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**Scientific and Technical
Information Branch**

Summary

Parametric tests were conducted in a high-speed bearing tester on three 118-mm-bore roller bearings, each with a three-piece inner ring. The bearings had different diametral clearances but were otherwise identical. They were manufactured from consumable-electrode-vacuum-melted (CVM) AISI M–50 steel with a room-temperature Rockwell C hardness of 60. Test parameters were radial loads of 2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). Outer-ring cooling flow rates were 0 to 3.22×10^{-3} m^3/min (0 to 0.85 gal/min) at an oil-inlet temperature of 366 K (200° F). The results of this study were compared with those of a previously tested similar 118-mm-bore roller bearing with a one-piece inner ring. The lubricant neopentylpolyol (tetra) ester met the MIL–L–23699 specifications.

A 118-mm-bore test bearing with a three-piece inner ring ran successfully at 3.0×10^6 DN for 20 hr. Bearing temperature increased with speed and decreased with increasing bearing clearance and oil flow rate. The difference between the outer- and inner-ring temperatures is small for a three-piece-inner-ring bearing compared with that of a one-piece-inner-ring bearing, indicating a marked difference in heat-transfer characteristics of the two configurations. Power loss within the bearing increased with both speed and total oil flow rates to the inner ring.

The outer-ring temperature of a three-piece-inner-ring bearing decreased by a maximum of 22 K (40° F) when outer-ring cooling was employed, whereas the inner-ring temperature remained essentially constant.

Cage slip decreased with increasing total lubricant flow rate. Cage slip can be greatly reduced or even eliminated by using a bearing with a very tight clearance at operating speed.

A visual inspection of the test bearing after runs with the larger clearance showed no appreciable wear. However, an after-test inspection of the bearing with the tightest clearance showed wear and plowed areas on the inner-ring track due to highly stressed areas at maximum speed conditions.

Introduction

It is anticipated that advanced aircraft turbine engines in the 1990's will require rolling-element bearings that operate at speeds to 3.0×10^6 DN. (DN is defined as the bearing bore in millimeters multiplied by the speed of the bearing shaft in rpm.) Centrifugal effects at these extreme conditions make bearing lubrication through the inner-ring mandatory. Successful operation of 120-mm ball bearings operating at 3.0×10^6 DN with this method of lubrication is reported in references 1 and 2.

Roller-bearing performance, in many cases, has been the limiting factor in the design of high-speed rotor systems because of a lack of understanding in certain aspects of roller-bearing behavior. Some basic parameters that can affect bearing performance are lubricant flow to the inner ring, inner-ring cooling, outer-ring cooling, oil-inlet temperature, bearing load, and inner-ring (shaft) speed. References 3 to 5 report experimental data on high-speed roller bearings. The parametric effects of speed and load on bearing performance were determined in reference 5 on a conventional 118-mm-bore roller bearing designed for the main shaft of an aircraft turbojet engine. The bearing had a one-piece flanged inner ring with radial holes to facilitate through-the-inner-ring lubrication. With this type of inner ring, the axial contraction of the distance between the flange thrust faces in which the rollers ride increases with speed. The primary influence causing this contraction is what might be termed the Poisson effect due to centrifugal forces; that is, as the flanges are expanded to a larger mean diameter they contract axially toward their inboard sides (ref. 6). Although the magnitude of the axial contraction is too small to actually squeeze the rollers, the fact that the geometry of the inner flange surfaces are changed from their unmounted configuration could influence bearing operation.

This potential problem can be eliminated by employing a three-piece inner ring. Also, the raceway, and flange inboard sides, of a three-piece-inner-ring bearing can be lapped or honed to result in better surface finishes than those obtained by grinding a conventional one-piece inner ring. On metallic surfaces finished by grinding, asperities as steep as 30° are common. On finely lapped or honed metal surfaces, it is possible to reduce the slopes to 1° to 3° (ref. 7). Another advantage of a three-piece inner ring is the possibility of using a design for under-ring lubrication without holes or slots in the load-carrying (center) ring, which reduces the chance of fracture problems with the inner ring.

The investigation reported herein was conducted to evaluate a three-piece-inner-ring roller bearing at speeds to 3.0×10^6 DN. The objective was to determine the parametric effects of speed, load, and clearance on bearing performance and to compare the data with those of a similar bearing having a one-piece inner ring (ref. 5). The tests were conducted in a high-speed bearing tester on a three-piece-inner-ring, 118-mm-bore roller bearing that was designed for lubrication through the inner ring.

A computer program called SHABERTH was used to predict bearing performance. This program is capable of calculating the thermal and kinematic performance of high-speed bearings, including determination of inner- and outer-ring temperatures, oil-outlet temperatures, cage speed, and bearing power loss. The program is described completely in reference 8. SHABERTH was run for several values of bearing unmounted diametral clearance to calculate the diametral clearance of the mounted bearing at operating speeds and temperatures. This clearance, which does not include the effects of bearing load, will be called the effective, hot-mounted clearance as listed in table I. The bearings were manufactured from consumable-electrode-vacuum-melted AISI M-50 steel. The room-temperature hardness was Rockwell C-60. Test parameters were radial loads of 2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb) and nominal shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature (for both the lubricating and the cooling oil) was 366 K (200° F). Oil was supplied through the inner ring for both lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). Outer-ring cooling rates were 0 to 3.1×10^{-3} m^3/min (0 to 0.82 gal/min). The lubricant, neopentylpolyol (tetra) ester, met the MIL-L-23699 specifications.

Apparatus and Procedure

High-Speed Bearing Tester

A cross-sectional drawing of the high-speed, roller-bearing test rig is shown in figure 1. The test shaft was mounted vertically and was supported radially by two roller bearings equidistant from the 118-mm-bore test roller bearing. The outer ring of the test bearing was assembled into a cylindrical housing suspended by rods that protruded from the top cover of the test rig. The housing could be moved axially to allow centering of the outer ring with the radial load line. The rollers of the test bearing were centered in the outer-ring length of the test bearing by adjusting the shaft vertically. Radial load was applied by a hydraulic cylinder attached through a linkage mechanism to a radial load strap that was wrapped partially around the housing. The test shaft was driven by an air turbine located at the bottom of the main test-shaft assembly.

TABLE I. - TEST-BEARING FITS AND CLEARANCES[a]

Test bearing	Shaft speed, rpm	Epicyclic cage speed, rpm	Bearing unmounted diametral clearance, mm (in.)	Outer-ring looseness in housing, mm (in.)	Total diametric looseness in free state, mm (in.)	Inner-ring interference fit on shaft, mm (in.)	Effective hot-mounted diametral clearance at operating speed (SHABERTH computer program), mm (in.)
1PT	0	0	0.1321 (0.0052)	0.0229 (0.0009)	0.1549 (0.0061)	0.0711 (0.0028)	
	10 000	4 564					0.0843 (0.0033)
	15 000	6 846					.0880 (.0035)
	20 000	9 128					.0527 (.0021)
	25 500	11 640					.0237 (.0009)
3PT	0	0	0.1143 (0.0045)	0.0279 (0.0011)	0.1422 (0.0056)	0.0737 (0.0029)	
	10 000	4 564					0.0597 (0.0024)
	15 000	6 846					.0421 (.0017)
	20 000	9 128					.0191 (.0008)
	25 500	11 640					-.0095 (-.0004)
3PM	0	0	0.1702 (0.0067)	0.0254 (0.0010)	0.1956 (0.0077)	0.0762 (0.0030)	
	10 000	4 564					0.1170 (0.0046)
	15 000	6 846					.1030 (.0041)
	20 000	9 128					.0827 (.0033)
	25 500	11 640					--------------
3PL	0	0	0.1880 (0.0074)	0.0254 (0.0010)	0.2134 (0.0084)	0.0737 (0.0029)	
	10 000	4 564					0.1390 (0.0055)
	15 000	6 846					.1200 (.0047)
	20 000	9 128					.0928 (.0037)
	25 500	11 640					.0585 (.0023)

[a]At 7.6×10^{-3} m^3/min (2.0 gal/min) total oil flow to bearing.

