NASA TECHNICAL NOTE

NASA TN D-7837

NASA TN D-7837

OPERATING CHARACTERISTICS OF 120-MILLIMETER-BORE BALL BEARINGS AT 3×10^6 DN

by Erwin V. Zaretsky, Eric N. Bamberger, and Hans Signer

Lewis Research Center
Cleveland, Ohio 44135

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • NOVEMBER 1974

1. Report No. NASA TN D-7837	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle OPERATING CHARACTERISTICS OF 120-MILLIMETER-BORE BALL BEARINGS AT 3×10^6 DN		5. Report Date November 1974
		6. Performing Organization Code
7. Author(s) Erwin V. Zaretsky, Lewis Research Center; Eric N. Bamberger, General Electric Co., Cincinnati, Ohio; and Hans Signer, Industrial Tectonics, Inc., Compton, California		8. Performing Organization Report No. E-7875
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		10. Work Unit No. 501-24
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		13. Type of Report and Period Covered Technical Note
		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

A parametric study was performed with split inner-race 120-mm-bore angular-contact ball bearings at a speed of 25 000 rpm (3×10^6 DN) at initial contact angles of 20° and 24°. Provisions were made for outer- and inner-race cooling and for injection of lubricant into the bearing through a number of radial holes in the split inner-race of the bearing. Oil flow and coolant rate to the bearing was controlled and varied for a total flow up to approximately 12×10^{-3} m^3/min (3.2 gal/min). Bearing temperature was found to decrease as the total lubricant flow to the bearing increased. However, at intermediate flow rates temperature began to increase with increasing flow. Power consumption increased with increasing flow rate. Bearing operating temperature, differences in temperatures between the inner and outer races, and bearing power consumption can be tuned to any desirable operating requirement. Cage speed increased by not more than 2 percent with increasing oil flow to the inner race.

17. Key Words (Suggested by Author(s)) High-speed bearings; Rolling element bearings; High-temperature lubrication; Heat transfer; Angular-contact ball bearings		18. Distribution Statement Unclassified - unlimited STAR category 15	
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 30	22. Price* $3.25

* For sale by the National Technical Information Service, Springfield, Virginia 22151

OPERATING CHARACTERISTICS OF 120-MILLIMETER-BORE BALL BEARINGS AT 3×10^6 DN

by Erwin V. Zaretsky, Eric N. Bamberger,* and Hans Signer†

Lewis Research Center

SUMMARY

A parametric study was performed with split inner-race 120-millimeter-bore angular-contact ball bearings optimally designed for 25 000 rpm (3×10^6 DN) operation. Bearings having nominal contact angles of 20° and 24° were tested. Provisions were made for outer- and inner-race cooling and for injection of lubricant into the bearing through a number of radial holes contained in the split inner-race of the bearing. Test conditions included a thrust load of 22 241 newtons (5000 lb), a speed of 25 000 rpm (3×10^6 DN), and an oil inlet temperature of 394 K (250° F).

Bearing inner-race temperature was found to decrease as the total lubricant flow to the inner race was increased. However, at an intermediate flow rate temperature began to increase with increasing flow. Outer-race temperatures also decrease with increasing total lubricant flow to the inner race and, in general, paralleled those of the inner race at lower temperatures. The magnitude of the outer-race temperature was a function of the outer-race cooling flow.

Bearing power consumption was a function of total lubricant flow to the bearing. As flow rate was increased, power consumption increased. Bearing operating temperature, differences in temperature between the inner and outer races, and bearing power consumption can be tuned to any desirable operating requirement by varying four parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race, and oil inlet temperature.

Ball orbital speed was found to increase with increased oil flow to the inner race. However, the increase for a given bearing was not more than 2 percent over the entire range of flow rates.

*General Electric Co., Cincinnati, Ohio.

†Industrial Tectonics, Inc., Compton, California.

INTRODUCTION

Advanced air breathing engines for high-speed aircraft for the 1980's are expected to operate with bearing temperatures near 492 K (425° F) and at speeds approaching 25 000 rpm (3×10^{6} DN). (DN is a bearing speed parameter and is equal to the product of the bearing bore in millimeters and the shaft speed in rpm.) In support of these engines, as well as for similar high performance oriented bearing applications, a reliable bearing-lubricant system is required. Such a system requires essentially three key items. These are, a suitable lubricant, a reliable bearing structural material, and an optimized bearing design coupled with the proper lubricant cooling flows needed to sustain ultrahigh speeds.

Over the past decade several new classes of lubricants were developed and evaluated, which extended the upper temperature range of lubricating fluids (refs. 1 to 4). Of these, the polyester and tetraester fluids have proven to be most useful and applicable in typical air breathing engine environments, and consequently have been widely accepted in current commercial and military aircraft applications (ref. 5). These fluids have good thermal stability at temperatures to 505 K (450° F). Bearing life at 492 K (425° F) with the tetraesters exceeded AFBMA-predicted (catalogue) life by a factor in excess of four (ref. 3).

With the tetraester lubricant and the AISI M-50 steel, two of the three key elements essential to successful large-diameter, high-load, and ultrahigh-speed bearing operation are now specified. However, high-speed bearing operation in the range of 25 000 rpm (3×10^{6} DN) requires more than the proper lubricant and bearing material. Heat generation within the bearing itself is extremely critical, as is component loading due to centrifugal effects. Jones (refs. 6 to 8) first considered speed effects on bearing life and dynamics without considering the effect of the lubricant. Subsequently, Harris (refs. 9 and 10) expanded these bearing analyses including lubricant effects.

Another problem of operating bearings at high speed is the need to adequately cool the bearing components because of excessive heat generation. A method which has been used successfully at bearing speeds to 25 000 rpm (3×10^{6} DN) is cooling lubricant applied under the inner race (refs. 11 and 12). In this method lubricant is centrifugally injected through the split inner race and shoulders of an angular-contact ball bearing by means of a number of rows of radial holes. As a result, both the cooling and lubricating function is accomplished.

The research reported herein was undertaken to investigate the performance of optimally designed 120-millimeter-bore angular-contact ball bearings at a speed of 25 000 rpm (3×10^{6} DN). The primary objective was to determine the operating characteristics of this bearing under varying cooling and lubricant flow rates utilizing inner- and outer-race cooling and inner-race injected lubrication. Split inner-race 120-millimeter-bore angular-contact ball bearings optimally designed for 3×10^{6} DN operation having nominal contact angles of 20° and 24° were used for testing. Provisions were made for outer- and inner-race cooling and for injection of lubricant into the bearing through a number of radial holes contained in the split inner race of the bearing. Test conditions included a thrust load of 22 241 newtons (5000 lb), a speed of 25 000 rpm (3×10^{6} DN) and an oil inlet temperature of 394 K (250° F). Oil flow and coolant rate were controlled and varied to the bearing inner and outer races. Measurements were made of power consumption, oil outlet temperatures, inner- and outer-race temperatures, and cage speed.

APPARATUS, MATERIALS, AND PROCEDURE

High-Speed Bearing Tester

A section view of the high-speed, high-temperature bearing tester used in these tests is shown in figure 1. This tester is described in detail in references 1 and 2 and has been subsequently modified to operate at speeds of 25 000 rpm (3×10^{6} DN). The tester consists of a shaft to which two test bearings are attached. Loading is supplied through a system of ten springs which apply a thrust load to the bearings. Dual flat belts drive the test spindle from a 75-kilowatt (100-hp) fixed speed electric motor. The drive motor is mounted to an adjustable base, so that drive pulleys for 12 000 to

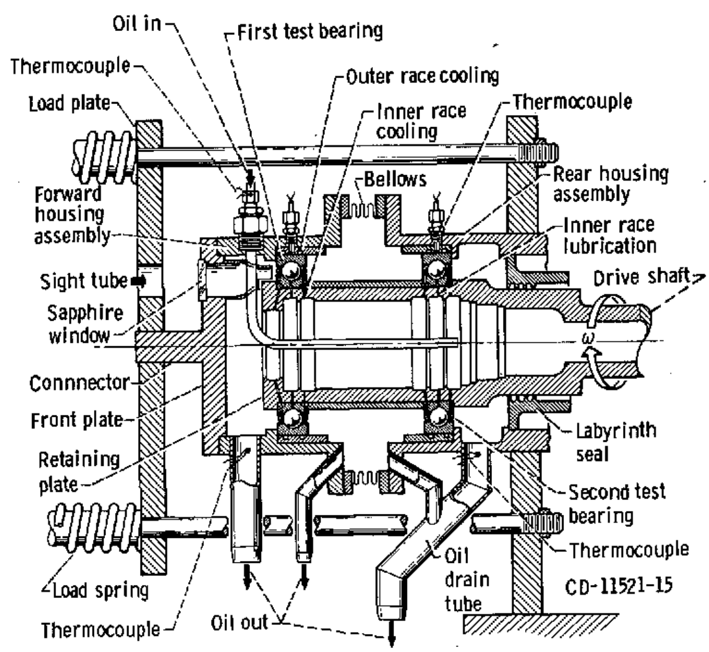


Figure 1. - High-speed, high-temperature bearing test apparatus.

25 000 rpm can be used with the same drive belts. The drive motor is controlled by a reduced voltage auto-transformer starter which permits a selection of the motor acceleration rate during startup. This control protects the bearings from undesirable acceleration during startup.

The lubrication system of the test rig delivers up to 2.8×10^{-2} cubic meter per minute (7.5 gal/min). There are three lubricant loops in the system. The oil flow in each loop is metered by adjustable flow control valves and can be individually measured by a flow rate indicator without interruption to the machine operation. Two of these loops are shown schematically in figure 2. The first of these loops supplies cooling oil to the

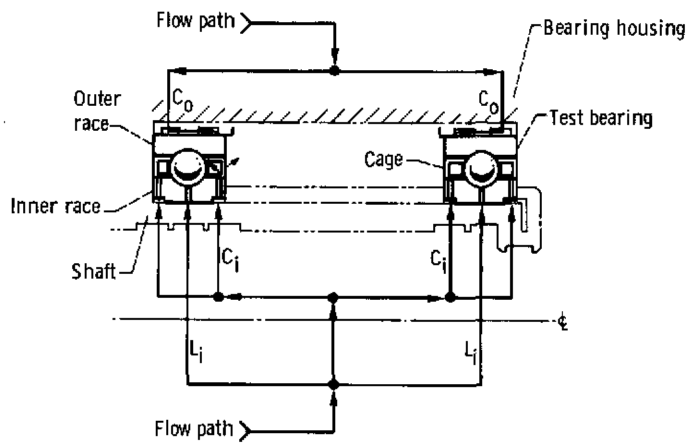


Figure 2. - Lubricant system for test bearings.

test bearing outer race and is designated C_o. The second loop is divided by a lubricant manifold which feeds individual annular grooves or channels at the shaft internal diameter proportioning the amount of oil which is to lubricate and/or cool each bearing inner race. The symbol L_i designates the oil flow to the bearing through a plurality of radial holes in the center of the split inner race, and C_i designates the lubricant utilized to cool the bearing inner race and lubricate the contact of the cage with the race land through a plurality of radial holes in the inner-race shoulder. The lubricant system permits a selection of various lubricant schemes, including bearing lubrication through the inner-race split, lubrication of the cage-race shoulder contact region, the application of inner- and/or outer-race cooling, and a selection of any desired flow ratio for cooling and lubrication as well as the conventional lubrication through jets. The third lubricant loop is fed into the slave bearing which supports the shaft (not shown in figs. 1 and 2). By the system of valves, orifices, and manifolds previously discussed, a large number of combinations of oil flows can be achieved to evaluate various conditions.

