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PERFORMANCE OF 110-MILLIMETER-BORE M-1 TOOL STEEL BALL BEARINGS AT HIGH SPEEDS, LOADS, AND TEMPERATURES

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SUMMARY

Eleven 110-millimeter-bore ball thrust bearings made of M-1 tool steel were operated over a range of DN values (product of bearing bore in mm times speed in rpm) from 0.4×10^6 to 1.54×10^6 at mean outer-race temperatures to 678° F. Thrust loads were varied from 1000 to 7000 pounds, radial loads from 1000 to 4900 pounds, oil flows from 6 to 18 pounds per minute, and oil inlet temperatures from 100° to 300° F. A synthetic lubricant of the diester type was used to lubricate the test bearings, which were equipped with either iron-silicon bronze, silver-plated iron-silicon bronze, or cast Inconel cages.

Eight of the eleven test bearings failed in fatigue. The life data for the failed bearings, although not statistically conclusive, indicate that the fatigue life of the M-1 tool steel and diester lubricant combination may be very much less than the life (based on catalog ratings) for the SAE 52100 steel and mineral oil combination. The fatigue failures were characterized by deep fissures running perpendicular to the surface rather than the normally shallow, horizontal pattern. Metallurgical examination of four of the failed bearings failed to uncover any inclusions or structural defects which might have caused the failures.

Under conditions of continuous oil flow, iron-silicon bronze was the best of the three cage materials tested. Both iron-silicon bronze and silver-plated iron-silicon bronze showed negligible wear under conditions of continuous oil flow, but the silver plate blistered on two of the four silver-plated cages tested, indicating that it may be difficult to consistently obtain a bond strong enough to withstand temperatures much above 450° F. Cast Inconel with a preformed surface film of nickel oxide is not satisfactory as a cage material for a bearing of this type operating at high DN values and temperatures below 600° F.

In several oil-interruption tests no significant differences in the time to failure were obtained in bearings equipped with iron-silicon bronze and silver-plated iron-silicon bronze cages.

INTRODUCTION

Higher flight speeds and higher engine power outputs have forced a steady rise in turbojet-engine bearing temperatures. The maximum bearing temperature in present engines is almost 500° F and will probably reach 700° F in engines of the near future (refs. 1 and 2).

A satisfactory bearing race material must have adequate hardness and dimensional stability over the complete range of operating temperatures. It should have a fatigue strength at elevated temperatures comparable to that of SAE 52100 steel at room temperature and good frictional properties with high-temperature cage materials. SAE 52100 steel is used in most present-day engines but is unsatisfactory at temperatures above 450° F because of the lack of hardness and dimensional stability. The existence of several high-speed tool steels with adequate hot hardness led to their trial as high-temperature bearing materials. However, little is known about the fatigue properties of these steels so that much experimental data must be accumulated before they can be properly evaluated. Fatigue tests of 6309-size ball bearings made of air-melt M-10 tool steel (4 percent chromium, 8 percent molybdenum, 2 percent vanadium, and 0.9 percent carbon) are reported in reference 3.

The data of reference 3 indicate that air-melt M-10 is inferior to SAE 52100 regarding fatigue strength and that the combination of high temperature and MIL-L-7808 synthetic oil (ref. 4) lubrication results in a further reduction in fatigue strength. Although the M-10 steel of reference 3 was metallurgically as clean as bearing-quality SAE 52100 steel, increased fatigue strength may possibly be obtained from vacuum-melting and closer control of the scrap used in M-10. Some fatigue data which indicate that 140-millimeter thrust bearings made of Latrobe MV-1 steel (4 percent chromium, 4 percent molybdenum, 1 percent vanadium, and 0.8 percent carbon) may have a greater capacity than similar bearings made of SAE 52100 steel are given in reference 5.

The exact effect of temperature on the fatigue life of a bearing material-lubricant combination cannot be evaluated with any certainty except by experiment. From theory, increasing temperature should have a detrimental effect on fatigue life because of the viscosity effect. Decreasing viscosity tends to lower fatigue life (ref. 5).

Much information remains to be obtained, not only on fatigue lives of high-temperature steels but on the performance of various cage materials at high temperatures as well. The cage problem can hardly be expected to become less severe with increasing temperature because of the deterioration in mechanical properties of cage materials and the fact that lubrication will undoubtedly become more marginal. This investigation was conducted to determine the operating characteristics of M-1 tool steel (4 percent chromium, 8.5 percent molybdenum, 1 percent vanadium, 1.5 percent tungsten, and 0.8 percent carbon) bearings with three different cage materials (iron-silicon bronze, silver-plated iron-silicon bronze, and cast Inconel). The ranges of operating variables were: DN (product of bearing bore in mm and shaft speed in rpm), 0.4×10^6 to 1.54×10^6; outer-race mean temperatures to 678° F; thrust load, 1000 to 7000 pounds; radial load, 1000 to 4900 pounds; oil flow, 6 to 18 pounds per minute; and oil inlet temperatures, 100° to 300° F.

PROCEDURE

Detailed descriptions of the bearing rig, drive equipment, lubrication systems, test-bearing heaters, temperature measurement, and test bearings are given in the appendix. The bearing test rig used in this investigation was designed to provide a test vehicle in which actual turbojet-engine bearing operating conditions could be duplicated. These operating conditions involve high speeds, high temperatures, and (in the case of the thrust bearing) high loads. A schematic drawing of the rig is shown in figure 1. Radial load was applied by means of a 5-inch-diameter piston and cylinder pressurized with oil and was transmitted from the piston through a yoke to the two radial-load roller bearings into the shaft. Thrust or axial load was applied by means of flexible stainless-steel bellows which, when pressurized with oil, exerted a force against a floating drum that transmitted the thrust load (essentially without friction) to the shaft through one of the angular contact ball bearings.

The lubricant was supplied to the test bearing through six nozzles, three on each side of the bearing. Two 15-kilowatt induction heaters, separately controlled, were used to heat the inner and outer races of the test bearing. Six thermocouples located in the test-bearing outer-race housing and one on the shaft were used to measure test-bearing outer- and inner-race temperatures.

All the test bearings were 222-size split inner-race ball bearings equipped with one-piece inner-race riding cages. A schematic drawing of a test bearing is shown in figure 2.

Tests Without Heat Addition

Before being subjected to operation with external heat addition, several of the test bearings were run at a range of DN values, radial and thrust loads, oil flows, and oil inlet temperatures to determine the effects of these variables on their equilibrium running temperatures. The ranges of controlled variables were generally as follows: DN values, 0.4×10^6 to 1.325×10^6; radial loads, 1000 to 4900 pounds; thrust loads, 1000 to 7000 pounds; oil flows, 6 to 18 pounds per minute, and oil inlet temperatures, 100° to 250° F.

Test With Heat Addition

At the conclusion of the tests without heat addition, several of the bearings were run at the range of oil flows and oil inlet temperatures given in the previous section with 3.4-kilowatt heat flow to the outer-race housing and somewhat less heat flow to the shaft. The exact heat input to the shaft could not be determined. The power output of the rectifier tubes was 3.5 to 4 kilowatts, and the power input to the shaft was probably about half this value.

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Following these tests, the test bearings were operated at mean outer-race temperatures up to 678° F and inner-race temperatures up to 569° F. The maximum outer-race temperature recorded was 838° F. For most of the test bearings, little testing was done at mean outer-race temperatures above 500° F because of recurrent failures of the outer-race heaters (see appendix). After the resistance heaters had been discarded in favor of an induction heater, no further trouble was encountered. The test bearing could then be operated at mean outer-race temperatures up to about 700° F which, at the DN values, loads, oil flows, and oil inlet temperatures used, required the full capacity of the outer-race heater. In order to avoid operating the test bearings without clearance, outer-race temperatures were maintained somewhat higher than inner-race temperatures as explained in the next section.

RESULTS AND DISCUSSION

Tests Without Heat Addition

The effects of DN, thrust load, oil flow, and oil inlet temperature on the equilibrium operating temperature of the test bearings are shown in figures 3 to 8. Figure 3 shows the effect of DN on test-bearing outer-race mean temperature. Over the range of DN values from 0.4×10^6 to 1.325×10^6, temperature increases at a rate slightly greater than linear. Since bearing temperature is related to the heat generated in a bearing, the bearing power loss could also be increasing with DN at a rate slightly greater than linear. This agrees with the results reported in reference 6 for 5-inch-bore angular contact bearings.

The effect of the thrust load on the test-bearing outer-race mean temperature is shown in figure 4. Here again, the rate of increase of temperature (and, thus, heat generation) with thrust load is greater than linear. The effect of thrust load on bearing temperature is quite strong as can be seen that for bearing 1008 the temperature rise above the oil inlet temperature increased from 101° F at a thrust load of 3000 pounds to 139° F at a thrust load of 7000 pounds.

The lower part of figure 5 shows the effect of oil flow on test-bearing outer-race mean temperature with no heat addition. A considerable increase in oil flow is needed to effect a significant reduction in bearing temperature. However, this decrease in bearing temperature is obtained at the expense of a greater heat load on the oil system. As shown in the lower part of figure 6, the drop in bearing temperature was accompanied by an increase in power rejected to the oil.

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The effect of oil inlet temperature on test-bearing outer-race mean temperature without heat addition is shown in figure 7. Increasing the oil inlet temperature results, of course, in an increase in bearing temperature; however, the heat load on the oil system decreases (lower part, fig. 8). Thus, at a penalty of an increase in bearing temperature, the oil-system heat load was decreased.

Tests With Heat Addition

The upper part of figure 5 shows the effect of oil flow on test-bearing outer-race mean temperature with the heat addition tabulated on the figure. The rate of decrease of bearing temperature with increasing oil flow is less for the case with heat addition than for the case without heat addition. In contrast to this, the upper part of figure 6 indicates that this increase in oil flow will result in an increase in the power rejected to the oil.

The upper parts of figures 7 and 8 show the effect of oil inlet temperature on bearing outer-race mean temperature and on the power rejected to the oil with the heat addition shown tabulated on the figures. These effects are similar to those obtained without heat addition.

Fatigue Failures

The results of the fatigue tests are summarized in table I. A total of eight bearings failed in fatigue, photographs of some of these failures are shown in figure 9. Six bearings had spalled inner races, one a spalled outer race, and one a spalled outer race and four spalled balls. The basic dynamic capacity of a similar bearing of SAE 52100 steel was calculated using the standard AFBMA method (ref. 7) and found to be 30,600 pounds. Using this capacity and the bearing manufacturer's catalog, a theoretical life at each load and speed condition was calculated. Use of this data together with the running times at each condition enabled calculation of a "life fraction." This life fraction (given in table I) is the fraction of the rating life of a similar SAE 52100 steel bearing that has been consumed in these tests. None of the failed components survived a life fraction of more than 0.395.

Although a sample size of eight bearings is not sufficient to yield statistically significant results, a Weibull plot of the eight failures is shown in figure 10 with life fraction instead of life as the abscissa. In this figure Johnson's median ranks (ref. 8) were used to plot the ordinates. Figure 10 shows that the expected life fraction with 90-percent survival for these bearings would be about 0.16. These bearings lubricated with MIL-L-7808 would then have a load capacity (for equal life) of 54 percent (ref. 7) of that of a similar SAE 52100 steel bearing lubricated with a mineral oil. These data cannot be considered conclusive because of the small number of bearings that failed, but they probably indicate a trend. The total running time and the temperature-time schedule are also shown in table I. The temperature-time schedule is a breakdown of the running time in five mean outer-race temperature ranges: (1) below 300° F, (2) 300° to 399° F, (3) 400° to 499° F, (4) 500° to 599° F, and (5) above 600° F. Although temperature does not enter into the life calculation, it probably does affect life through its effect on lubricative effectiveness and material properties.

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The fatigue failures obtained were somewhat different from those usually obtained. Ordinarily, fatigue failures develop from cracks that originate slightly below the surface and propagate to the surface producing a spall. The spalled area is usually very shallow compared with its dimensions parallel to the surface. In this investigation, however, it was noted that several of the failures had deep cracks or fissures running approximately normal to the surface. A typical example of this type of failure is shown in figure 11. The photograph shows a section parallel to the bearing face through an inner-race spall on bearing 1001. The causes of this peculiar failure characteristic are not yet known.

Metallurgical examinations of four of the failed bearings were made. No evidence could be found to explain the origin of any of the failures except in the case of a ball failure in bearing 1007, which originated at a small slag streak. However, the magnitude of the failures was such that minute inclusions could easily have been lost in the spalled material. The results of the metallurgical examination (table II) showed that the material cleanliness, hardness, carbide segregation, and structure were within the limits allowed for SAE 52100 bearings. Other factors may be responsible for the early fatigue failures obtained for this material-lubricant combination. Perhaps more stringent metallurgical tolerances must be imposed on tool steels. Either or both the tool steel and lubricant could be responsible for the deficient fatigue life. The data of reference 3 seem to indicate that it may be a combination of a poorer material and a poorer lubricant. In reference 3, M-10 tool steel showed a decrease in fatigue life from that of SAE 52100 steel when lubricated with a mineral-oil grease and showed a further decrease in fatigue life when lubricated with a synthetic lubricant of the diester type.