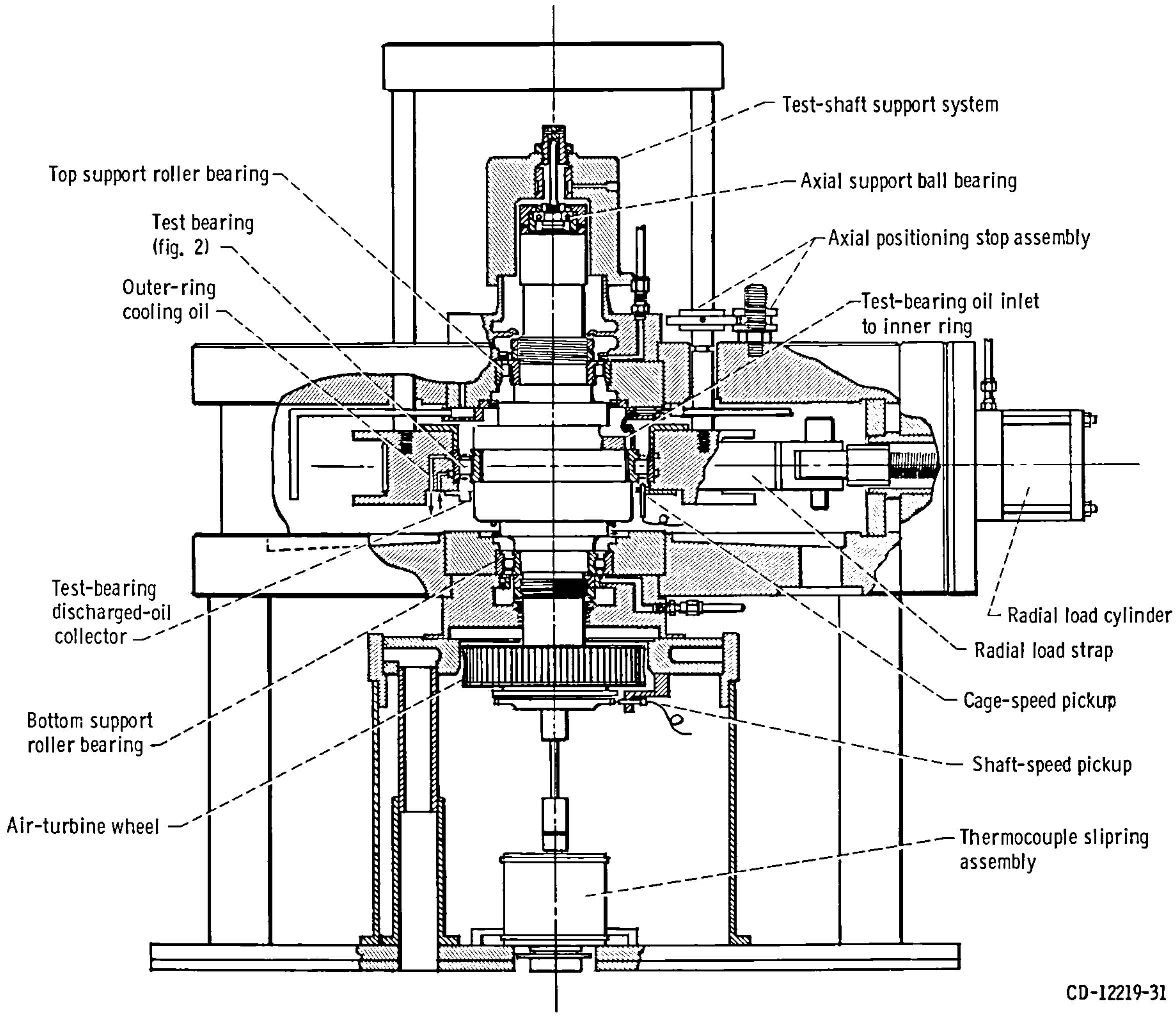


Figure 1.—High-speed roller bearing test rig.

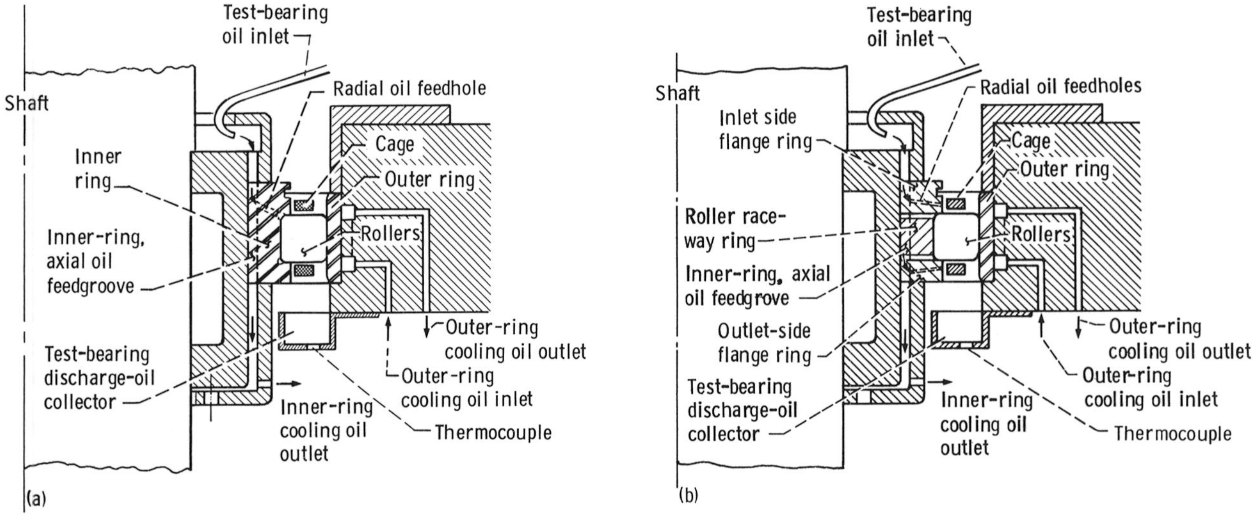
Measured lubricant flow through an inline turbine flowmeter provided oil to the test bearing by means of two jets that fed an annular groove adjacent to the bearing (fig. 2). Oil was pumped by centrifugal force through the grooves in the test-bearing bore and through a series of small radial slots to the rolling elements. Those axial grooves in the bearing bore that did not have radial slots allowed oil to flow axially under the ring for inner-ring cooling. Measured cooling oil was supplied through an inline turbine flowmeter to the outer ring by means of holes and grooves in the bearing housing (figs. 1 and 2).

Shaft speed (inner-ring speed) was measured with a magnetic probe. Cage speed was measured with an induction probe on the face of the test-bearing cage. Inner-ring temperature was measured with two spring-loaded thermocouples that projected radially outward from the shaft center at the test-bearing location. Outer-ring temperature was measured with three equally spaced spring-loaded thermocouples in the test housing, one 15° from the center of the bearing load zone. The temperature of the oil supplied to the test-bearing inner ring was measured near the jets that supplied the annular groove that fed the bearing. The temperature of the oil discharged from the test bearing was measured in an oil collector that had a thermocouple at the bottom of its drain hole (figs. 1 and 2). In this way accurate readings of the oil-inlet and -outlet temperatures were obtained.

Test Bearings

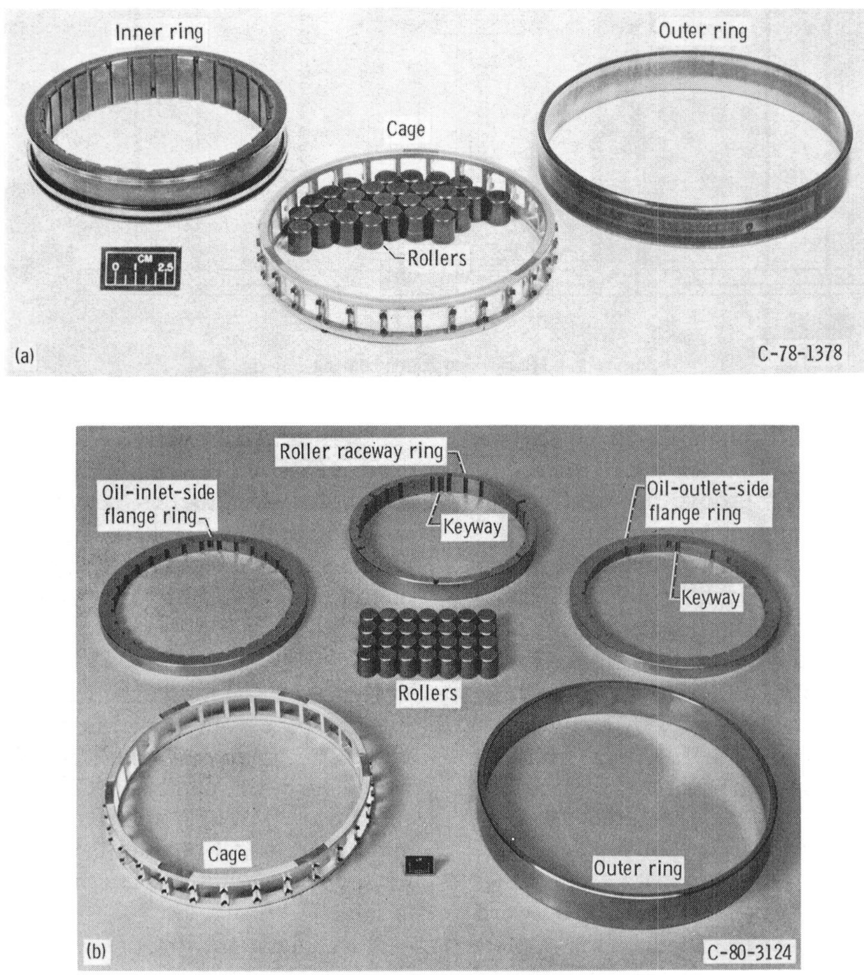
The test bearing in each case was an ABEC-5 grade, 118-mm-bore roller bearing with an inner-ring riding cage. The inner ring consisted of three separate pieces keyed together at the inside diameter. The bearing contained 28 rollers, approximately 12.6 mm (0.498 in.) in diameter and 14.6 mm (0.573 in.) long. Specifications for the one- and three-piece-inner-ring bearings are given in table II.

A schematic drawing of the one- and three-piece-inner-ring test bearings and their lubrication and cooling systems is shown in figures 2(a) and (b), respectively. The two different bearing configurations are shown disassembled in figure 3. Each bearing design permitted



(a) One-piece inner ring.
(b) Three-piece inner ring.

Figure 2.—Test-bearing lubrication and cooling for two different bearing inner-ring configurations.



(a) One-piece inner ring.
(b) Three-piece inner ring.

Figure 3.—118-Millimeter-bore, high-speed roller bearing.

TABLE II. - TEST BEARING SPECIFICATIONS

Bearing configuration	One-piece inner ring (ref. 5), 1P	Three-piece inner ring		
		3PT	3PM	3PL
Bearing bore[a], mm-(in.)	118 (4.6454)	118 (4.6454)	118 (4.6454)	118 (4.6454)
Bearing inner-ring, outer-ring, and roller material	CVM M-50 steel	CVM M-50 steel	CVM M-50 steel	CVM M-50 steel
Cage material	Silver-plated AMS 6414 steel	Silver-plated AMS 6414 steel	Silver-plated AMS 6414 steel	Silver-plated AMS 6414 steel
Number of rollers	28	28	28	28
Bearing pitch diameter, mm (in.)	145.00 (5.70)	145.00 (5.70)	145.00 (5.70)	145.00 (5.70)
Number of grooves in bearing bore	31	28	30	28
Number of radial lubricant feedholes in inner ring	16	14	16	14
Oil flow to bearing inner ring for lubrication, percent of total[b]	52	50	53	50
Oil flow to bearing inner ring for inner ring cooling, percent of total[b]	48	50	47	50
Roller diameter[a], mm (in.)	12.647/12.649 (0.4979/0.4980)	12.644/12.647 (0.4978/0.4979)	12.652/12.654 (0.4981/0.4982)	12.644/12.647 (0.4978/0.4979)
Roller length[a], mm (in.)	14.562/14.564 (0.5733/0.5734)	14.564/14.569 (0.5734/0.5736)	14.567/14.575 (0.5735/0.5738)	14.564/14.569 (0.5734/0.5736)
Roller end clearance in raceway[a], mm (in.)	0.33 (0.0013)	0.33 (0.0013)	0.25 (0.0010)	0.33 (0.0013)
Roller axial pocket clearance in cage[a], mm (in.)	0.272 (0.0107)	0.300 (0.0118)	0.300 (0.0118)	0.300 (0.0118)
Roller tangential pocket clearance in cage[a], mm (in.)	0.218 (0.0086)	0.241 (0.0095)	0.239 (0.0094)	0.241 (0.0095)

[a]Measured dimensions before testing.
[b]Based on bearing geometry.

lubrication through the inner ring by means of axial grooves and radial paths machined in the inner ring. Various grooves and small holes radiating from the bearing bore through the inner ring formed flow paths for lubrication of the rolling elements. There were 14 or 16 radial oil paths and 28 or 30 axial grooves in the bearing bore of the test bearings. Therefore, it was assumed that, for all test speeds, approximately 51 percent of the oil supplied to the inner ring lubricated the bearings, and the remainder flowed axially through those grooves that contained no radial flow paths. The latter flow cooled the inner ring.