The machine instrumentation includes protective circuits which shut down a test when a bearing failure occurs, or if any of the test parameters deviate from the programmed conditions. Measurements were made of bearing inner-race speed, bearing cage speed, test spindle excursion, oil flow, test bearing inner- and outer-race and lubricant temperatures, and machine vibration level. The speed and spindle excursion measurements were made with proximity probes and displayed by numerical readout and oscilloscope, respectively. The oil flow was measured by a flowmeter, and bearing outer-race and lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded by a strip chart recorder. The inner-race temperature of the front test bearing was measured with an infrared pyrometer.

Test Bearings

The test bearings were ABEC-5 grade, split inner-race 120-millimeter-bore ball bearings. The inner and outer races, as well as the balls, were manufactured from one heat of double vacuum-melted (vacuum-induction melted consumable electrode vacuum remelted) AISI M-50 steel. The chemical analysis of the particular heat is shown in table I. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15 balls, 2.0638 centimeters (13/16 in.) in diameter. The cage was a one piece inner-land riding type, made out of an iron base alloy (AMS 6415) heat treated to a Rockwell C hardness range of 28 to 35 and having a 0.005-

TABLE I. - CHEMICAL ANALYSIS OF VACUUM-INDUCTION MELTED, CONSUMABLE-ELECTRODE VACUUM REMELTED AISI M-50 BEARING STEEL

Element	Composition of races and balls, wt. %
Carbon	0.83
Manganese	.29
Phosphorus	.007
Sulfur	.005
Silicon	.25
Chromium	4.11
Molybdenum	4.32
Vanadium	.98
Iron	Balance

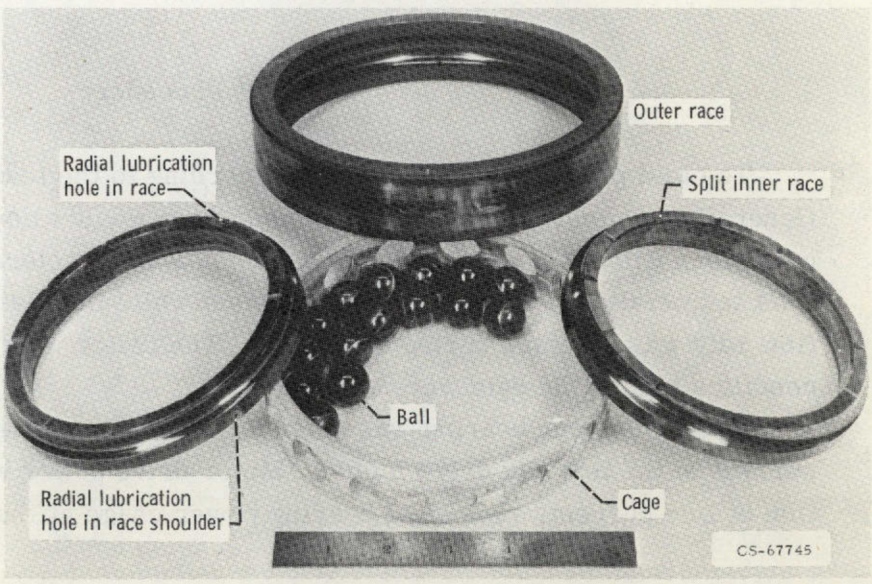


Figure 3. - Unfailed 120-millimeter-bore angular-contact high-speed test ball bearing. Running time, 1000 hours; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); temperature, 492 K (425^0 F).

centimeter (0.002-in.) maximum thickness of silver plate (AMS 2410). The cage balance was 3 gram-centimeters (0.042 oz-in.).

The retained austenite content of the ball and race material was less than 3 percent. The inner- and outer-race curvatures were 54 and 52 percent, respectively. All components with the exception of the cage were matched within ±1 Rockwell C point. This matching assured a nominal differential hardness in all bearings (i.e., the ball hardness minus the race hardness, commonly called ΔH) of zero (ref. 13). Surface finish of the balls was 2.5 microcentimeters (1 μin.) AA and the inner and outer raceways were held to a 5 microcentimeters (2 μin.) AA maximum surface finish.

A photograph of the test bearing is shown in figure 3. The bearing design permitted under-race lubrication by virtue of radial slots machined into the halves of the split inner races. It had been shown in references 11 and 12 that this was the most reliable technique for lubricating high-speed bearings. Provision was also made for inner-race land to cage lubrication, by the incorporation of several small diameter holes radiating from the bore of the inner race to the center of the inner shoulder.

Lubricant

The oil used for these studies was a 5-centistoke neopentylpolyol (tetra) ester. This is a Type II oil, qualified to MIL-L-23699 as well as to the oil specifications of most major aircraft-engine producers. The major properties of the oil are presented in table II. A temperature-viscosity curve is shown in figure 4.

Test Procedure

The test procedure was adjusted according to the test conditions to be evaluated. Generally, a program cycle was defined which would allow the evaluation of a number of conditions without a major interruption. Test parameters such as load, speed, and oil inlet temperature were maintained constant while the tester was in operation. Lubricant flow rate was adjusted during operation. The tester was allowed to reach an equilibrium condition before the data were recorded.

Power loss per bearing was determined by measuring line to line voltage and line current to the test-rig drive motor. Motor drive power was then calculated as a function of line current, reflecting bearing power usage at the various operating speeds.

TABLE II. - PROPERTIES OF

TETRAESTER LUBRICANT

Additives	Antiwear oxidation inhibitor antifoam
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flash point, K ($^\circ$F)	533 (500)
Fire point, K ($^\circ$F)	Unknown
Autoignition temperature, K ($^\circ$F)	694 (800)
Pour point, K ($^\circ$F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt. %	3.2
Specific heat at 477 K (400° F), J/(kg)(K) (Btu/(lb)($^\circ$F))	2340 (0.54)
Thermal conductivity at 477 K (400° F), J/(m)(sec)(K) (Btu/(hr)(ft)($^\circ$F))	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

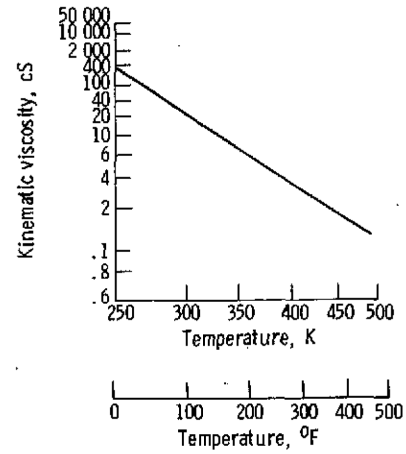


Figure 4. - Viscosity as a function of temperature for tetraester (Type II) lubricant.

RESULTS AND DISCUSSION

Effect of Lubricant Flow on Bearing Temperature

The effect of lubricant flow into the bearing and the cooling flow to the inner and outer races was determined. The test bearings were operated at a speed of 25 000 rpm (3×10^6 DN) and a thrust load of 22 241 newtons (5000 lb). This condition was chosen on the basis of approaching maximum Hertzian stresses [2.07×10^9 N/m^2 ($\sim$300 000 psi)] and speeds which can reasonably be anticipated in advanced state-of-the-art turbojet engines. Outer-race cooling flow C_o was 9.5×10^{-4}, 1.9×10^{-3}, 3.8×10^{-3}, and 5.7×10^{-3} cubic meter per minute (0.25, 0.5, 1.0, and 1.5 gal/min). Inner-race lubricant flow L_i ranged from 7.6×10^{-4} cubic meter per minute (0.2 gal/min) to approximately 5.7×10^{-3} cubic meter per minute (1.5 gal/min). Inner-race cooling C_i was varied as a function of L_i.

The data for these inner- and outer-race temperatures are summarized in figures 5 to 9. Referring to figure 5(e), a summary of test results are shown for the 20° contact-

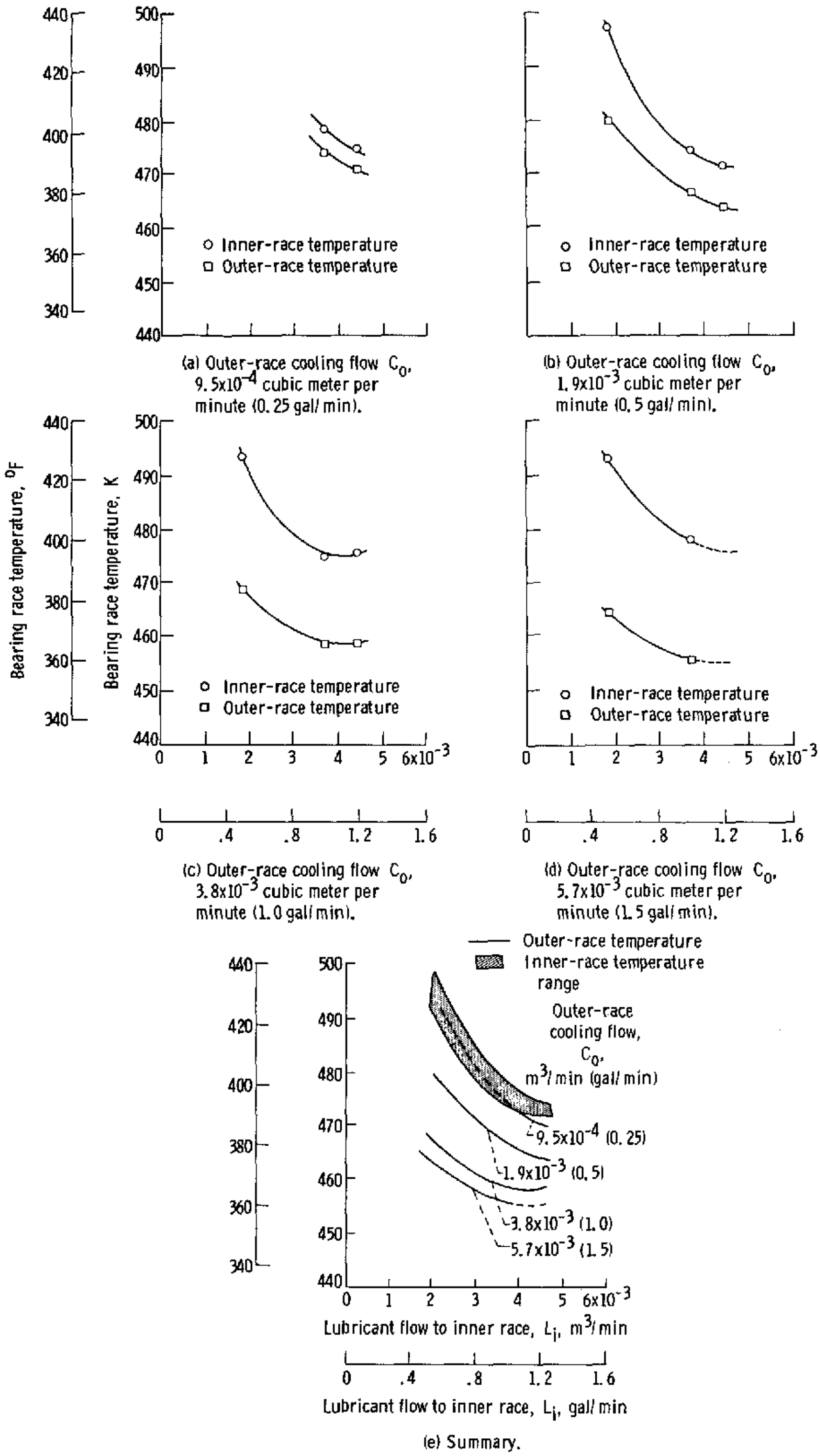


Figure 5. - Bearing race temperature as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_0. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 20°; inner-race cooling flow C_i, 0.