The pressure-viscosity characteristics of the lubricant may also play a significant role. In reference 9 poor ball-bearing fatigue life was obtained using several combustion-resistant hydraulic fluids as lubricants. The fact that these synthetic fluids have poor pressure-viscosity characteristics may have been partly responsible for the poor bearing fatigue lives obtained. Dorr (ref. 10) determined the effect of the lubricant in the contact zone for the two-dimensional case (line contact). The presence of a lubricant with good pressure-viscosity characteristics can appreciably lower the maximum compressive stress in the contact area. Since the life of a bearing varies inversely as the ninth power of the maximum stress, small decreases in stress mean large gains in life. Much research, not only on the fatigue life of tool steels but on the effect of lubricant type on fatigue life, remains to be done. The pertinent properties of these lubricants must also be obtained before a conclusive analysis can be made.

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A disturbing characteristic of the test results is the fact that several of the ball tracks and fatigue failures were low on the races, indicating a contact angle of about 15° rather than the design value of 20° to 26°. A subsequent bench test of the unmounted contact angle of an unused bearing showed that it was between 19° and 20°, so that the low-running contact angle resulted from test conditions and was not due to a manufacturing error. In a bearing of this type a low-running contact angle may result from a radial preload or from operation under an external load that is almost purely radial. The latter case could develop only as a result of a malfunction of the thrust-load device and was quickly eliminated as a possible cause when the thrust-load device was found to be functioning properly. It was determined experimentally that the radial preload had developed under certain test conditions because of unequal expansion of the inner and outer races. The outer race and its housing were restrained from expanding freely by the relatively cold framework in which they were held, while the inner race expanded freely to the point where the radial clearance was reduced to zero.

An experimental investigation was made to determine the operating temperatures under which radial preload existed. Dial indicators were set up to show the vertical position of the shaft on either side of the test bearing. With the shaft stationary, the radial-load device and both the inner- and outer-race heaters were turned on. With the outer-race mean temperature minus the inner-race temperature (hereinafter called ΔT) held constant, the shaft movement at the test bearing under a reversed radial load of 1000 pounds was determined as a function of outer-race temperature. Three curves of shaft radial movement against outer-race temperature were determined at ΔT values of -50°, 0°, and 50° F. The intersection of each of these curves with the theoretical shaft radial movement at zero bearing radial clearance gave three outer-race temperatures at which the bearing radial clearance became zero.

These three "zero clearance" temperatures were then plotted against ΔT to define the region of operating conditions that will produce radial preload. The resulting curve is shown in figure 12. To avoid radial preload, testing must be done at increasingly large ΔT values as the bearing outer-race mean temperature is increased. At a bearing outer-race mean temperature of 550° F, the outer-race mean temperature must be at least 50° F higher than the inner-race temperature to avoid radial preloads. Since the curve of figure 12 resulted from a static investigation, it does not include the effect of inner-race expansion due to rotation. This will serve to reduce operating clearances further and make the ΔT values required to avoid radial preload larger. 4247

By using the curve of figure 12 and the original data sheets for each bearing, it was determined that bearings 1001, 1002, 1003, 1004, 1006, and 1008 had been operated for a portion of their test time with slight radial preload. These times are as follows:

Bearing	Total running time, hr	Running time with preload, hr
1001	35.9	1.4
1002	50.0	.5
1003	50.1	7.6
1004	16.9	.9
1006	13.5	3.4
1008	55.6	9.7

In addition, the extent of the preload at each running condition was determined qualitatively by the position of the point when plotted on figure 12. None of the temperature conditions to which the previously mentioned bearings were subjected resulted in more than a slight preload. Since the bearings which were not preloaded did not exhibit longer fatigue lives than the bearings which were preloaded, it is concluded that the operation under preload did not significantly affect the fatigue results.

Performance of Cage Materials

A summary of the test-bearing cage-material performance data is given in table I. Two bearings were tested with iron-silicon bronze cages, four with silver-plated iron-silicon bronze cages (bearings 1002 and 1013 had the same cage), and four with cast Inconel cages. One of the cast Inconel cages was oxide coated by heating in air at 1200° F, two were oxide coated by immersion in molten sodium hydroxide, and one was left untreated. Iron-silicon bronze was the best cage material tested. In bearing 1001 practically no cage wear was evident. In bearing 1009 some 15 hours running time was accumulated at temperatures of 501° to 678° F without the cage incurring any appreciable wear. During several oil-interruption tests, which cumulatively consumed 17 minutes and 22 seconds running time (table III), severe wear of the cage and extremely severe wear of both the inner and outer races occurred. The depth of wear on the cage inside and outside diameters was less than 1/64 inch, but the cage wore the inner and outer races to uniform depths of 7/64 and 1/16 inch, respectively, over the surfaces contacting the cage. A photograph of the cage of bearing 1009 is shown in figure 13(a). The inner race of bearing 1009 is shown in figure 9 and the outer race in figure 14.

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The silver-plated cages showed very little wear under continuous oil flow conditions but on two of these cages the silver plate flaked and blistered. The same cage was used in bearings 1002 and 1013 and was run for over 100 hours, almost 50 of which were at temperatures over 540° F, without incurring any appreciable wear. The silver plate on the cage of bearing 1008 flaked badly on one face and one side of the inside diameter (fig. 13(b)). The cage of bearing 1013 was subjected to a much higher temperature without the silver plate failing, but the cage used in bearing 1005A, which was run at outer-race mean temperatures over 501° F for about 28 hours, showed some signs of blistering on its faces. Wear was very low on the inside diameter and no signs of failure of the silver plate were found. The results of the tests of the silver-plated cages indicate that it might be difficult to obtain a bond between the silver and base metal good enough to withstand temperatures above 450° F.

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Cast Inconel cages were used in bearings 1004, 1005, 1006, and 1007. In reference 11 a tenfold reduction in wear of Inconel specimens sliding against tool steel was accomplished by preforming a nickel oxide coating on the Inconel specimen. Accordingly, three of the four Inconel cages tested herein were oxide coated, while the fourth was left uncoated for comparison purposes.

It is difficult to determine how well the cage of bearing 1007 performed because of the damage which resulted from the outer-race and ball fatigue failures. The cage fractured in several places because of the high loads imposed by the balls after the fatigue failure. Some wear was evident on the cage inside diameter and on the inner-race cage locating diameter (fig. 13(c)). Severe wear was evident in the sides of some of the ball pockets, indicating that large axial forces between the balls and cage had developed as a result of the fatigue spalls. The worn areas on the cage had broken through the oxide film to expose relatively clean metal. The cages of bearings 1006 (oxide coated by chemical means) and 1004 (untreated) sustained severe wear in relatively short running times. The cage locating surfaces on the inner races of these bearings were also severly worn and galled. A photograph of the cage of bearing 1006 is shown in figure 13(d). Note the galled area resulting from a surface weld in the ball pocket. No significant surface damage on the balls was noted. Testing of bearing 1005 was stopped after $3\frac{1}{2}$ hours to observe cage wear. The cage, which had been oxide coated by heating in air at 1200° F, had begun to wear on the cage inside diameter (fig. 13(e)). Many small surface welds were distributed over the wear area. Figure 13(f) shows a macrophotograph of a typical galled area. Even in this short running time the oxide coating had been worn through, indicating that the life of the oxide coating under these operating conditions is very short. As soon as the surface oxide film is removed, high wear and widespread surface damage result. It is therefore imperative that the nickel oxide film be reformed as it is worn away if Inconel is to be a satisfactory cage material. Inconel might prove to be satisfactory at temperatures in the region of 1000° F, depending strongly on whether the rate of reformation of nickel oxide was sufficient to keep the surface film intact. Cast Inconel proved to be a satisfactory cage material for 20-millimeter-bore ball bearings operating at 1000° F (ref. 12). However, cast Inconel with a preformed surface film of nickel oxide cannot be considered a satisfactory cage material for bearings of this type operating at high DN values and at temperatures below 600° F. This agrees qualitatively with the results obtained in reference 11.

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Oil-Interruption Tests

In addition to bearing 1009, bearings 1005A and 1013 were subjected to oil-interruption tests in order to determine the time to failure after oil interruption. A summary of the oil-interruption test data is given in table III. Breaking of a shear pin, which occurred at about 250 percent of normal drive torque, constituted failure of the test bearing.

Bearing 1005A survived for 5 minutes and 23 seconds under conditions almost identical to those under which bearing 1009 survived for 1 minute and 48 seconds. However, bearing 1013 survived only 36 seconds under similar operating conditions, so that the silver plate does not seem to be beneficial in prolonging the time to failure during an interruption in oil flow. However, it is more effective than iron-silicon bronze in reducing the wear of the mating surfaces on the races. From the character of the cage wear in bearings 1005A and 1013 (photographs of these cages are shown in figs. 13(g) and (h), respectively) it appears that the silver at the cage locating surface acted as a lubricant. In these figures the light areas at which the arrows are pointing are thin smears of silver on the bronze base metal. Since the wear depth is many times the original thickness of the silver plate, silver was picked up by the inner race and smeared on the cage inside diameter as it wore. The appearance of these silver smears indicates that some silver at the cage locating surface may have become molten during the oil interruption. If this is so, interface temperatures of at least 1760° F were present during oil interruption.

Tests 1, 2, and 3 of bearing 1009 indicate that initial internal bearing clearance (changed by varying ΔT) may be important (table III). Tests 4, 5, and 7 of bearing 1009 indicate that survival time is decreased with increasing DN.

It was observed that in the first few seconds after an oil interruption, the drive motor power decreased slightly while bearing temperatures remained essentially constant. The momentary decrease in drive motor power is caused by the fact that a copious supply of oil is no longer being churned and sheared within the bearing. Throughout most of the period between the oil interruption and the failure, the drive motor power remains at or slightly below the normal running value. Just before failure occurs there is a sharp rise in both the drive motor power and in bearing temperatures. These tests were designed to simulate the conditions in an engine so a constant amount of heat was fed into the outer-race housing and the shaft during the oil interruption. After the oil interruption, the heat that is ordinarily removed by the oil from the bearing and its immediate environment remains to produce a rise in bearing temperaure. Since the outer race cannot expand as freely as the balls or the inner race, complete loss of clearance and seizure result.

The damage incurred in each test did not seem to influence the time to failure in succeeding tests. The shear pin stopped the test in time to prevent surface damage on the balls and ball tracks so that the only damage incurred was in the form of cage and inner-race wear.

SUMMARY OF RESULTS

An experimental investigation was conducted to determine the performance characteristics of 110-millimeter-bore ball thrust bearings operated over a range of DN values from 0.4×10^6 to 1.54×10^6, at bearing temperatures to 678° F, thrust loads of 1000 to 7000 pounds, and radial loads of 1000 to 4900 pounds. Bearing races and balls were made of M-1 tool steel while cages were made of iron-silicon bronze, silver-plated iron-silicon bronze, and cast Inconel. A synthetic lubricant of the diester type at inlet temperatures of 100° to 300° F and at flow rates of 6 to 18 pounds per minute was used to lubricate the test bearings. The following results were obtained:

1. Eight of the eleven test bearings failed in fatigue. Although the life data for the failed bearings are not extensive enough to be statistically conclusive, they indicate that the fatigue life of the M-1 tool steel-diester lubricant combination may be very much less than the life (based on catalog ratings) of the SAE 52100 steel-mineral oil combination. A plot of these data gave a value of 0.16 for the ratio of life (with 90-percent probability of survival) of the M-1 tool steel-diester lubricant combination to that for a SAE 52100 steel-mineral oil combination. The fatigue failures were unusual in that they were characterized

by deep fissures running normal to the surface. Normally spalled areas are very shallow in appearance. Inclusion ratings were within the limits specified for SAE 52100 steel in the four bearings examined.

2. Under conditions of continuous oil flow, iron-silicon bronze was the best of the three cage materials tested. Both iron-silicon bronze and silver-plated iron-silicon bronze showed negligible wear under conditions of continuous oil flow, but the silver plate blistered on two of the four silver-plated cages tested. Although one of the silver-plated cages showed no signs of blistering after 100 hours of running (almost 50 of which were at temperatures over 540° F), it may be difficult to consistently obtain a bond strong enough to withstand temperatures above 450° F. Cast Inconel with a surface film of nickel oxide preformed as it was in these experiments is not satisfactory as a cage material for a bearing of this type operating at high DN values and temperatures below 600° F. Experimental evidence indicates that the life of the nickel oxide surface film was very short under these operating conditions, and extensive wear and surface damage of both the cage and its locating race occurred after the film was worn away.

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3. In several oil-interruption tests no significant differences in the time to failure were obtained with bearings equipped with iron-silicon bronze and silver-plated iron-silicon bronze cages. The iron-silicon bronze cage caused severe wear of the tool steel races, whereas the silver plate effectively reduced wear by acting as a lubricant at the cage locating surface.

4. In a series of tests conducted without external heat addition it was found that:

(a) Bearing temperature increased at a rate slightly greater than linear with either increasing DN or increasing thrust load, other variables being held constant.

(b) An increase in oil flow effected a drop in bearing temperature and an increase in the oil system heat load.

(c) Increasing the oil inlet temperature decreased the oil system heat load at a penalty of a rise in bearing temperatures.

