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PARAMETRIC STUDY OF THE LUBRICATION OF THRUST LOADED 120-MM BORE BALL BEARINGS TO 3 MILLION DN

by H. Signer, E. N. Bamberger, and E. V. Zaretsky
Lewis Research Center
Cleveland, Ohio 44135

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by H. Signer*, E. N. Bamberger**, and E. V. Zaretsky‡

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ABSTRACT

A parametric study was performed with 120-mm bore angular-contact ball bearings under varying thrust loads, bearing and lubricant temperatures, and cooling and lubricant flow rates. Contact angles were nominally 20° and 24° with bearing speeds to 3 million DN. Endurance tests were run at 3 million DN and a temperature of 492 K (425° F) with 10 bearings having a nominal 24° contact angle at a thrust load of 22241 N (5000 lb). Bearing operating temperature, differences in temperatures between the inner and outer races, and bearing power consumption can be tuned to any desirable operating requirement by varying 4 parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race and oil inlet temperature. Preliminary endurance tests at 3 million DN and 492 K (425° F) indicate that long term bearing operation can be achieved with a high degree of reliability.

*Industrial Tectonics, Inc., Compton, California

**General Electric Co., Cincinnati, Ohio

‡NASA Lewis Research Center, Cleveland, Ohio - Member ASME

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 3 million DN. (DN is a bearing speed parameter and is equal to the product of the bearing bore in millimeter 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 operational parameters 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 [1-4][1]. Of these, the poly-ester and tetra-ester fluids have proven to be most useful and applicable in typical air-breathing environments, and consequently have been widely accepted in current commercial and military applications [5]. These fluids have good thermal stability at temperatures to 505 K (450° F). Bearing life at 492 K (425° F) with the tetra-esters exceeded AFBMA-predicted (catalogue) life by a factor in excess of four [3]. When test oil in [3] was replaced at a rate approximating the replenishment rate in

[1] Numbers in brackets designate References at end of paper.

actual commercial engine usage, no significant increase in lubricant viscosity or acidity with time was observed at 492 K (425° F).

Research reported in [6] indicated that AISI M-50 steel can produce the most favorable life results at elevated temperatures when compared with other high-speed or high-temperature steels such as AISI M-1, AISI M-10, WB-49, etc. Rolling-element bearing tests to 589 K (600° F) with the AISI M-50 steel and a synthetic paraffinic oil produced lives in excess of 13 times the AFBMA predicted life.

In conventional rolling-element bearings, both metallic and nonmetallic cages have found widespread use. Precision bearings, such as those used for aircraft applications, are usually equipped with cages machined from iron or copper base alloys. Where marginal lubrication is suspected during operation, silver plating of the cage material has been used to attenuate the detrimental effects of intermittent or momentary metal to metal contact. The bearing tests reported in [1-4] successfully utilized a cage material made of a nickel base alloy (AMS 4892). In most critical high-temperature, high-load and high-speed aircraft applications, the cages are a carbon steel (AISI 4340 or similar) hardened to a Rockwell C hardness of approximately 33 and silver plated for extra protection.

With the tetra-ester 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 3 million 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 [7-9] first considered speed effects on bearing life and dynamics without considering the effect of the lubricant. Subsequently, Harris [10, 11] expanded these bearing analyses including lubricant effects.

At high speed, the effect of centrifugal loading of the rolling elements against the outer race of the bearing becomes extremely important. Theoretical life calculations for a 150-mm bore angular-contact ball bearing operating at 3 million DN (20 000 rpm) predict that this bearing has approximately 20% AFBMA-calculated life [12]. This decrease in predicted life is due to the increased stress in the outer race caused by centrifugal effects. The expected final result is extremely short bearing life at speeds much above 2 million DN both in actual running time (hr) and in total bearing inner-race revolutions.

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 to 3 million DN is cooling lubricant applied under the race [13]. In this method lubricant is centrifugally injected through the split inner-race and shoulders of an angular-contract ball bearing by means of a plurality of radial holes. As a result, both the cooling and lubricant function is accomplished.

The research reported herein, which is based on the work reported in [14, 15], was undertaken to investigate the performance of optimally designed 120-mm bore angular-contact ball bearings at speeds to 3 million DN. The primary objectives were to (a) determine the operating characteristics under variable lubricant flow conditions at 3 million DN, (b) determine the effect of speed and load on bearing performance and (c) conduct preliminary bearing endurance tests at 3 million DN and at a temperature of 492 K (425° F).

HIGH-SPEED BEARING TESTER

A schematic of the high-speed, high-temperature bearing tester used in these tests is shown in Fig. 1. This tester is described in detail in [1, 2] and has been subsequently modified to operate at speeds of 25 000 rpm. 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 kW (100 hp) fixed speed electric motor. The drive motor is mounted to an adjustable base, so that drive pulleys for 12 000 to 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 start-up. This control protects the bearings efficiently from undesirable acceleration during start-up. The lubrication system of the test rig delivers up to 2.8×10^{-2} cubic meters (7.5 gallons). 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 in Fig. 2. The first of these loops supplies cooling oil to the 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. L_i designates the oil flow to the bearing through a plurality of radial holes in the center of the split inner race. 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 and manifolds previously discussed an unlimited number of combinations of oil flows can be achieved to evaluate various conditions. Consequently, values of L_i, C_i, and C_o can be independent of each other.

The machine instrumentation includes the standard 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 read-out and oscilloscope, respectively. The oil flow was established by a flow meter, 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-mm 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 1. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15 balls, 2.0638 cm (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 cm (0.002 in.) maximum thickness of silver plate (AMS 2410). The cage balance was 3 gm-cm (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 ± one 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 [16]. Surface finish of the balls was 2.5 μcm (1 microinch) AA and the inner and outer raceways were held to a 5 μcm (2 microinch) AA maximum surface finish.

A photograph of the test bearing is shown in Fig. 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 [13] 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 race shoulder.

LUBRICANT

The oil used for the parametric studies as well as for the subsequent long-time high-speed (3 million DN) endurance tests, was a 5-centistoke neopentylpolyol (tetra) ester. This is a Type II oil, qualified to MIL-L-23699 as well as to the internal oil specifications of most major aircraft-engine producers. The major properties of subject oil are presented in Table 2 and a temperature-viscosity curve is shown in Fig. 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. With the exception of speed, all test parameters such as load, lubricant flow rate and oil temperature could be adjusted while the tester was in operation. During operation, the tester was allowed to reach equilibrium condition before the data were recorded. For the long term endurance test at 3 million DN the procedure varied only to the extent that once the preset test parameters had been achieved no further adjustments were made.