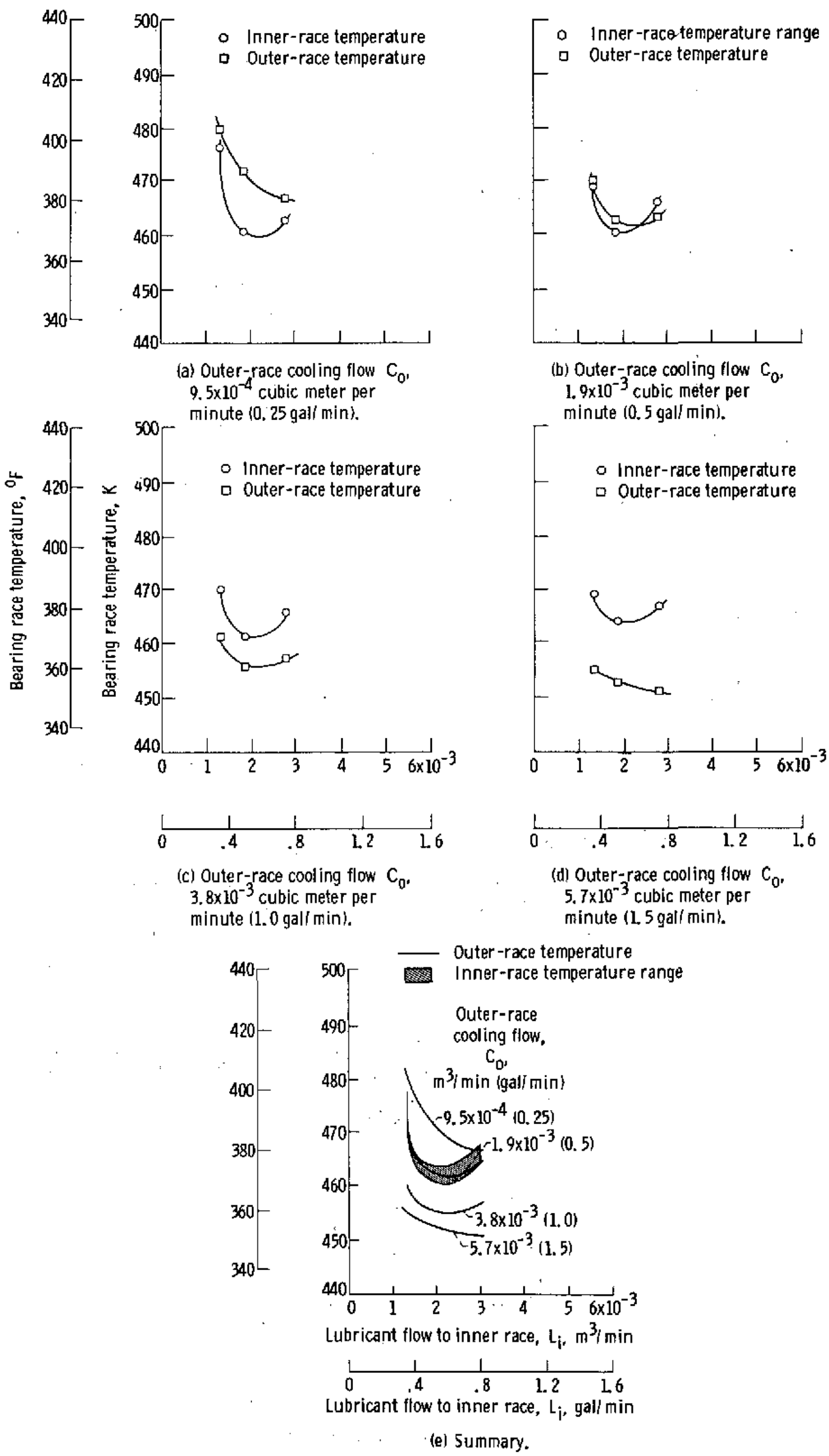


Figure 6. - Bearing race temperature as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 20°; inner-race cooling flow C_i, 1.33 L_i.

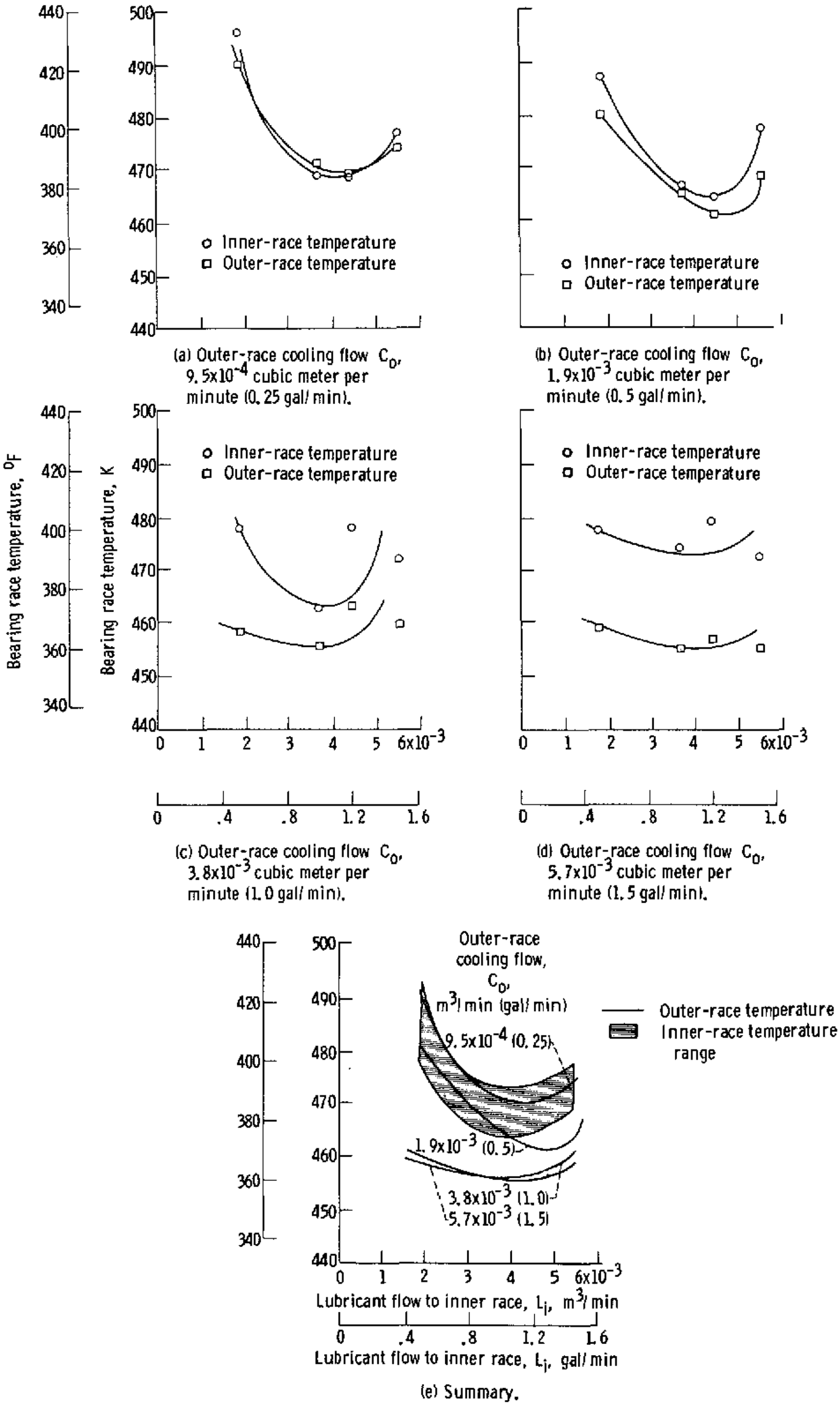


Figure 7. - Bearing race temperature as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 0.

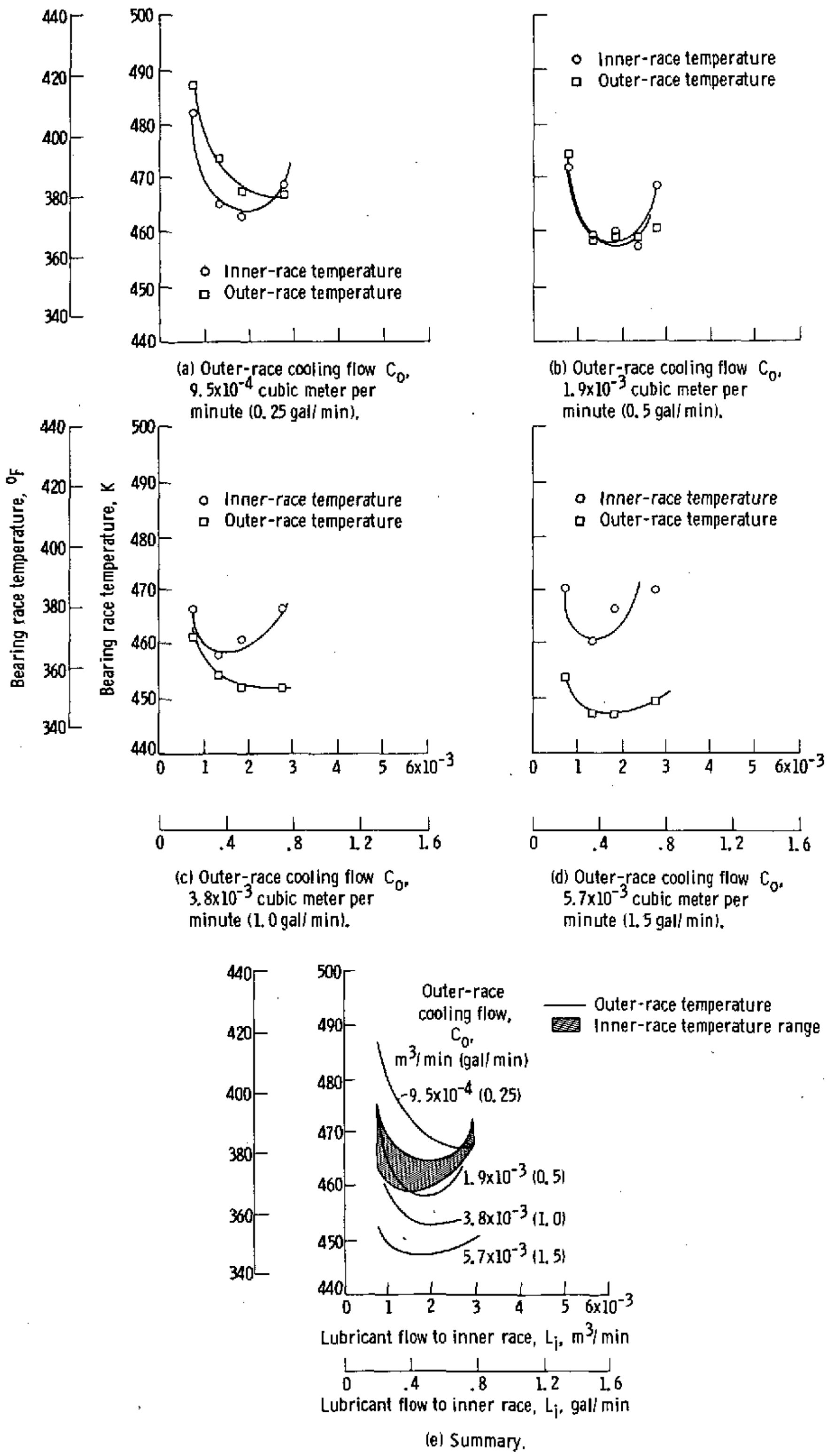


Figure 8. - Bearing race temperature as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 1.33 L_i.