5. With 3.35 kilowatts of heat fed into the outer-race housing and from 3.5 to 4 kilowatts fed into the inner-race coil, oil flow and oil inlet temperature affected bearing temperature and oil system heat load in about the same way they did without external heat addition.

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APPENDIX - APPARATUS

Bearing Rig

A schematic drawing of the rig is shown in figure 1. As stated previously, radial load was applied by means of a 5-inch-diameter piston and cylinder that was pressurized with oil and was transmitted from the piston through a yoke to the two radial-load roller bearings into the shaft. Approximately half the piston force was taken by the test bearing and half by the roller bearing in the right housing. Either the cylinder chamber above or that below the piston could be pressurized to reverse the direction of the radial load. Thrust, or axial load, was applied by means of flexible stainless-steel bellows which, when pressurized with oil, exerted a force against a floating drum that transmitted the thrust load (essentially without friction) to the shaft through one of the angular contact ball bearings. The drum floated in an externally pressurized oil bearing. The direction of the thrust load was determined by whichever bellows was pressurized. Both the thrust and radial loads could be cycled automatically at any desired time interval from 15 seconds to 15 minutes.

Although all the test bearings whose performances are reported herein were 222-size (110-mm bore) ball bearings, shafts and housings were also available to test 219- (95-mm bore), 226- (130-mm bore) and 126- (130-mm bore) size bearings.

Drive Equipment

The high-speed drive equipment consists of an 80-horsepower direct-current motor coupled to a 7 to 1 ratio speed increaser. (A 50-hp motor would have been ample.) The high-speed shaft of the speed increaser was connected to the test shaft with a floating spline coupling. The low-speed coupling between the drive motor and the speed increaser was fitted with a shear pin to prevent damage to the test rig. The maximum possible speed of the test shaft was 30,000 rpm, and an electronic speed controller was used to control the speed to within ±2 percent.

Lubrication Systems

Test-bearing lubrication system. - The test-bearing lubrication system was of the circulating type. In addition to a positive displacement pump, the system contained a remotely controlled pressure regulator for controlling flow, a 29.5-kilowatt oil heater, an oil cooler, a mixing valve with an automatic controller for maintaining a set oil inlet temperature, suitable pressure gages, a full flow filter, and a scavenge pump for returning scavenge oil to the main sump.

Lubricant at 100° to 300° F was supplied to the test bearing through 0.065-inch-diameter nozzles. Three nozzles, spaced 120° apart, were on each side of the test bearing, with one group of three staggered 60° with respect to the second group.

A synthetic lubricant of the diester type (which met the MIL-L-7808 specification except for low-temperature viscosity) was used to lubricate the test bearings. It had a viscosity of 20.5 centistokes at 100° F and 5.2 centistokes at 210° F.

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Support bearing lubrication and hydraulic oil system. - This system was also of the circulating type. A positive-displacement pump supplied oil to the center and rear support bearings through preset hand valves in the lines leading to these bearings. A 1500-pound-per-square-inch piston pump was used to supply high-pressure oil for the radial-load piston, the thrust-load bellows, and the hydrostatic bearing. The pressure in each one of these elements was controlled by hydraulic pressure regulators. The scavenge oil from the support bearings and the bleed oil from the load devices and hydrostatic bearing were collected and pumped back to the main sump by a scavenge pump. An oil cooler, a sump heater, and suitable pressure gages, including precision gages for measuring the pressure in the radial-load piston and thrust-load bellows, were included in the system. Two four-way solenoid valves operated by automatic timing devices were installed in the lines leading to the two thrust-load bellows and to the upper and lower chambers of the radial-load cylinder to enable the thrust and radial loads to be cycled.

All the support bearings were lubricated with an SAE 30 mineral oil introduced at about 80° F. The two roller bearings in the center housing received a total of 20 pounds per minute of oil, while the three support bearings in the rear housing received a total of 15 pounds per minute. The hand valves in the oil lines to these two sets of bearings were calibrated before the test and preset to give the above oil flows.

Test-Bearing Heaters

Two separately controlled test-bearing heaters were used to enable the heat flows to the inner and outer races to be varied independently. The inner-race heater was a 15-kilowatt induction heater which supplied heat to the shaft through a coil located in the shaft cavity at the test bearing (fig. 1). This induction heater was an electronic type of 400,000 cycles per second. A stationary thermocouple located very close to the surface of the shaft near the test bearing was used to provide the signal for the inner-race induction heater controller. The controller setting was adjusted until the desired test-bearing inner-race temperature was obtained. The controller maintained this temperature by turning the heater on and off.

Initially, resistance heaters were used to heat the test-bearing outer race. Nichrome wires insulated by ceramic beads were wound in the outer-race housing grooves. This system was abandoned because adequate power could not be dissipated to the housing without incurring burnouts of the heating elements. Tubular commercial resistance heaters were tried next, but heater burnouts continued despite efforts to improve heat transfer using, first, aluminum shims wedged between the heater tubes and the groove walls, and later by casting the tubular heaters in the grooves with molten aluminum. Casting eliminated burnouts, but only two heaters of 6.7-kilowatt total capacity could be wound in the available groove space. Since this heat input was insufficient to provide an average test-bearing outer-race temperature greater than 550° F under the most severe operating conditions investigated, an induction heater was installed. A second 15-kilowatt induction heater, employing a motor generator set to generate 10,000-cycle-per-second frequency for heating, was installed. The heating unit consisted of seven turns of 1/4-inch-diameter copper tubing insulated with fiberglass sleeving and wound in the outer-race housing grooves. Cooling water was circulated through the coils. In the initial trials of this heater, the test-bearing housing buckled because of the high stresses caused by the restraint of the relatively cool surrounding framework. A new housing, which could expand in both the axial and circumferential directions, was designed. This housing-heater combination has been very successful, and no further trouble has been experienced.

Temperature Measurement

Six iron-constantan thermocouples were located at 60° intervals around the outer-race periphery of the test bearing at its axial centerline. The thermocouples were embedded in the outer-race housing flush with the housing bore and made contact with the outer-race outside diameter (fig. 2). Five iron-constantan thermocouples were located on the shaft, three on the outside diameter and two on the surface of the induction heater coil cavity. Those on the outside diameter were located in line 3/4 inch apart with the center one at the axial centerline of the test bearing. These were embedded in the shaft metal at the surface. The electromotive forces generated by the rotating thermocouples were taken from the shaft through copper slip rings and copper brushes (ref. 13) located on the rear side of the speed increaser as shown in figure 1. A steam box provided a constant temperature atmosphere for all dissimilar metallic junctions. Temperatures of rotating thermocouples were read by energizing a solenoid which brought the brushes into contact with the slip rings for the period of time required to determine temperatures.

Test Bearings

All the test bearings were 222-size split inner-race ball bearings equipped with one-piece inner-race riding cages (fig. 2). The over-all dimensions of the test bearings were: bore, 110 millimeters; outer diameter, 200 millimeters; and width, 38 millimeters. The nominal no-load contact angle of these bearings was 20° to 26°. The outer-race curvature (ratio of groove radius to ball diameter) was 52 percent; inner-race curvature, 51.5 percent. Races and balls were made of AISI M-1 tool steel hardened to Rockwell-C hardness values of 62 to 65. The capacity of the test bearings was calculated using the AFBMA method (ref. 7) and was found to be 30,600 pounds.

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Cage materials used were iron-silicon bronze (2.5 to 4 percent silicon, 1.0 to 2.0 percent iron, 1.5 to 4.0 percent zinc, 1.0 percent manganese, and the remainder copper), silver-plated iron-silicon bronze (0.001 plate), and both untreated and oxide-coated cast Inconel. The oxide coatings on the cast Inconel cages were applied either by heating in an air atmosphere at 1200° F for 4 hours and air-cooling or by immersing the cage in molten sodium hydroxide for 52 hours (table I). The temperature of the sodium hydroxide was kept slightly above its melting point (604° F).

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TABLE I. - TEST-BEARING FATIGUE AND CAGE WEAR DATA

Test bearing	Cage material	Total running time, hr	Life fraction	Temperature, °F: Outer race, Mean	Temperature, °F: Outer race, Max.	Temperature, °F: Inner race	Time at temperature, hr	Fatigue remarks	Cage wear remarks
1001	Iron-silicon bronze	35.9	0.277	163-299 318-394 400-468 507-527	165-300 386-418 432-575 602-645	163-287 237-363 332-473 493-569	18.5 5.1 8.3 3.0	Four spalled areas ranging in size to 3/8 x 1/4 in. on inner-race ball path. Ball path and flakings low in groove.	Light wear.
1002	Silver-plated iron-silicon bronze	50.0	0.396	167-298 300-397 400-457	168-299 303-418 460-520	188-278 283-399 389-455	5.1 43.9 1.0	One fatigue crack about 3/8 in. long across inner-race ball path. Crack and ball path low in groove.	Very little wear.
1003	Silver-plated iron-silicon bronze	50.1	0.379	156-292 307-398 406-491 510	170-338 342-419 418-536 563	164-305 286-409 325-428 435	6.3 27.5 14.6 1.7	One fatigue crack about 5/16 in. long across inner-race ball path. Crack and ball path low in groove.	Very light cage wear.
1004	Cast Inconel (non-oxide coated)	16.9	0.121	166-299 301-392 402-485 515-556	167-310 304-416 410-520 544-580	164-256 255-370 368-407 Inoperative	7.2 4.8 3.6 1.3	No fatigue failure.	Operation rough and motor torque high from start. Cage inside diameter and inner-race outside diameter sustained severe wear.
1005	Cast Inconel (oxide coated by heating at 1200 °F in air and air-cooling)	3.5	0.016	165-289 302	166-298 309	151-277 276	3.1 .4	No fatigue failure.	Test stopped to observe first evidence of wear at cage locating surface. Welds evident on cage and inner race. Oxide coating on cage worn through in spots.
1008	Silver-plated iron-silicon bronze	55.6	0.395	158-298 314-398 400-492 517-521	160-300 316-405 412-519 534-538	141-269 264-403 208-426 220-328	15.3 17.1 22.2 1.0	Two spalled areas, 1/32 and 3/32 in. in diameter on outer-race ball path. Ball path and flakings low in groove.	Light wear at a DN of 1.275×10^6, oil flows of 7-10 lb/min, and mean outer-race temperatures up to 479° F. At DN of 1.34 to 1.54×10^6, oil flow of 7 lb/min, oil inlet temperature of 250° F, and mean outer-race temperature of 481° to 524° F, silver plate flaked at cage locating diameter. Large silver flakes deposited on races and balls.
1006	Cast Inconel (oxide coated by chemical means)	13.5	0.096	162-295 318-381 400	163-296 320-386 411	152-269 294-420 390	5.3 8.0 .2	No fatigue failure.	Severe wear of cage inside diameter and inner-race outside diameter. Incipient surface welding evident in cage pocket wear areas, but no surface damage on balls.
1007	Cast Inconel (oxide coated by chemical means)	9.2	0.069	361-380 407-441	362-398 420-484	305-340 332-350	3.2 7.4	One spalled area about $4\frac{1}{4}$ in. long in outer-race ball path. Four balls spalled, each with one spalled area ranging from 3/32 in. in diam. to 9/32 x 3/32 in.	Fatigue failure of outer race necessitated stopping test. Multiple fracture of cage precipitated by outer-race spall. Cage inside diameter worn with evidence of surface welding. Oxide coating on cage worn through over some of cage inside diameter.
1009	Iron-silicon bronze	31.7	0.206	280-285 315-386 410-494 501-593 603-678	285-289 320-396 428-508 522-668 755-838	324-330 312-470 302-339 310-506 480-550	1.9 3.6 11.2 13.3 1.7	Two spalled areas, 9/32 x 7/16 in. and 3/16 x 1/16 in. and 10 small fatigue cracks across inner-race ball path.	Cage wear light up to oil-interruption tests. Oil-interruption tests caused severe wear of cage and extremely severe wear of inner and outer races. Depth of wear on cage inside and outside diameters less than 1/64 in. but on inner race was 7/64 in. and on outer race, 1/16 in. Initially inner-race located cage became outer-race located after wearing inner race until it contacted the outer race. Ratio of weight loss of races to that of cage was 7.
1005A	Silver-plated iron-silicon bronze	35.1	0.236	151-278 331 418-460 501-594 601-604	163-289 342 425-494 539-678 679-686	163-282 340 358-420 374-507 490-494	2.3 .7 4.3 26.3 2.2	Two spalled areas, 1/16 and 3/16 in. in diameter on inner-race ball path.	Some evidence of blistering of silver plate on sides of cage. Cage wear before oil-interruption test was low. Oil-interruption test caused cage inside diameter to wear several times the thickness of the plate, but there was still considerable silver smeared on the surface. This silver probably came from the ball pocket plate exposed as the inside diameter wore.
[a]1013	Silver-plated iron-silicon bronze	50.8	Outer race, 0.704 Inner race, 0.326 Balls, 0.703	462 540-599 600-633	472 567-687 680-716	405 379-405 375-456	1.0 36.3 13.3	One spalled area, $1\frac{5}{16}$ in. circumferentially and 5/16 in. axially on inner-race ball path.	No evidence of blistering of silver plate on cage. Cage wear before oil-interruption test was low. Wear due to oil-interruption test very similar to that of bearing 1005A. Varnish deposits on all surfaces showed a crack network similar to china crazing or brittle lacquer cracking.