Different sized outer rings were used to obtain the zero speed bearing clearances listed in table I for each bearing.

Lubricant

The oil used for the parametric studies was a neopentylpolyol (tetra) ester, a type II oil qualified to the MIL-L-23699 specifications. The major properties of the oil are presented in table III.

Procedure

The test shaft was brought to the desired speed with a radial load of 2200 N (500 lb) on the bearing. When the inner-ring oil-inlet temperature stabilized at 366 K (200° F), the desired total oil flow rate was set. When all bearing temperatures stabilized (after about 20 min), conditions were set for data acquisition. Data were

TABLE III. - PROPERTIES OF TETRAESTER LUBRICANT

Additives	Corrosion, oxidation, wear, and foam inhibitors
Kinematic viscosity, cS,[a] at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flashpoint, K (°F)	533 (500)
Autogenous ignition temperature, K (°F)	694 (800)
Pourpoint, K (°F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt.%	3.2
Specific heat at 372 K (210° F), J/kg•K (Btu/lb•°F)	2140 (0.493)
Thermal conductivity at 372 K (210° F), J/m•sec•K (Btu/hr•ft•°F)	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

[a] 10^{-4} Stoke = 1 m^2/sec.

subsequently taken at four loads—2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb)—at a constant flow rate. This procedure was repeated at four shaft speeds—10 000, 15 000, 20 000, and 25 500 rpm—at oil flow rates to the inner ring from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). (A running time of at least 20 min was allowed at each condition before data were acquired.) Some of the tests were run with outer-ring cooling oil flow rates from 0 to 3.1×10^{-3} m^3/min (0 to 0.82 gal/min). Because of the physical geometry of the inner-ring segments, it was assumed that approximately 51 percent of the total oil flow supplied to it fed the rollers and cage. The remainder flowed through the inner-ring axial grooves for cooling.

After the test runs just described, the resulting data were compared with those of a similar bearing with a conventional one-piece inner ring (ref. 5). Data comparisons were also made with results obtained from tests on two other three-piece-inner-ring bearings with increased clearances, bearings 3PM and 3PL (table I).

Results and Discussion

Parametric tests were conducted in a high-speed bearing tester on three 118-mm-bore roller bearings, each having a three-piece inner ring. The bearings had different diametral clearances but were otherwise identical. Test parameters were radial loads of 2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). Outer-ring cooling flow rates were 0 to 2.65×10^{-3} m^3/min (0 to 0.7 gal/min).

Fits and Clearances

The test-bearing fits and clearances are shown in table I. One three-piece-inner-ring bearing, 3PT (table I), was designed to result in a bearing with an effective, hot-mounted diametral clearance of −0.0095 mm (−0.0004 in.) at 3.0×10^{-6} DN. This clearance value, which does not contain the effects of bearing load, was expected to be sufficiently tight to insure minimum cage slip at high speed. Tests were also run on bearings with greater clearances than bearing 3PT—namely, bearings 3PM and 3PL (table I). Results of all tests were compared to observe the effect of clearance changes on bearing performance. Figure 4 shows the effect of shaft speed on diametral clearance of four 118-mm roller bearings. Each bearing has a different unmounted diametral clearance. The data in figure 4 were obtained from the computer program SHABERTH (ref. 8). SHABERTH considers interference fit effects, centrifugal force effects, and thermal effects on bearing clearance.

Table IV lists some limitations that were encountered in certain of the bearing tests. The one-piece-inner-ring bearing 1P (ref. 5) ran without any difficulties to the maximum speed of 25 500 rpm (3.0×10^6 DN). Bearing 3PT, one of the three-piece-inner-ring bearings, ran well up to 25 500 rpm (3.0×10^6 DN) but the thermocouple that measured oil-out temperature failed early in the test and therefore calculations for heat transfer could not be made. Bearings 3PM and 3PL ran with outer-ring temperatures that were so low that outer-ring cooling was impractical and therefore omitted. Bearing 3PM was limited to a maximum speed of 20 000 rpm when excessive bearing noise was encountered as higher speeds were attempted.

Bearing Temperature

Effect of load and clearance.—The effect of load on bearing temperature at four shaft speeds is shown in figure 5. Oil flow to the inner ring was constant at 7.6×10^{-3} m^3/min (2.0 gal/min) for a constant temperature of 366 K (200° F). No outer-ring cooling was used in these tests. Load was varied from 2200 to 8900 N (500 to 2000 lb). Temperatures of both the inner and

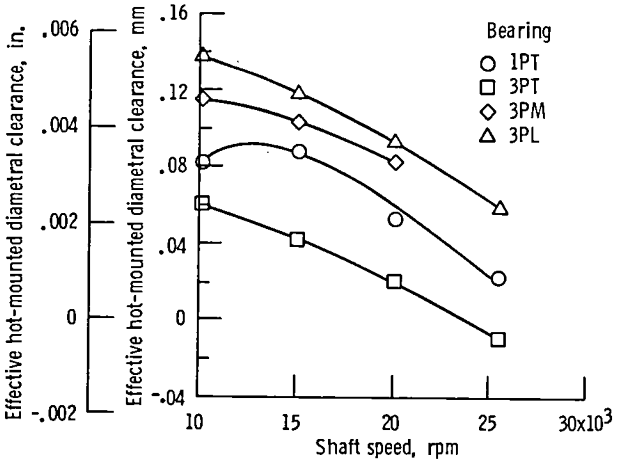
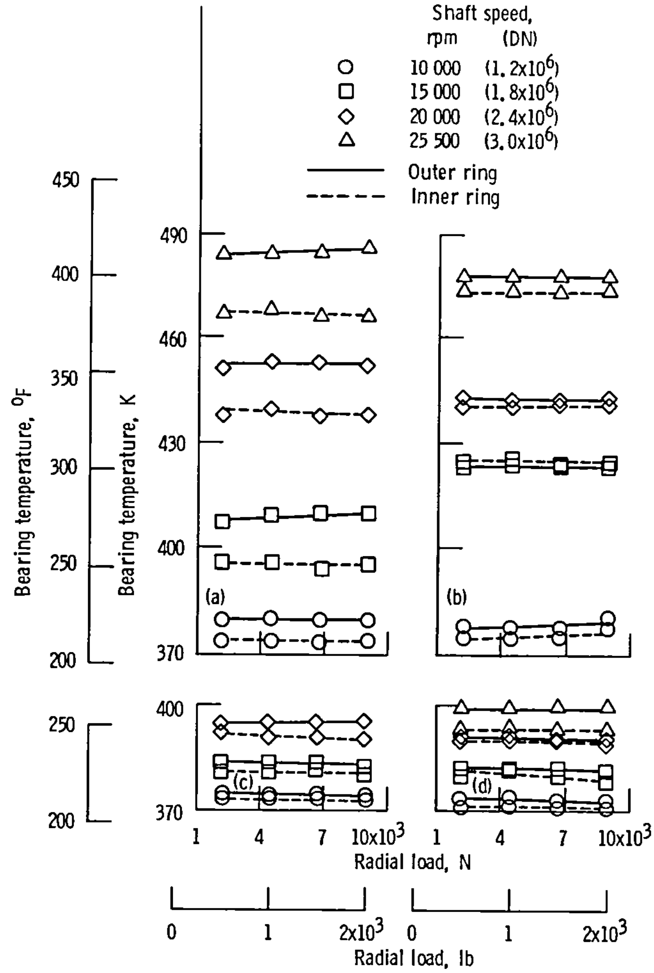


Figure 4.—Effect of shaft speed on effective hot-mounted diametral clearance of four 118-mm roller bearings with different assembled diametral play at zero rpm.

outer bearing rings for all test bearings remained constant as the load was increased at each speed tested. Since the changes in load did not have a significant effect on bearing temperature, most of the remaining tests were conducted at the maximum load of 8900 N (2000 lb). There was less temperature difference between the inner and outer ring of the three-piece-inner-ring bearing 3PT than there was for the one-piece-inner-ring bearing 1P (fig. 5). This was the case at all four speeds employed. The inner-ring temperature of bearing 3PT was higher than that of bearing 1P at each of the four speeds tested. This was probably due, in part, to the heat-transfer restrictions between the flange inner faces and the raceway ends of the three-piece-inner-ring configuration. The one-piece inner ring is one homogeneous configuration lending itself to a more efficient heat transfer away from the raceway and to the flanges. Temperature differences at identical test conditions between bearings 3PT and 1P could also be attributed to differences in bearing clearance between the two bearings.

Increasing the bearing clearance of the three-piece-inner-ring bearing markedly decreased bearing



(a) One-piece-inner-ring bearing 1P (ref. 5). Effective hot-mounted clearance at 3.0×10^6 DN, 0.0237 mm (0.0009 in.).
(b) Three-piece-inner-ring bearing 3PT. Effective hot-mounted clearance at 3.0×10^6 DN, −0.0095 mm (−0.0004 in.) (interference).
(c) Three-piece-inner-ring bearing 3PM. Effective hot-mounted clearance at 2.4×10^6 DN, 0.0827 mm (0.0033 in.).
(d) Three-piece-inner-ring bearing 3PL. Effective hot-mounted clearance at 3.0×10^6 DN, 0.0585 mm (0.0023 in.).