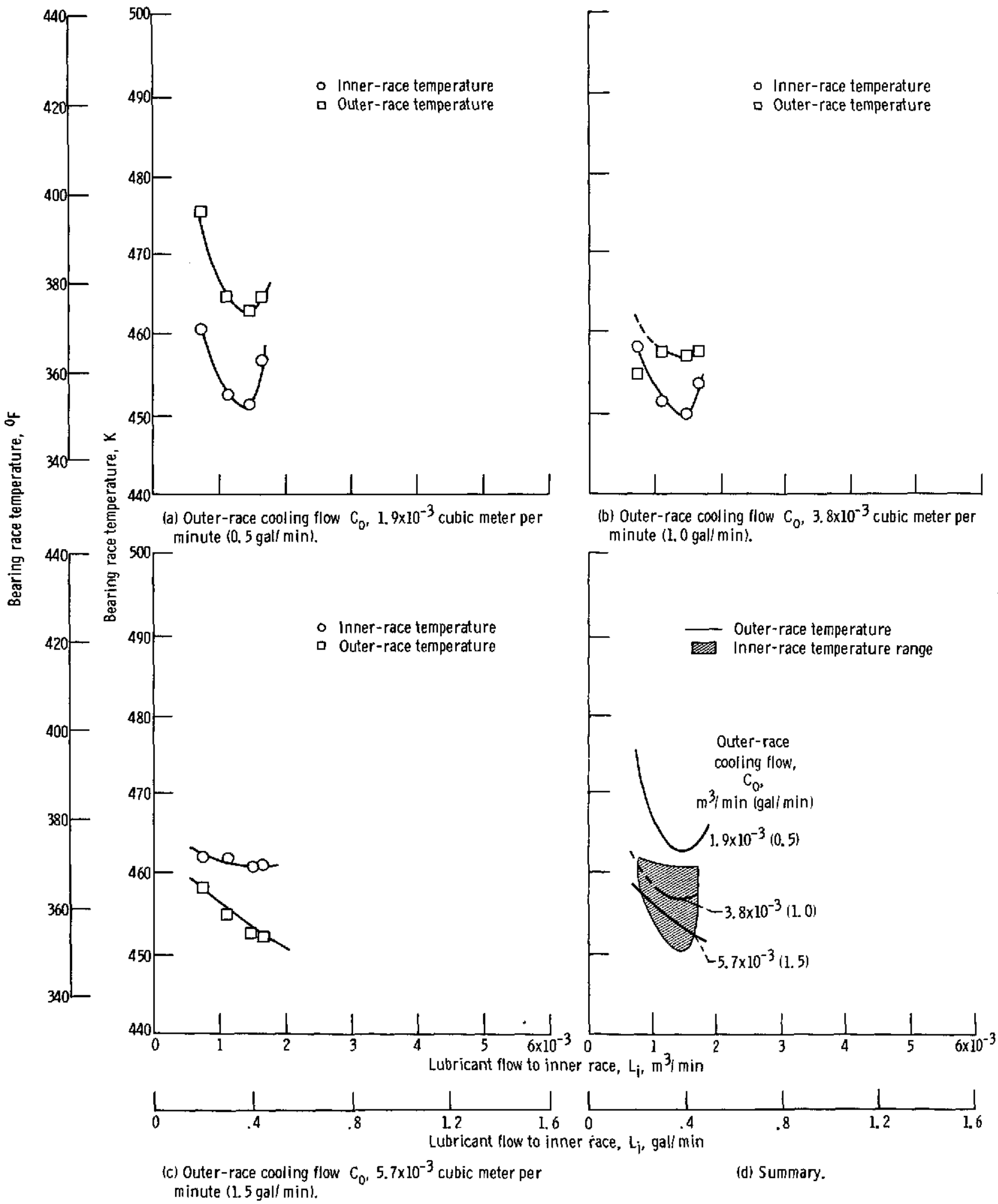


Figure 9. - Bearing race temperature as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_0. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 3 L_i.

angle bearing with no inner-race cooling flow ($C_i = 0$). Cooling of the inner race, if any was performed by lubricant entering the split inner race through the slots in the mating surfaces, designated L_i. The variation in inner-race temperatures was small, generally from 6 to 11 K (10° to 20° F), at a given lubricant flow L_i for the range of outer-race cooling rates C_o investigated. For example, at a L_i value of approximately 2.3×10^{-3} cubic meter per minute (0.6 gal/min), the temperature of the inner race ranged from approximately 486 to 497 K (415° to 435° F) at an oil inlet temperature of 394 K (250° F). (The actual temperature depended upon the outer-race cooling flow C_o.) At an increased lubricant flow rate to the inner race L_i of approximately 4.5×10^{-3} cubic meter per minute (1.2 gal/min), the inner-race temperature ranged from approximately 472 to 475 K (390° to 395° F).

At an outer-race cooling flow C_o of 9.5×10^{-4} cubic meter per minute (0.25 gal/min) (fig. 5(a)), the temperature of the outer race was nearly equal to that of the inner race. As the flow rate to the outer race C_o was increased, the outer-race temperature decreased (fig. 5(e)). At an outer-race flow C_o of 5.7×10^{-3} cubic meter per minute (1.5 gal/min) (fig. 5(d)), the temperature of the outer race was approximately 22 K (40° F) lower than the inner-race temperature. The amount of decrease in outer-race temperatures, with increasing inner-race flow L_i for all values of C_o, generally paralleled those of the inner race. What is significant is that the internal clearances of the bearing will be affected with the changes in the outer-race cooling flow C_o.

Figure 6(e) summarizes the data for the 20° contact-angle bearing wherein the inner-race cooling flow rate C_i was equal to 1.33 L_i. The temperature of the inner race ranged from approximately 478 K (400° F) at a L_i value of 1.1×10^{-3} cubic meter per minute (0.3 gal/min) to approximately 461 K (370° F) when the inner-race flow rate L_i was doubled to 2.3×10^{-3} cubic meter per minute (0.6 gal/min). Beyond this value of L_i, the temperature of the inner race increased again. In general, the outer-race temperature paralleled the inner-race temperature for the various outer-race cooling flow rates C_o. Although the outer-race temperatures were decreased with outer-race cooling flow rates C_o, the inner-race temperatures were not significantly affected.

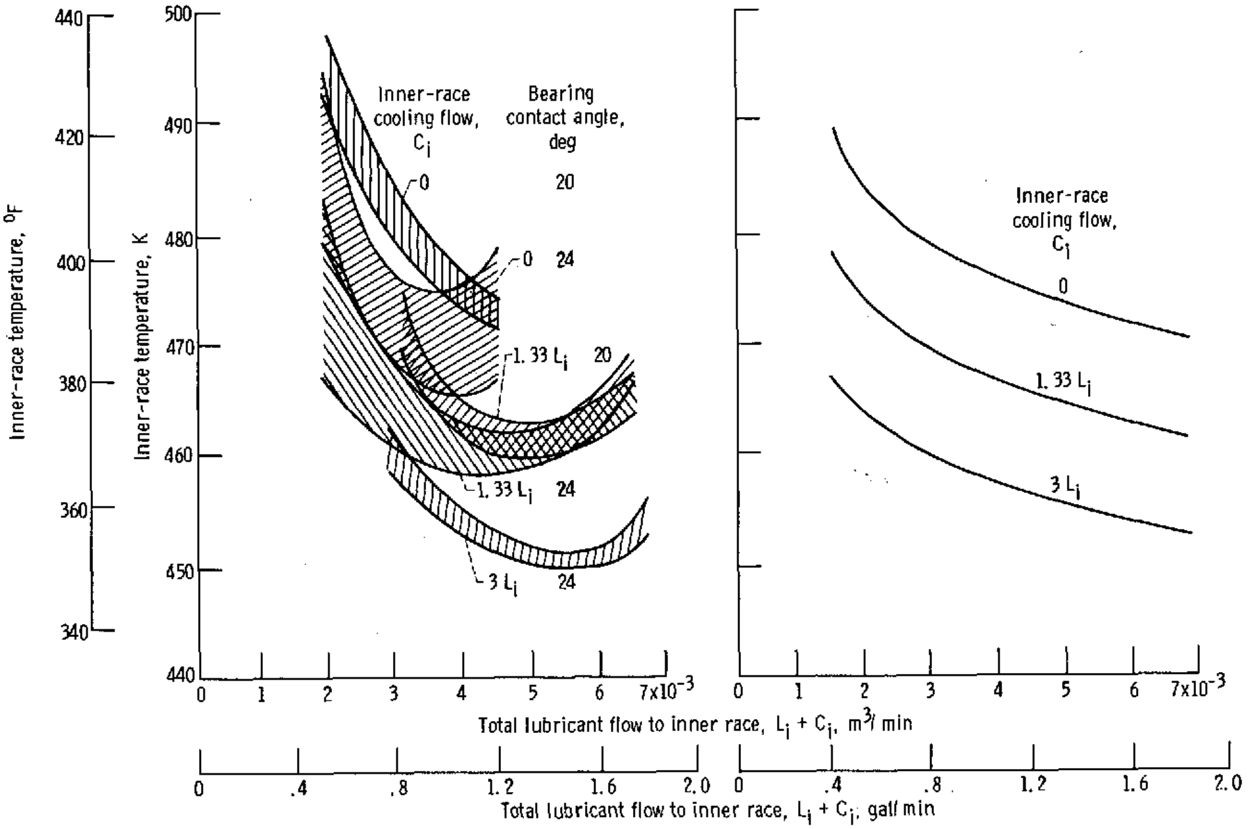
The data obtained with the 24° contact-angle bearing and a value of $C_i = 0$ are shown in figure 7. At a lubricant flow to the inner race L_i of approximately 1.9×10^{-3} cubic meter per minute (0.5 gal/min), the temperature ranged from approximately 478 to 494 K (400° to 430° F) for various values of outer-race cooling flow rate C_o. This temperature of the inner race decreased with increasing inner-race lubricant flow rate L_i. At a value of L_i equal to approximately 3.8×10^{-3} cubic meter per minute (1.0 gal/min), inner-race temperatures appear to be at a minimum for the corresponding C_o value, ranging between 464 to 479 K (375° to 390° F) (fig. 7(e)). Beyond the 3.8×10^{-3} cubic meter per minute (1.0 gal/min) flow rate temperatures appear to increase as L_i increased. This rise in temperature was probably due to the increased heat generation caused by churning effects as the result of an excessive quantity of lubricant within the bearing cavity. Again, the amount of decrease in the outer-race temperatures closely paralleled the decrease in inner-race temperatures and decreased with increasing values of C_o. The minimum outer-race temperature, achieved with a C_o of 5.7×10^{-3} cubic meter per minute (1.5 gal/min), was approximately 455 K (360° F).