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.

Approximately 1.9×10^{-3} cubic meters (0.5 gallon) of oil was replenished in every twenty-four hours of operation. This amount of oil was removed from the sump and replaced with an equal amount of fresh oil. The rate of replenishment was approximately equal to 0.3 percent per hour of the entire sump capacity. This operation was performed without interruption of the test. By closely monitoring the oil replenished, it has been shown that no significant increases in either viscosity or acid number occur, even in tests on the order of 1000 hours duration.

The rationale for this replenishment is that it serves to maintain the lubricant properties, and specifically, viscosity and acid number, at a relatively constant level. If no oil changes were made, the lubricant would present a continuously changing parameter with indeterminate effects on bearing performance. The selected replenishment rate is based on actual oil consumption rates in turbojet engines.

RESULTS AND DISCUSSION

Effect of Speed

The effect of speed on the operating characteristics of the 120-mm bore angular-contact ball bearings are shown in Figs. 5 and 6 for three thrust loads. The 20° and 24° contact angle bearings were run at nominal speeds of 12 000, 16 000, 20 000, and 25 000 rpm. Test conditions included an oil-inlet temperature of 428 K (310° F); a lubricant flow rate through the inner race, L_i, of 1.2×10^{-3} cubic meters per minute (0.313 gpm); an inner-race cooling rate, C_i, of 3.6×10^{-3} cubic meters per minute (0.94 gpm) or 3 L_i; and an outer-race cooling flow rate, C_o, of 1.9×10^{-3} cubic meters per minute (0.5 gpm). Generally, throughout the range of speeds for the 6672- and 13345-N (1500- and 3000-lb) thrust loads, the bearing inner-race temperatures were 3 to 8 K (5 to 15 F) higher than the outer-race temperatures (Figs. 5(a) and (b)). For the 22241-N (5000-lb) thrust load, the temperatures of the inner and outer races were generally within 3 K (5 F) of each other. Over the range of speeds, the temperature of the bearing races increased from the range of 441 to 453 K (335° to 355° F), to temperatures in the range of 478 to 494 K (400° to 430° F) or an increase of approximately 36 to 42 K (65 to 75 F). In general, the 24° contact angles bearings ran approximately 3 to 8 K (5 to 10 F) cooler than the 20° contact angle bearings over the entire speed range. This difference in temperature was not considered significant.

The power loss for the bearings over the speed range is shown in figure 6. As would be expected, bearing power consumption increases linearly with speed for the three loads shown. At the thrust load of 22241 N (5000 lb) and a speed of 25 000 rpm, the power loss is approximately 15 kW (20 horsepower) per bearing for the two contact angles. At the speed of 12 000 rpm and a thrust load of 6672 N (1500 lb), the power loss is approximately 3 kW (4 horsepower) or 20 percent of the power loss at the high-speed, high-load condition.

Comparing the power consumption of the two contact-angle bearings at the 22241N (5000 lb) load and at low speeds, the 24^{o} contact-angle bearing had a power loss approximately 25-percent greater than the 20^{o} contact-angle bearing. At the higher speeds and for all three loads, the difference in power loss between the two contact angles was essentially insignificant.

Another effect of speed on bearing operation is the centrifugal effect on outer-race load or stress and dynamic contact angle. The effect of speed on contact stress and operating contact angle using the methods of Harris [10, 11] are shown in Table 3. For these calculations, a bearing operating temperature of 478 K (400^{o} F) was assumed. These calculated data show that there is a more marked increase in stress at the outer-race ball contact at the lower thrust load of 6672 N (1500 lb) than at the 22241-N (5000-lb) thrust load. This difference in the relative increase in stress at the outer-race ball contact, probably accounts for the slightly higher rate of increase in power consumption with speed with the lower thrust-loaded bearing tests. Contact-angle measurements were made on bearings having an initial contact angle of 24^{o} and tested at 12 000, 16 000, and 25 000 rpm at a load of 22241 N (4000 lb). The actual measurements were in excellent agreement with the prediction made in Table 3, with the exception of the inner-race contact angle at the maximum speed condition. Here the measured angle was approximately 25^{o} versus a predicted operating angle of 34^{o}. The reason for this discrepancy is not understood at the present time.

Effect of Load

The effect of load on the bearing temperature is shown in Fig. 7. Bearing temperatures increased nearly linearly with load for most of the speeds. However, a temperature rise of only 6 to 11 K (10 to 30 F) occurs when the load was increased from 6672 to 22241 N (1500 to 5000 lb). The difference in temperature between the inner and outer race for the 24^{o} contact-angle bearings was insignificant and generally was within ± 3 K (± 5 F). For the 20^{o} contact-angle bearings, the variation in temperature between the inner and outer races was slightly greater at the lower loads. This difference in temperature, however, did not exceed 8 K (15 F). Bearing power loss as a function of load for the two contact angles is shown in figure 8. The greatest increase in power consumption at any speed was approximately 3 kW (4 horsepower) from the 6672-N (1500-lb) thrust load to the 22241-N (5000-lb) thrust load. The increase in power consumption with thrust is considered negligible for most practical applications.

Effect of lubricant flow. - The effect of lubricant flow into the bearing, and that used for cooling of the inner and outer races was determined. A test condition was selected which incorporated a speed of 25 000 rpm (3 million DN) and a thrust load of 22241 N (5000 lb). This condition was chosen on the basis of approaching maximum Hertzian stresses [2068×10^{6} N/m^{2} (~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 meters per minute (0.25, 0.5, 1.0, and 1.5 gpm). Inner-race lubricant flow (L_i) ranged from 7.6×10^{-4} cubic meters per minute (0.2 gpm) to approximately 5.7×10^{-3} cubic meters per minute (1.5 gpm). Inner-race cooling (C_i) was varied as a function of L_i. The results of these tests are shown in Figs. 9 and 10.

