.)

The results of this test (fig. 12) are inconclusive, since two factors were unknown. The length of time that the lubricant flow of 0.42 pound per minute passed through the bearing during the first run was unknown, inasmuch as at the conclusion of the first run it was found that because of circumferential creep of the outer race the oil could not pass through the hole in the outer-race flange during the entire run, but could only wash over one bearing face. Secondly, it was not known whether the bearing was damaged during the first run. (It is believed the bearing was damaged, since the operating temperatures of the second run were greater than those of the first.) In order to determine whether oil churning was responsible for the high temperatures, the oil was shut off

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completely at a DN value of 0.755×10^6 during the second run (see point A, fig. 12); however, bearing temperatures did not attain the low values of the first run. Ordinarily the steep rise in bearing temperatures from a DN of 1.0×10^6 to 1.2×10^6 during the first run would have meant incipient bearing failure. However, when the rig was shut down, and it was found that the outer race had crept circumferentially, it was believed that the sharp temperature rise was due to lack of oil supply. This fact was substantiated by the free turning of the test shaft. However, the results of the second run indicate that a major change had occurred in the test bearing, perhaps an incipient failure not of sufficient severity to prevent free shaft rotation.

The bearing failed when running at 11,300 rpm, 10 pounds load, and 0.42-pound-per-minute oil flow. Post-test examination of the bearing (fig. 13) showed the outer race cracked, all the load rollers badly chipped along the edges, one small roller broken, and the inner race track surface roughened. Chipping of the ends of the brittle tungsten carbide load rollers is believed to have been the first event leading to complete bearing failure.

In view of the uncontrolled variables of oil flow and bearing failure, the data of figure 12 may not be significant.

Test 5. - Test 5 was conducted with bearing 5a, which was a special cageless roller bearing having a pitch diameter of 4.223 inches. This bearing was in general similar to bearing 4a, except that it was made of SAE 52100 steel and had a few design modifications (see fig. 4). The bearing was operated over a speed range from 4000 to 16,000 rpm (DN range, 0.3×10^6 to 1.2×10^6), load range from 10 to 365 pounds, and lubricant-flow rate from 1.4 to 5.9 pounds per minute.

The effect of speed on outer-race-maximum and inner-race bearing temperatures of special cageless roller bearing 5a and of the conventional bearing 6 are shown in figure 14 at a load of 368 pounds and an oil flow of 2.75 pounds per minute. Both raceway bearing temperatures increased approximately linearly with an increase in speed up to a DN of 1.0×10^6. (The bearing failed at a DN of 1.2×10^6.) It may be seen that bearing 5a operated at somewhat higher inner- and outer-race equilibrium temperatures (particularly at the higher speeds) than did the conventional bearing, the operating characteristics of which are included on figure 14 for comparative purposes. Bearing 5a is shown after test in figure 15. The inner race was damaged severely and cracked, the large rollers were out of round and badly worn, and the small rollers were bent at the ends; in some cases the small ends were fractured.

Evidently failure was caused by one or more of the small rollers escaping radially beyond the pitch diameter of the large rollers, causing jamming. The failure was not caused by breakage of the small rollers at the small diameter because of differential expansion of the clearance ring against them, as this surface showed no sign of distress (see fig. 15).

Extensive research regarding critical bearing tolerances must be conducted if a valid evaluation of bearing types 4a and 5a is to be made.

Test 6. - Test 6 was conducted with bearing 6, which was a 75-millimeter-bore cylindrical roller bearing equipped with a one-piece inner-race-riding brass cage. This bearing is the same as reported in reference 8 and is one of the better bearings tested to date. The operating characteristics of this bearing are taken from reference 8 and are shown for comparative purposes in figure 14. 2679

Full-Complement Cageless Roller Bearings

Test 7. - Test 7 was conducted with bearing 17, which was a full-complement cageless roller bearing equipped with 22 cylindrical rolling elements each having a length-diameter ratio of 1. The circumferential clearance per roller was 0.011 inch. The bore of the test bearing was 75 millimeters, and the pitch diameter was 4.033 inch.

The bearing was operated over a speed range from 4000 to 16,000 rpm (DN range from 0.3×10^{6} to 1.2×10^{6}), a load range from 7 to 1113 pounds, and a lubricant-flow rate from 2.8 to 12.7 pounds per minute. The operating characteristics of this bearing are compared with other full-complement cageless roller bearings and with the conventional roller bearing (bearing 6) in later figures.

Test 8. - Test 8 was conducted with bearing 19, which was a full-complement cageless roller bearing having 23 cylindrical rolling elements with a length-diameter ratio of 1. The circumferential clearance per rolling element was 0.0019 inch. The bore of the test bearing was 75 millimeters, and the pitch diameter was 4.032 inches.

The bearing was operated over a speed range from 4000 to 9785 rpm (DN range of 0.3×10^{6} to 0.735×10^{6}), a load of 368 pounds, and a lubricant-flow rate of 2.5 pounds per minute.

Test 9. - Test 9 was conducted with bearing 20, which was a full-complement cageless cylindrical roller bearing having 22 cylindrical rolling elements with a length-diameter ratio of 1. The circumferential clearance per roller was 0.023 inch. The bearing was of 75-millimeter bore and 4.036 inches pitch diameter.

The bearing was operated over a speed range from 4000 to 16,000 rpm (DN range 0.3×10^{6} to 1.2×10^{6}), a load of 368 pounds, and a lubricant-flow rate of 2.5 pounds per minute.

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Comparison of Large Bearings

Comparison of full-complement cageless roller bearings. - The outer-race-maximum and inner-race bearing temperatures of test bearings 17, 19, and 20 are shown in figure 16 over a DN range of 0.3×10^6 to 1.2×10^6. It is seen that both raceway temperatures of each bearing increased approximately linearly with an increase in speed. A direct comparison cannot be made, because the lubricant-flow rate to bearing 17 was 2.8 pounds per minute, whereas the flow rate to bearings 19 and 20 was 2.5 pounds per minute.

It was impossible to maintain an equilibrium temperature of test bearing 19 above a DN value of 0.735×10^6, in spite of the fact that the inner-race operating temperature of this bearing was considerably less than the inner-race temperatures of either bearing 17 or 20.

Because of the effect of running time on full-complement bearing operating temperature (see fig. 17) the data of figure 16 should be evaluated with caution.

One of the most important results of this test series is the fact that bearing 17 showed negligible wear after the tests, whereas bearings 19 and 20 showed appreciable wear. The effect of running time on outer-race bearing operating temperature is shown indirectly for bearings 19 and 20 in figure 17, where bearing operating temperature is plotted against DN value. It is interesting to note that bearing operating temperature decreases for the second and the third runs for each bearing at equivalent operating conditions. The decrease in the operating temperatures of bearings 19 and 20 with running time might be explained by an increase in the running clearance. The increase in running clearance represents either a substantial decrease in heat generated or a substantial increase in the deflected-to-transmitted oil-flow ratio and therefore a substantial increase in the effective cooling of a given total flow of oil. The decrease in bearing operating temperature is associated with abnormal roller and raceway wear of the "washboard" type. This roller wear for bearing 20 is shown clearly in figure 18.

A stroboscopic investigation of the operation of the full-complement cageless roller bearings was conducted at high speed, and it was observed that the rolling elements were continually bouncing off one another as they proceeded around the circumference of the bearing from the loaded region to the unloaded region. This action, together with inherent sliding at very high speeds in a full-complement bearing, produced the roller wear. Another variable which may be of extreme importance in high-speed cageless roller bearings is uniformity of size among rollers. A lack of uniformity among rollers might cause "roller bouncing" of the type observed, because the small rollers would bounce loosely back and forth between two large or driving rollers. The rollers in these tests were measured to within 0.0001 inch. Roller uniformity to within a few millionths of an inch may be required for satisfactory operation. The negligible wear of bearing 17 may be explained by the fact that the circumferential clearance per roller might have been an optimum value, or that greater roller uniformity prevailed, or both. Experimental verification of the effect of circumferential clearance per roller and of roller uniformity on high-speed operating characteristics of cageless roller bearings might bring fruitful results.

<u>Comparison of conventional and full-complement bearings</u>. - The outer-race-maximum and inner-race bearing temperatures for test bearings 6 and 17 are compared in figure 19(a) over a range of speeds from 4000 to 16,000 rpm (DN range 0.3×10^6 to 1.2×10^6). There is little difference in operating temperatures for the two bearings up to a DN value of 1.0×10^6.

The outer-race-maximum and inner-race bearing temperatures of bearings 6 and 17 are compared in figure 19(b) over a load range from 7 to 1113 pounds with DN as a parameter. There is only a slight effect of load on raceway temperatures, except at the very low loads. Both raceway temperatures are seen to be lower at a given DN value for the full-complement cageless roller bearing than for the conventional roller bearing at the lowest DN value. This trend is reversed at the higher DN values, where the conventional bearing shows superior operating performance.

Although deflected- and transmitted-oil flows for the test bearings were not investigated (since the bearing tests were conducted early in the high-speed bearing program investigation before the significance of these variables was appreciated), it is expected that the ratio of deflected-to-transmitted oil flows is relatively low for the full-complement cageless roller bearing compared with the conventional roller bearing. This statement is based on the restricted oil-entry path of the conventional, inner-race-riding cage-type roller bearing compared with the relatively unrestricted oil-flow path of the full-complement cageless roller bearing. It has been found in recent high-speed bearing research (ref. 17) that the outer-race bearing operating temperature decreases significantly with a decrease in deflected-to-transmitted oil-flow ratio for a given total flow. This fact leads to the tentative conclusion that considerably more heat is generated within the full-complement cageless roller bearing than in the conventional bearing, inasmuch as the outer-race-maximum bearing temperature was greater over the entire lubricant-flow range investigated in spite of its relatively greater cooling.

In spite of the fact that the cageless roller bearings were found to be comparable to the conventional bearing under certain conditions of operation, the number of variables involved and the small quantity of experimental data obtained to date indicate that the cageless bearing types may be fully evaluated for general high-speed service only after an extensive research program has been conducted. Until that time the conventional roller bearing appears to be better suited for high speeds;

its lower operating temperatures, lower wear, and greater reliability result from fewer critical sliding contacts.

SUMMARY OF RESULTS

Conventional and special designs of 17- and 75-millimeter-bore cageless roller bearings were compared by means of a single test of each bearing type over a DN range of 0.035×10^6 to 1.2×10^6 (DN is the product of the bearing bore in mm times the shaft speed in rpm), loads from 2 to 1120 pounds, and lubricant-flow rates from 0 to 12.7 pounds per minute.

1. This preliminary experimental investigation indicates that conventional cage-type cylindrical roller bearings are in general better suited for high-speed operation than are either full-complement cageless cylindrical roller bearings or special cageless roller bearings of the type investigated.

2. Full-complement cageless roller bearings as well as special cageless roller bearings of the type investigated have been designed to operate successfully at DN values to 1.0×10^6.

3. Significant variables regarding roller and raceway wear in full-complement cageless cylindrical roller bearings appear to be the circumferential clearance per rolling element and uniformity of rollers within a given bearing.

4. It is postulated that severe rolling element and raceway wear of full-complement cageless cylindrical roller bearings are camouflaged during operation by lubricant cooling due to low deflected-to-transmitted flow ratios associated with this bearing type.

5. Needle-type bearings and special cageless roller bearings of the type investigated operate at higher temperatures and with somewhat greater friction torque than do conventional cage-type cylindrical roller bearings.

6. The results of investigating a bonded tungsten carbide bearing containing 11 percent cobalt indicate that this material may be too brittle to be used as the element of cylindrical roller bearings.

7. A special 1.112-inch-pitch-diameter cageless roller bearing ran without oil supply for 2 hours 3 minutes at 16,600 rpm and a load of 600 pounds per square inch without showing distress or visible damage.

8. Under particular conditions of operation, both the full-complement cageless cylindrical roller bearing and the special cageless roller

bearing have favorable operating characteristics. Before definite conclusions can be drawn for these particular operating conditions, more extensive experimental investigation must be obtained.