At an inner-race cooling flow of C_i equal to 1.33 L_i for the 24° contact angle shown in figure 8, the minimum inner-race temperature was approximately 458 K (365° F). A minimum temperature at the outer race of approximately 447 K (345° F) was obtained for a C_o of 5.7×10^{-3} cubic meter per minute (1.5 gal/min (fig. 8(d)). At a value of L_i in excess of 1.5×10^{-3} cubic meter per minute (0.4 gal/min), temperature at the inner race appears to increase. Likewise, the temperatures at the outer race which parallel those of the inner race, except at a C_o value of 9.5×10^{-4} cubic meter per minute (0.25 gal/min) (fig. 8(a)) and 3.8×10^{-3} cubic meter per minute (1.0 gal/min) (fig. 8(c)) appears to increase.

For a C_i equal to 3.0 L_i for the 24° contact angle (fig. 9) the range of inner-race temperatures was very narrow, decreasing from approximately 460 K (370° F) at a L_i of 7.6×10^{-4} cubic meter per minute (0.2 gal/min) to 350° F at L_i equal to 1.5×10^{-3} cubic meter per minute (0.4 gal/min). Beyond this point, temperature increased at the inner race. The matching of inner- to outer-race temperatures whereby the inner- and outer-race temperatures were nearly equal was obtained with outer-race lubricant cooling rates C_o ranging between 3.8×10^{-3} to 5.7×10^{-3} cubic meter per minute (1.0 to 1.5 gal/min). As a result, the thermal effects on the bearing geometry would be minimized.

Examination of the data of figures 5 to 9 would indicate that there is relatively little change in inner-race temperature with outer-race cooling flow. Of course, the outer-race temperature is dependent on the outer-race cooling flow C_o.

It is desirable for engineering applications to define the total amount of lubricant flow required and its distribution within the bearing. In figure 10(a) the inner-race temperature range is plotted as a function of total lubricant flow to the inner race $L_i + C_i$. From these data it may be concluded that the 24° contact-angle bearing ran slightly cooler than its 20° contact-angle equivalent. However, this difference in temperature was only about 8 K (15° F) over most of the oil flows investigated. As the inner-race cooling flow C_i increased, temperatures decreased significantly. A difference in inner-race temperature of approximately 50 K (90° F) occurs between $C_i = 3.0\ L_i$ and $C_i = 0$. For nearly all conditions of inner-race cooling except where $C_i = 0$, race temperature decreases with increased total lubricant flow and then begins to increase at an intermediate flow. This temperature increase can be attributed to increased heat generation resulting from churning losses within the bearing cavity with the increased lubricant flow. It was found that for a given total flow of lubricant into the bearing inner race the lowest inner-race temperatures are obtained with 75 percent of the lubricant



(a) Experimental temperature range.

(b) Inner-race temperature calculated from emperical relation.

Figure 10. - Bearing inner-race temperature as a function of total lubricant flow to bearing inner race $L_i + C_i$ for varying outer-race cooling rates C_0. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F).

flow used primarily for cooling and lubrication of the cage-land contact ($C_i = 3\ L_i$). From these results it can be concluded that at high bearing speeds, bearing temperatures can be "tuned" to any desirable temperature by varying the inner- and outer-race lubricant flow rates and proportioning the oil flow which is used for inner-race cooling and lubrication.

For the purpose of engineering approximations, the data for both the 20° and 24° contact angle bearings were plotted on log-log coordinates. From these plots, the data for the bearing inner-race temperatures can be represented by the following empirical relation:

$$T_{Bi} = T_{oil\ in} + K_1\left[\frac{1}{F_{(L_i+C_i)}}\right]^n \tag{1}$$

where

T_{Bi}	inner-race temperature, K (^oF)
$T_{oil\ in}$	lubricant inlet temperature, K (^oF)
K_1	proportionality factor, K $(m^3/min)^n$ (^oF $(gal/min)^n$)
$F_{(L_i+C_i)}$	lubricant flow to inner race, m^3/min (gal/min)
n	exponent

The value for n can be taken as 0.14. For temperature in K,

$$K_1 = \begin{cases} 38 \text{ for } C_i = 0 \\ 33 \text{ for } C_i = 1.33\ L_i \\ 29 \text{ for } C_i = 3\ Li \end{cases}$$

For temperature in ^oF

$$K_1 = \begin{cases} 149 \text{ for } C_i = 0 \\ 131 \text{ for } C_i = 1.33\ L_i \\ 114 \text{ for } C_i = 3\ L_i \end{cases}$$

Using the aforementioned relation, bearing inner-race temperatures were calculated as a function of $L_i + C_i$ for the three inner-race cooling flows for both the 20^o and 24^o contact angle bearings. These temperatures are plotted in figure 10(b).

Effect of Lubricant Flow on Bearing Power Loss

The effect of lubricant flow L_i on bearing power loss for varying inner-race C_i and outer-race C_o race cooling flows are shown in figures 11 to 15 for both the 20^o and 24^o contact angle bearings. The power loss increased with increased L_i and total flow to the inner race $L_i + C_i$. For outer-race cooling flows from 9.5×10^{-4} to 3.8×10^{-3} cubic meter per minute (0.25 to 1.0 gal/min) the difference in power loss did not appear to be significant. At an outer-race cooling flow C_o of 5.7×10^{-3} cubic meter per minute (1.5 gal/min) the power loss was approximately 0.75 to 3 kilowatts (1 to 4 hp) greater than at the lower values of C_o. This difference in power loss can be attributed to the reduction in bearing internal clearances due to temperature differences between the inner and outer races. Hence, from these data it appears that bearing power loss is

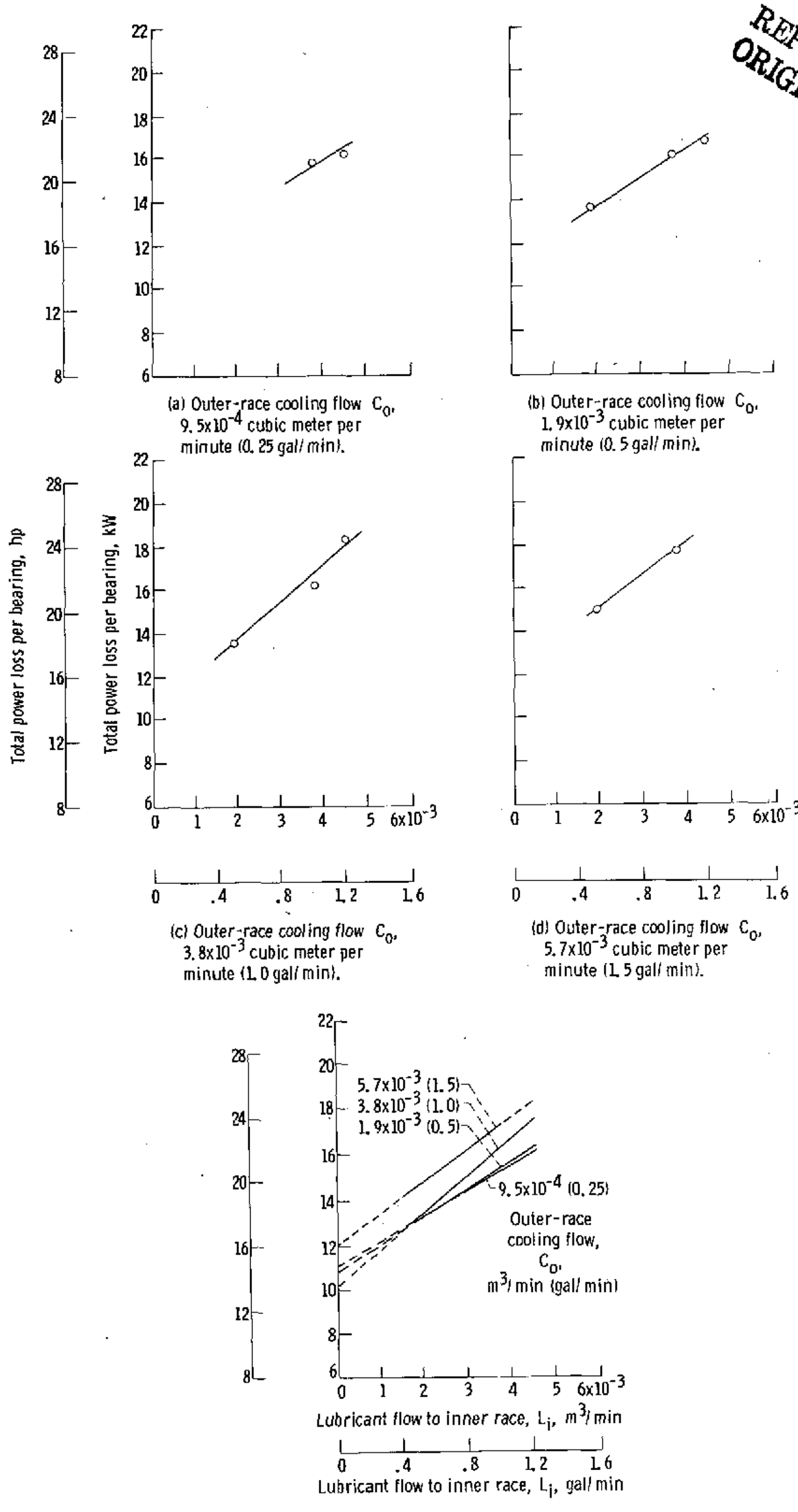


Figure 11. - Bearing power loss as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 20°; inner-race cooling flow C_i, 0.