[a]This bearing was assembled using the outer race of bearing 1002, the balls from bearing 1008, and the inner race from an unused bearing. Thus the different life fractions for the various bearing parts.

TABLE II. - TEST-BEARING FATIGUE-FAILURE EXAMINATION

[Data obtained from a commercial source.]

Bearing	Failed component	Inclusion ratings								Fracture size	Average hardness, Rockwell C-	Carbide segregation	Structure	Remarks
		Thin series				Heavy series								
		A	B	C	D	E	F	G	H					
1001	Inner race	0	1.0	1.5	1.5	0	0	1.25	1.0	$8\frac{1}{2}$	64.8	Moderate	Good	Ball path and flakings low in groove
1003	Inner race	0	1.25	0	1.5	0	0	0	0	9	64.6	Moderate	Good	Fatigue crack across ball path about 5/16 in. long
1007	Outer race	0	1.5	0	1.5	0	1.25	0	0	(a)	[b]58.7 to 65.0	Slight	Good	
	Ball	-	----	---	---	-	----	----	---	---	64.4	---------	----	
	Ball	Satisfactory								(a)	64.6	---------	Good	
	Ball	Satisfactory								(a)	65.2	---------	Good	
	Ball	Satisfactory								(a)	66.0	---------	Good	
1008	Outer race	0	1.5	0	2.0	0	0	0	0	(a)	63.2	Slight	Good	Ball path and flakings low in groove

[a]Within specified limits.

[b]Low hardness attributed to localized decarb on side surface.

TABLE III. - OIL-INTERRUPTION TEST CONDITIONS

Bear-ing	Test order	DN	Load		Oil flow, lb/min	Oil inlet temp., °F	Outer-race mean temp., °F	Inner-race temp., °F	Mean outer-race temp. - inner-race temp., ΔT, °F	Time to failure (a)
			Thrust	Radial						
1009	1	1.27×10^6	5000	1000	10	246	411	316	95	1 Min, 48 sec
	2	1.27				246	326	326	0	1 Min
	3	1.27				256	461	330	131	2 Min, 25 sec
	4	.389				247	469	315	154	5 Min, 54 sec
	5	.785				247	467	319	148	2 Min, 8 sec
	6	.785				247	467	307	160	2 Min, 6 sec
	7	1.09				247	464	316	148	2 Min, 1 sec
1005a	1	1.27×10^6	5000	1000	10	262	444	348	96	5 Min, 23 sec
	2	1.27	5000	1000	10	259	441	362	79	2 Min, 22 sec
1013	1	1.27×10^6	5000	1000	7	268	449	335	114	36 sec

[a]Failure is defined as the breaking of a shear pin that occurred at about 250 percent of normal drive torque.

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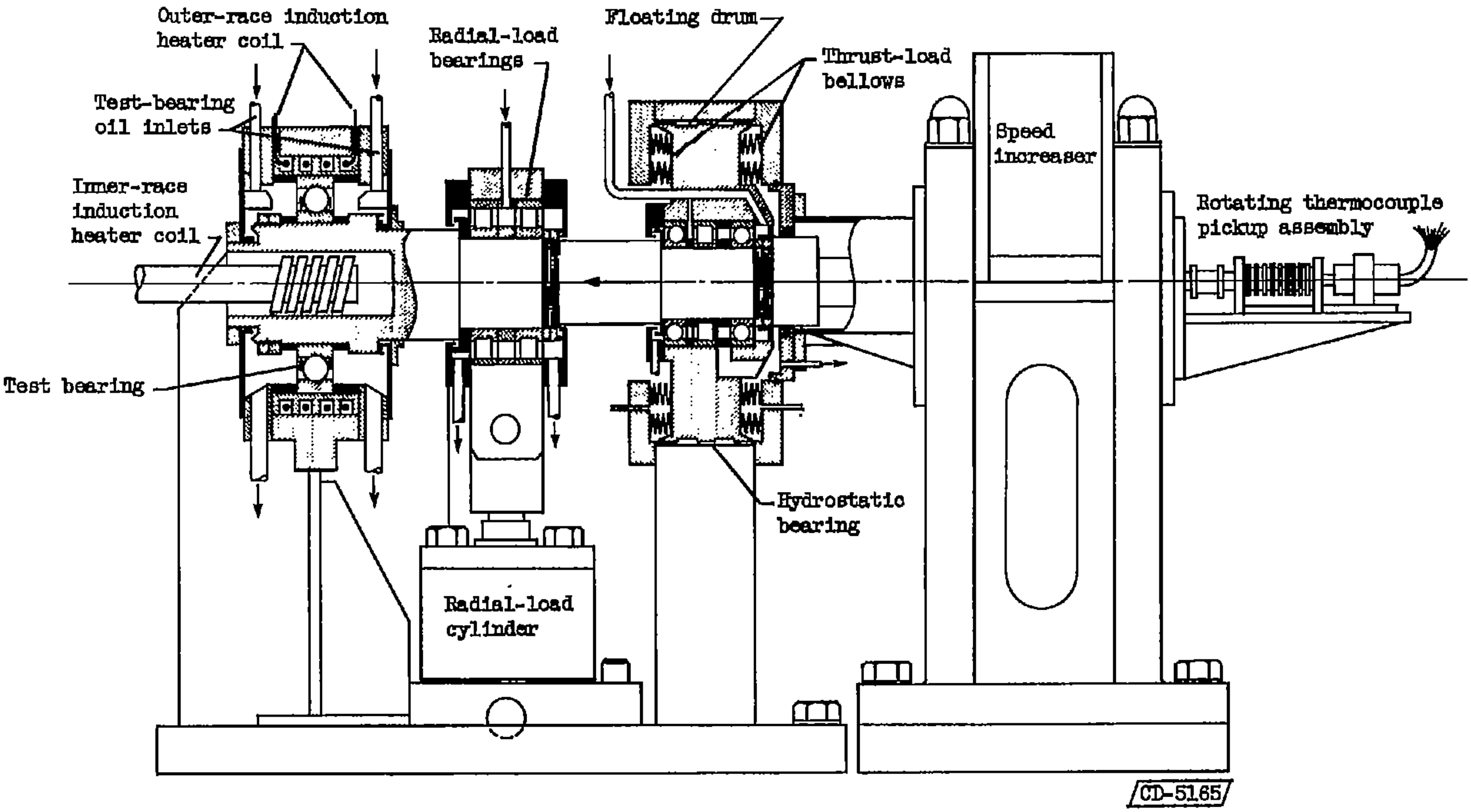

Figure 1. - High-temperature bearing test rig.

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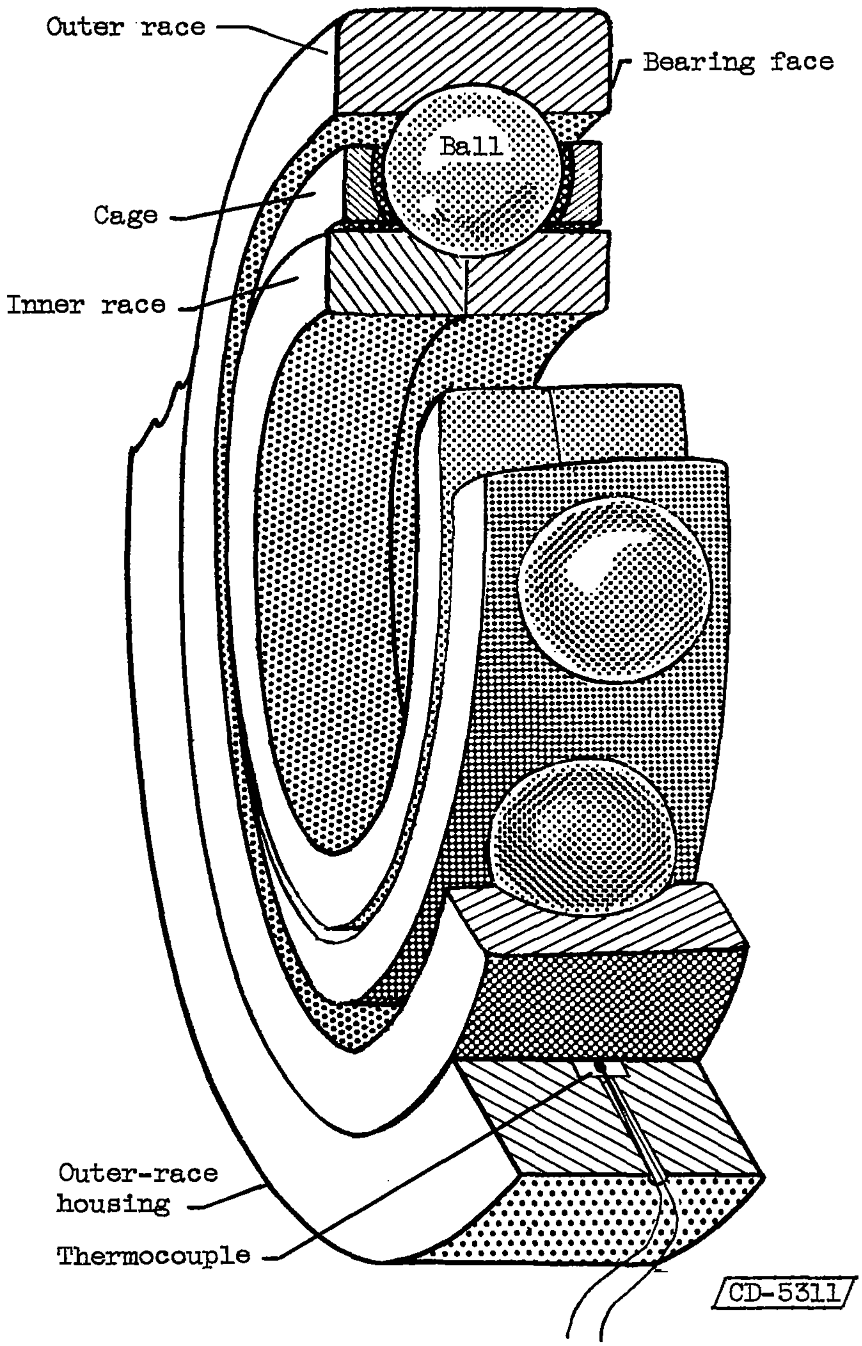

Figure 2. - Schematic drawing of test bearing.

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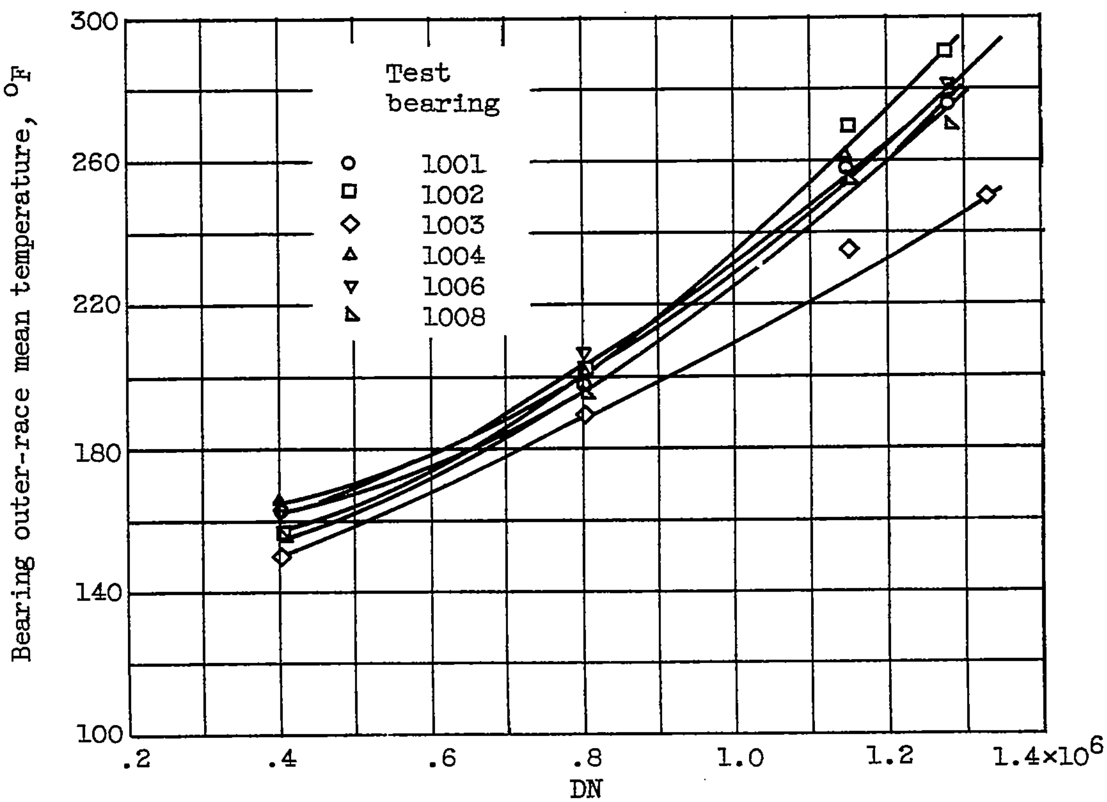

Figure 3. - Effect of DN value on mean outer-race temperature of test bearings with no external heat added. Radial load, 1000 pounds; thrust load, 5000 pounds; oil flow, 10 pounds per minute; and oil inlet temperature, 150° F.