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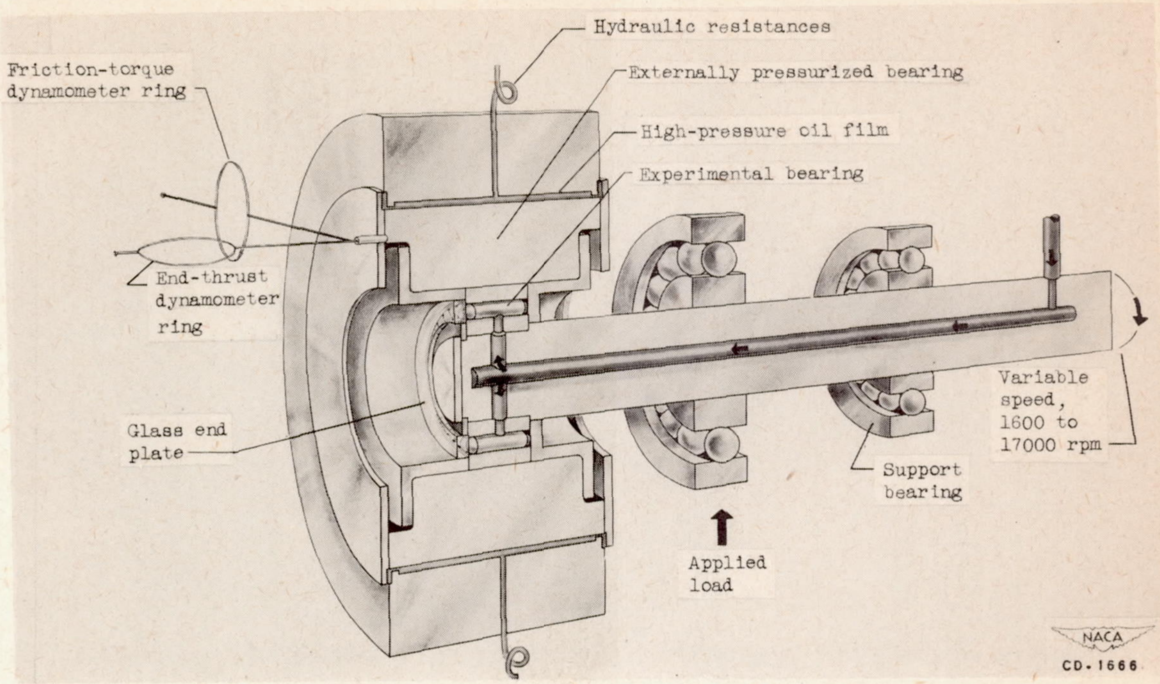
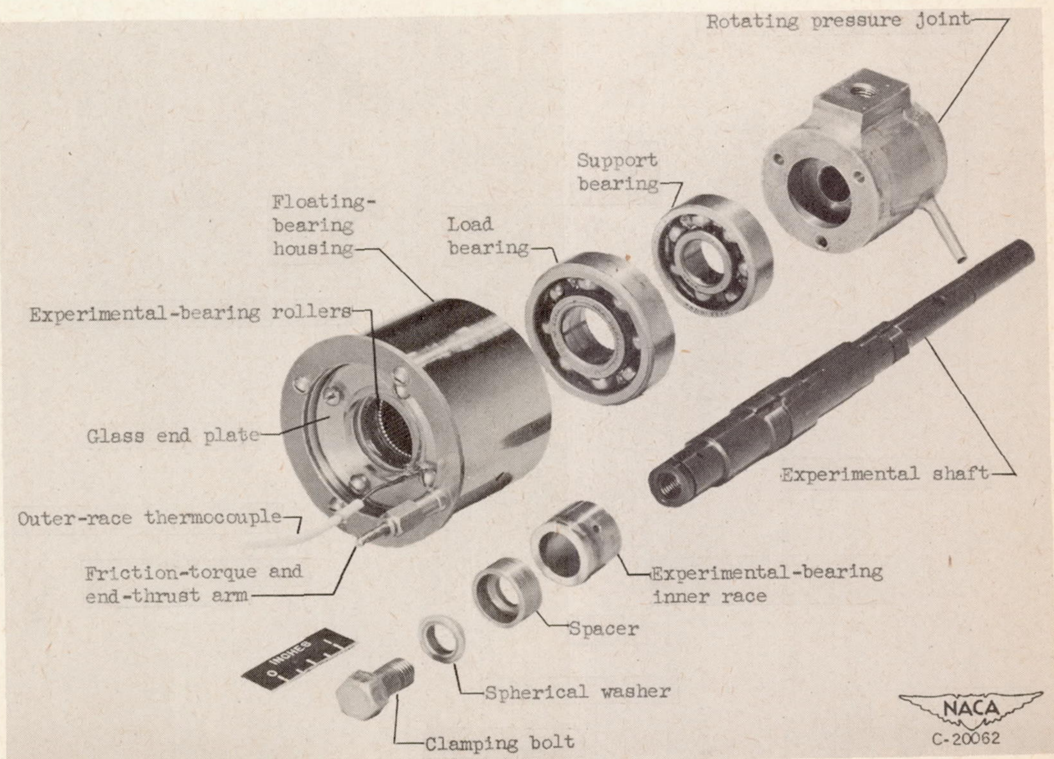
TABLE I - PHYSICAL CHARACTERISTICS OF TEST BEARINGS AND RUNNING DATA

Bearing size		Small			Large					
Bearing number		1a	2a	3a	4a	5a	6	17	19	20
Construction		Two-piece inner-race-riding cage	Special cageless with true rolling action	Needle type	Special cageless with true rolling action	Special cageless with true rolling action	One-piece inner-race-riding cage	Full-complement cageless	Full-complement cageless	Full-complement cageless
Bearing material		SAE 52100	Paragon tool steel (SAE 52100 rollers)	Paragon tool steel (SAE 52100 rollers)	Tungsten carbide (11 percent cobalt)	SAE 52100	SAE 52100	SAE 52100	SAE 52100	SAE 52100
Bearing bore, in.		0.6693	0.6693	0.75	2.953	2.953	2.953	2.953	2.953	2.953
Bearing width, in.		0.472	0.689	0.22	1.387	1.387	0.984	0.984	0.984	0.984
Bearing O.D., in.		1.575	1.575	1.500	5.118	5.118	5.118	5.118	5.118	5.118
Pitch diameter of bearing, in.		1.112	1.112	1.117	4.223	4.223	4.036	4.033	4.032	4.036
Roller diameter, in.		0.250	0.250 Load .063, .042 Idler	0.0625	0.563 Load .125, .101 Idler	0.563 Load .125, .101 Idler	0.551	0.564	0.548	0.551
Roller length, in.		0.250	0.250 Load .260, .480 Idler	0.22	0.660 Load .571, 1.039 Idler	0.560 Load .571, 1.042 Idler	0.551	0.563	0.551	0.551
Circumferential clearance per roller, in.		Not applicable	Not applicable	0.0001	Not applicable	Not applicable	Not applicable	0.011	0.0019	0.023
No. of rollers		9	12 Load 12 Idler	56	20 Load 20 Idler	20 Load 20 Idler	18	22	23	22
Bearing clearance, in.	Mounted	0.001	0.0004	0.0016	0.002	0.003	0.0009	0.0007	0.002	0.001
	Unmounted	0.001	0.0004	0.0016	0.002	0.003	0.0020	0.0017	0.003	0.002
Radial load, lb		2 to 169.5	2 to 169.5	2.5 to 147	10	10 to 365	7 to 1113	7 to 1113	368	368
Unit radial load[a], lb/sq in.		7 to 600	7 to 600	10 to 600	4.5 to 500	5.0 to 182	3.5 to 583	3.5 to 554	193	192
Maximum operating speed	DN	0.282×10^6	0.282×10^6	0.316×10^6	1.2×10^6	1.2×10^6	1.2×10^6	1.2×10^6	0.735×10^6	1.2×10^6
	rpm	17,000	17,000	16,600	16,000	16,000	16,000	16,000	9,785	16,000
Lubricant-flow rate		3 to 5 lb/hr	0 to 5 lb/hr	0.5 to 5 lb/hr	0 to 0.42 lb/min	1.4 to 5.9 lb/min	0.5 to 12 lb/min	2.8 to 12.7 lb/min	2.5 lb/min	2.5 lb/min
Remarks		Satisfactory operation	Satisfactory operation	Satisfactory operation	Bearing failed	Bearing failed	Satisfactory operation	Satisfactory operation	Bearing failed	Bearing failed

[a]Based upon product of effective load roller length and bearing pitch diameter.

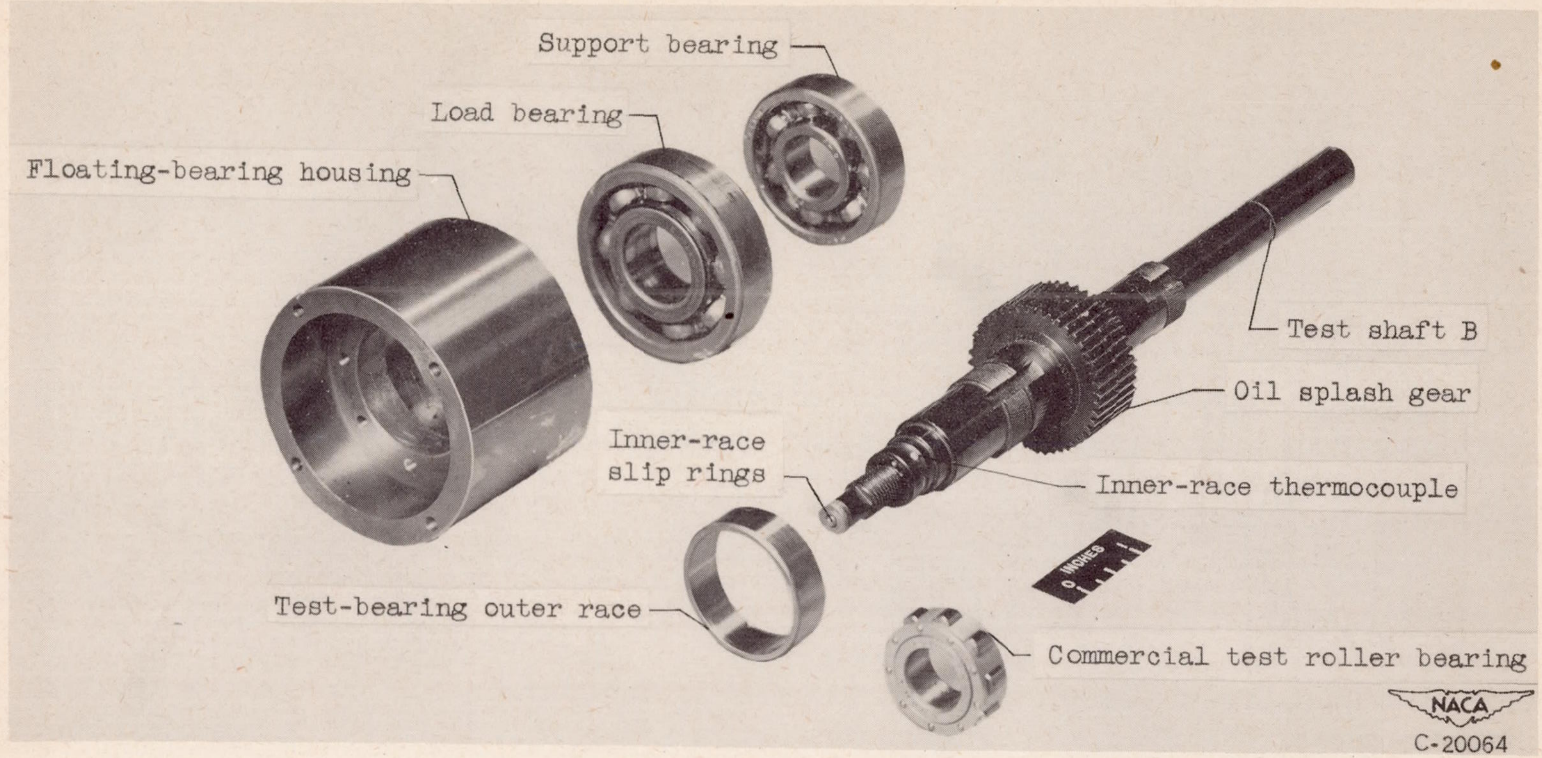


(a) Schematic diagram of shaft assembly.