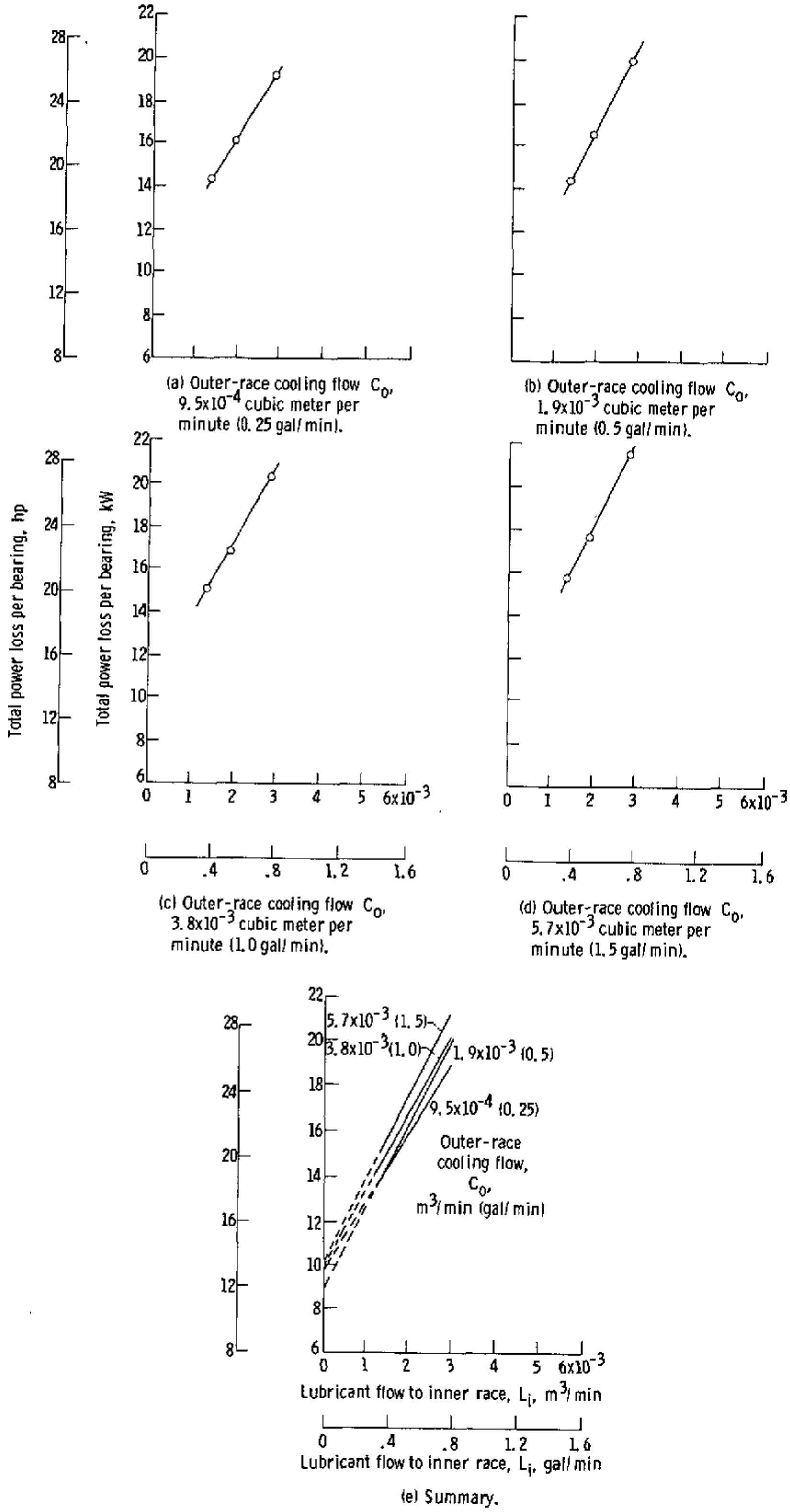


Figure 12. - Bearing power loss as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 20°; inner-race cooling flow C_i, 1.33 L_i.

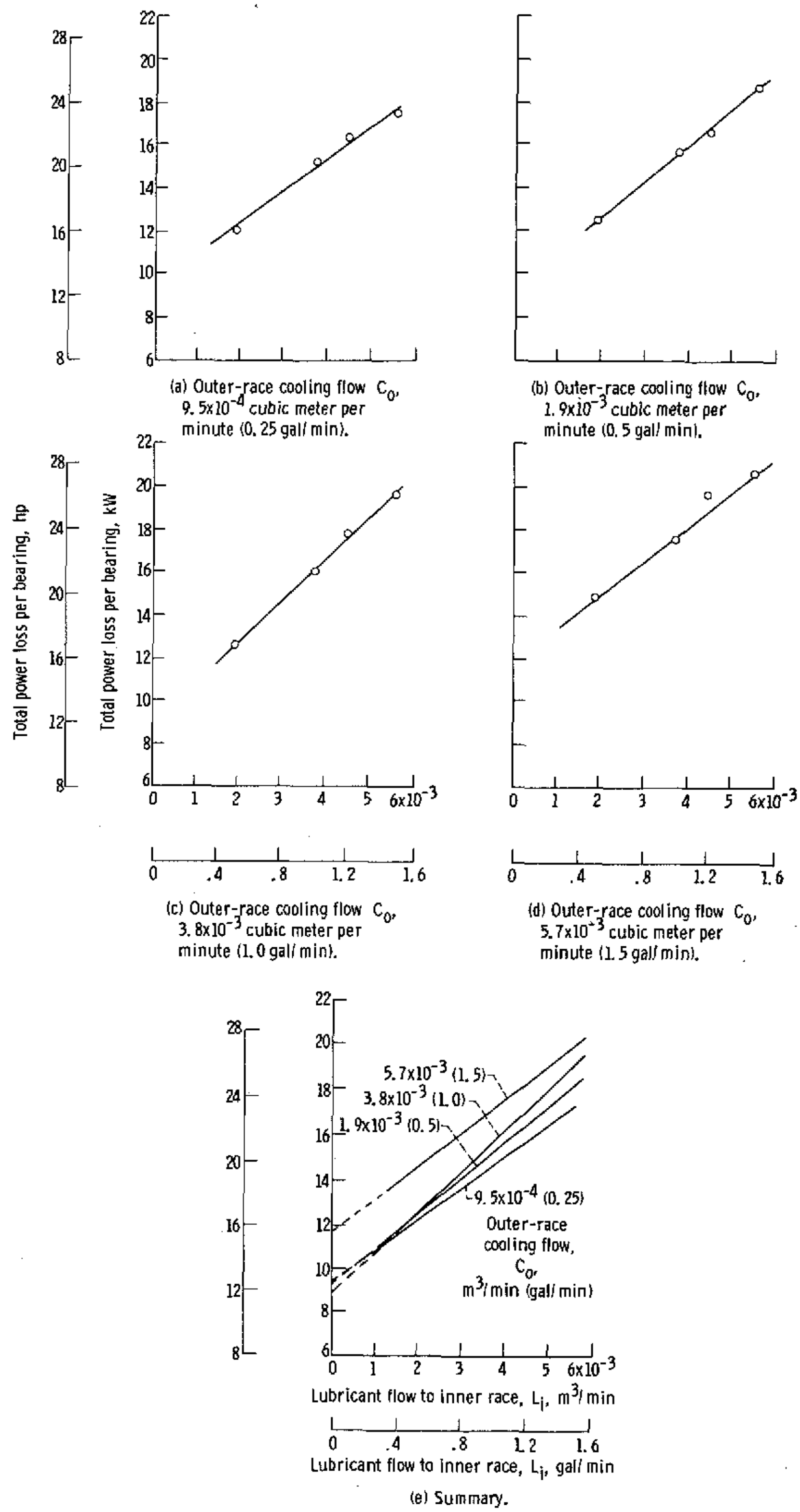
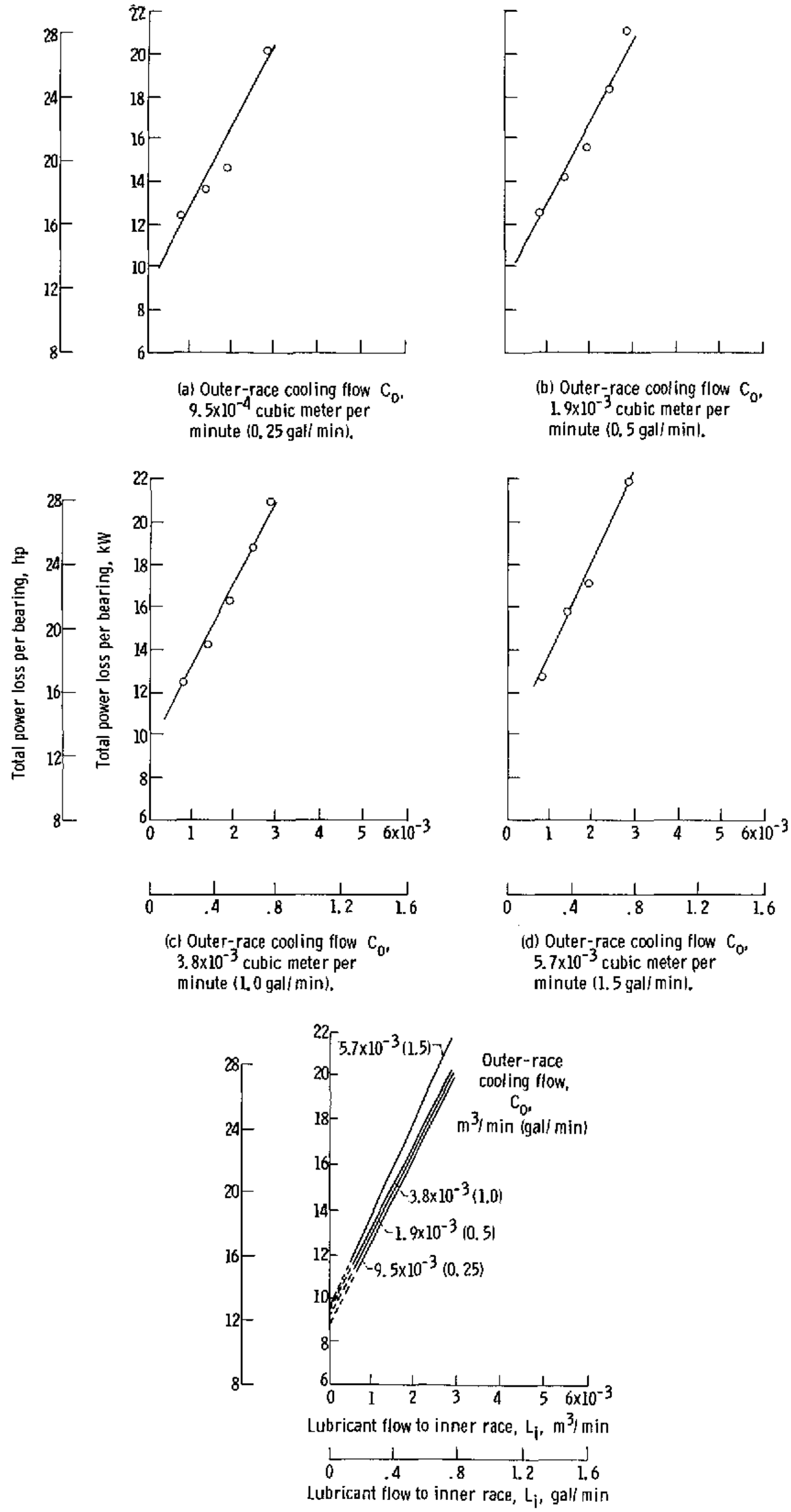


Figure 13. - Bearing power loss as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 0.



(a) Outer-race cooling flow C_o, 9.5x10^{-4} cubic meter per minute (0.25 gal/min).

(b) Outer-race cooling flow C_o, 1.9x10^{-3} cubic meter per minute (0.5 gal/min).

(c) Outer-race cooling flow C_o, 3.8x10^{-3} cubic meter per minute (1.0 gal/min).

(d) Outer-race cooling flow C_o, 5.7x10^{-3} cubic meter per minute (1.5 gal/min).

(e) Summary.

Figure 14. - Bearing power loss as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3x10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 1.33 L_i.