Referring to Fig. 9(a), the test results are shown for the 20° contact-angle bearing with no inner-race cooling flow (C_i) except for that lubricant entering the inner race through the slots in the mating surfaces of the split inner race. The temperatures of the inner race varied generally from 6 to 11 K (10 to 20 F) for a particular lubricant flow (L_i) over the range of outer-race cooling rates (C_o). At a lubricant flow (L_i) of approximately 2.3×10^{-3} cubic meters per minute (0.6 gpm) to the inner race, the temperature of this component ranged from approximately 486 to 497 K (415° to 435° F) at an oil inlet temper-perature of 394 K (250° F). The actual temperature depends 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 meters per minute (1.2 gpm), the temperature ranged from 472 to 475 K (390° to 395° F).

At an outer-race cooling flow (C_o) of 9.5×10^{-4} cubic meters per minute (0.25 gpm)(fig. 9(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. At an outer-race flow (C_o) of 5.7×10^{-3} cubic meters per minute (1.5 gpm) the temperature of the outer race was approximately 17 K (30 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).

Referring to Fig. 9(b) wherein the inner-race cooling flow rate (C_i) was 1.33 L_i, the temperature of the inner race ranged from 478 K (400^o F) at an inner-race lubricant flow rate (L_i) of $1.1x10^{-3}$ cubic meters per minute (0.3 gpm) to approximately 461 K (370^o F) when the inner-race flow rate was doubled to $2.3x10^{-3}$ cubic meters per minute (0.6 gpm). 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). However, the inner-race temperatures were not significantly affected by the outer-race flow rates (C_o). At an outer-race cooling flow (C_o) of $1.9x10^{-3}$ cubic meters per minute (0.5 gpm), the temperature of the inner and outer race was approximately equal (fig. 9(b)). However, at an L_i value of $2.3x10^{-3}$ cubic meters per minute (0.6 gpm) and with inner-race cooling (C_i), a decrease of as much as 36 K (55 F) can be achieved over the same bearing having no lubricant cooling flow to the inner-race lands (Fig. 9(a)). It may be concluded, that inner-land cooling (C_i) can play a significant role in reducing the detrimental thermal effects on the bearing and specifically under marginal lubrication conditions.

The data obtained with the 24^o contact-angle bearing and a value of $C_i = 0$ are shown in Fig. 9(c). The results under this operating condition were generally similar to those obtained for the 20^{o} contact angle bearing shown in Fig. 9(a). However, the inner-race temperatures of the 24^{o} contact-angle bearing varied over a greater range. At a lubricant flow to the inner race (L_i) of approximately $1.9x10^{-3}$ cubic meters per minute (0.5 gpm), the temperature ranged from approximately 478 to 494 K (400^{o} to 430^{o} F) with various levels 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 $3.8x10^{-3}$ cubic meters per minute (1 gpm), operating temperatures were at a minimum ranging between 464 to 479 K (375^{o} to 390^{o} F). Beyond this flow rate temperatures began to increase as lubricant flow (L_i) increased. This rise in temperature was probably due to the increased quantity of lubricant within the bearing cavity and to the resultant churning effects. Again, the outer-race temperature closely paralleled the inner-race temperature and decreased with increased outer-race cooling flow (C_o). The minimum outer-race temperature, achieved with an outer-race cooling flow (C_o) of $5.7x10^{-3}$ cubic meters (1.5 gpm), was approximately 455 K (360^{o} F).

At an inner-race cooling flow of C_i equal 1.33 L_i shown in Fig. 9(d), the minimum inner-race temperature was approximately 458 K (365^{o} F). The minimum temperature at the outer race was obtained at an outer-race cooling flow (C_o) of $5.7x10^{-3}$ cubic meters per minute (1.5 gpm). This temperature was approximately 447 K (345^{o} F). At a value of L_i in excess of $1.5x10^{-3}$ cubic meters per minute (0.4 gpm), temperature at the inner race began to increase. Likewise, the temperatures at the outer race which parallel those of the inner race, except at an outer-race flow rate of 9.5×10^{-4} cubic meters per minute (0.25 gpm), began to increase. For a lubricant inner-race flow C_i equal to 3.0 L_i (Fig. 9(e)) the range of component temperatures was very narrow, decreasing from 460 K (370° F) at a flow rate of L_i equal to 7.6×10^{-4} cubic meters per minute (0.2 gpm) to 350° F at L_i equal to 1.5×10^{-3} cubic meters per minute (0.4 gpm). Beyond this point, temperature rapidly increased at the inner race. The optimum matching of inner- to outer-race temperatures was obtained with outer-race lubricant cooling rates (C_o) ranging between 3.8×10^{-3} to 5.7×10^{-3} cubic meters per minute (1 to 1.5 gpm).

A summary of the inner-race temperatures is shown in Fig. 9(f). 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 comparable operating conditions. As the inner-race cooling flow (C_i) increased, temperatures decreased significantly. The maximum difference in temperature 50 K (90 F) was observed between C_i equal to 3.0 L_i and C_i equals zero. In the case where no cooling was attempted of the inner race except for that provided by the inner-race lubricant flow (L_i), most of this difference can be accounted for by the fact that a major cause of the heat generation within the bearing is likely due to churning of the lubricant within the bearing cavity.

Power loss as a function of inner-race cooling (C_i) and lubrication (L_i) is presented in Fig. 10. The power loss increases linearly with increased flow to the inner race. For outer-race cooling flows from 9.5×10^{-4} to 3.8×10^{-3} cubic meters per minute (0.25 to 1.0 gpm) there was essentially no significant difference in power loss. At an outer-race cooling flow (C_o) of 5.7×10^{-3} cubic meters per minute (1.5 gpm) the power loss was approximately 0.75 to 3 kW (1 to 4 horsepower) greater than at the lower values of C_o. This difference in power consumption can be attributed to the reduction in bearing internal clearances due to temperature differences between the inner and outer races. The rate of increase in power loss with L_i is due to both the amount of oil used for the inner-race cooling flow (C_i) and that used for the primary lubricant flow, L_i.

The rapid increase in power consumption with increased C_i can be attributed to a larger quantity of lubricant being entrapped within the bearing cavity at the higher lubricant flow rates. For all values of C_o, the extrapolated value of power loss where L_i was zero was in the range of 9 to 12 kW (12 to 16 horsepower).