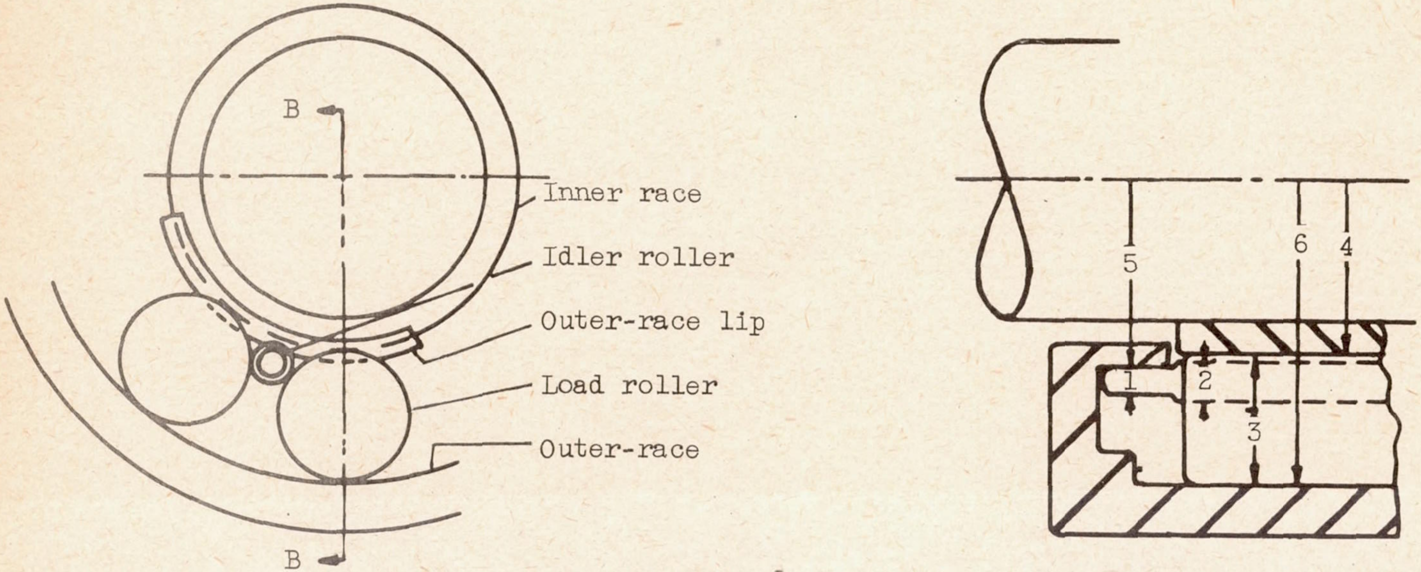
(b) Shaft A.

Figure 1. - Small test rig.



(c) Shaft B.

Figure 1. - Concluded. Small test rig.



Section B-B

If $\frac{r_1}{r_2} = \frac{r_5}{r_6}$ then the surface velocity of 1 will equal the surface velocity of 5 and pure rolling will occur

Figure 2. - Special cageless roller bearing, 2a.

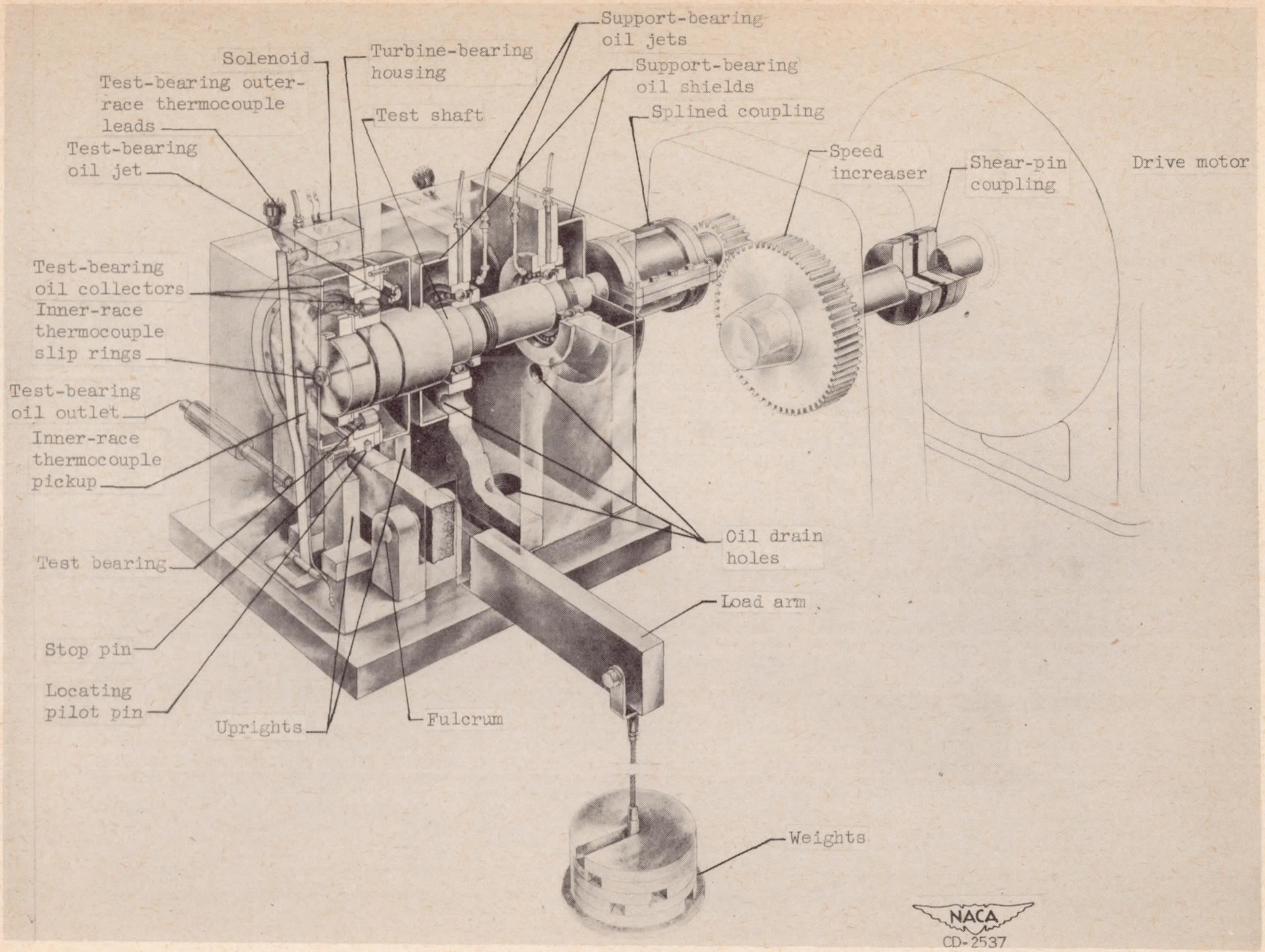
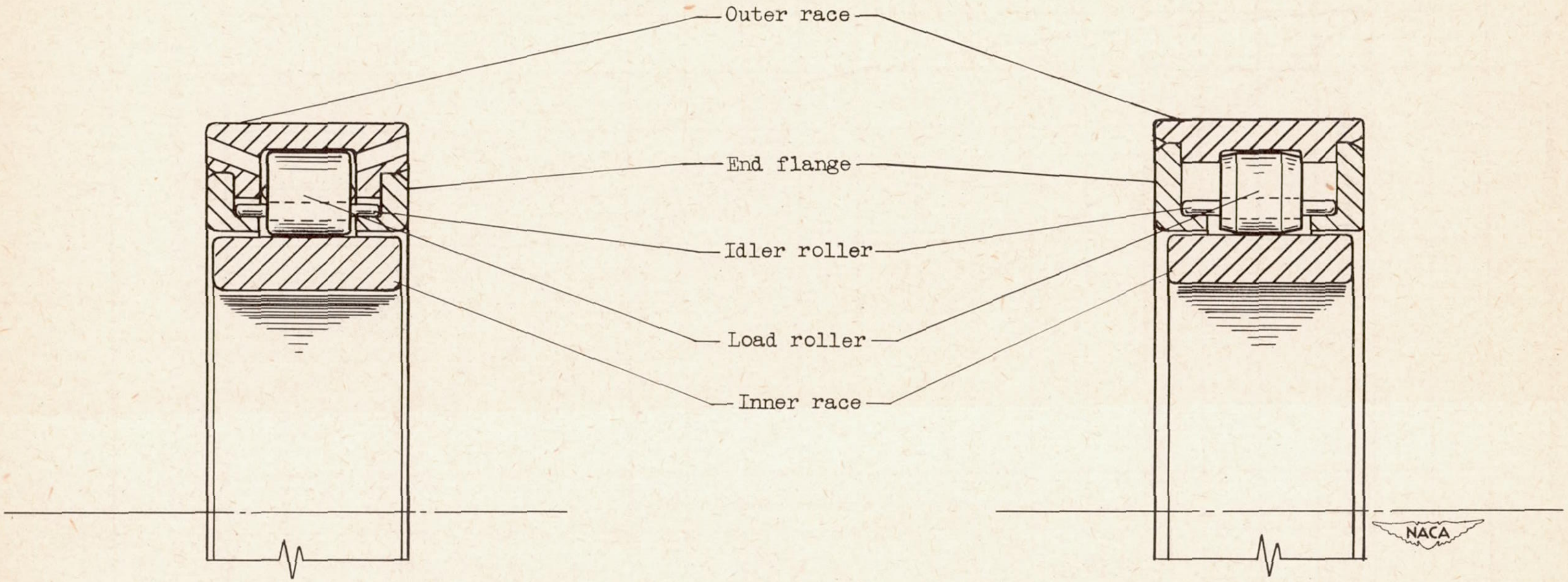


Figure 3. - Large test rig.



(a) Tungsten carbide design
(test bearing 4a).

(b) SAE 52100 design
(test bearing 5a).

Figure 4. - Large special cageless roller-bearing designs.