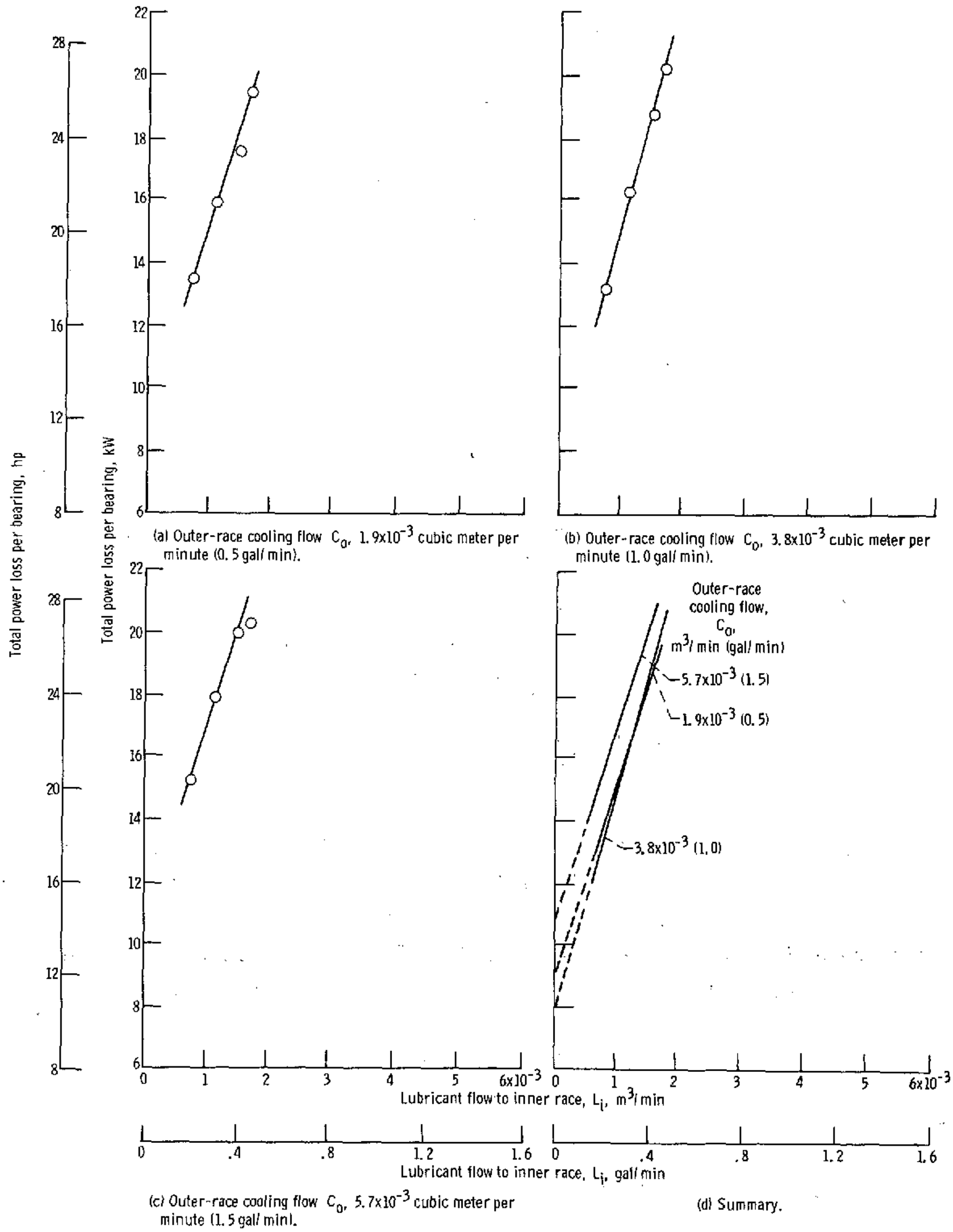


Figure 15. - Bearing power loss as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F); contact angle, 24°; inner-race cooling flow C_i, 3 L_i.

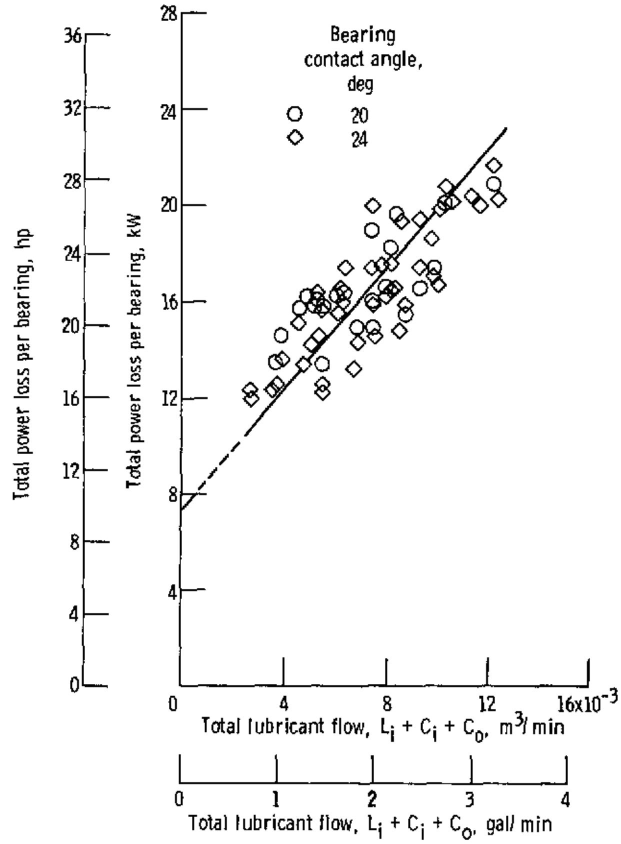


Figure 16. - Bearing power loss as a function of total lubricant flow to bearing $L_i + C_i + C_o$. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F).

some function of total lubricant flow to the bearing $L_i + C_i + C_o$. The power loss data of figures 11 to 15 are replotted in figure 16 as a function of total lubricant flow to the bearing. From this figure power loss increases nearly linearly with lubricant flow to the bearing regardless of bearing contact angle. From a least-squares analysis the relation of power loss and flow rate for the data presented can be expressed as follows:

$$P_B = P_T + K_2(L_i + C_i + C_o) \tag{2}$$

where

P_B bearing power loss, kW (hp)

P_T bearing tare power loss, kW (hp)

K_2 proportionality factor, kW-min/m^3 (hp-min/gal)

For power in terms of kilowatts and flow in terms of cubic meters per minute,

$$P_B = 7.5 + 1192(L_i + C_i + C_o)$$

For power in terms of horsepower and flow in terms of gallons per minute,

$$P_B = 10 + 6(L_i + C_i + C_o)$$

From the measurement of bearing temperatures, it was reported in the previous section that, by varying both inner- and outer-race flow rate, bearing temperature can be tuned to any desirable value which, of course, depends also on lubricant inlet temperature. It must be also realized that, at very low flow rates to the bearing, there is a good probability that the bearing would not even operate because of a lack of sufficient cooling. Further, providing lubricant to the outer race only, without providing for lubrication into the ball-race contacts, would prove to be catastrophic. Hence, the relation provided in equation (2) would be applicable only to the lubrication flow combinations reported.

The power loss from the bearing is dissipated in the form of heat by conduction to the lubricant and by convention and radiation to the surrounding environment. Lubricant outlet temperature from the bearing was measured for all conditions of flow. Power transferred to the lubricant was calculated using the standard heat transfer equation

$$Q_T = MC_p(t_{out} - t_{in}) \tag{3}$$

where

Q_T total heat transfer to lubricant, J/min (Btu/min)

M mass flow, kg/min (lb/min)

C_p specific heat, J/(kg)(K) (Btu/(lb)(^{o}F))

t_{out} oil outlet temperature, K (^{o}F)

t_{in} oil inlet temperature, K (^{o}F)

The result of these heat transfer calculations are shown in figure 17 as a function of total lubricant flow to the bearing $L_i + C_i + C_o$. (For convenience, heat values were converted from J/min to kW.) From figure 17(d), a larger amount of heat is transferred to the lubricant at $C_i = 0$ than at $C_i = 3\ L_i$. The following general relation represents the total heat transferred to the lubricant in terms of lubricant flow for the conditions specified

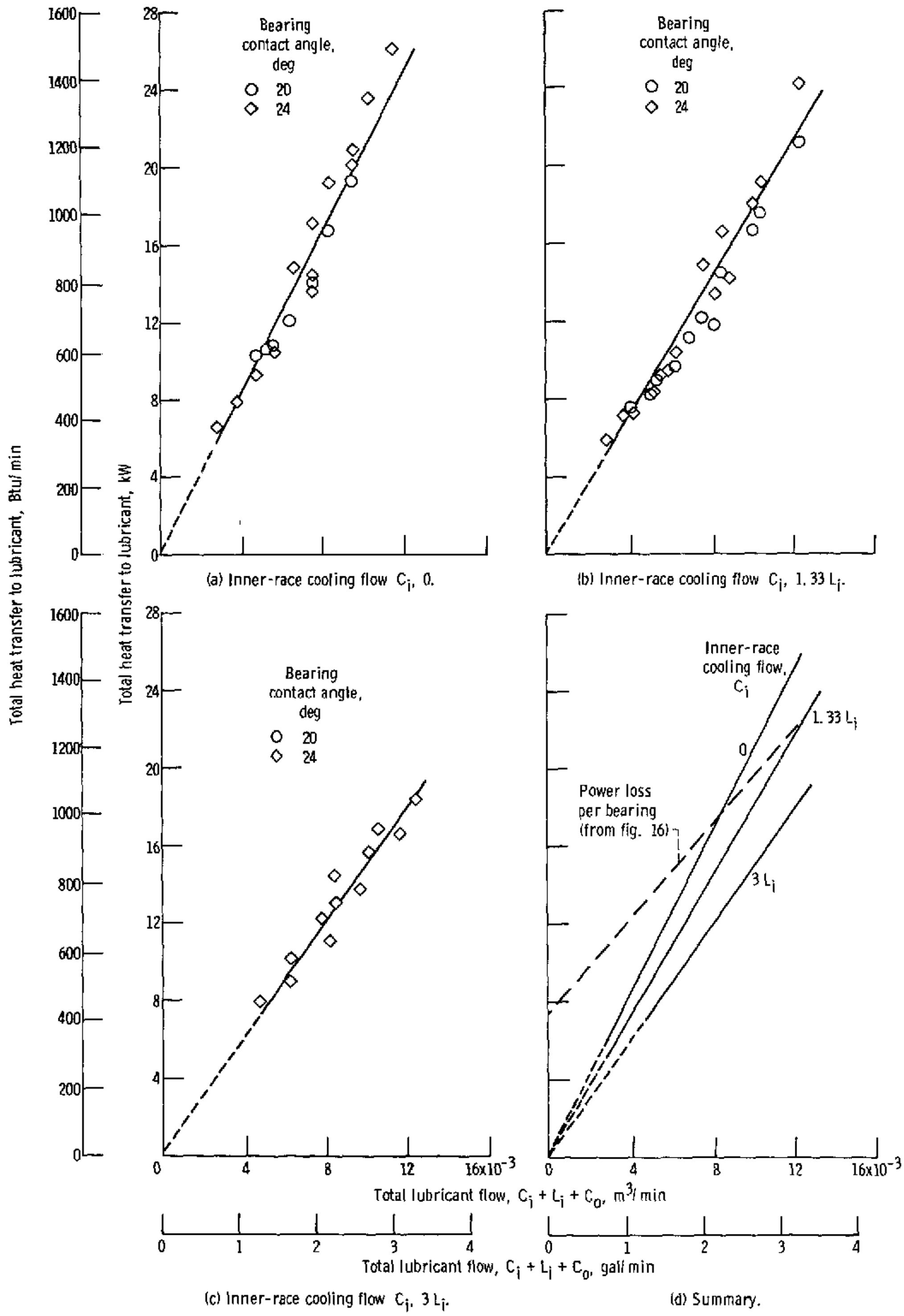


Figure 17. - Heat generated by a bearing transferred to lubricant as a function of total lubricant flow to bearing $L_i + C_i + C_o$ for varying inner-race cooling rates C_i. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250° F).