Referring to Fig. 10(f) which is a summary of the range of power loss for the 20° and 24° contact angles, there appears to be no significant difference between the power loss with either of these contact angles for a particular inner-race cooling flow (C_i). This is expected since power loss increases when the amount and viscosity of oil within the bearing cavity is increased. It must be recognized, however, that all power loss data presented herein are based solely on shaft horsepower measurements. As such, they do not necessarily encompass the power requirement for pumping the lubricant, and, more specifically, for circulating the oil used to cool the outer race.

The above results indicate that the bearing can be temperature and power tuned to any specific operating condition depending upon the lubricant characteristics. The concept of Bearing Thermal Management proposed herein, 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.

Bearing endurance at 3 million DN. - Preliminary bearing endurance tests were conducted with the 24^{o} contact-angle bearing at a speed of 25 000 rpm (3 million DN) and a thrust load of 22241 N (5000 lb). Under these conditions the maximum Hertz stresses in the inner and outer races are 1965×10^{6} and 2096×10^{6} N/m^2 (285 000 and 304 000 psi), respectively. For these long-time, high-speed bearing tests the cooling-flow rate per bearing to the outer race (C_o) was 2.8×10^{-3} cubic meters per minute (0.75 gpm). Lubricant flow to the inner race (L_i) was 1.3×10^{-3} cubic meters per minute (0.35 gpm) and inner-race cooling flow (C_i) was approximately 3.9×10^{-3} cubic meters per minute (1 gpm).

The standard AFBMA (catalog)-life calculation predicts a bearing ten-percent (B_{10}) life under these operating conditions of approximately 16 hours. (The bearing 10-percent (B_{10}) life is the operating time (life) at which 90% of a group of bearings will survive.) However, using the material, lubricant and speed factors given in [17] a B_{10} life of about 175 hours would be a more reasonable prediction. Of the ten bearings initially tested all ran for 1000 hours without failure. These results show that long-term bearing operation at 3 million DN can be achieved with a high degree of reliability using sophisticated but currently available state-of-the-art bearing materials and designs, lubricants, and lubrication techniques.

SUMMARY

A parametric study was performed with 120-mm bore angular-contact ball bearings having nominal 20° and 24° contact angles under varying thrust load, bearing and lubricant temperature, and cooling and lubricant flow rates at speeds to 3 million DN. Endurance tests were run at 3 million DN and a temperature of 492 K (425° F) with 10 bearings having a nominal 24° contact angle at a thrust load of 22241 N (5000 lb) producing a maximum Hertz stress of 1965×10^{6} and 2096×10^{6} N/m^{2} (285 000 and 304 000 psi) on the bearing inner and outer races, respectively. The following results were obtained:

1. Bearing inner- and outer-race temperatures and power consumption were found to vary with load, speed, lubricant flow rate into the bearing, and lubricant cooling to the inner race. Lubricant cooling flow to the outer race was found to affect outer-race temperatures significantly, but had only a small effect on the measured inner-race temperature. Power loss due to change in lubricant cooling flow to the outer race was relatively insignificant.

2. Bearing operating temperature, differences in temperatures between the inner and outer races, and bearing power consumption can be

tuned to any desirable operating requirement by varying 4 parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race and oil inlet temperature.

3. All ten bearings which were endurance tested ran for times in excess of 1000 hours without failure. These results indicate that long-term bearing operation at 3 million DN can be achieved with a high degree of reliability using sophisticated but currently available state-of-the-art bearing materials, designs, lubricants, and lubrication techniques.

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TABLE 1 - CHEMICAL ANALYSIS OF VACUUM INDUCTION, CONSUMABLE-ELECTRODE VACUUM REMELTED AISI M-50 BEARING STEEL

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

TABLE 2 - PROPERTIES OF TETRAESTER LUBRICANT

Additives	Antiwear Oxidation Inhibitor Antifoam
Kinematic viscosity, cS, at - 311 K (100° F) 372 K (210° F) 477 K (400° F)	 28.5 5.22 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

TABLE 3 - CALCULATED OPERATING CONTACT ANGLES AND STRESSES AS A FUNCTION OF INITIAL CONTACT ANGLE, SPEED AND LOAD

Speed, rpm	Thrust load, N (lb)	Contact angle, deg			Maximum Hertz stress, N/m^2 (ksi)	
		Unloaded	Operating		Outer race	Inner race
			Outer race	Inner race		
12 000	6672 (1500)	20	13	26	1365×10^6 (198)	1420×10^6 (206)
	13345 (3000)		18	26	1572 (228)	1806 (262)
	22241 (5000)		21	27	1779 (258)	2130 (309)
	6672 (1500)	24	14	30	1344 (195)	1351 (196)
	3345 (3000)		19	30	1531 (222)	1731 (251)
	22241 (5000)		23	31	1731 (251)	2048 (297)
16 000	6672 (1500)	20	9	27	1517×10^6 (220)	1400×10^6 (203)
	3345 (3000)		14	28	1682 (244)	1779 (258)
	22241 (5000)		18	28	1855 (269)	2096 (304)
	6672 (1500)	24	9	32	1503 (218)	1330 (193)
	3345 (3000)		15	32	1648 (239)	1703 (247)
	22241 (5000)		19	32	1813 (263)	2020 (293)
20 000	6672 (1500)	20	6	28	1682×10^6 (244)	1386×10^6 (201)
	3345 (3000)		11	29	1813 (263)	1758 (255)
	22241 (5000)		15	29	1958 (284)	2068 (300)
	6672 (1500)	24	6	33	1682 (244)	1324 (192)
	3345 (3000)		12	33	1923 (260)	1682 (244)
	22241 (5000)		16	33	1924 (279)	1993 (289)
25 000	6672 (1500)	20	5	29	1889×10^6 (274)	1365×10^6 (198)
	3345 (3000)		8	30	1993 (289)	1731 (251)
	22241 (5000)		12	30	2110 (306)	2041 (296)
	6672 (1500)	24	5	34	1862 (270)	1310 (190)
	3345 (3000)		8	34	1993 (289)	1662 (241)
	22241 (5000)		12	34	2096 (304)	1965 (285)