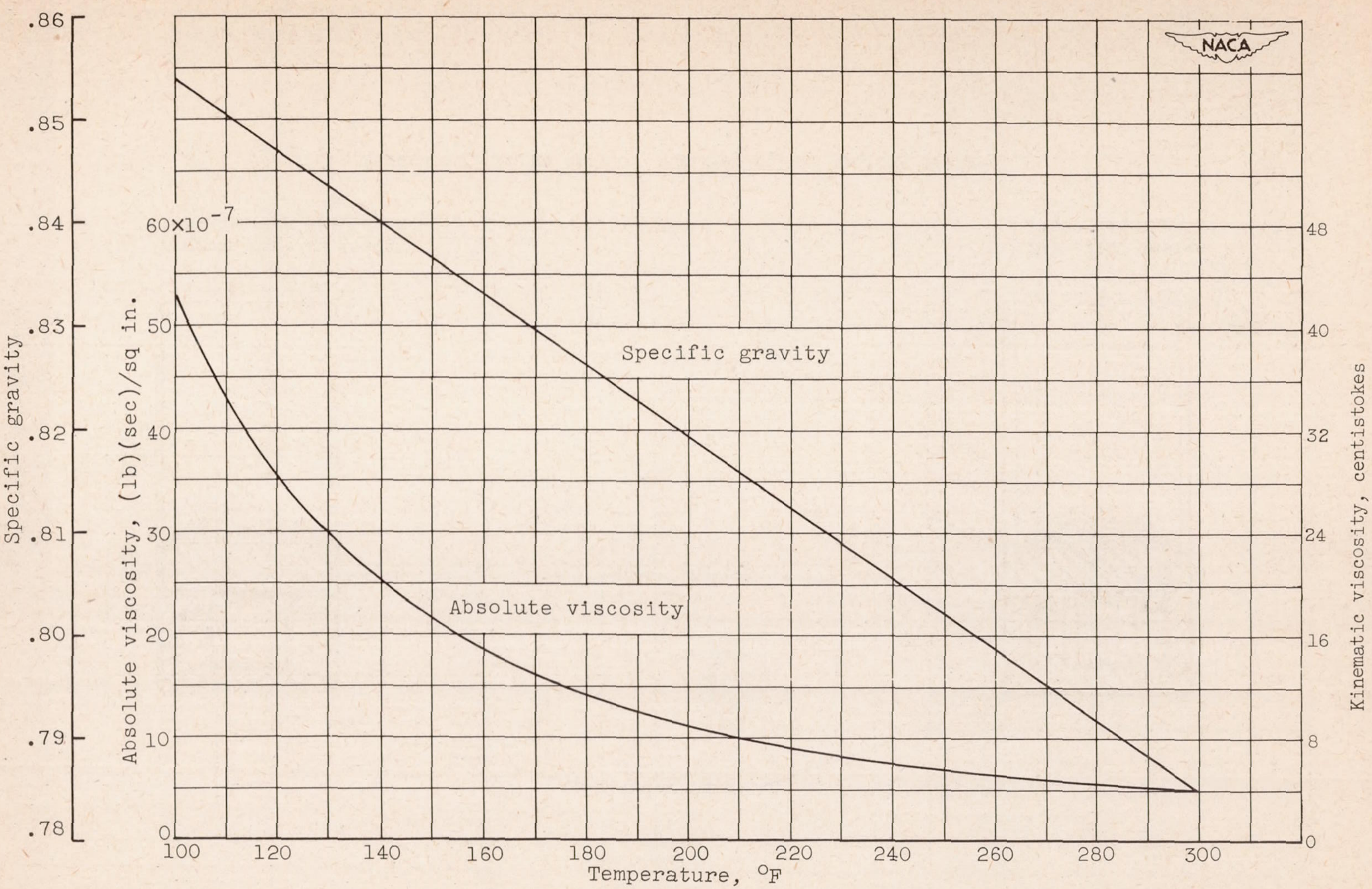
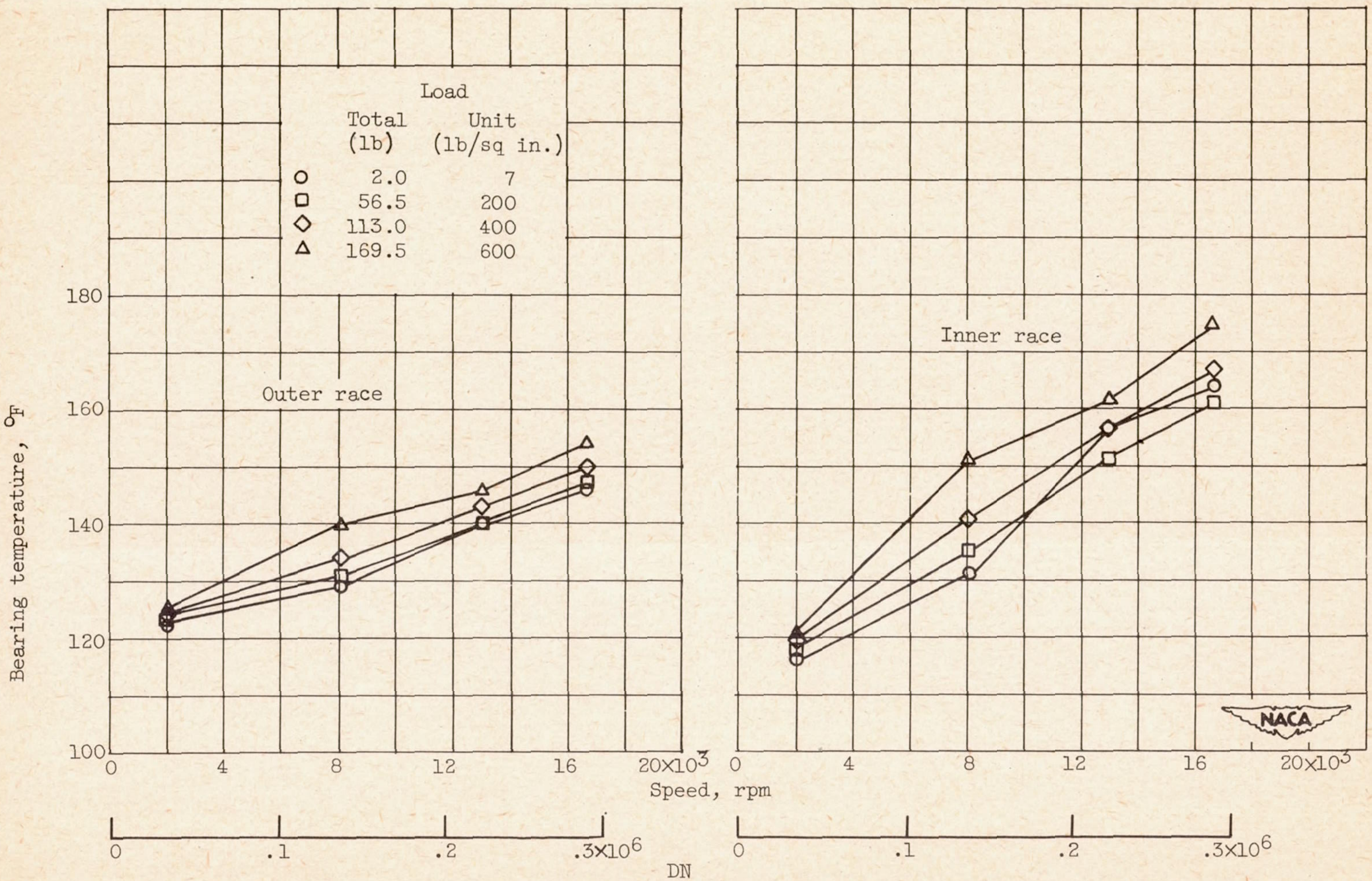
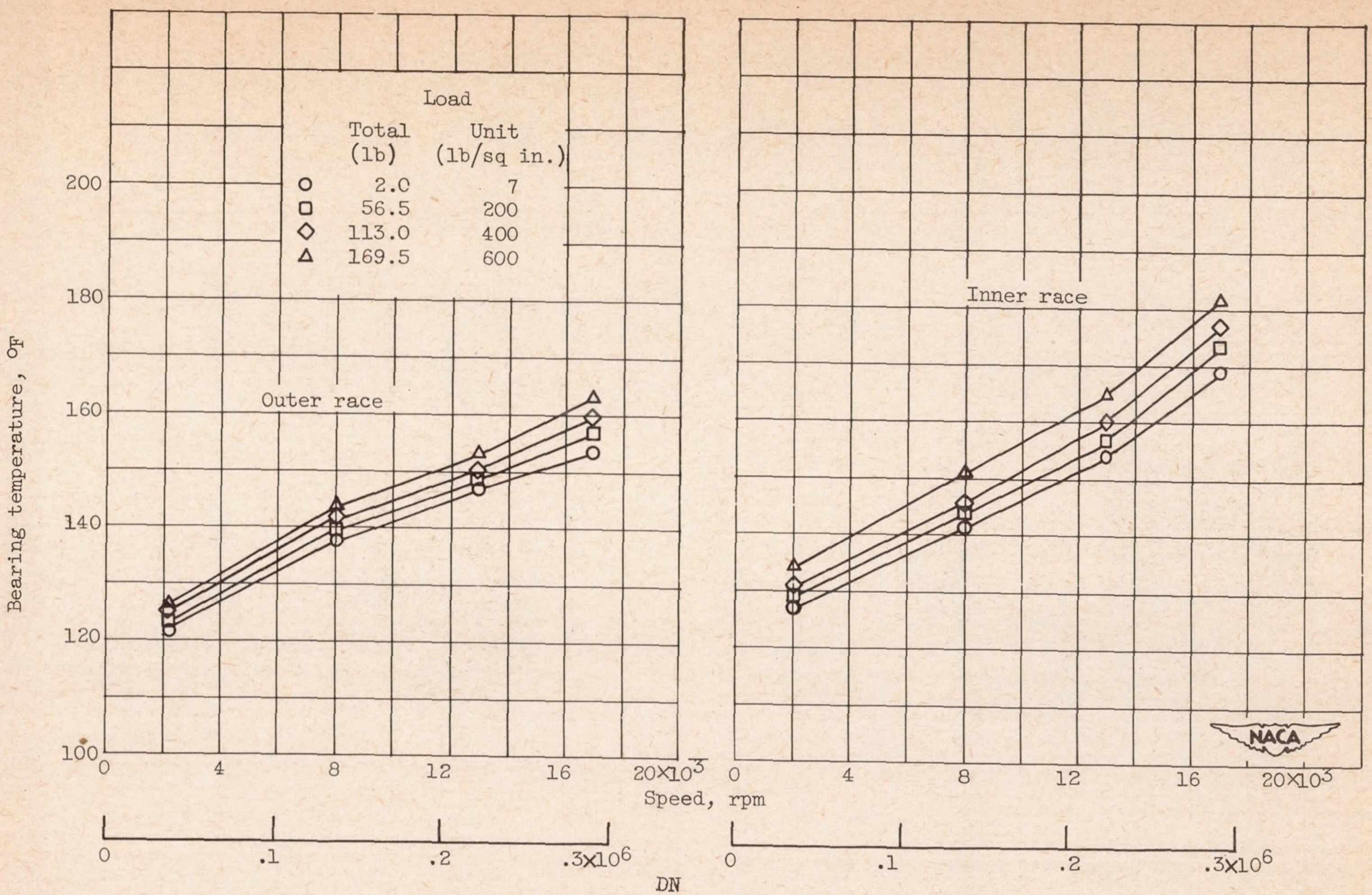


Figure 5. - Absolute viscosity and specific gravity of lubricant. Pour point, -50° F; flash point, 310° F; viscosity index, 150.



(a) Conventional bearing, 1a.

Figure 6. - Effect of speed and load on inner- and outer-race operating temperatures of roller bearings of 1.112-inch pitch diameter. Lubricant-flow rate, 3 pounds per hour; lubricant inlet temperature, 100° F.



(b) Special cageless bearing, 2a.

Figure 6. - Concluded. Effect of speed and load on inner- and outer-race operating temperatures of roller bearings of 1.112-inch pitch diameter. Lubricant-flow rate, 3 pounds per hour; lubricant inlet temperature, 100° F.

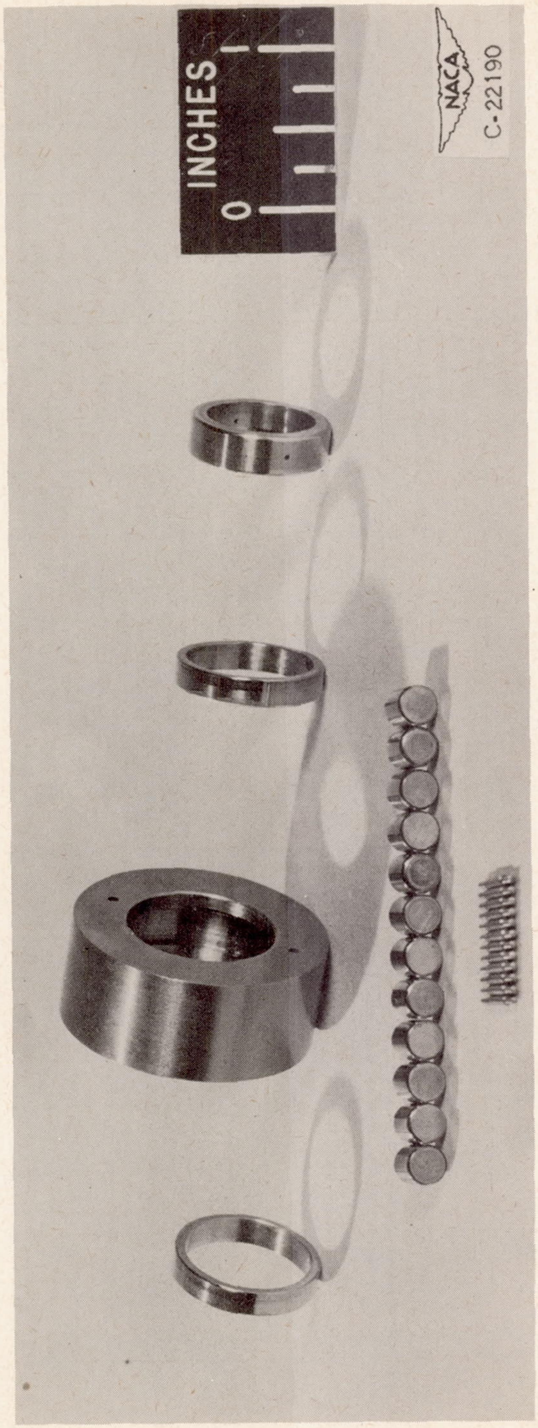


Figure 7. - Bearing 2a after test.