Figure 5.—Effect of load on bearing temperature for various shaft speeds for one- and three-piece-inner-ring bearings. Total oil flow rate to inner ring, 7.6×10^{-3} m³/min (2.0 gal/min); oil-inlet temperature, 366 K (200° F).

TABLE IV. - SOME BEARING TEST CONDITIONS AND LIMITATIONS

Bearing	Inner-ring configuration	Effective hot-mounted diametral clearance at 3.0×10^6 DN[a], mm (in.)	Maximum speed attained, rpm	Outer-ring cooling	Heat-transfer data
1PT	One piece	0.0237 (0.0009) clearance	25 500	Yes	Yes
3PT	Three piece	-.0095 (-.0004) interference	25 500	Yes	No
3PM	Three piece	.0827 (.0033) clearance[b]	20 000	No	Yes
3PL	Three piece	.0585 (.0023) clearance	25 500	No	Yes

[a]From SHABERTH computer program.
[b]At 2.4×10^6 DN.

temperatures, especially at the higher speeds (fig. 5). At an effective, hot-mounted clearance of −0.0095 mm (−0.0004 in.), using bearing 3PT, the outer-ring temperature was 478 K (400° F) at 25 500 rpm (3.0×10^6 DN). At the largest bearing clearance (bearing 3PL), the temperature was 397 K (255° F) at 25 500 rpm, a decrease of 81 K (145° F) from bearing 3PT. The effect of larger clearances on cage slip will be discussed later.

Effect of total oil flow to the inner ring.—The effect of the total oil flow to the inner ring (includes radial flow component for lubrication and the axial component for inner-ring cooling) on test-bearing temperature is shown in figure 6. No direct outer-ring cooling was used in these tests. Oil flow rate was varied from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). Speed was varied from 10 000 to 25 500 rpm.

Bearing temperature decreased with increasing total oil flow rate for all speeds tested; the decrease was more pronounced at the tight clearances of bearings 1P and 3PT than at the larger clearances of bearing 3PM and 3PL (fig. 6). Bearing temperature increased with speed. The rate of increase was greater at the tighter clearances. The difference between the outer- and inner-ring temperatures, in figures 5 and 6, is very small for the three-piece-inner-ring bearing 3PT when compared with that of the one-piece-inner-ring bearing 1P. The three-

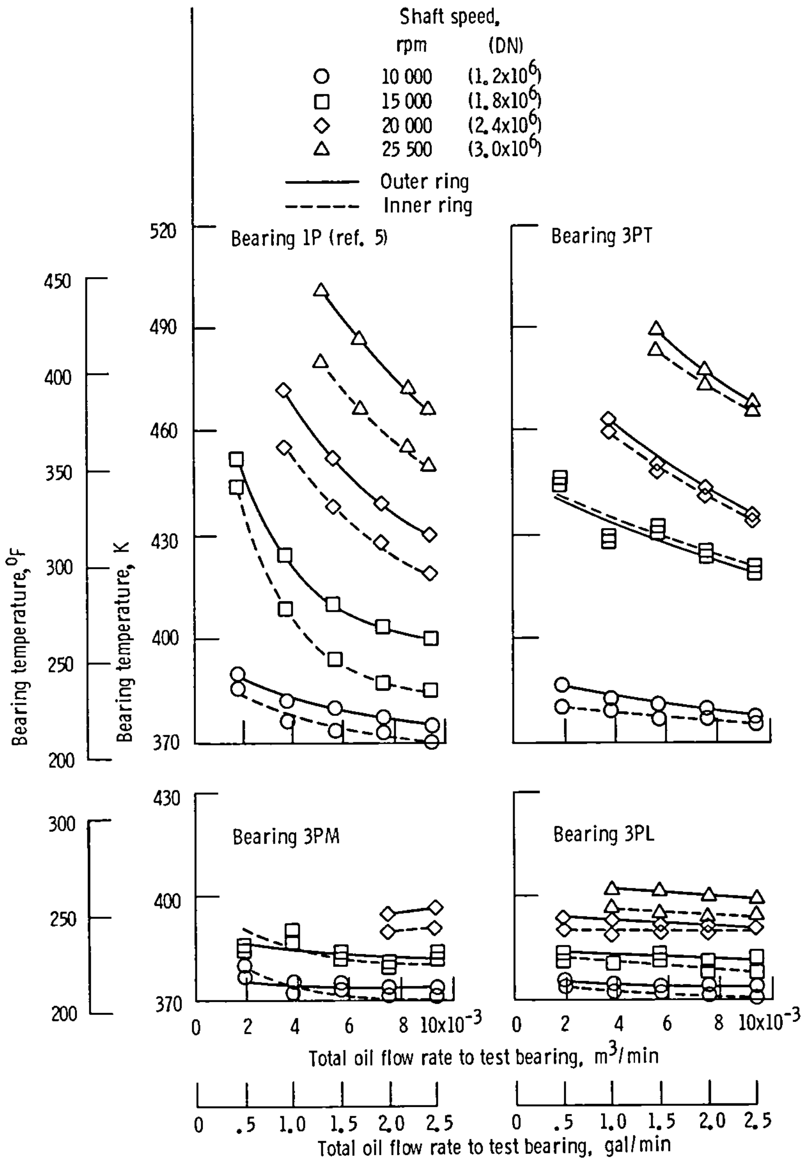


Figure 6.—Effect of total oil flow rate through inner ring on bearing temperature at various shaft speeds and diametral clearances. Oil-inlet temperature, 366 K (200° F); radial load, 8900 N (2000 lb).

piece-inner-ring bearing operates at a much cooler temperature at large clearance conditions (bearings 3PM and 3PL) than with a tight clearance (bearing 3PT).

Power Loss

Bearing power loss is dissipated in the form of heat transfer to the surrounding environment by conduction, convection, and radiation. To obtain a measure of this heat transfer and power loss within the bearing, the oil-inlet and -outlet temperatures were obtained for all lubricant flow conditions. Total heat absorbed by the lubricant was obtained from the standard heat-transfer equation

$$Q_T = MC_p(T_{\text{out}} - T_{\text{in}})$$

where

Q_T	total heat-transfer rate to lubricant, J/min (Btu/min)
M	mass flow rate, kg/min (lb/min)
C_p	specific heat, J/kg-K (Btu/lb-°F)
T_{out}	oil-outlet temperature, K (°F)
T_{in}	oil-inlet temperature, K (°F)

Approximately 51 percent of the total flow to the bearing passed radially through the bearing to lubricate the moving elements. The remainder flowed axially through those grooves (in the bearing bore) that contained no radial holes. The function of the latter flow was to cool the inner ring.

The outlet temperature of this cooling oil was not measured in these tests. Therefore, that portion of the flow is not included in the heat-transfer calculations. It is assumed that the heat-transfer to the cooling oil is small relative to that to the oil passing radially through the bearing. Some unpublished experimental data indicate that this assumption is valid. The calculated heat-transfer values are considered to be indicative of the major portion of the power loss within the bearing, and at the very least should provide valid relative effects of variations in shaft speed and lubricant flow rate.

Effect of shaft speed and total oil flow to the inner ring.—The results of the heat-transfer calculations are shown in figures 7 and 8 as a function of shaft speed and total oil flow to the inner ring, respectively. (For convenience, heat-transfer values were converted from J/min to kW). Since heat-transfer data were not obtained for the three-piece, tight-clearance bearing 3PT, because of a failure of the oil-out temperature thermocouple, data for bearings 3PL and 3PM were compared with those of the one-piece-inner-ring bearing 1P. Since the average temperatures of bearings 1P and 3PT were

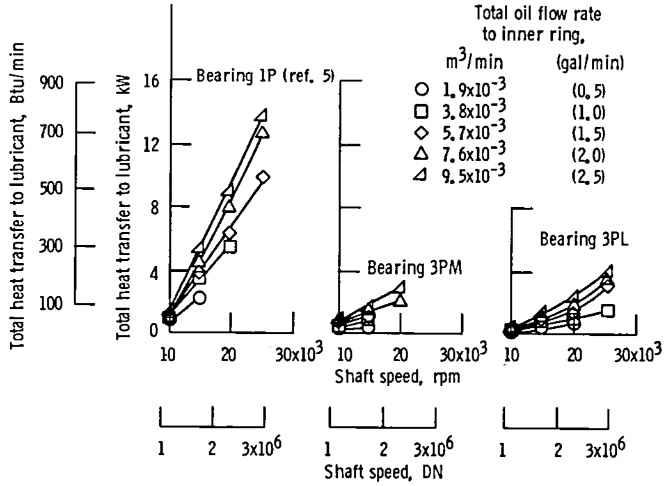


Figure 7.—Effect of shaft speed on heat transfer to lubricant for various total oil flow rates to inner ring. Oil-inlet temperature, 366 K (200° F); radial load, 8900 N (2000 lb).

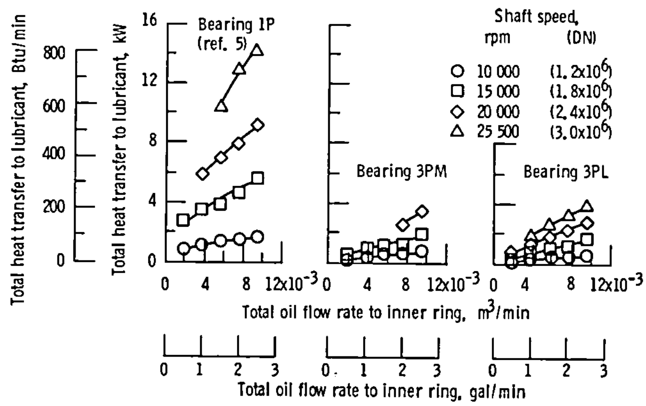


Figure 8.—Effect of total oil flow rate to inner ring on heat transfer to lubricant for various shaft speeds. Oil-inlet temperatures, 366 K (200° F); radial load, 8900 N (2000 lb).