$$Q_T = K_3(L_i + C_i + C_o) \tag{4}$$

where

K_3 proportionality factor, kW-min/m^3 (Btu/gal)

For Q_T in terms of kilowatts

$$K_3 = \begin{cases} 2.08\times10^3 \text{ for } C_i = 0 \\ 1.8\times10^3 \text{ for } C_i = 1.33\ L_i \\ 1.5\times10^3 \text{ for } C_i = 3\ L_i \end{cases}$$

For Q_T in terms of Btu per minute,

$$K_3 = \begin{cases} 451 \text{ for } C_i = 0 \\ 390 \text{ for } C_i = 1.33\ L_i \\ 325 \text{ for } C_i = 3\ L_i \end{cases}$$

For values of $C_i = 1.33\ L_i$ and $3\ L_i$, the bearing power loss is in excess of the heat transferred to the lubricant. However, at $C_i = 0$, the heat transferred to the lubricant exceeds the bearing power loss where the total lubricant flow $L_i + C_i + C_o$ exceeds 8×10^{-3} cubic meter per minute (2.2 gal/min). This, of course, is a physical impossibility. It is estimated that the lubricant pump can add approximately 0.1 kilowatt (6 Btu/min) of heat energy to the lubricant. This value compares to a 3.1 kilowatts (178 Btu/min) higher heat rejection rate to the lubricant at a flow rate of 12×10^{-3} cubic meter per minute (3.2 gal/min) than was indicated by the bearing power loss data (which is based upon shaft horsepower measurements). The maximum power available to pump the lubricant for its cooling and lubrication function including pump losses was 0.56 kilowatt (0.75 hp). It must be concluded that there may be sufficient inaccuracies in the determination of tare losses in the test rig and/or shaft horsepower measurements to account for the power differences discussed herein.

The previous results indicate that the bearing can be temperature and power tuned to any specific operating condition depending upon the heat transfer and lubricating characteristics of the lubricant. The concept of Bearing Thermal Management proposed is believed to be the proper technological approach to high-speed bearing operation. The basis of this is the recognition that total and flexible thermal control over all of the bearing components is essential to achieve a reliable high-speed, highly loaded bearing. This in turn requires a lubrication scheme of sufficient sophistication to achieve the thermal controls and still permit its practical use in actual flight hardware.

Effects on Ball Orbital Speed

In general, ball orbital speed, of which cage speed is a function, can be affected by lubricant and coolant flow rates as well as lubricant viscosity. Basically, two variables can affect ball speed. The first variable is the drag of the balls through the viscous fluid medium. This drag can be a function of the amount of lubricant in the bearing cavity and its viscosity (ref. 9). Second, the temperature difference between the inner and outer races of the bearing result in changes in bearing clearance and, thus, contact angle.

Figure 18 contains a summation of cage speed for various values of lubricant flow L_i into the inner face for various inner-race C_i and outer-race C_o race cooling flows. For a given bearing contact angle, the cage speed does not vary more than 2 per-

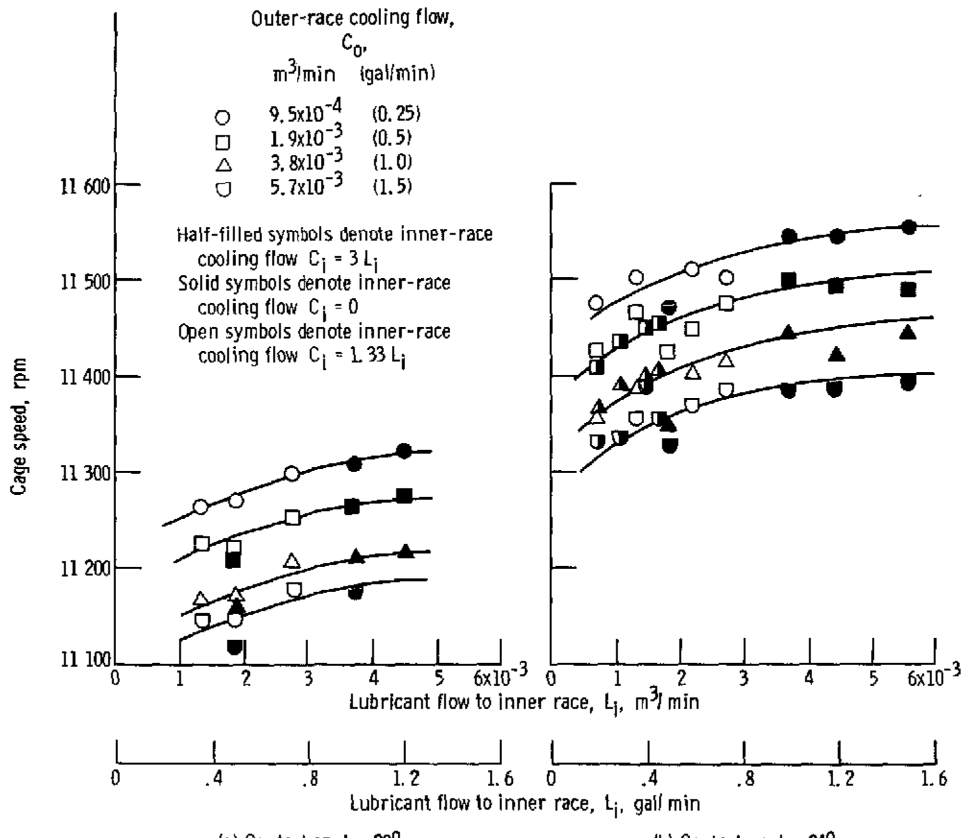


Figure 18. - Bearing cage speed as a function of lubricant flow into bearing L_i for varying outer-race cooling rates C_o. Bearing type, 120-millimeter-bore angular-contact ball bearing; bearing thrust load, 22 241 newtons (5000 lb); speed, 25 000 rpm ($3x10^6$ DN); oil inlet temperature, 394 K (250° F).

cent over the entire range of flow rates. This range of speed difference is not considered significant. However, the trends in cage speed are interesting and may serve as a guide to determine the correctness of various assumptions used in bearing dynamic analysis.

What is initially apparent from these data is that, as outer-race cooling C_o increases (resulting in lowering of outer-race temperature (see figs. 5 to 9)), cage speed decreases. This can be readily explained by the fact that bearing clearances decrease which in turn lowers the ball orbital speed.

As the lubricant flow into the inner race L_i increases, cage speed increases. This probably occurs because of cooling of the bearing races which tends to increase clearance. For the 24^o contact angle bearing, the increase in cage speed appears to be tapering off at the higher values of L_i. This is probably caused by the inner-race temperature beginning to increase at intermediate values of L_i causing a decrease in bearing clearance.

SUMMARY OF RESULTS

A parametric study was performed with split inner-race 120-millimeter-bore angular-contact ball bearings optimally designed for 25 000 rpm (3×10^6 DN) operation. The bearings had nominal contact angles of 20^o and 24^o. Provisions were made for outer- and inner-race cooling and for injection of lubricant into the bearing through radial holes contained in the inner-race split of the bearing. Test conditions included a thrust load of 22 241 newtons (5000 lb), a speed of 25 000 rpm (3×10^6 DN) and an oil inlet temperature of 394 K (250^o F). Oil flow and coolant rate were controlled and varied to the bearing inner and outer races. Measurements were made of power consumption, oil outlet temperatures, inner- and outer-race temperatures, and cage speed. The following results were obtained:

1. Bearing inner-race temperature was found to decrease as the total lubricant and cooling flow to the inner race was increased. However, at intermediate flow rates temperature appears to increase with increasing flow.

2. Outer-race temperatures, also decreased with increasing total lubricant flow to the inner race and, in general, paralleled those of the inner race at lower temperatures. The magnitude of the temperature was a function of the outer-race cooling flow.

3. Bearing power consumption was a function of total lubricant flow to the bearing. As flow rate was increased, power consumption increased.

4. Bearing operating temperature, differences in temperatures between the inner and outer races, and bearing power consumption can be tuned to any desirable operation requirement by varying four parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race and oil inlet temperature.

5. Ball orbital speed was found to increase with increased oil flow to the inner race. However, the increase for a given bearing was not more than 2 percent over the entire range of flow rates.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 25, 1974,
501-24.

REFERENCES

1. Bamberger, Eric N.; Zaretsky, Erwin V.; and Anderson, William J.: Fatigue Life of 120-mm Bore Ball Bearings at 600° F with Fluorocarbon, Polyphenyl Ether and Synthetic Paraffinic Base Lubricants. NASA TN D-4850, 1968.
2. Bamberger, E. N.; Zaretsky, E. V.; and Anderson, W. J.: Effect of Three Advanced Lubricants on High-Temperature Bearing Life. J. Lub. Tech., Trans. ASME, Ser. F, vol. 92, no. 1, Jan. 1970, pp. 23-33.
3. Zaretsky, Erwin V.; and Bamberger, Eric N.: Advanced Airbreathing Engine Lubricants Study with a Tetraester Fluid and a Synthetic Paraffinic Oil at 492 K (425° F). NASA TN D-6771, 1972.
4. Parker, R. J.; and Zaretsky, E. V.: Effect of Oxygen Concentration on an Advanced Ester Lubricant in Bearing Tests at 400° and 450° F. NASA TN D-5262, 1969.
5. D'Orazio, A. J.: Development and Utilization of Specification Mil-L-23699 Synthetic Lubricating Oils for Aircraft Gas Turbine Engines. NAPTC-AED-1868, Naval Air Propulsion Test Center (AD-828064L), 1968.
6. Jones, A. B.: Life of High-Speed Ball Bearings. ASME Trans., vol. 74, no. 5, July 1952, pp. 695-703.
7. Jones, A. B.: Ball Motion and Sliding Friction on Ball Bearings. J. Basic Eng., Trans. ASME, Series D, vol. 81, no. 1, Mar. 1959, pp. 1-12.
8. Jones, A. B.: General Theory for Elastically Constrained Ball and Radial Roller Bearings under Arbitrary Load and Speed Conditions. J. Basic Eng., Trans. ASME, Ser. D, vol. 82, no. 2, June 1960, pp. 309-320.
9. Harris, T. A.: An Analytical Method to Predict Skidding in Thrust-Loaded, Angular-Contact Ball Bearings. J. Lub. Tech., Trans. ASME, Ser. F, vol. 93, no. 1, Jan. 1971, pp. 17-24.

10. Harris, T. A.: An Analytical Method to Predict Skidding in High Speed Roller Bearings. ASLE Trans., vol. 9, no. 3, Jul. 1966, pp. 229-241.

11. Holmes, P. W.: Evaluation of Drilled-Ball Bearings at DN Values to Three Million. I - Variable Air Flow Tests. NASA CR-2004, 1972.

12. Holms, P. W.: Evaluation of Drilled-Ball Bearings at DN Values to Three Million. II - Experimental Skid Study and Endurance Tests. NASA CR-2005, 1972.

13. Zaretsky, E. V.; Parker, R. J.; and Anderson, W. J.: Component Hardness Differences and Their Effect on Bearing Fatigue. J. Lub. Tech., Trans. ASME, Ser. F, vol. 89, no. 1, Jan. 1967, pp. 47-62.