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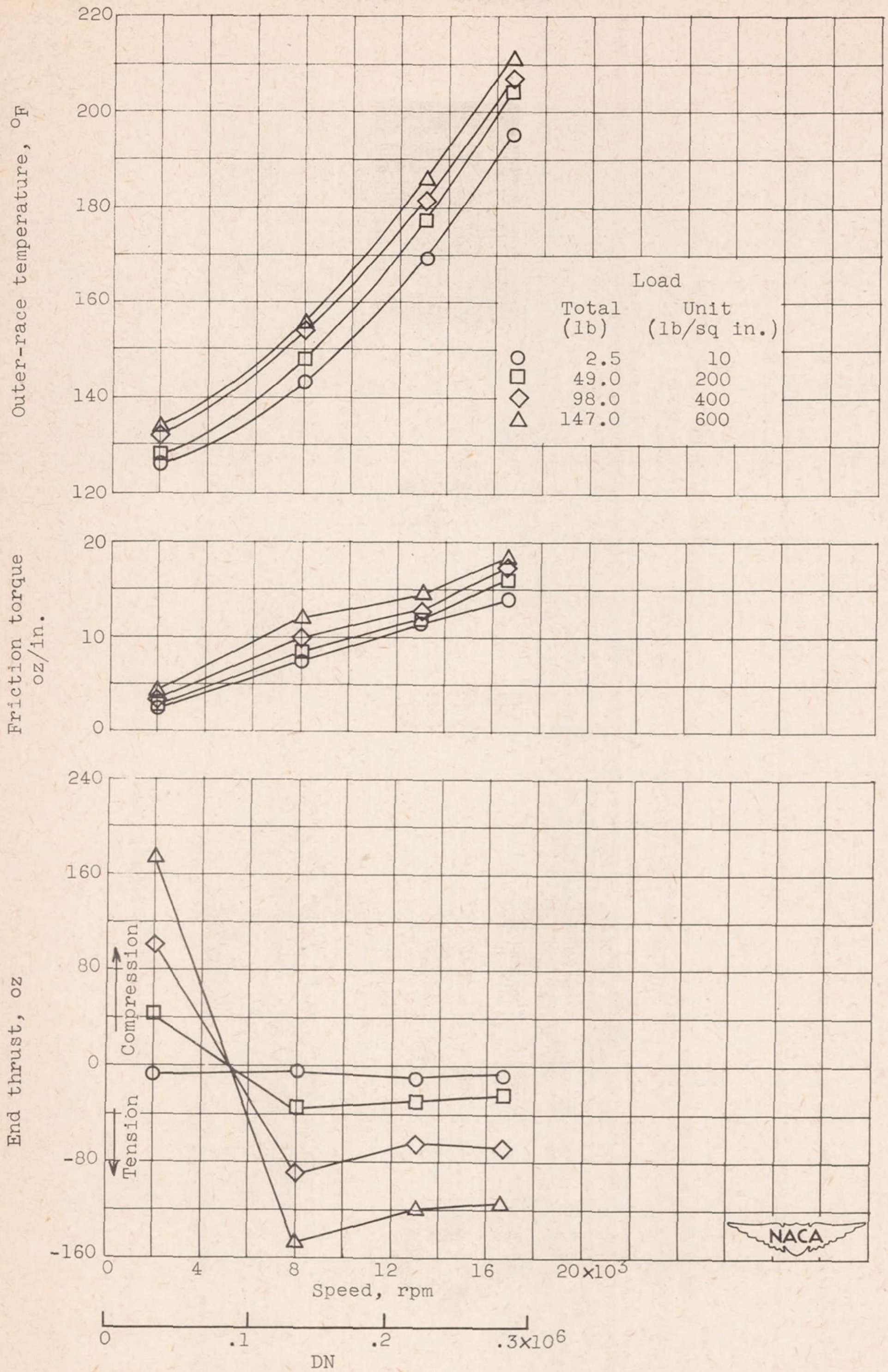
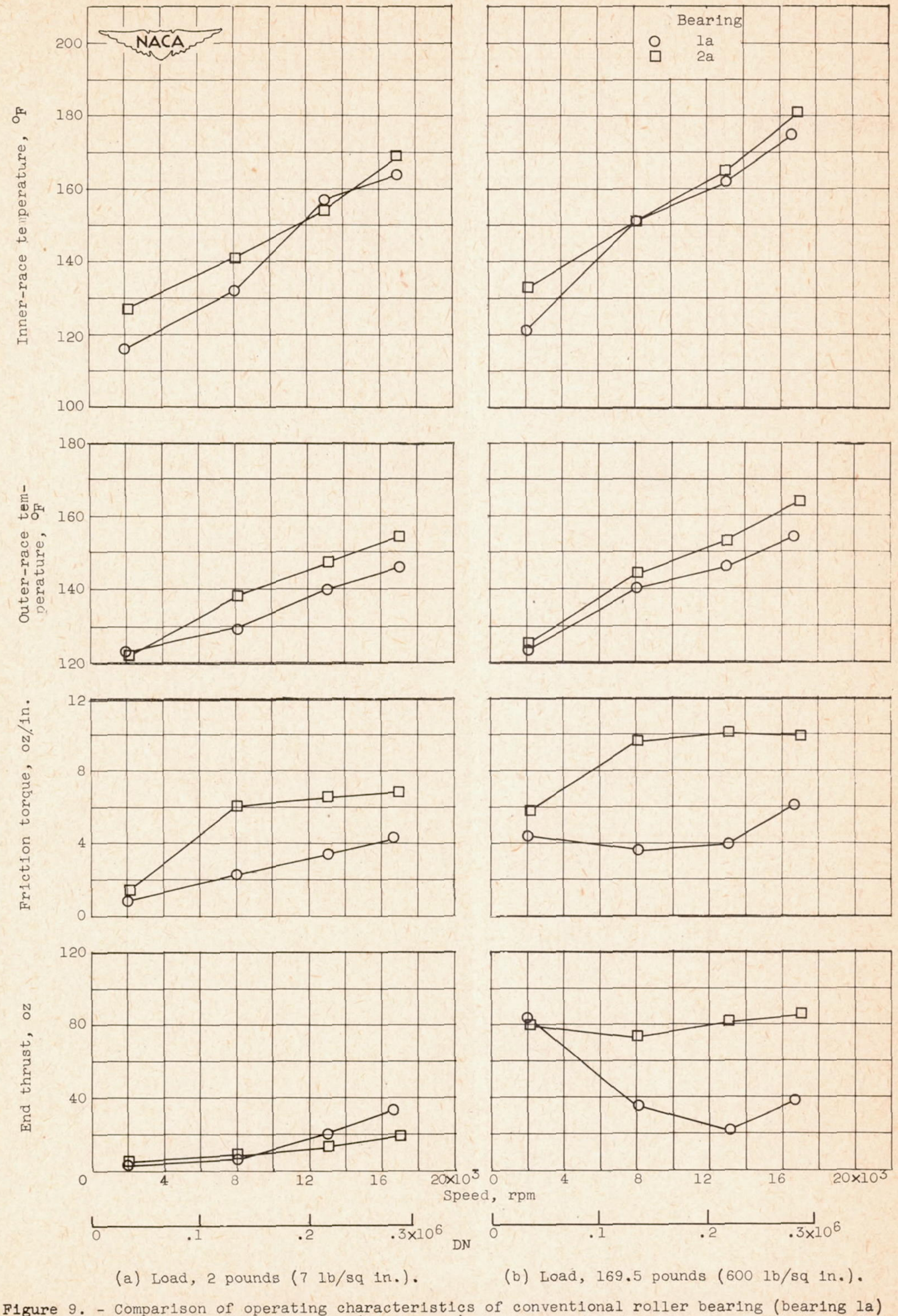


Figure 8. - Effect of speed and load on outer-race temperature, friction torque, and end thrust of a needle-type bearing (bearing 3a). Lubricant-flow rate, 0.5 pound per hour; lubricant inlet temperature, 100° F.



(a) Load, 2 pounds (7 lb/sq in.). (b) Load, 169.5 pounds (600 lb/sq in.).

Figure 9. - Comparison of operating characteristics of conventional roller bearing (bearing 1a) and special cageless roller bearing (bearing 2a) of 1.112-inch pitch diameter. Lubricant-flow rate, 3 pounds per hour; lubricant inlet temperature, 100° F.

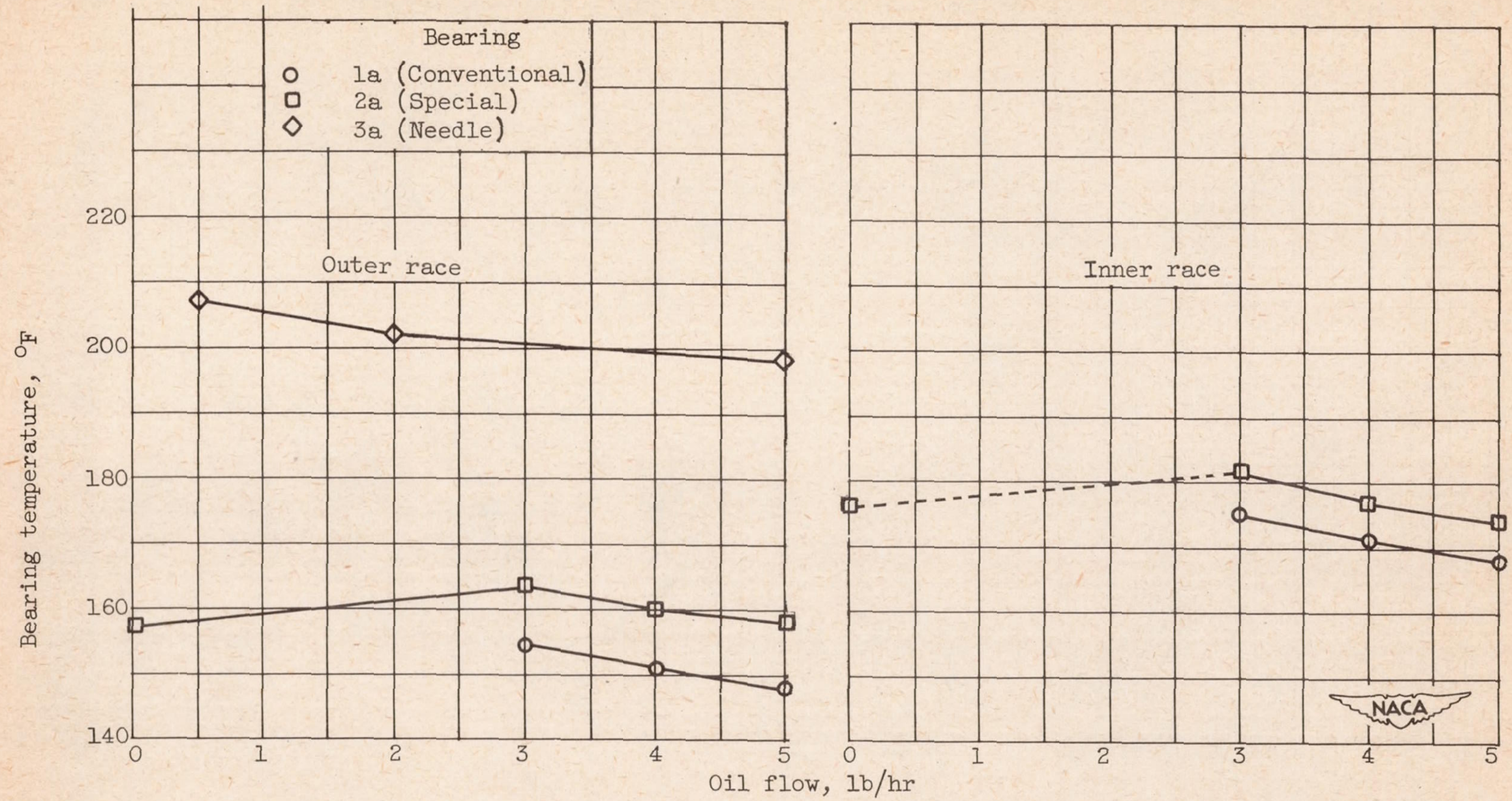


Figure 10. - Comparison of operating temperatures of bearings 1a, 2a, and 3a over a range of lubricant flows. Speed, 16,600 rpm; load, 600 pounds per square inch (bearings 1a and 2a), 400 pounds per square inch (bearing 3a); lubricant inlet temperature, 100° F.