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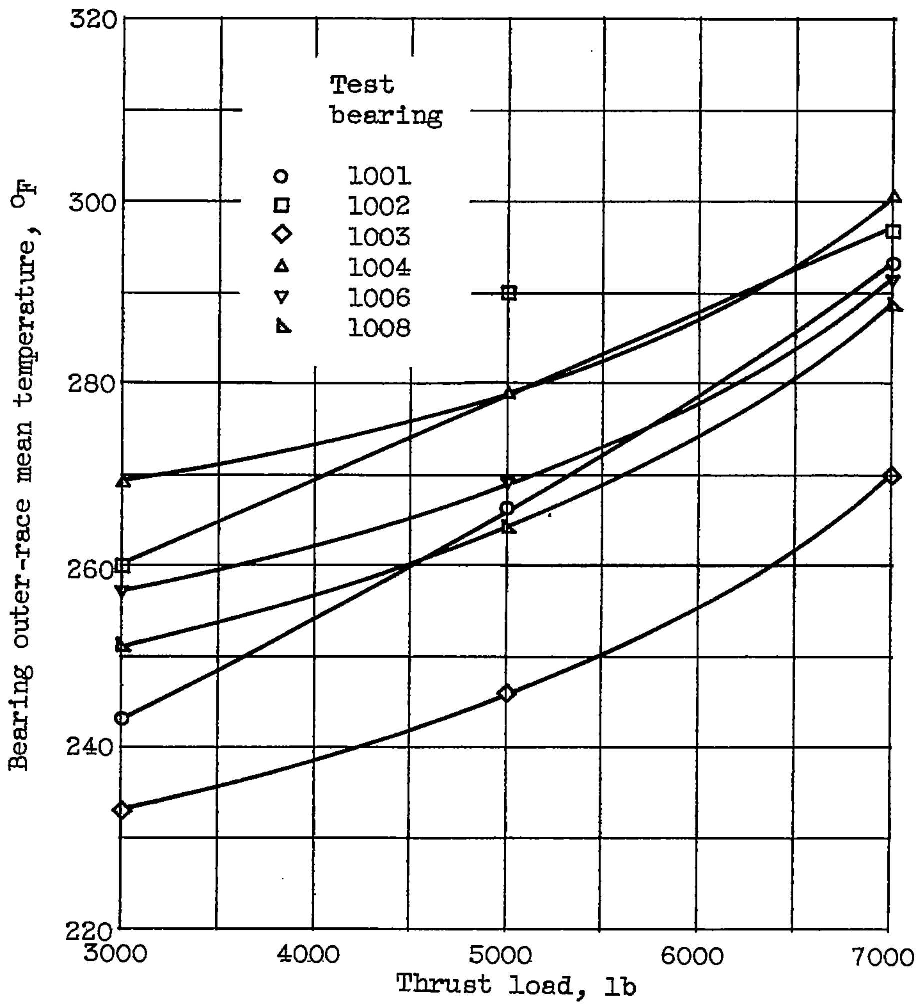

Figure 4. - Effect of thrust load on mean outer-race temperature of test bearings with no external heat added. DN, 1.27×10^6; radial load, 1000 pounds; oil flow, 10 pounds per minute; and oil inlet temperature, 150° F.

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CU-4

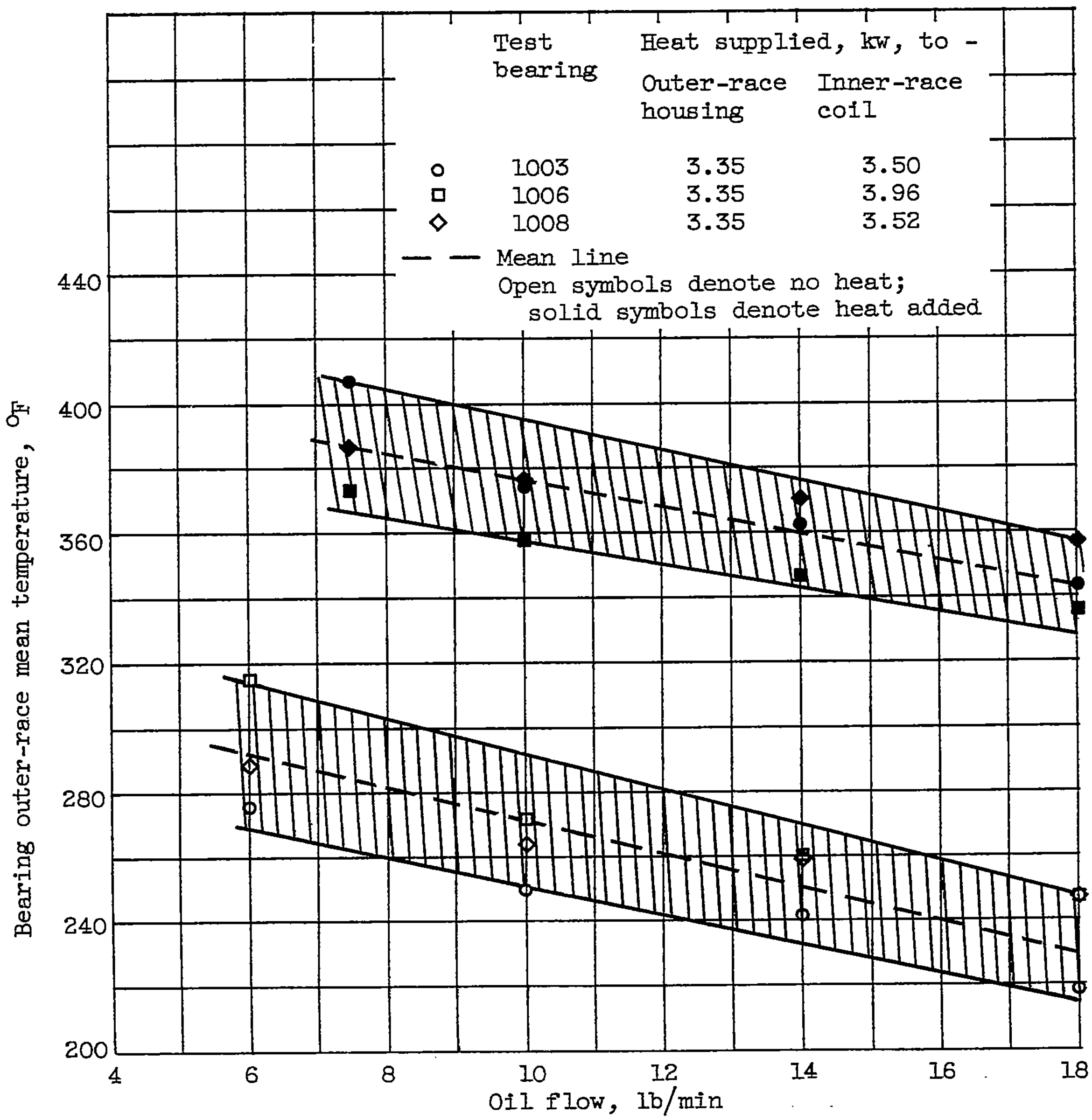

Figure 5. - Effect of oil flow on mean outer-race temperature of test bearings with and without external heat added. DN, 1.27×10^6; radial load, 1000 pounds; thrust load, 5000 pounds; and oil inlet temperature, 150° F.

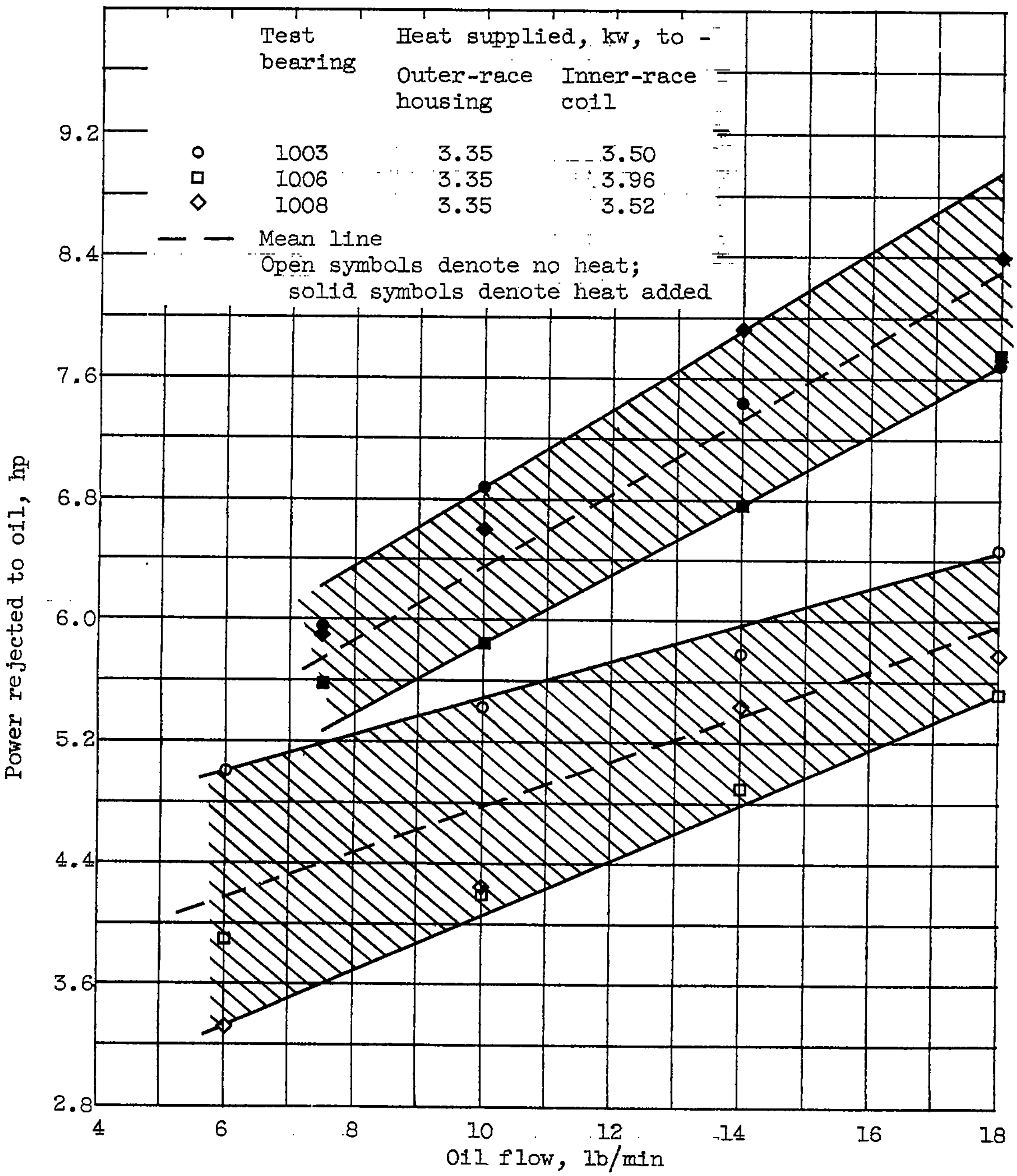

Figure 6. - Effect of oil flow on power rejected to oil for test bearings with and without external heat added. DN, 1.27×10^6; radial load, 1000 pounds; thrust load, 5000 pounds; and oil inlet temperature, 150° F.