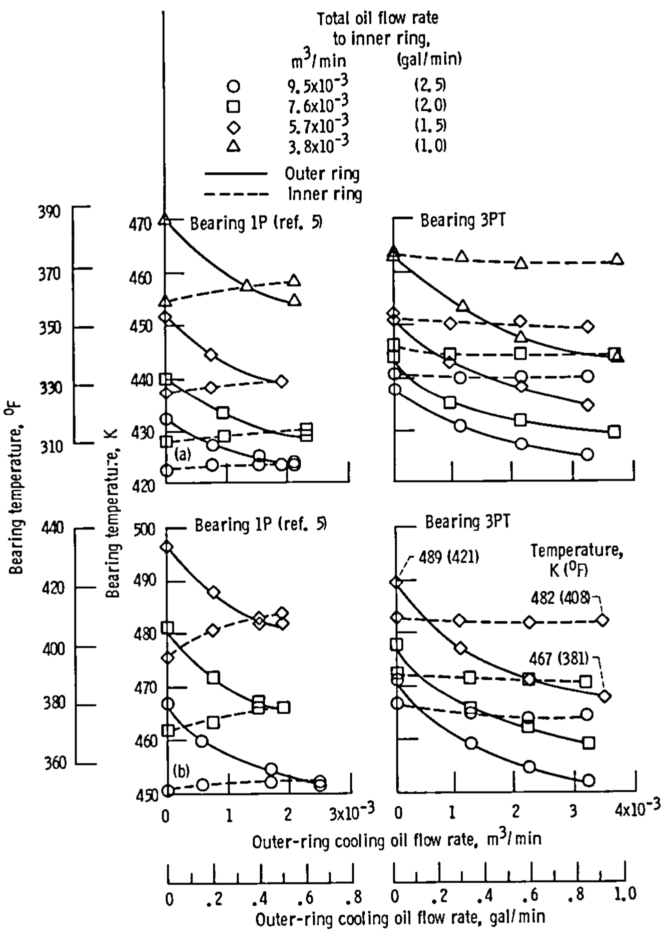
generally similar (fig. 6), this comparison is valid. The heat transfer to the lubricant (power loss within the bearing) increased with speed (fig. 7) for all three bearing clearances tested. The maximum heat transferred was 13.7 kW (790 Btu/min) at a shaft speed of 25 500 rpm at a total oil flow rate of 9.5×10^{-3} m^3/min (2.5 gal/min) for the one-piece-inner-ring bearing of reference 5. The power loss at 20 000 rpm was lower by a factor of about four when the test results of the large clearance bearings 3PM and 3PL and the tight-clearance bearing 1P were compared. The values for heat transfer (power loss within the bearing) were essentially identical at similar conditions for the three-piece-inner-ring bearings 3PL and 3PM over the range of oil flow rates to the inner ring tested (fig. 7).

The heat transfer to the lubricant increased with total oil flow rate to the inner ring (fig. 8). The rate of increase in heat transfer increased with shaft speed. From these figures it is apparent that a high-speed bearing should not always be operated at high oil flow rates since the power loss in the bearing can become extremely large. This is especially true for a tight clearance bearing as shown by the data (fig. 8) for the one-piece bearing of reference 5. However, these higher flow rates may be required to keep bearing temperatures within acceptable limits.

Outer-Ring Cooling

Effect on bearing temperature.—The effect of outer-ring cooling oil flow rate on bearing temperature for various oil flow rates to the inner ring is shown in figure 9 for shaft speeds of 20 000 and 25 500 rpm. Data for the one- and three-piece-inner-ring bearings with tight clearances are shown in the figure. Because of the relatively low operating temperature of the larger clearance bearings (3PM and 3PL, fig. 6), the highest of which was only 401 K (262° F) at the maximum 25 500 rpm operating speed, outer-ring cooling was not used in these tests. In figure 9, the bearing outer-ring temperature of both bearings 1P and 3PT decreased as flow to the outer ring increased for all conditions. The total oil flow rate to the inner ring was 3.8×10^{-3} to 9.5×10^{-3} m³/min (1.0 to 2.5 gal/min) and shaft speeds were 20 000 and 25 500 rpm. The inner-ring temperature of bearing 3PT decreased slightly as outer-ring cooling flow increased, whereas the inner-ring temperature of bearing 1P increased with increasing outer-ring cooling flow. This was the case for both 20 000 and 25 500 rpm. It is obvious from the data in figure 9 that the heat-transfer characteristics of a one-piece inner ring are apparently different than those of a three-piece inner ring. At the maximum speed of 25 500 rpm, bearing 3PT had an effective hot-mounted diametral clearance of −0.0095 mm (−0.0004 in.), whereas bearing 1P at identical test conditions had a clearance of 0.0239 min (0.0009 in.) (table III). Both bearings ran successfully to a maximum speed of 3.0 million DN.

At 25 500 rpm (3.0×10^{6} DN) and 5.7×10^{-3} m³/min (1.5 gal/min) total oil flow rate to the inner-ring of bearing 3PT (fig. 9(b)), the outer-ring temperature was 489 K (421° F) with no outer-ring cooling. When the outer ring was cooled with a cooling oil flow rate of 3.5×10^{-3} m³/min (0.92 gal/min), the outer-ring temperature dropped to 467 K (381° F), a ΔT of 22 K (40° F). This was the highest temperature change recorded. The inner ring remained at a relatively constant temperature of about 482 K (408° F) regardless of what outer-ring cooling flow rate was used.



(a) 20 000 rpm.
(b) 25 500 rpm.

Figure 9.—Effect of outer-ring cooling oil flow rate on bearing temperature for various total oil flow rates to inner ring. Oil-inlet temperature (lubricating and cooling), 366 K (200° F); radial load, 8900 N (2000 lb).

Cage Slip

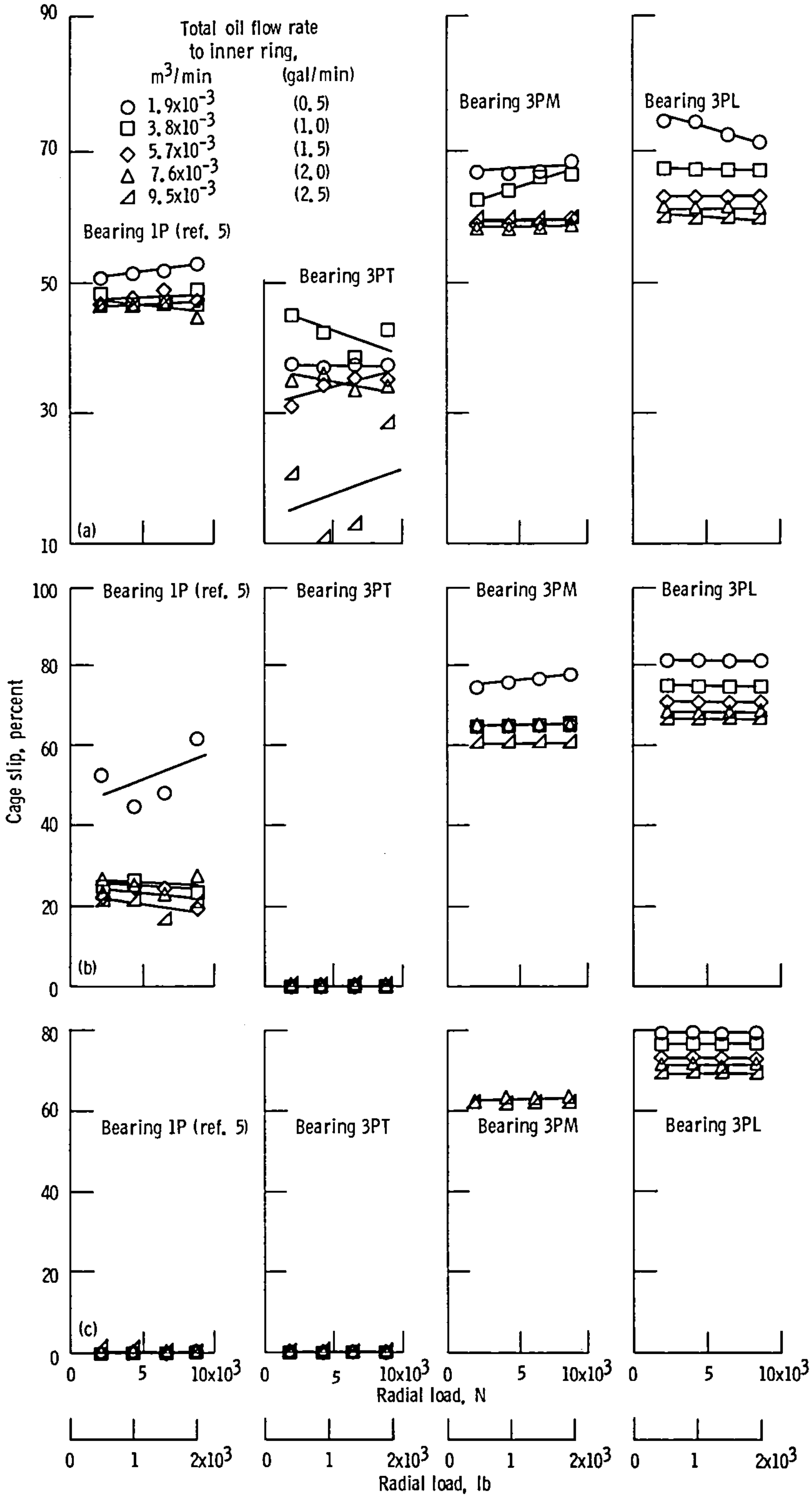
Effect of radial load.—The effect of radial load on cage slip is shown in figure 10 for various total oil flow rates to the inner ring. The outer ring was not directly oil cooled in these experiments. Cage slip expressed as a percentage is obtained from the following equation:

$$C_s = 100\left[1 - \frac{w_c}{w}(2 + 2S)\right]$$

where

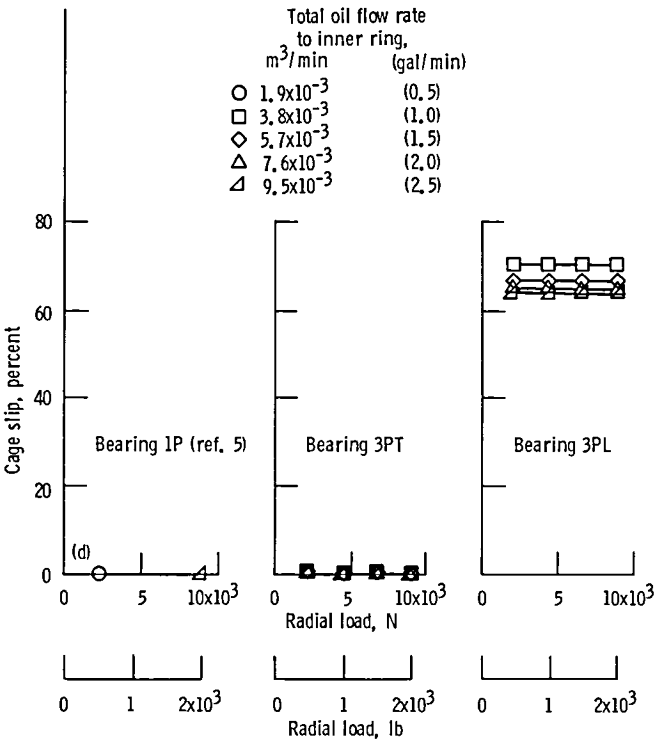
C_s	cage slip, percent
w_c	cage speed
w_s	shaft speed (inner-ring speed)
S	radius ratio, r/R
r	radius of roller
R	radius of inner-ring roller track (raceway)