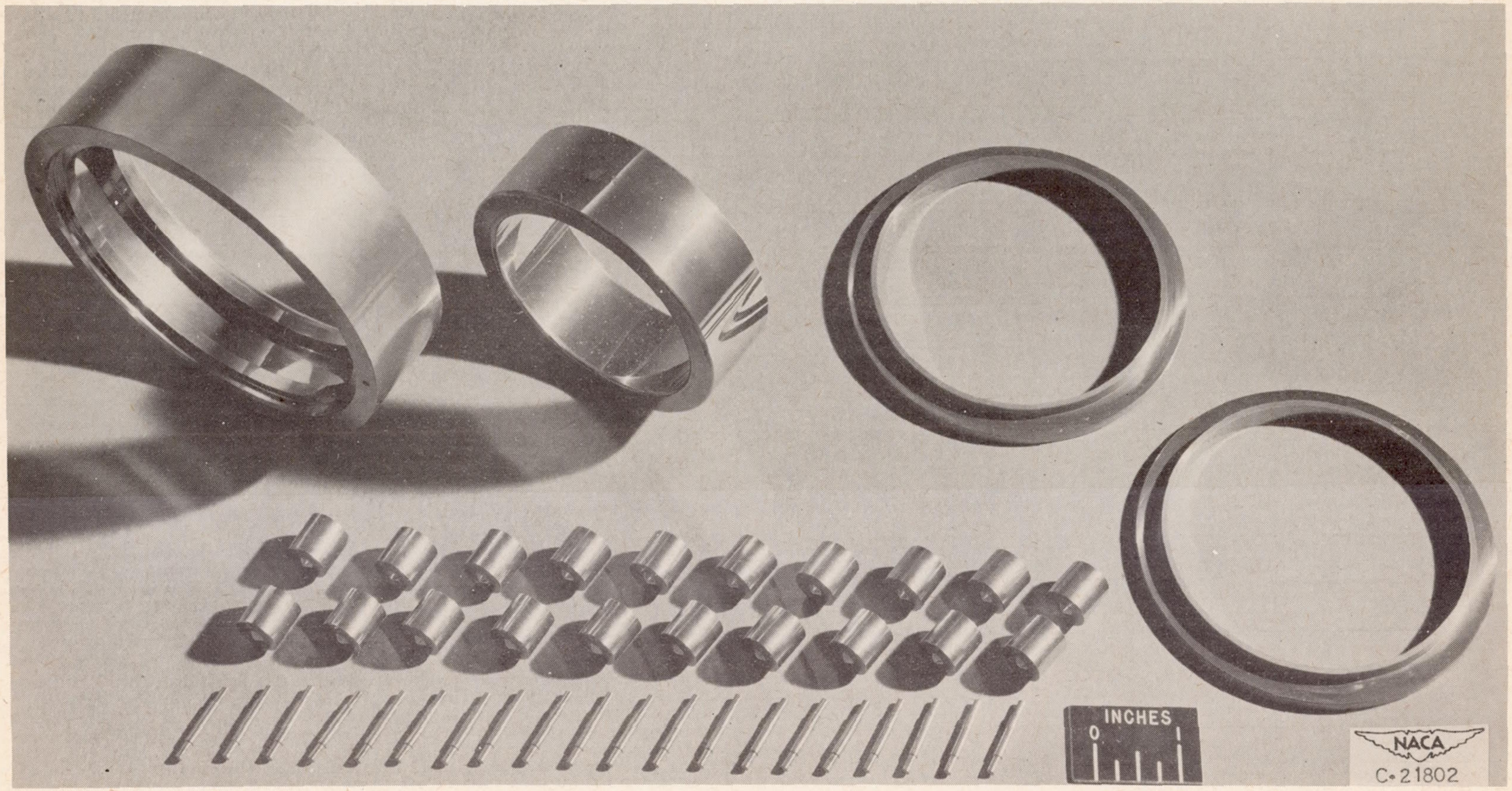


Figure 11. - Special tungsten carbide cageless roller bearing (bearing 4a).

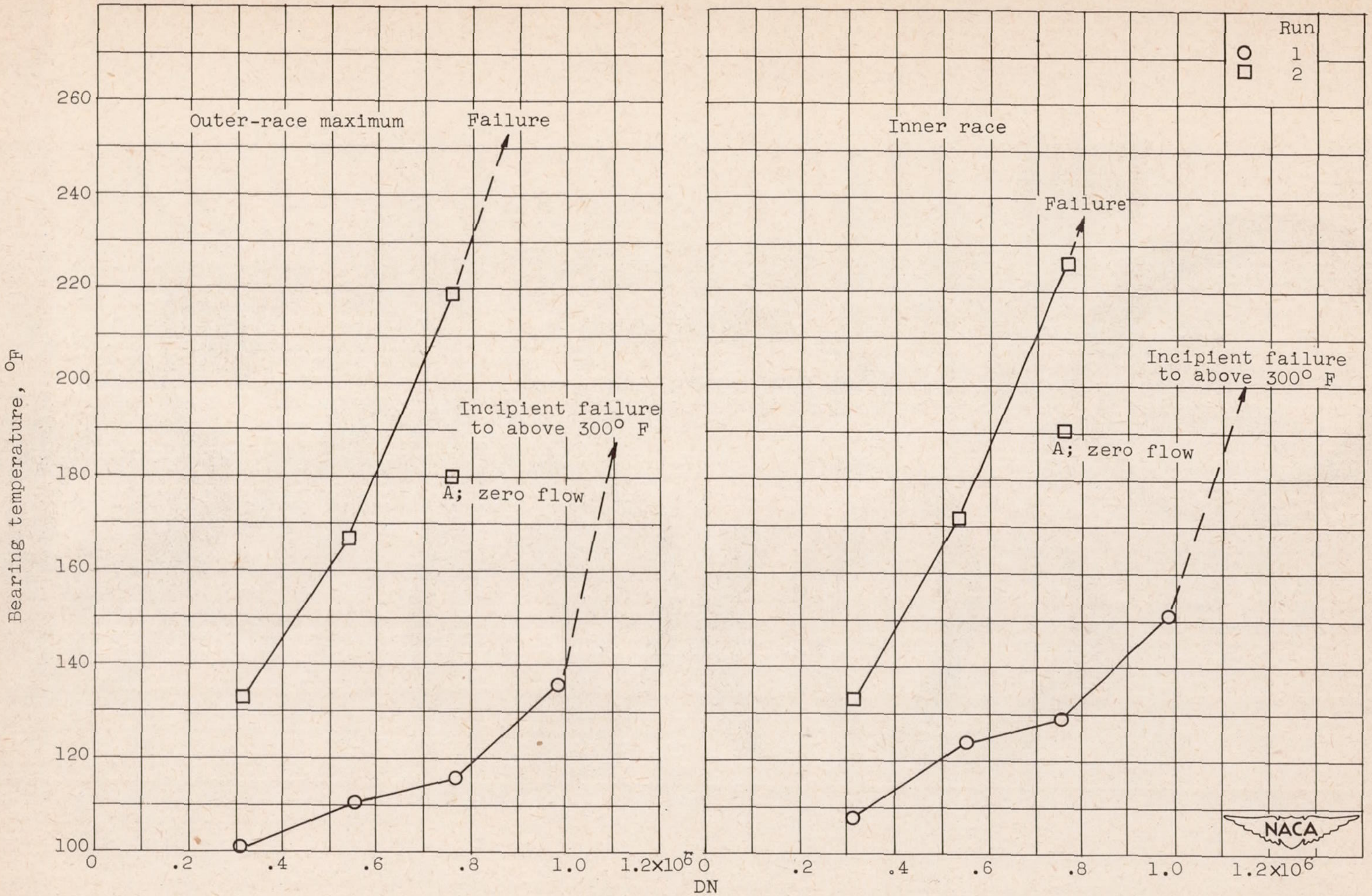


Figure 12. - Effect of speed on inner- and outer-race operating temperatures of tungsten carbide special cageless roller bearing (bearing 4a). Load, 10 pounds; lubricant-flow rate, 0.42 pound per minute (except point A); lubricant inlet temperature, 100° F.

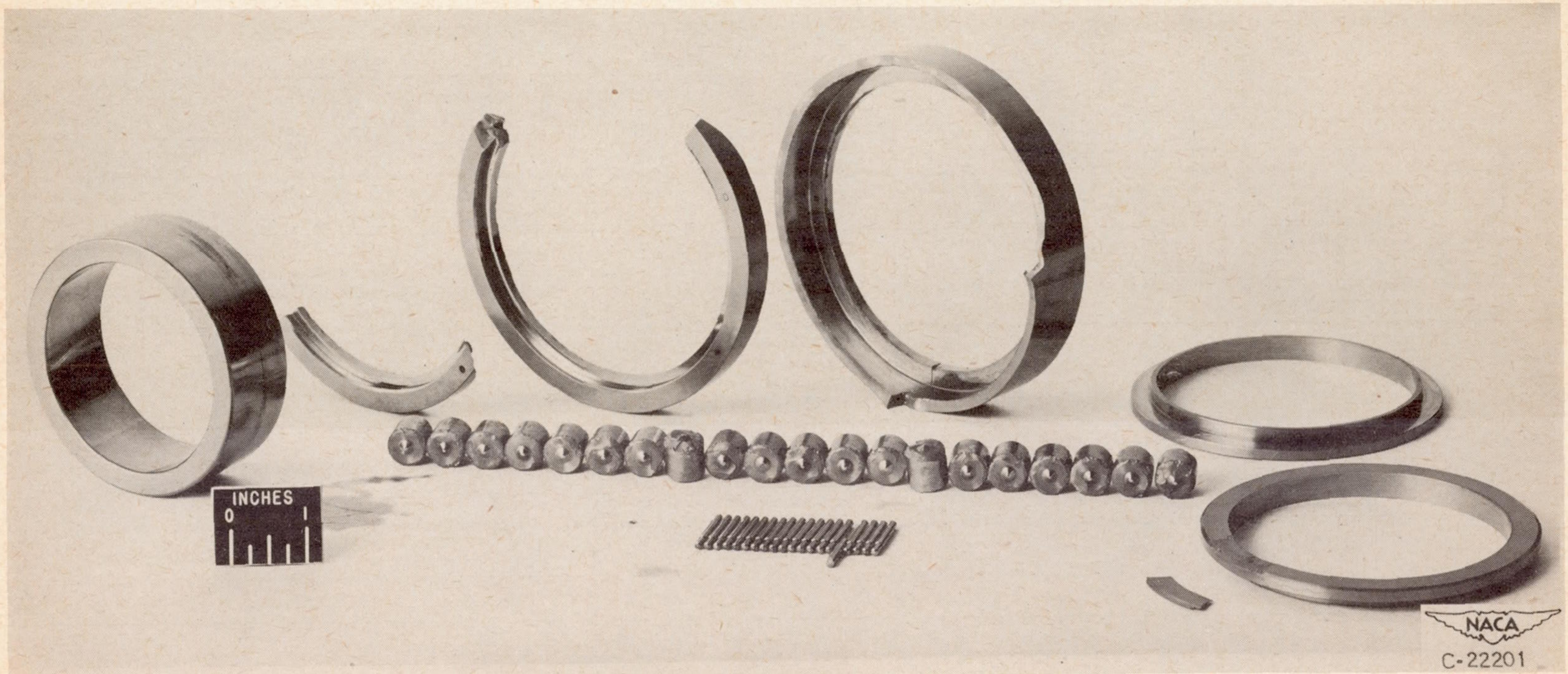


Figure 13. - Special tungsten carbide cageless roller bearing after test (bearing 4a).

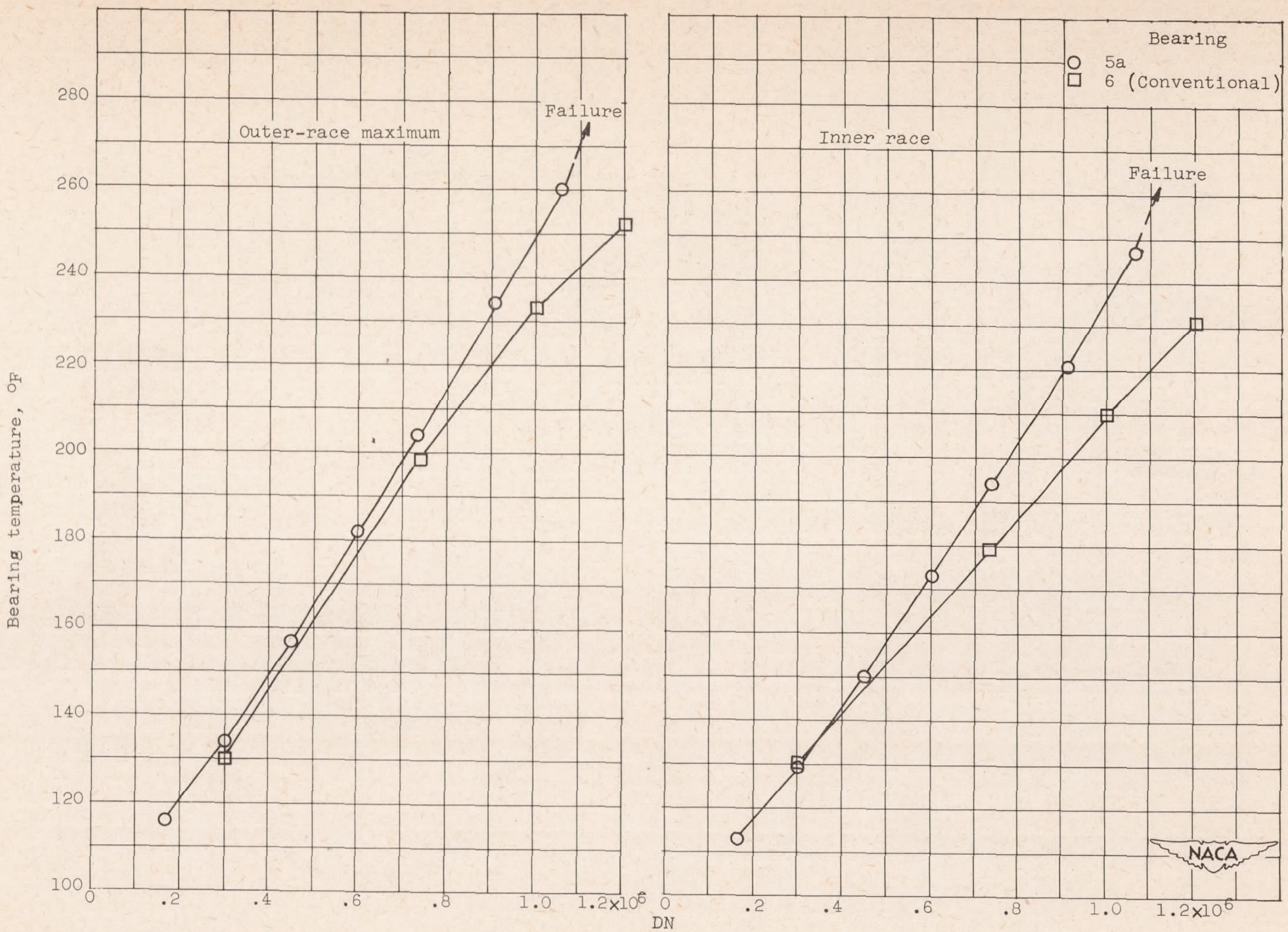


Figure 14. - Effect of speed on inner- and outer-race operating temperatures of special cageless roller bearing (bearing 5a) compared with effect of speed on operating temperatures of conventional inner-race-riding cage-type roller bearing (bearing 6, data from ref. 6). Load, 368 pounds; lubricant-flow rate, 2.75 pounds per minute; lubricant inlet temperature, 100° F.