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CU-4 back

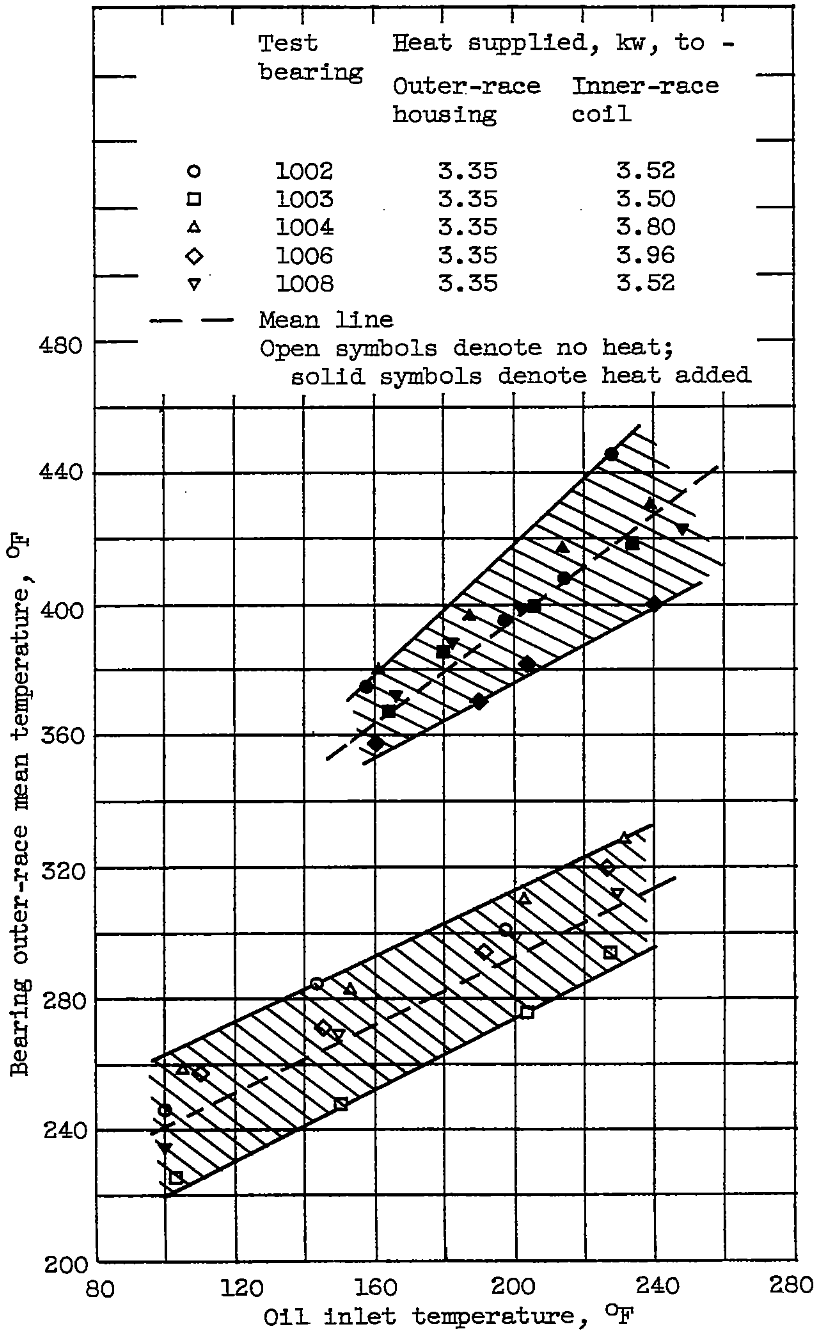

Figure 7. - Effect of oil inlet temperature on mean outer-race temperature of test bearings with and without external heat added. DN, 1.27×10^{6}; radial load, 1000 pounds; thrust load, 5000 pounds; and oil flow, 10 pounds per minute.

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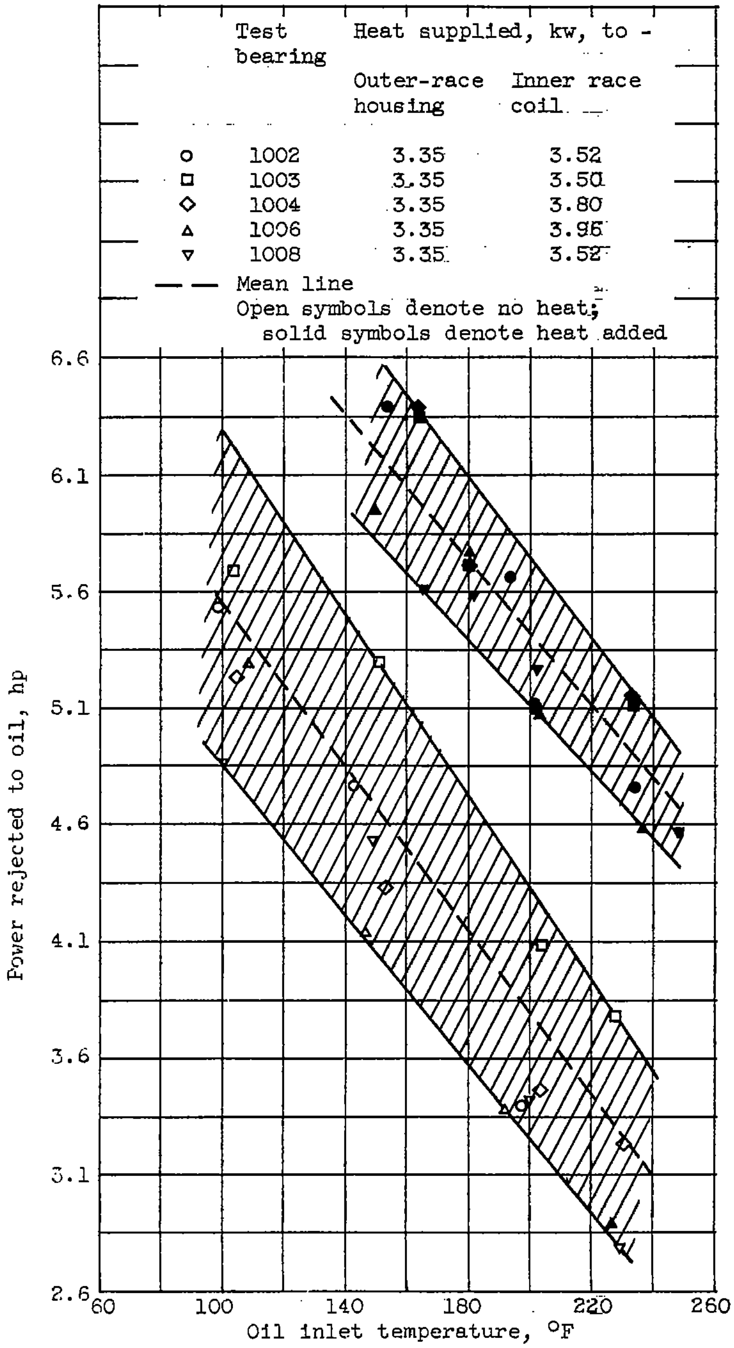

Figure 8. - Effect of oil inlet temperature on power rejected to oil for test bearings with and without external heat added. DN, 1.27×10^6; radial load, 1000 pounds; thrust load, 5000 pounds; and oil flow, 10 pounds per minute.

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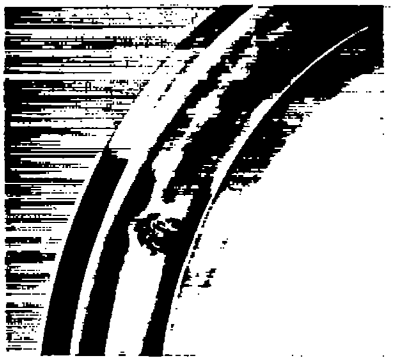
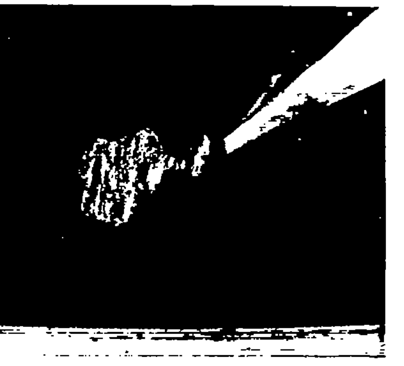
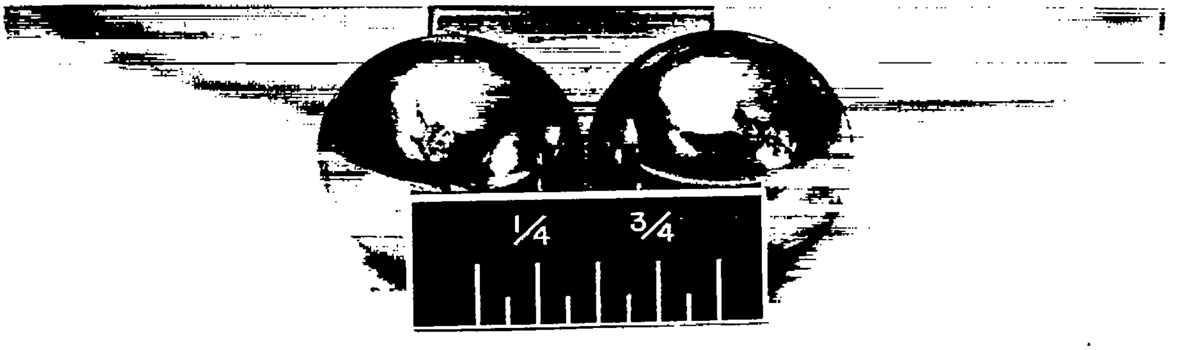
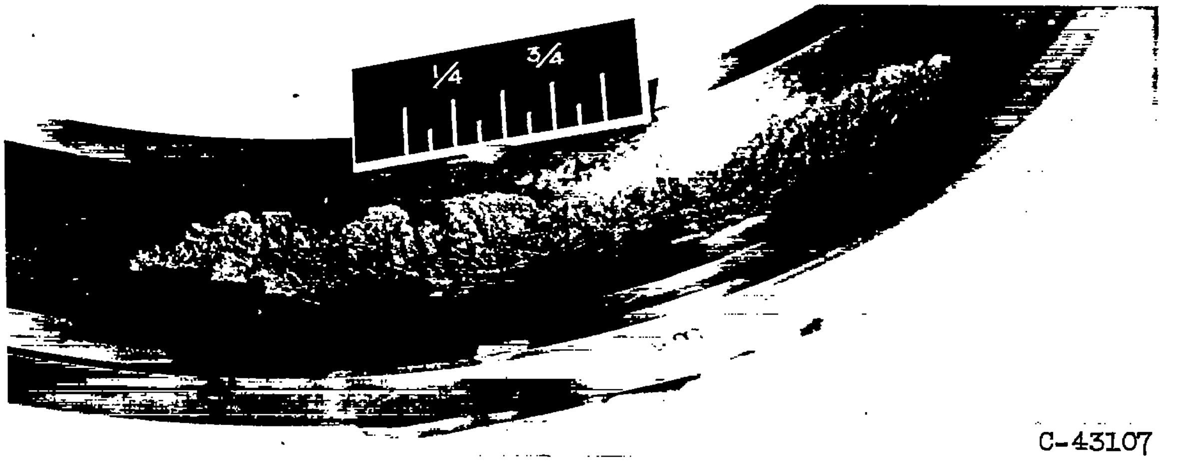

Bearing 1001


Bearing 1003


Bearing 1005A



Bearing 1007



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Bearing 1007

Figure 9. - Fatigue spalls and cracks on test bearings.

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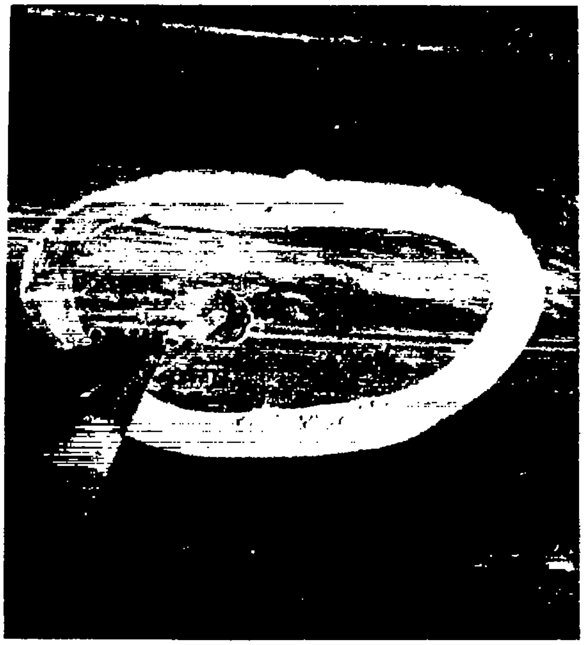
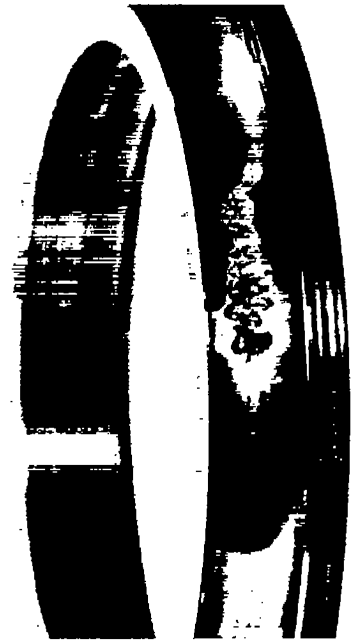


Bearing 1008



Bearing 1013

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Bearing 1009

Figure 9. - Concluded. Fatigue spalls and cracks on test bearings.