For each size of roller bearing, there is a ratio of cage speed to shaft speed at which pure rolling occurs. Any



(a) Shaft speed, 10 000 rpm.
(b) Shaft speed, 15 000 rpm.
(c) Shaft speed, 20 000 rpm.

Figure 10.—Effect of radial load on cage slip for various total oil flow rates to inner ring. Oil-inlet temperature, 366 K (200° F).



(d) Shaft speed, 25 500 rpm.

Figure 10.—Concluded.

value below this ratio indicates roller slip (more commonly referred to as cage slip). The ratio for pure rolling R_{ns} is obtained from the following equation:

$$R_{ns}=\frac{1}{2+2S}$$

For the 118-mm-bore roller bearing, this ratio is 0.4564. Present opinion (ref. 9) is that a 10-percent cage slip is tolerable. Although this value has not been experimentally verified, it can be used as a basis for determining whether the experimental bearing may be approaching a condition that can lead to bearing failure.

Although there was some data scatter at the lower speeds (10 000 and 15 000 rpm), figure 10 shows that the percent cage slip was not appreciably affected by loads from 2200 to 8900 N (500 to 2000 lb) and shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. At 10 000 rpm (fig. 10(a)), the tight clearance bearings (1P and 3PT) showed somewhat lower percent cage slip than the larger clearance bearings (3PM and 3PL). At 15 000 rpm (fig. 10(b)), bearing 3PT, which had the tightest clearance, showed zero percent cage slip at all load and flow conditions. The cage slip of bearing 1P dropped from approximately 50 percent at 10 000 rpm to 25 percent at 15 000 rpm, except at the lowest total oil flow rate. The percent cage slip of the two large clearance bearings (3PM and 3PL) ranged from 60 to 80 percent over the 10 000 to 25 500 rpm speed range. The total oil flow was 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). At 20 000 and 25 500 rpm (figs. 10(c) and (d)) both the one- and three-piece-inner-ring bearings with the tight bearing clearances (1P and 3PT) showed zero percent cage slip.

Computer generated data from the SHABERTH program are shown in figure 11. Curves for a tight clearance bearing 3PT and a large clearance bearing 3PL are compared. The radial and centrifugal loads on these bearings are distributed among the rollers as shown in figure 11. The bowed portion of these curves represents the centrifugal and radial load shared by the rollers in Hertzian contact with both bearing rings. The data points at either end show the centrifugal load of the rollers on the outer ring of an otherwise unloaded roller. A minimum number of rollers (5 to 7) are in elastohydrodynamic Hertzian contact with both bearing

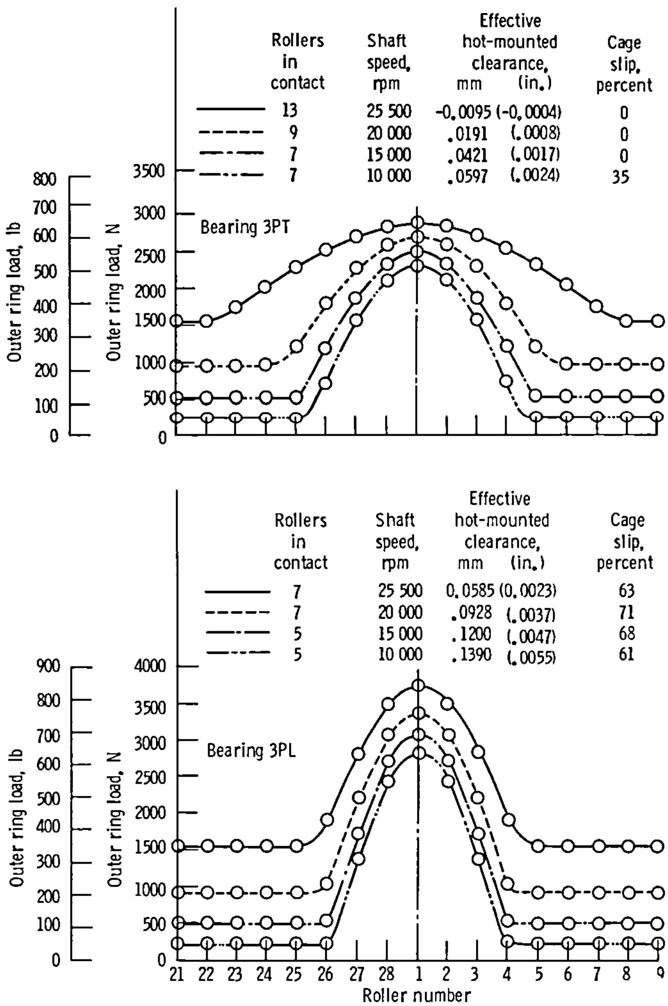


Figure 11.—Outer-ring loads generated from SHABERTH computer program for three-piece-inner-ring bearings 3PT and 3PL. Radial load, 8900 N (2000 lb); total oil flow, 7.6×10^{-3} m^3/min (2.00 gal/min).

rings at all speeds for bearing 3PL and at 10 000 rpm for bearing 3PT, minimizing traction forces which allow the rollers and cage to slip at a high rate. An exception was at 15 000 rpm for bearing 3PT where zero percent cage slip was recorded with 7 rollers in contact. The tight, effective, hot-mounted diametral clearance of 0.042 mm (0.0017 in.) at 15 000 rpm (table I) and resultant higher bearing temperature (45 K (80° F) higher than that at 10 000 rpm (fig. 6)) could have reduced the oil viscosity enough to allow the cage to increase in speed. This condition reduced cage slip to zero percent even with 7 rollers in contact. Percent cage slip is affected by the number of rollers in contact with the inner and outer rings (a function of bearing clearance) and the bearing operating temperature.

Effect of lubricant flow rate.—The effect of lubricant flow rate on percent cage slip for the one- and three-piece-inner-ring roller bearings is shown in figure 12. In all cases where sufficient clearance was present to allow slip to occur, the percent cage slip decreased (cage speed increased) with an increase in lubricant flow rate. An explanation for this might be that increasing the flow rate cools the oil within the bearing with resultant increased viscosity of the oil between the bearing lands and cage mating surfaces. The more viscous oil increases the traction force driving the cage to higher speeds (decreasing cage slip). The amount of decrease in cage slip is dependent on the operating speed of the bearing whose temperature is a function of bearing clearance. Because bearing running temperatures are low at high bearing clearance, increases in oil flow rates have little effect on oil viscosity within the bearing, and cage slip decreases only slightly with increased oil flow rates. Conversely, because tight-clearance bearings (1P and 3PT) operate at higher heat levels than do the large-clearance bearings (3PM and 3PL) (fig. 6), increases in oil flow rate have a more pronounced effect on oil viscosity and show greater reductions in cage slip. For test conditions resulting in bearing temperatures under about 400 K (260° F) (fig. 6), the bearing clearances remained virtually unchanged with increases in oil flow, as indicated in figure 12. Therefore, clearance change was not the cause of the decrease in cage slip with increased oil flow.

Effect of shaft speed.—Figure 13 shows how cage slip varied with shaft speed for a one-piece and for three, three-piece-inner-ring bearing configurations at different bearing clearances. For the two bearings with the tight clearances (namely, 1P and 3PT), the percent cage slip decreased as shaft speed increased and reached zero percent beyond shaft speeds of 20 000 and 15 000 rpm, respectively. For speeds above 20 000 rpm the lubricant