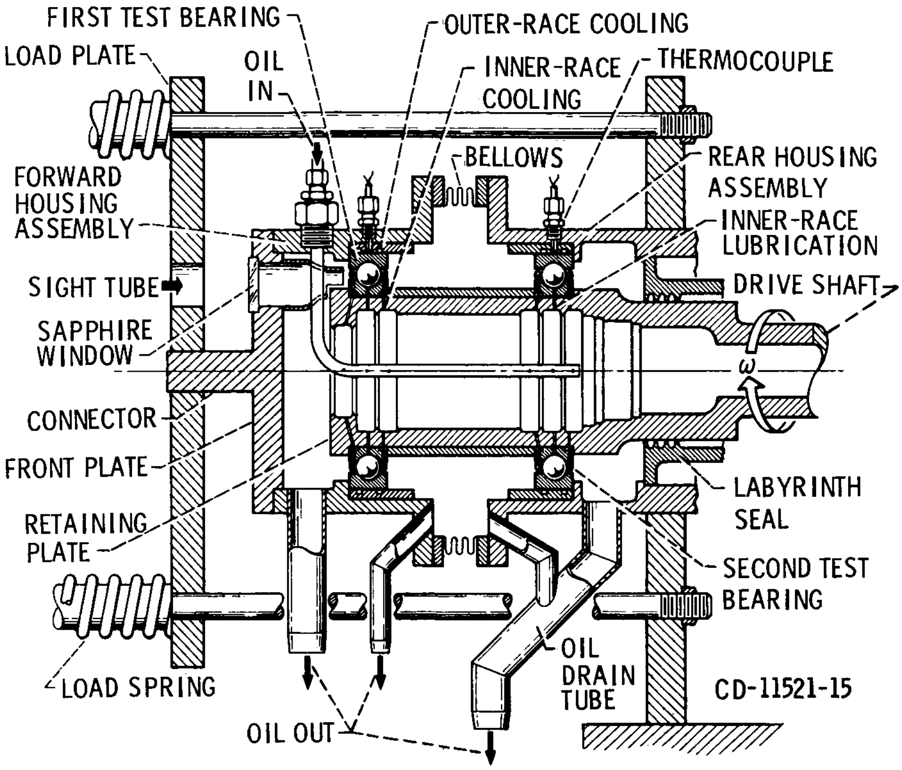

Fig. 1 High-speed, high-temperature bearing test apparatus.

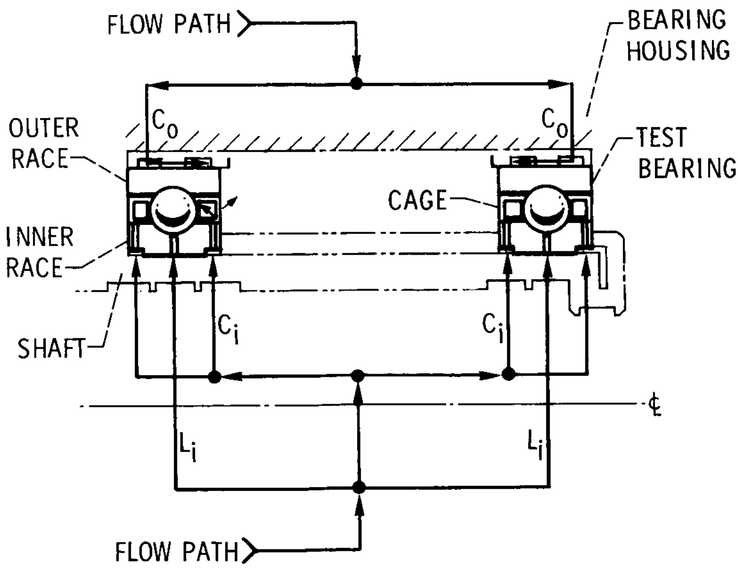

Fig. 2 Lubricant system for test bearings.

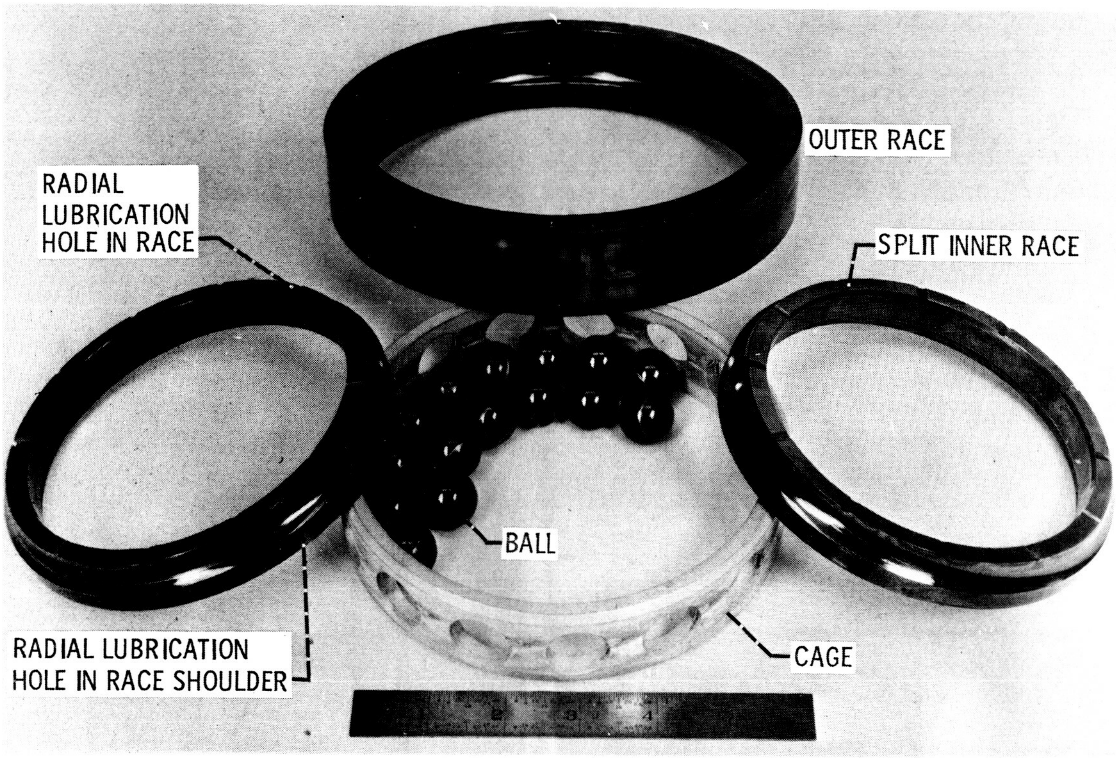

Fig. 3 Unfailed 120-mm bore angular-contact high-speed test ball bearing. Running time, 1000 hours; speed, 25 000 rpm (3 million DN); temperature, 492 K (425^{0} F); thrust load, 22 241 N (5000 lb).