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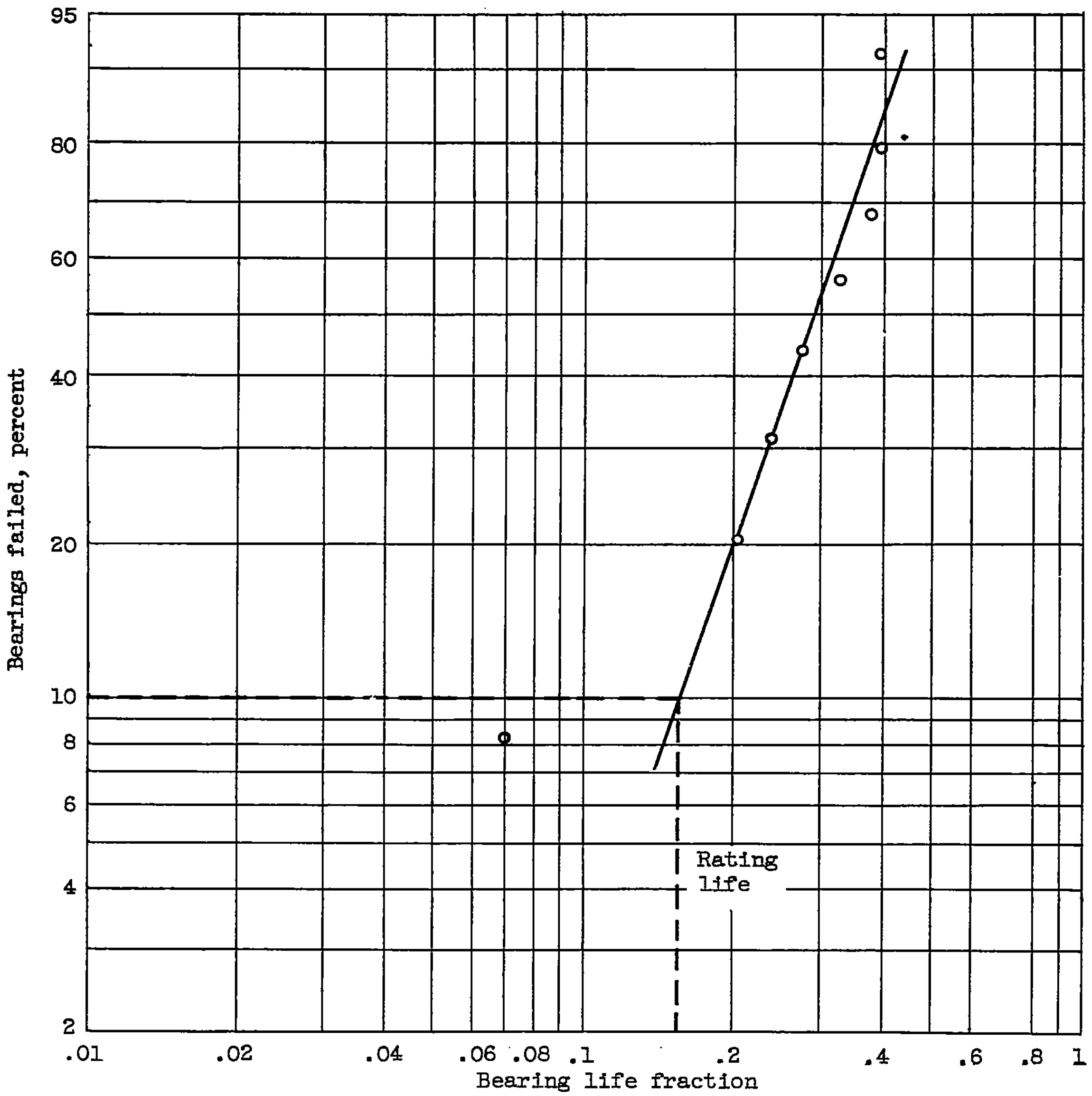

Figure 10. - Weibull plot of eight fatigue failures obtained in M-1 tool steel test bearings.

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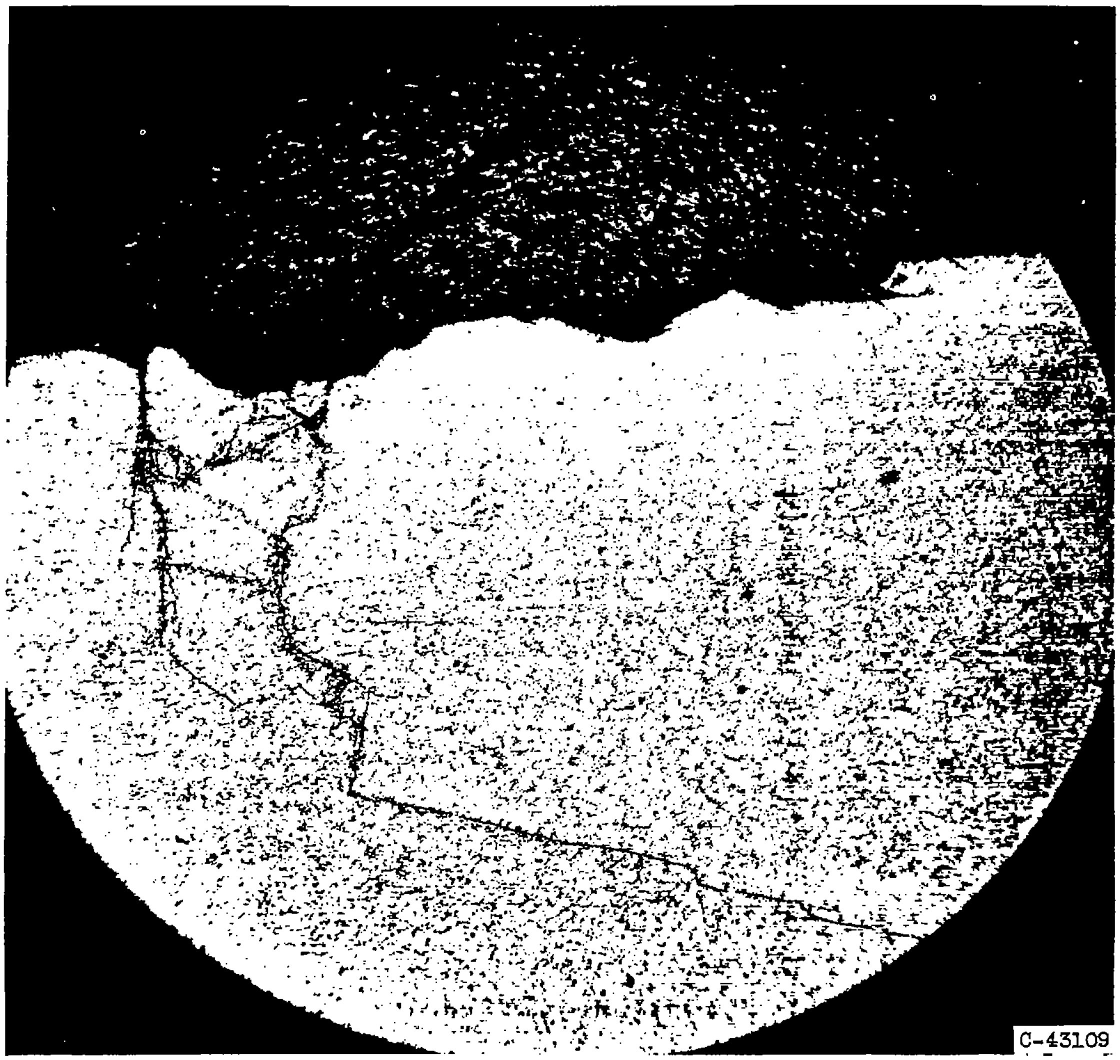


Figure 11. - Section through inner-race fatigue spall of bearing 1001 parallel to bearing face. X40.

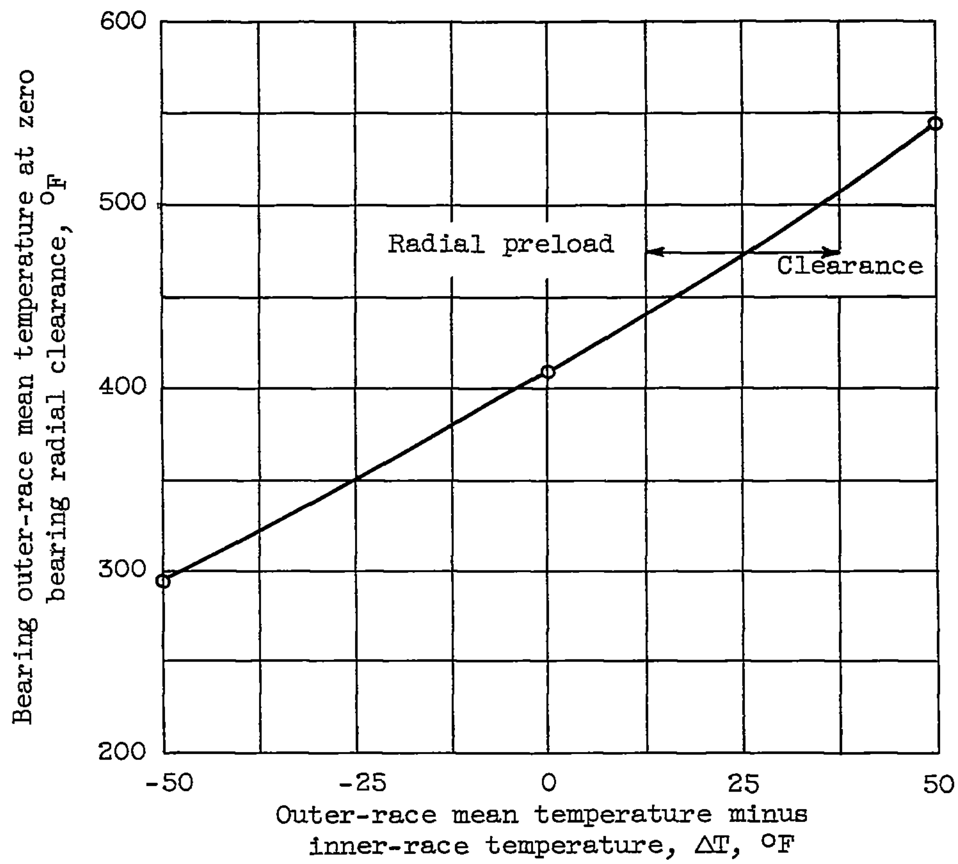
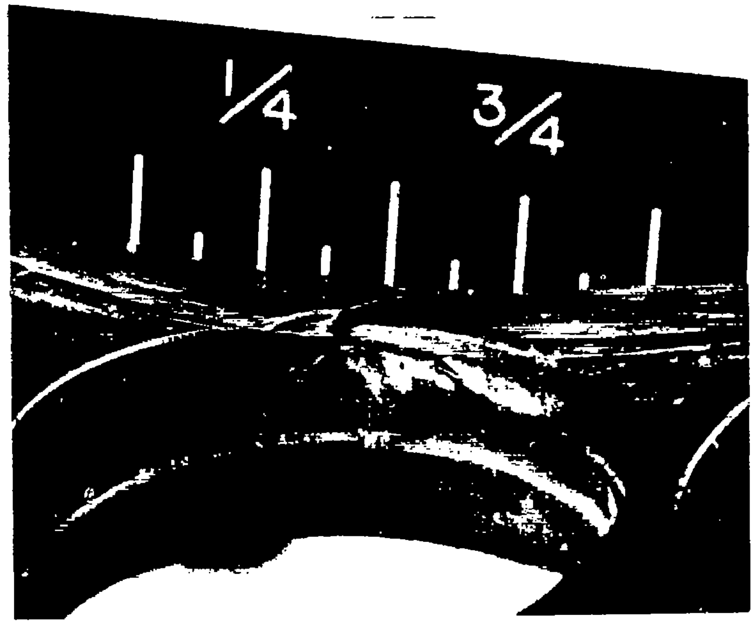
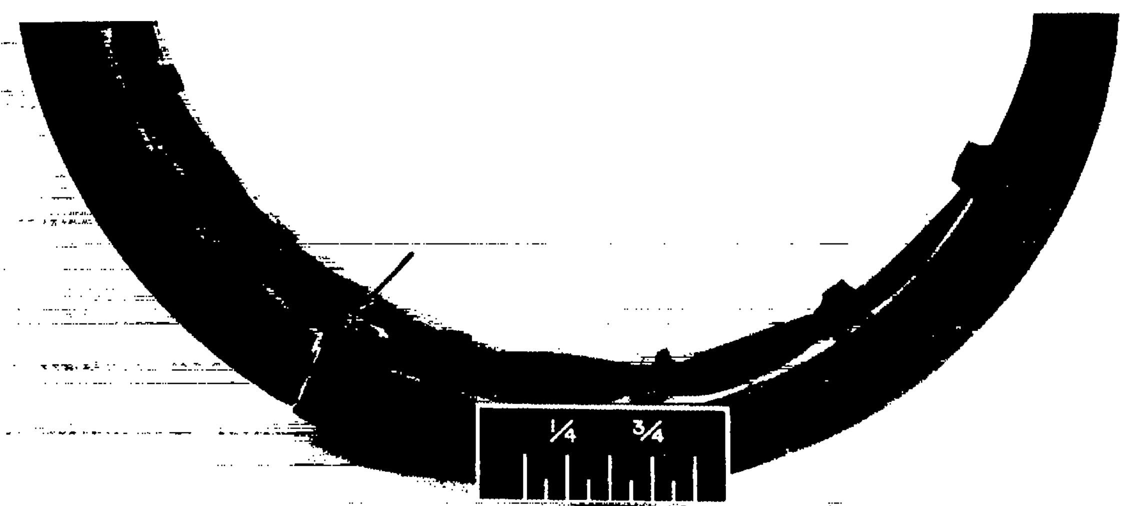


Figure 12. - Test-bearing outer-race mean temperature at which zero bearing radial clearance exists as function of difference in outer- and inner-race temperatures. Determined experimentally with bearing 1009.

(a) Bearing 1009. Cage wear resulting from several oil-interruption tests.



(b) Bearing 1008. On this cage silver plate broke down and came off in large flakes on one side.