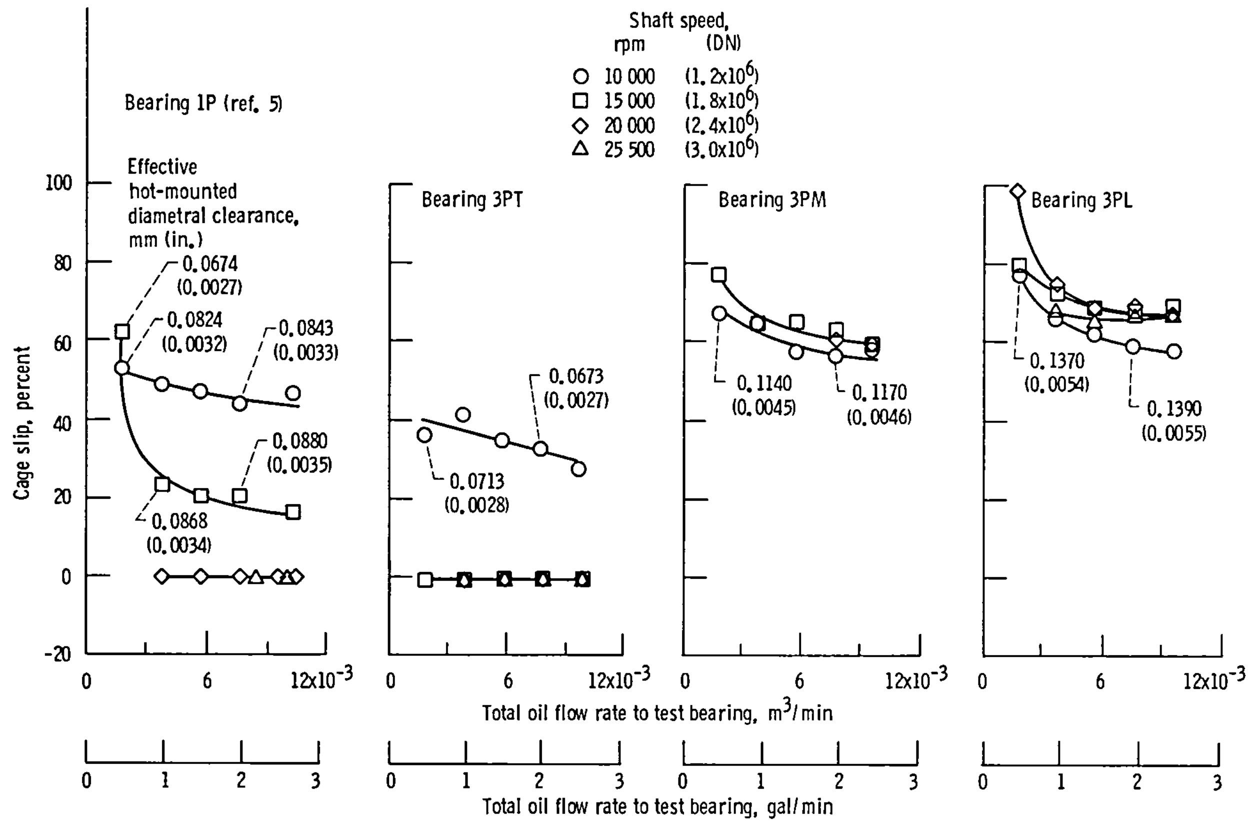


Figure 12.—Effect of total oil flow rate to test bearing on cage slip at four different shaft speeds. Radial load, 8900 N (2000 lb); oil-inlet temperature, 366 K (200° F).

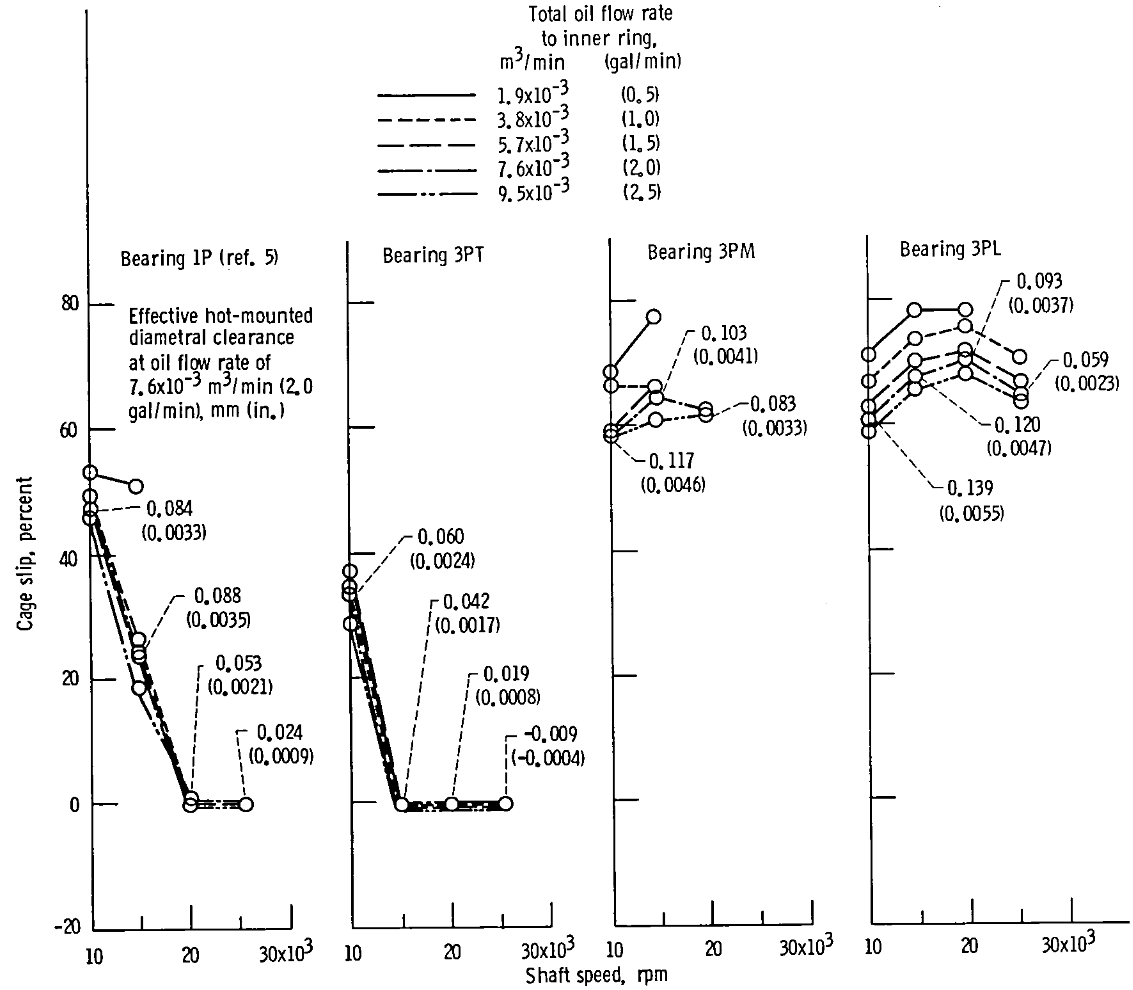


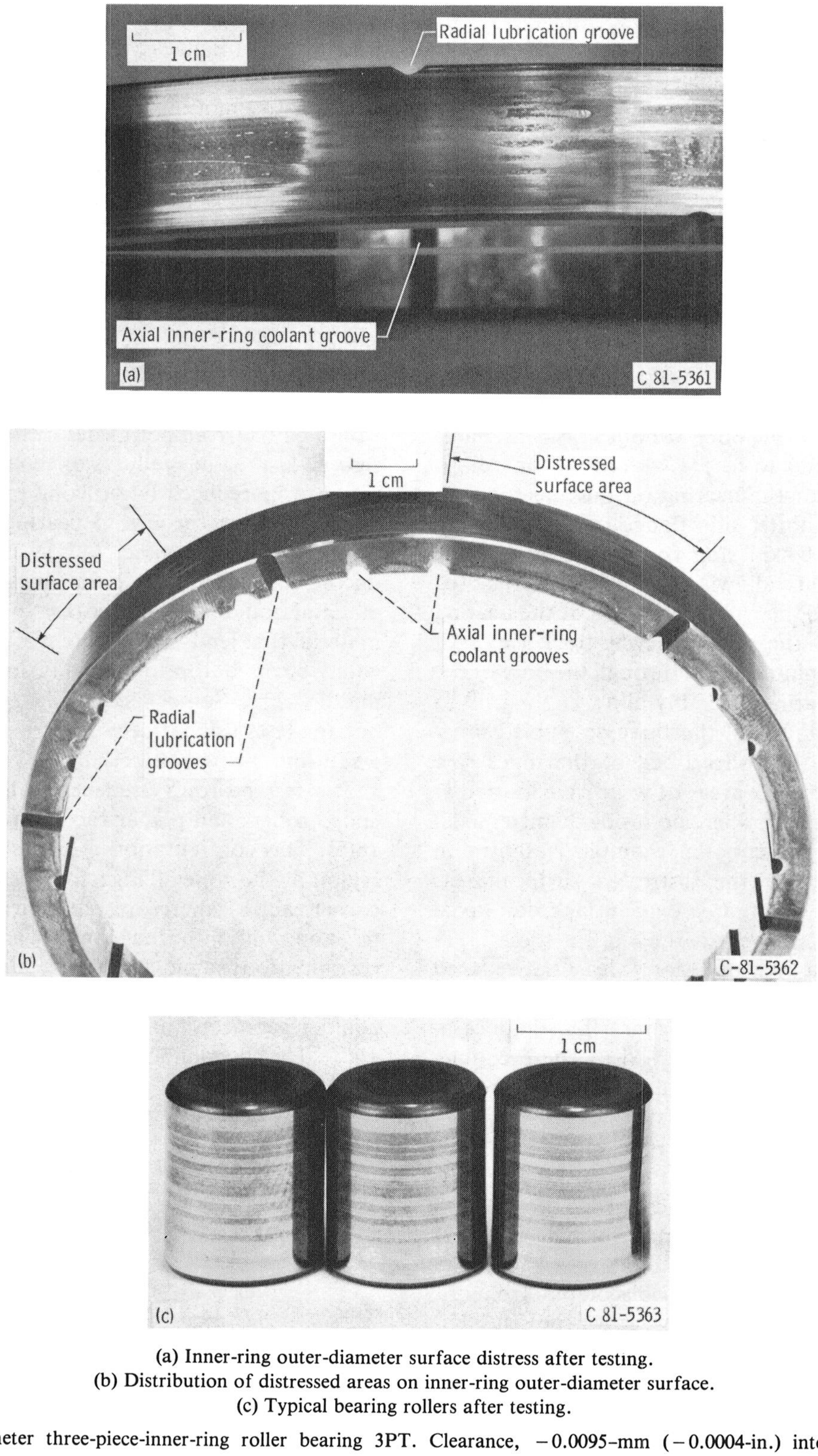
Figure 13.—Effect of shaft speed on cage slip at five different total lubricant flow rates. Radial load, 8900 N (2000 lb); oil-inlet temperature, 366 K (200° F).

flow rate for successful bearing operation was limited to oil flows above 5.7×10^{-3} m³/min (1.5 gal/min) for both bearings. For the bearing with the largest clearance (3PL, fig. 13), percent cage slip increased with shaft speed at 20 000 rpm and then decreased at the maximum speed of 25 500 rpm for all flow rates tested. The lowest flow rate of 1.9×10^{-3} m³/min (0.5 gal/min) was too low for successful 25 500-rpm speed operation.