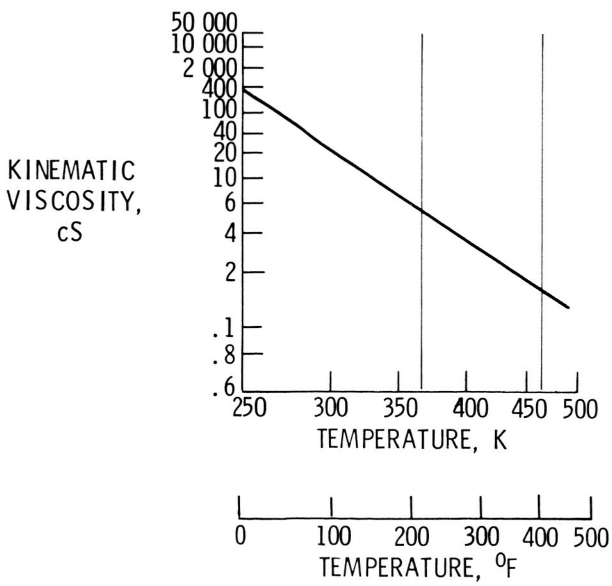

Fig. 4 Viscosity as function of temperature for tetra-ester (Type II) lubricant.

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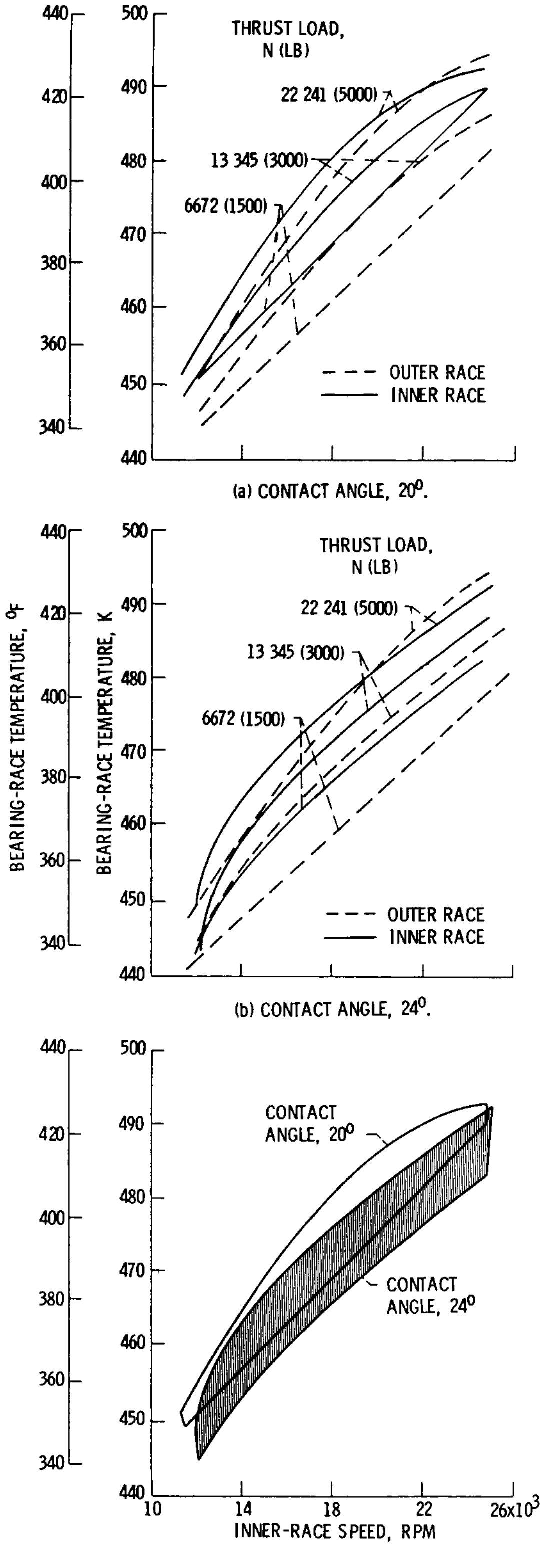


Fig. 5 Range of bearing race temperature as a function of speed for various thrust loads. Bearing type, 120-mm bore angular-contact ball bearing; lubricant flow to inner race, L_i, 1.2×10^{-3} m^3/min (0.313 gpm); inner-race cooling flow, C_i, 3.6×10^{-3} m^3/min (0.94 gpm); outer-race cooling flow, C_o, 1.9×10^{-3} m^3/min (0.5 gpm); oil inlet temperature, 428 K (310° F).