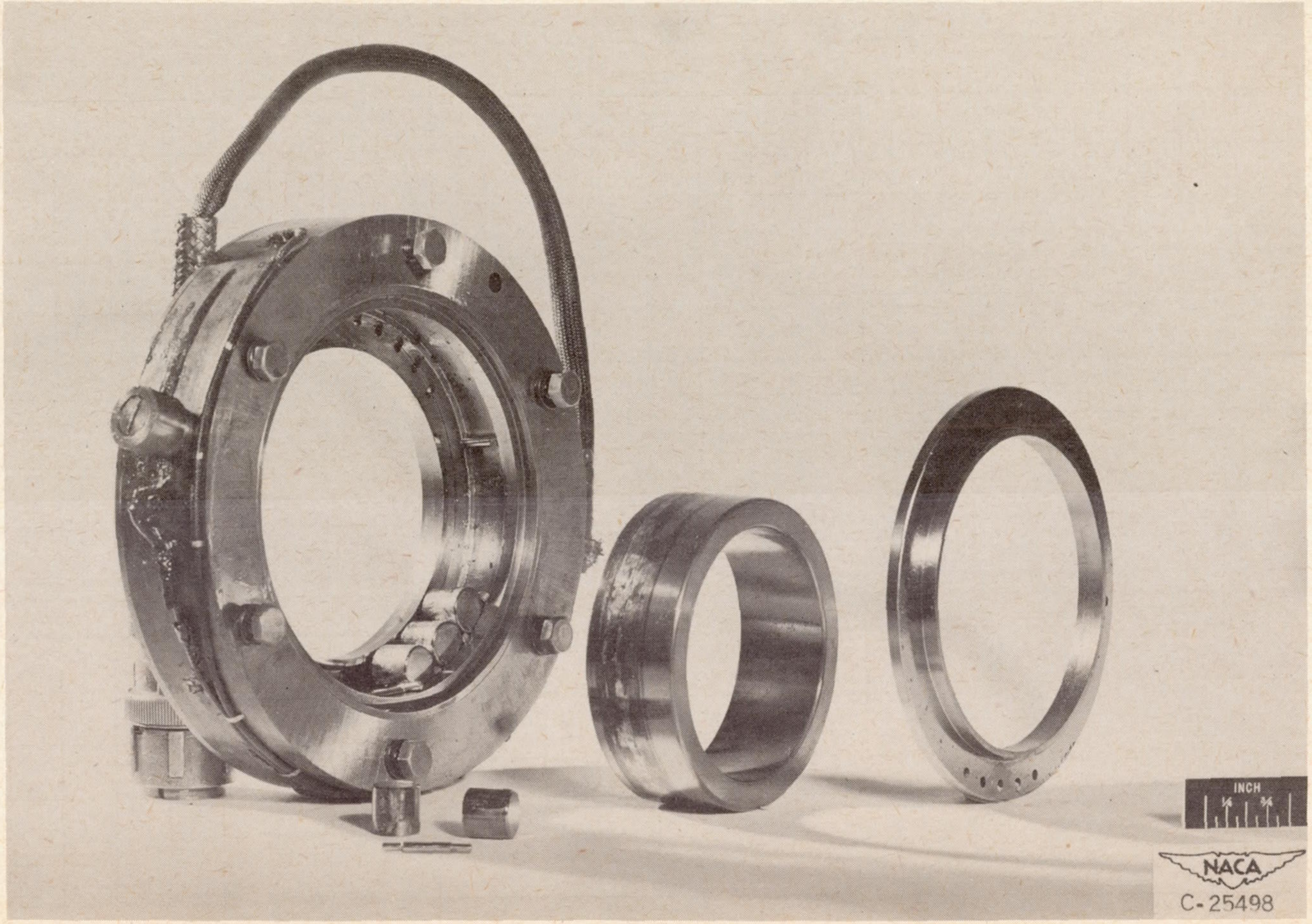


Figure 15. - Special cageless roller bearing after test (bearing 5a).

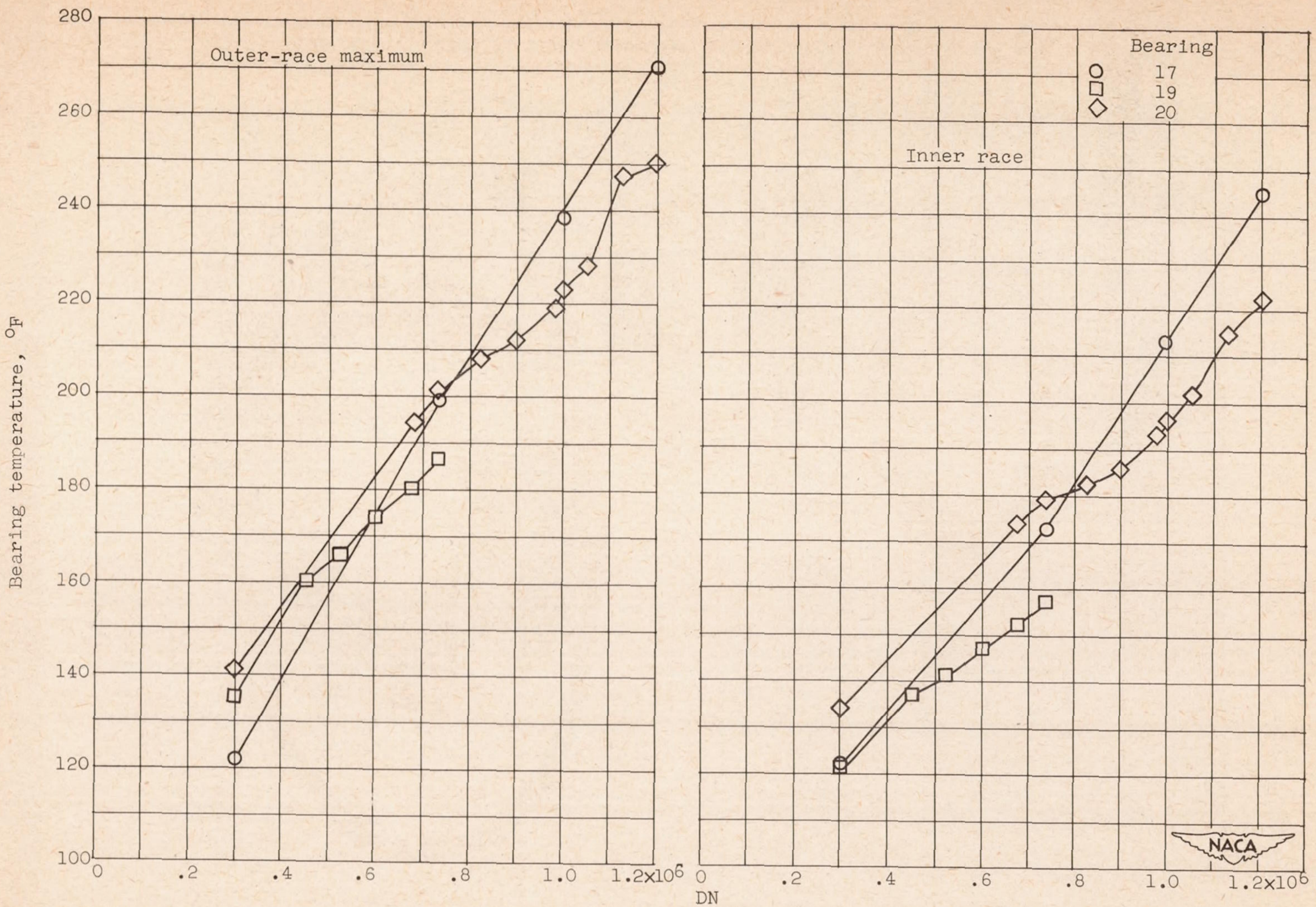
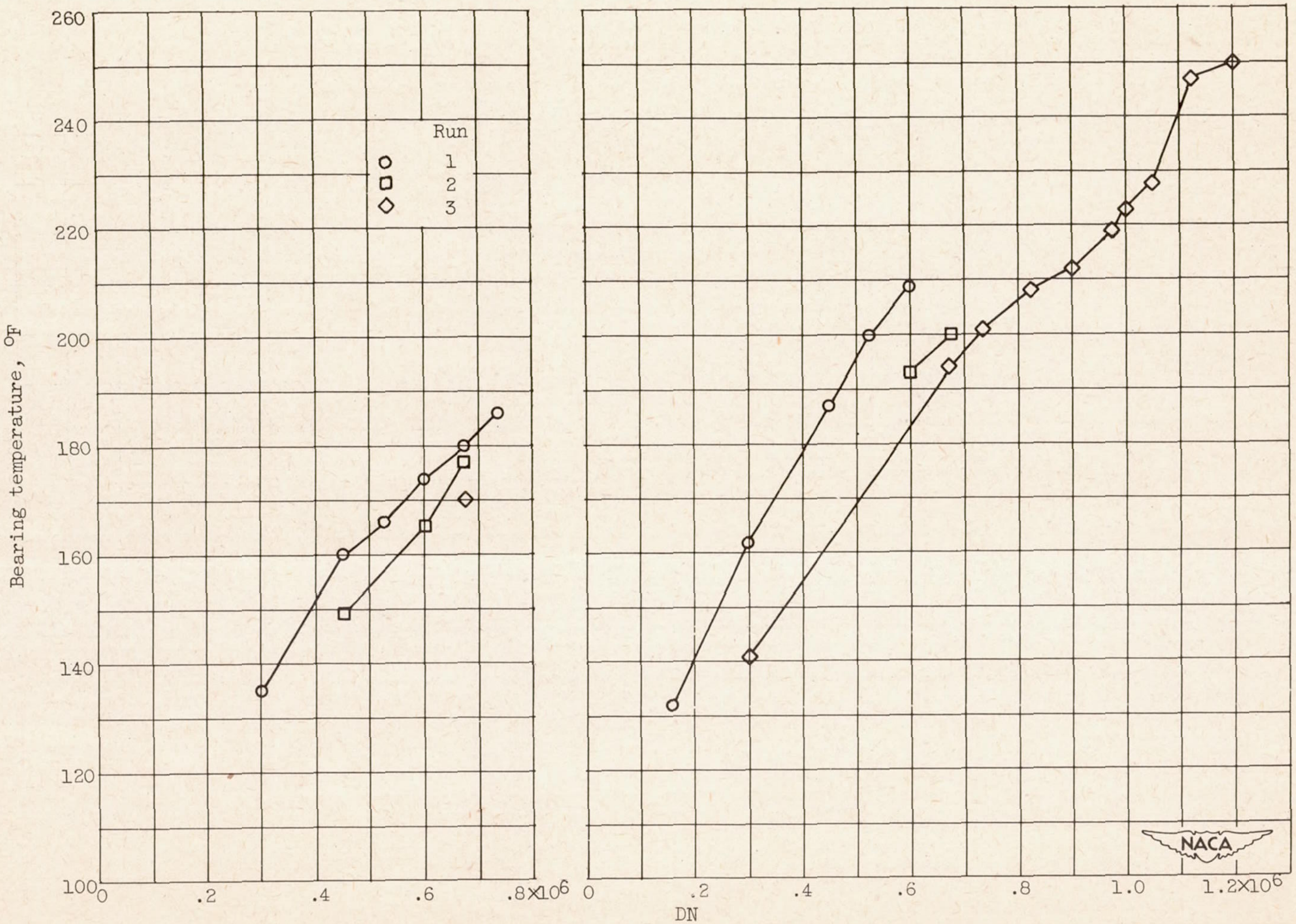


Figure 16. - Effect of speed on inner- and outer-race operating temperatures of full-complement cageless roller bearings 17, 19, and 20. Load, 368 pounds; lubricant-flow rate, 2.8 pounds per minute for bearing 17, 2.5 pounds per minute for bearings 19 and 20; lubricant inlet temperature, 100° F.