Effect of clearance.—Some of the data points in figure 13 have calculated bearing clearances labelled for specific shaft speeds. The data show that zero percent cage slip can be obtained at tight clearances without external preload. This was true for bearing 1P at 20 000 rpm where the clearance was 0.053 mm (0.0021 in.) and for bearing 3PT at 15 000 and 20 000 rpm where the clearances were 0.042 and 0.019 mm (0.0017 and 0.0008 in.), respectively. The effect of clearance on cage slip is best illustrated by comparing the data of two bearings in figure 13 at the same speed, but at different clearance values. Bearing 3PT at 15 000 rpm had zero percent cage slip at a relatively small clearance of 0.042 mm (0.0017 in.) and an oil flow rate of 7.6×10^{-3} m³/min (2.0 gal/min); bearing 3PL at the same speed and flow rate indicated a cage slip value of 67 percent at a large clearance of 0.120 mm (0.0047 in.). The amount of bearing clearance affects the percent cage slip, and preload is not necessarily a prerequisite for zero percent cage slip.

Bearing Condition After Testing

Because of the failure of the test-bearing oil-out thermocouple when testing bearing 3PT, a second test run was attempted after repairing the defective thermocouple. At 25 500 rpm bearing tests were terminated because of excessively high bearing-inner-ring temperatures and bearing noise. The bearing was disassembled and inspected. Figure 14 shows the results



(a) Inner-ring outer-diameter surface distress after testing.
(b) Distribution of distressed areas on inner-ring outer-diameter surface.
(c) Typical bearing rollers after testing.

Figure 14.—118-Millimeter three-piece-inner-ring roller bearing 3PT. Clearance, −0.0095-mm (−0.0004-in.) interference (preload) at 25 500 rpm.

of this inspection. Figure 14(a) shows the area of maximum distress on the outside diameter (track) of the inner ring. This area shows a series of short, circumferential lines that appear to be the result of plowing the surface metal as the rollers passed over it. There were six similar wear areas, although less severe than that shown, spaced unevenly around the circumference of the track. Each area was about 25 mm (1 in.) long. The uneven distribution of wear areas can be explained by studying the geometry of the inner-ring axial grooving.

Figure 3(b) shows the oil-inlet and -outlet side flange rings and the roller raceway ring. The oil-inlet flange has 30 equally spaced through-axial grooves on its inside surface plus a keyway that does not extend the full width of the ring. The roller raceway ring contains 22 axial slots that allow oil to flow axially under the raceway ring. The sections in the raceway ring bore without slots form dams to permit 8 radial paths to be fed with lubricating oil to the rolling elements of the bearing on the inlet side. In like manner, the oil-outlet side flange ring contains 14 axial slots permitting 8 axial slots from the roller raceway ring to be dammed, thus allowing 8 radial paths to be fed with lubricating oil to the rolling elements of the bearing on the outlet side of the roller raceway track ring. All axial slots that pass uninterrupted through the bore of the three rings of the bearing assembly allow cooling oil to flow through them and cool the three assembled inner rings. Inspection of the roller track of the inner ring showed that each of the six areas of wear were located at a portion of the inner ring where no inside diameter axial cooling groove was present. An example is shown in figure 14(b). (Note that the distressed surface areas appear directly above an area which lacks an axial groove.) The heat generated in the ring in these areas during operation is not removed, or at least not reduced by any axial cooling oil flow. The wall of the inner ring expands at these uncooled areas more than it does at areas where axial cooling grooves exist. These cooling grooves create decreased clearance areas where the rollers are momentarily squeezed excessively.

The bearing clearance at 25 500 rpm was calculated to be -0.0095 mm (-0.0004 in.) (table I). Under these extreme stress conditions the lubricant film broke down and metal to metal contact occurred, forming the wear and plowed areas shown in figure 14(a).

The outer-ring roller track was discolored from the heat generated but no discernable wear was present. The rollers (fig. 14(c)) showed burnish marks corresponding to the ploughed surface marks of figure 14(a), but no discernable wear. The cage also showed no measurable wear in the pockets or on the land-guided inside-diameter areas. The silver plating was not worn through.

Concluding Remarks

The test-bearing inner-ring assembly used in this investigation consisted of three separate rings, two of which formed flanges along the sides of the third ring on which the bearing rollers rode. The rings were restrained from relative motion among themselves during operation by a key in an axial keyway machined in the bore of each piece. The keyway plus the axial lubrication grooves in the bore of the rings make the radially loaded middle ring especially susceptible to cracking. This design, however, was necessary since the lubricant was introduced to only one side of the bearing. A preferred design would be one in which the middle radially loaded ring is a cylinder without any axial grooves or radial holes to weaken it. The adjacent flange rings contain the necessary grooving and holes for bearing lubrication. In this case the oil would be fed from both sides of the bearing as shown in figure 15. Fracture failures of the radially loaded middle ring can be reduced by utilizing case-carburized bearing raceways. Case-carburized bearings have hard surfaces for good rolling contact fatigue life and relatively soft ductile cores for fracture toughness. AISI M-50 bearing material could still be used for the flanges and another material that lends itself more readily to case-carburizing could be used for the radially loaded, highly stressed middle ring. Some success was reported (ref. 11) in bearing tests using a case-carburized alloy (CBS 600) for a split-inner-ring ball bearing.

The test bearings used herein had conventional flat-ended rollers and planar race flanges as shown in figure 16(a). This configuration provides no lubricant entrance region at the roller-flange contact (other than the roller corner radius); therefore, the roller has very little ability to pump oil into the contact area. In order for an elastohydrodynamic contact (which functions most efficiently in pure rolling) to be present in the bearing, the contact geometry may be modified as shown in figure 16(b). The coned inboard surface of the guide flanges can

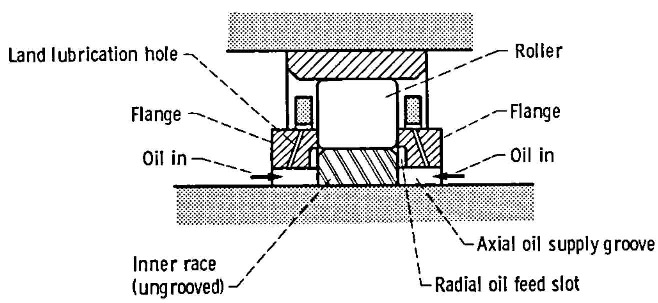
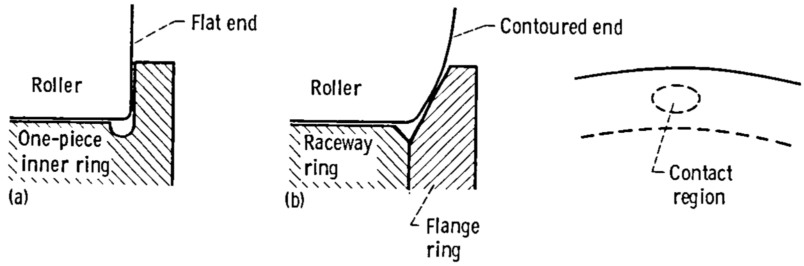


Figure 15.—Three-piece-inner-ring bearing with inner race of ungrooved geometry.



(a) Conventional.
(b) Modified configuration.

Figure 16.— Roller end-contact geometry of one- and three-piece inner-ring bearings.

be honed, and the accuracy of cone angle maintained, if the flange pieces are made separately as part of a three-piece-inner-ring assembly. This design also makes it possible for the roller bearing to accept some thrust loading.

Summary of Results

Parametric tests were conducted in a high-speed bearing tester on three 118-mm-bore roller bearings each having a three-piece inner ring. The bearings had different diametral clearances but were otherwise identical. They were manufactured from consumable-electrode-vacuum-melted (CVM) AISI M-50 steel with a room-temperature Rockwell C hardness of 60. Test parameters were radial loads of 2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 9.5×10^{-3} m^3/min (0.5 to 2.5 gal/min). Outer-ring cooling flow rates were 0 to 3.22×10^{-3} m^3/min (0 to 0.85 gal/min) at an oil inlet temperature of 366 K (200° F). The results of this study were compared with those of a previously tested similar 118-mm-bore roller bearing having a one-piece inner ring. The lubricant was a neopentylpolyol (tetra) ester that met the MIL–L–23699 specification. The following results were obtained:

(1) A 118-mm-bore test bearing with a three-piece inner ring ran successfully at 3.0×10^6 DN with a radial load of 8900 N (2000 lb) for 20 hr.

(2) Bearing temperature decreased with increased clearance and oil flow rate, and increased with shaft speed.

(3) The difference between the outer- and inner-ring temperatures was small for a three-piece-inner-ring bearing, without outer-ring cooling, when compared with that of a one-piece-inner-ring bearing, indicating a difference in bearing heat-transfer characteristics of the two configurations.

(4) Power loss within the bearing increased with both speed and total oil flow rates to the inner ring. The rate of increase in power loss increased with shaft speed.

(5) Outer-ring temperature of a three-piece-inner-ring bearing at 25 500 rpm was 22 K (40° F) cooler with outer-ring cooling than it was without outer-ring cooling, whereas the inner-ring temperature remained essentially constant.

(6) Cage slip was greatly reduced or even eliminated by using a bearing with a very tight clearance at operating speed. Cage slip decreased with increasing total lubricant flow rate.

(7) The three-piece-inner-ring test bearings with the two largest clearances showed no visible wear after testing. The bearing with the tightest clearance showed surface distress in the track area of the inner ring due to high Hertzian stresses generated. The outer-ring roller track was discolored from the excessive heat generated but showed no measurable wear.

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15. Supplementary Notes

16. Abstract

A 118-mm-bore roller bearing with a three-piece inner ring ran successfully at 3.0×10^6 DN for 20 hr. Provisions were made for lubrication and cooling through the inner ring. In some tests the outer ring was also cooled. Power loss within the bearing increased with both speed and total oil flow rate to the inner ring. Outer ring temperature decreased by as much as 22 K (40° F) when outer ring cooling was employed whereas inner ring temperature remained essentially constant. Cage slip was greatly reduced or even eliminated by using a bearing with a very tight clearance at operating speed. A three-piece-inner-ring bearing had higher inner-ring temperatures and less temperature difference between the inner and outer rings than a conventional one-piece-inner-ring bearing.

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