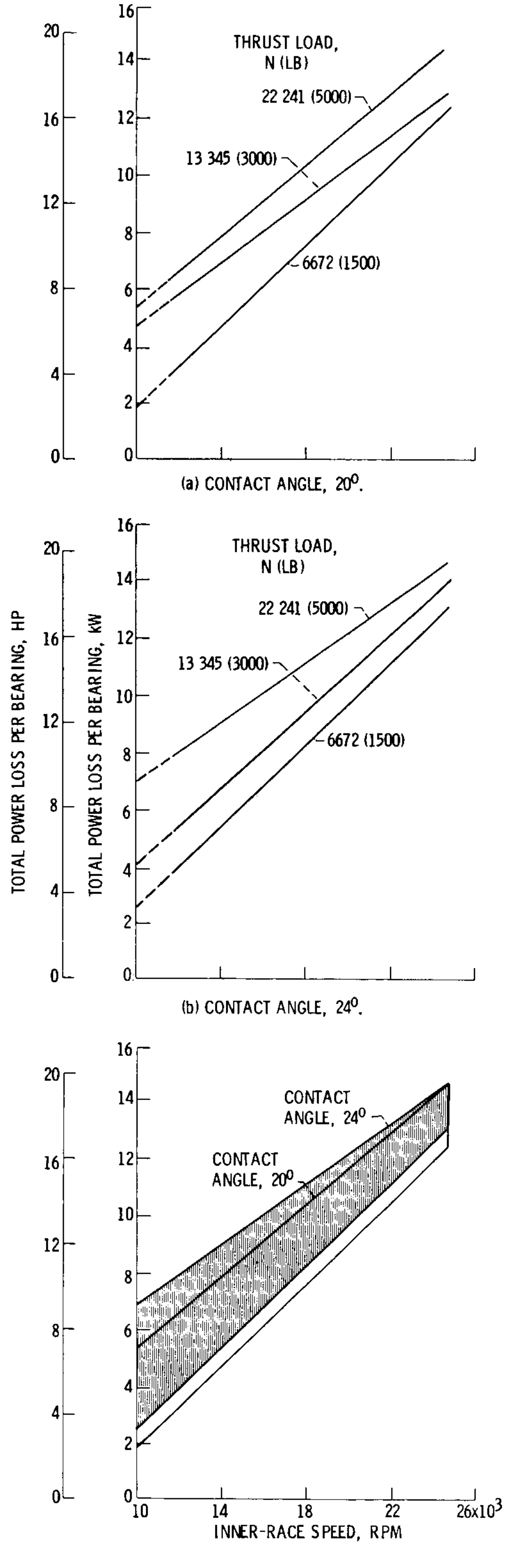


Fig. 6 Range of power loss as a function of speed for varying thrust loads. Bearing type, 120-mm bore angular-contact ball bearing; lubricant flow to inner race, L_i, 1.2×10^{-3} m^3/min (0.313 gpm); inner-race cooling flow, C_i, 3.6×10^{-3} m^3/min (0.94 gpm); outer-race cooling flow, C_o, 1.9×10^{-3}/min (0.5 gpm); oil inlet temperature, 428 K (310° F).

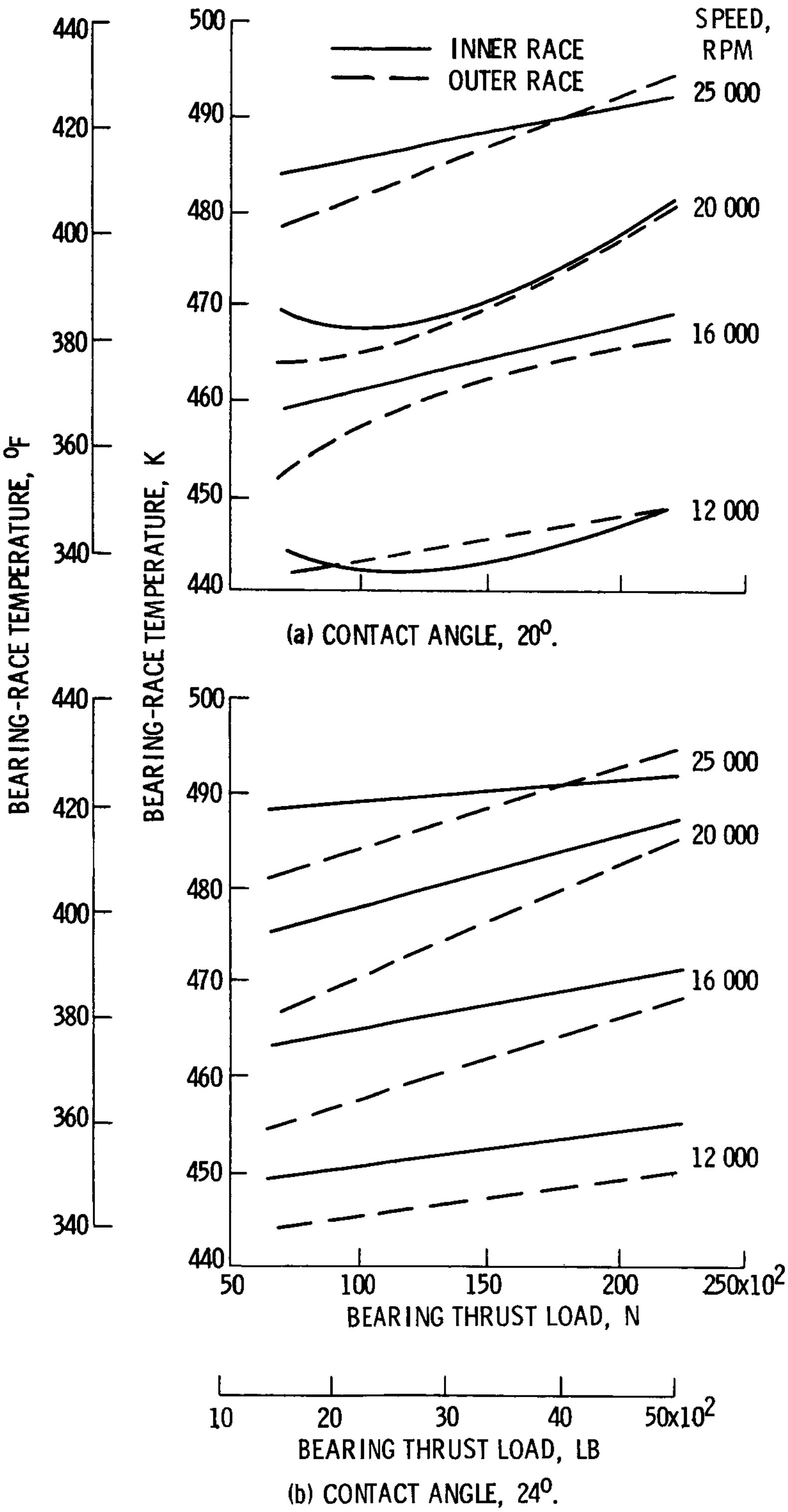
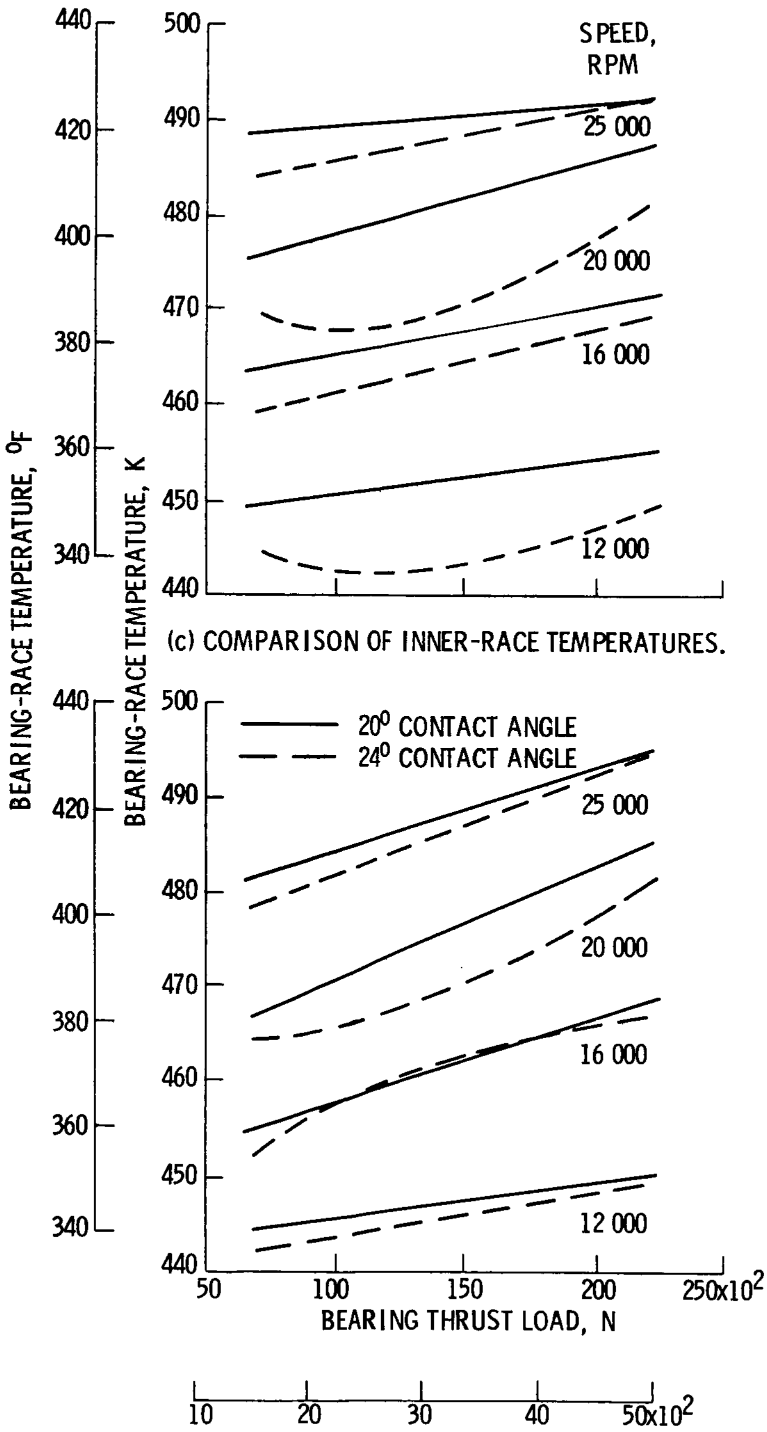