(a) Bearing 19.

(b) Bearing 20.

Figure 17. - Effect of speed on outer-race maximum bearing temperature of bearings 19 and 20 for first, second, and third runs. Load, 368 pounds; lubricant flow, 2.5 pounds per minute; lubricant inlet temperature, 100° F.

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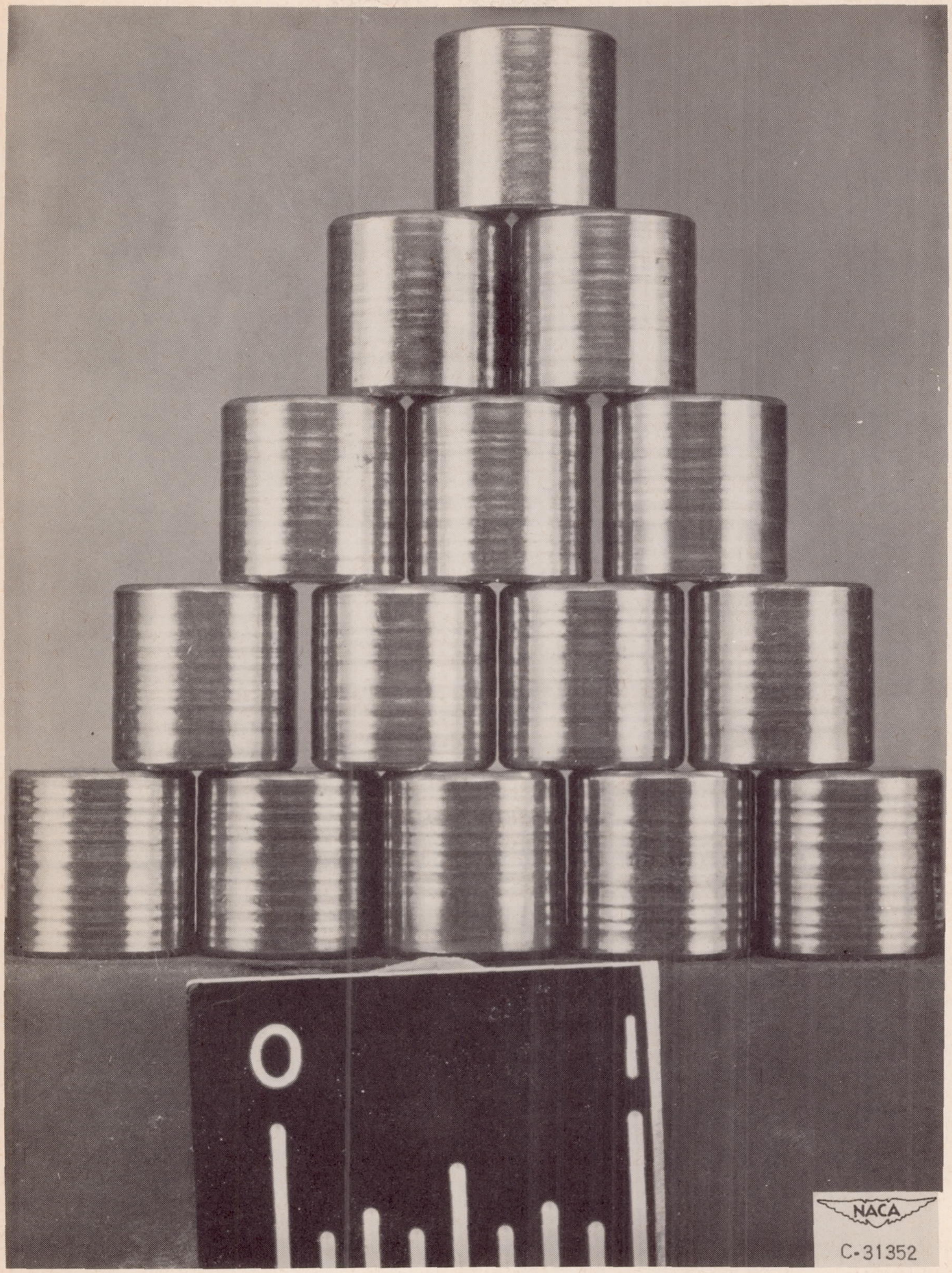
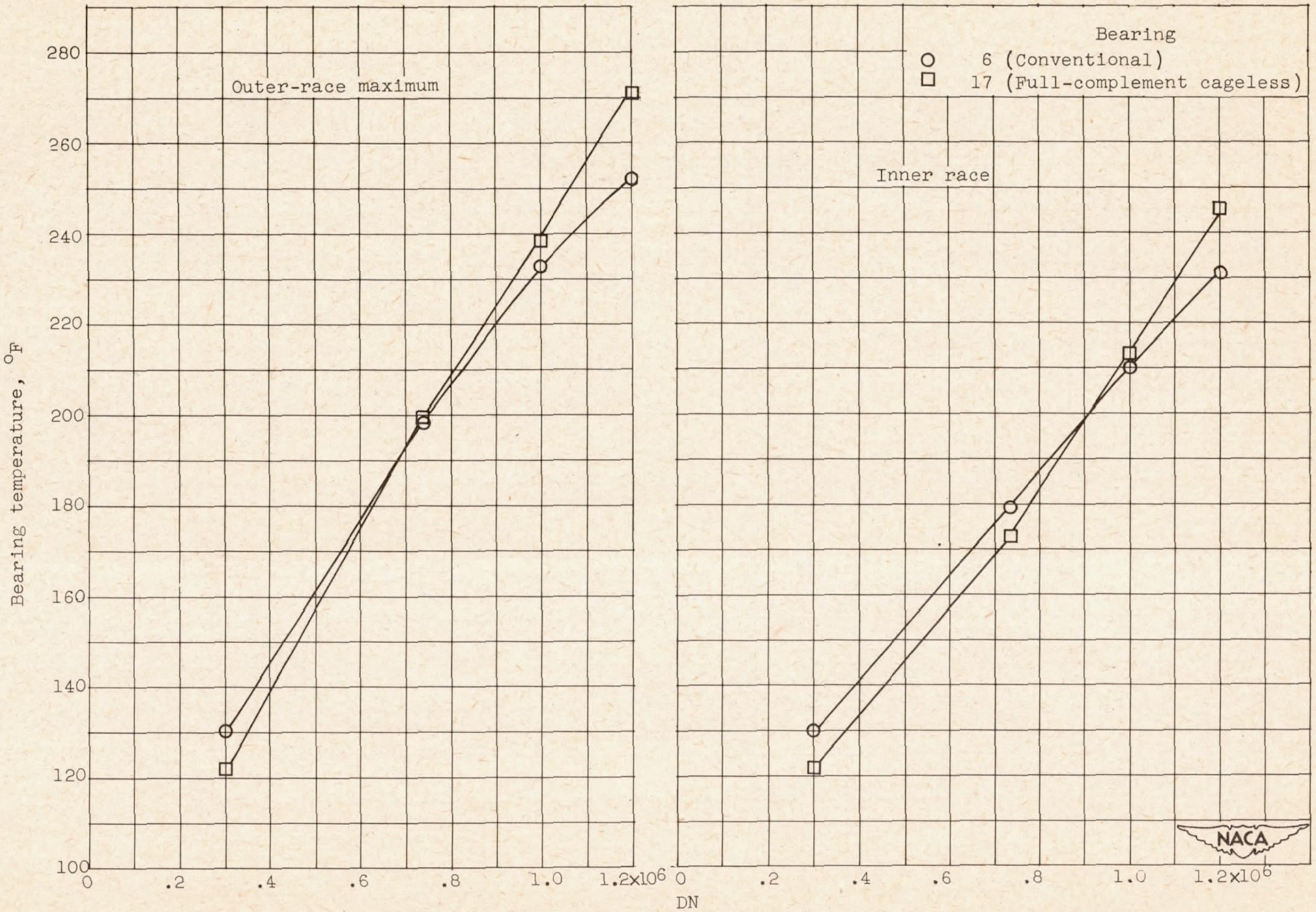
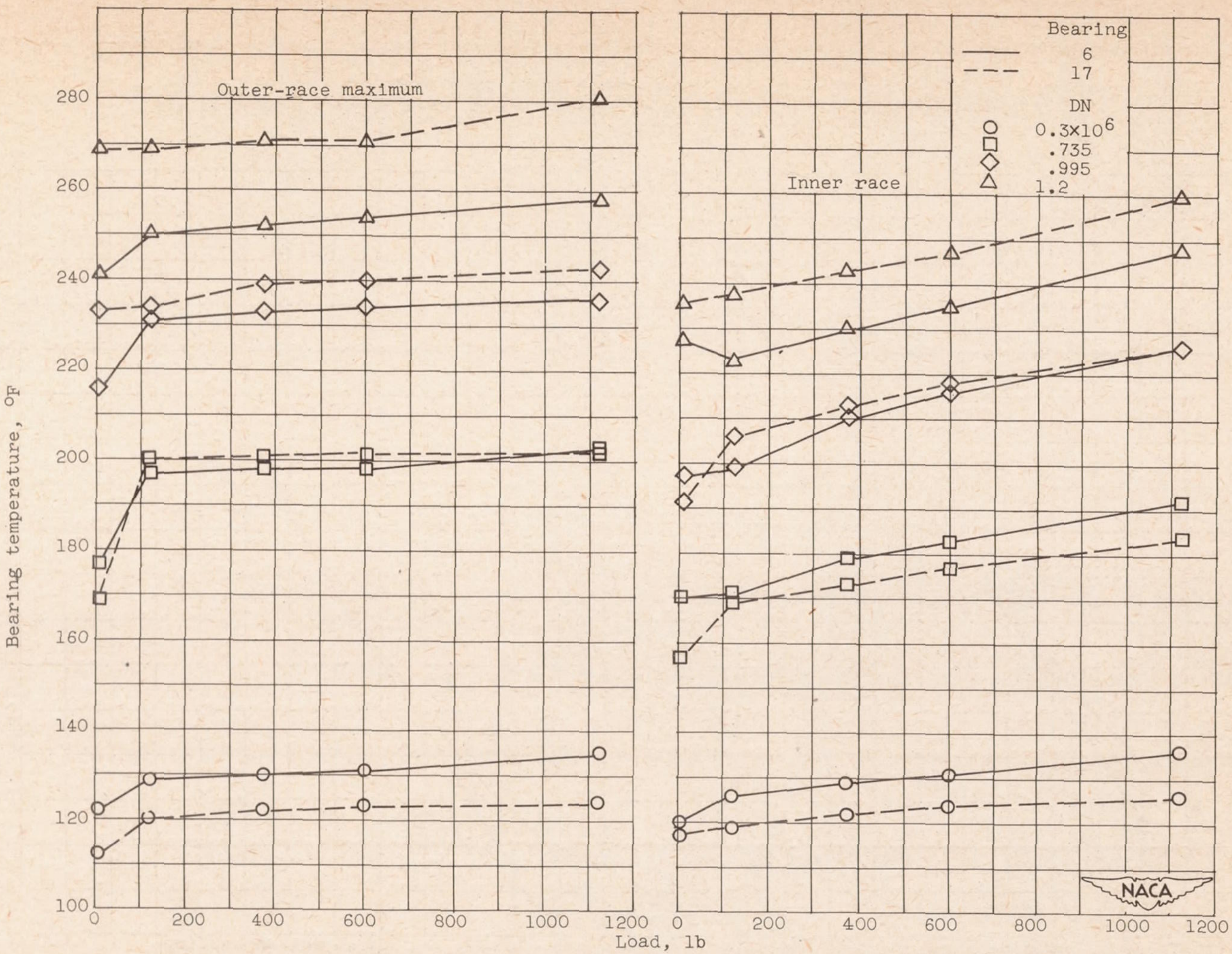


Figure 18. - "Wash board" roller wear of bearing 20 after test.



(a) Effect of speed. Load, 368 pounds; lubricant flow, 2.8 pounds per minute; lubricant inlet temperature, 100° F.

Figure 19. - Comparison of effect of speed, lubricant flow, and load on outer-race maximum and inner-race temperatures of bearings 6 and 17.



(b) Effect of load. DN value, 1.2×10^6; lubricant flow, 2.8 pounds per minute; lubricant inlet temperature, 100° F.

Figure 19. - Concluded. Comparison of effect of speed, lubricant flow, and load on outer-race maximum and inner-race temperatures of bearings 6 and 17.

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