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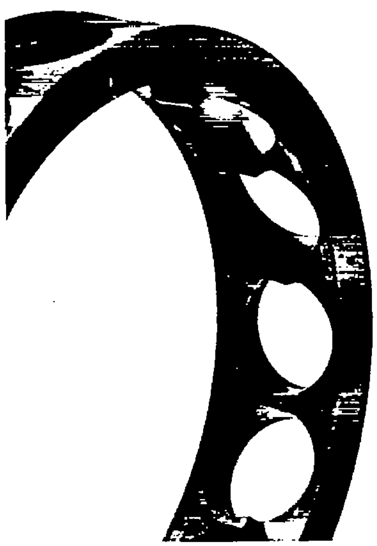
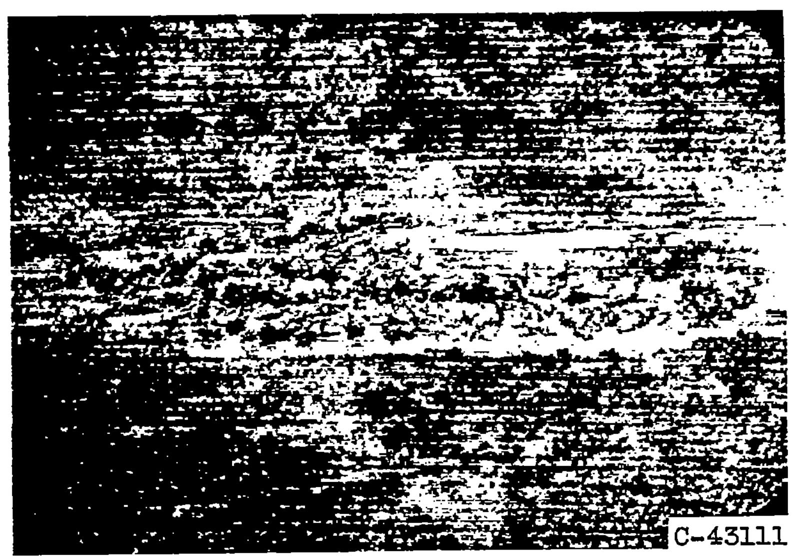
(c) Bearing 1007. Cage wear and fractures resulting from fatigue spalls.

Figure 13. - Test-bearing cage wear characteristics.

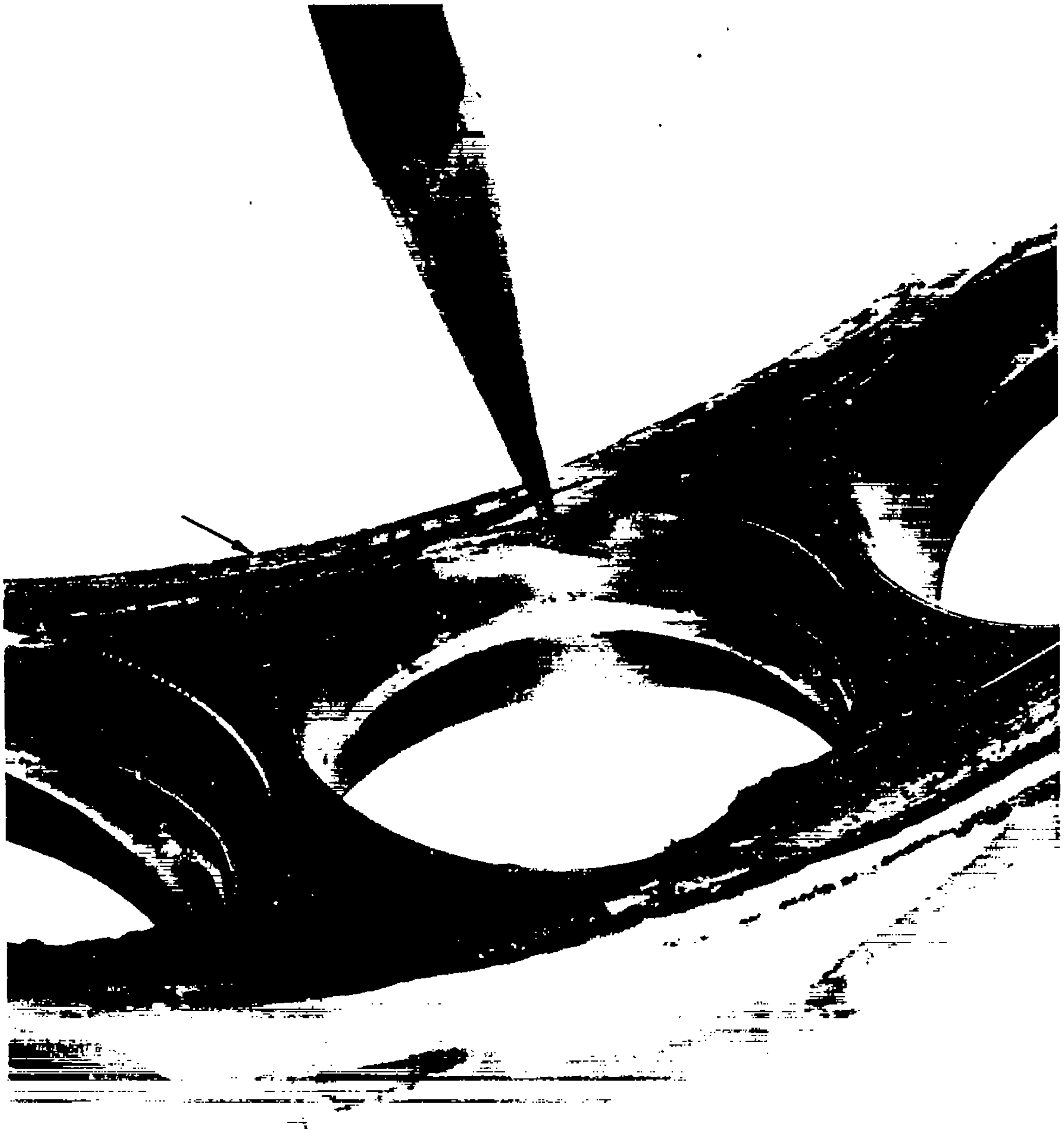
(d) Bearing 1006. Cage locating surface and pocket wear. Note surface weld in pocket.





(e) Bearing 1005. (f) Bearing 1005. Cage inside diameter showing incipient surface weld. X15.

Figure 13. - Continued. Test-bearing cage wear characteristics.



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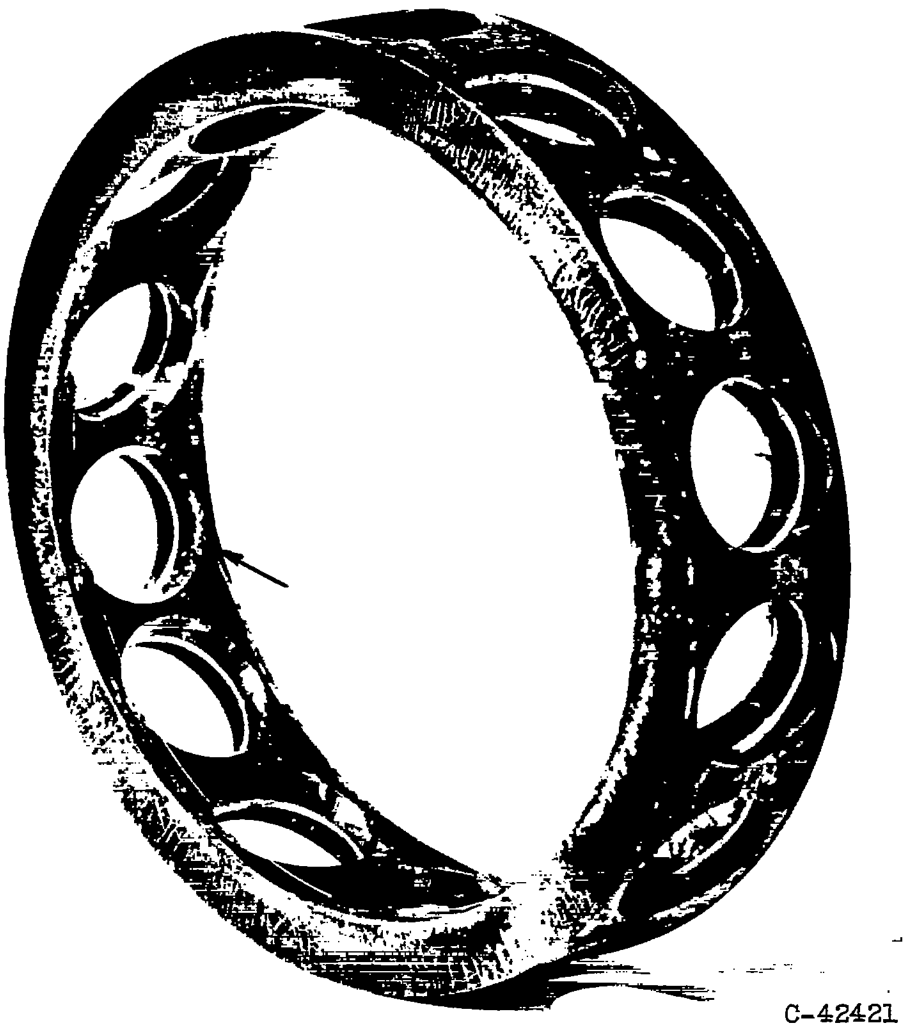
(g) Bearing 1005A. Cage locating surface and pocket wear resulting from an oil-interruption test.

Figure 13. Continued. Test-bearing cage wear characteristics.



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(h) Bearing 1013.

Figure 13. - Concluded. Test-bearing cage wear characteristics.

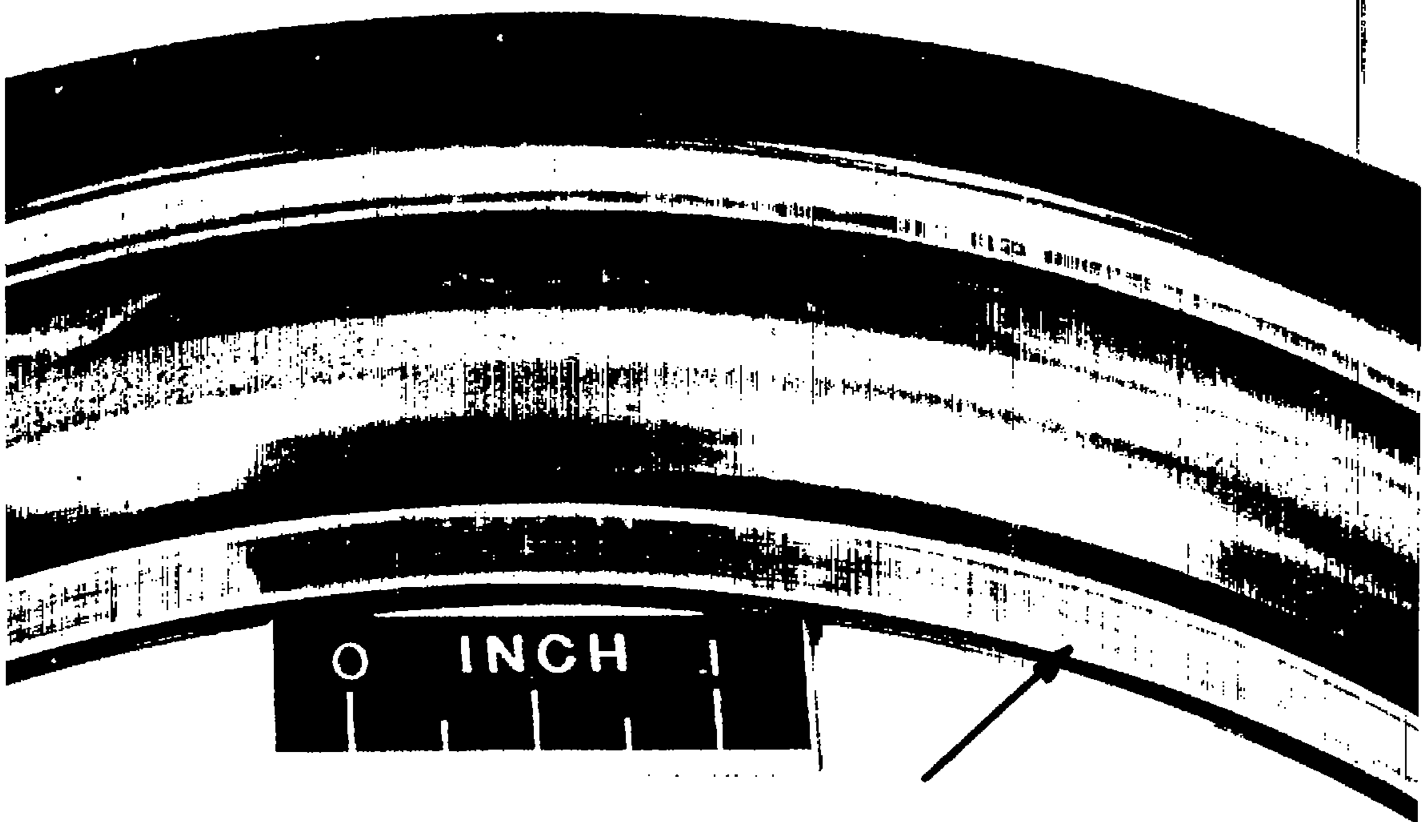

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Figure 14. - Outer race of bearing 1009 showing wear resulting from cage contact during oil-interruption tests.

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