Fig. 7 Range of bearing race temperature as a function of bearing thrust load for various speeds. Bearing type, 120-mm bore angular-contact ball bearing; lubricant flow to inner race, L_i, 1.2×10^{-3} m^3/min (0.313 gpm); inner-race cooling flow, C_i, 3.6×10^{-3} m^3/min (0.94 gpm); outer-race cooling flow, C_o, 1.9×10^{-3} m^3/min (0.5 gpm); oil inlet temperature, 428 K (310° F).



(c) COMPARISON OF INNER-RACE TEMPERATURES.

(d) COMPARISON OF OUTER-RACE TEMPERATURES.

Fig. 7 Concluded.

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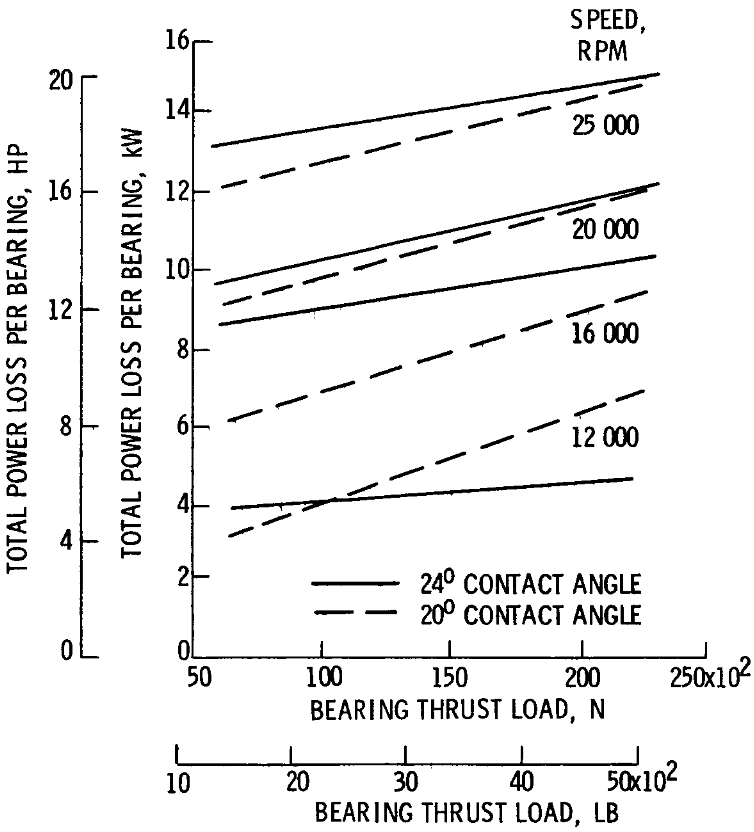

Fig. 8 Bearing power loss as a function of bearing thrust load for various speeds. Bearing type, 120-mm bore angular-contact ball bearing; lubricant flow to inner race, L_i, 1.2×10^3 m^3/min (0.313 gpm); inner-race cooling flow, C_i, 3.6×10^{-3} m^3/min (0.94 gpm); outer-race cooling flow, C_o, 1.9×10^{-3} m^3/min (0.5 gpm); oil inlet temperature, 428 K (310° F).

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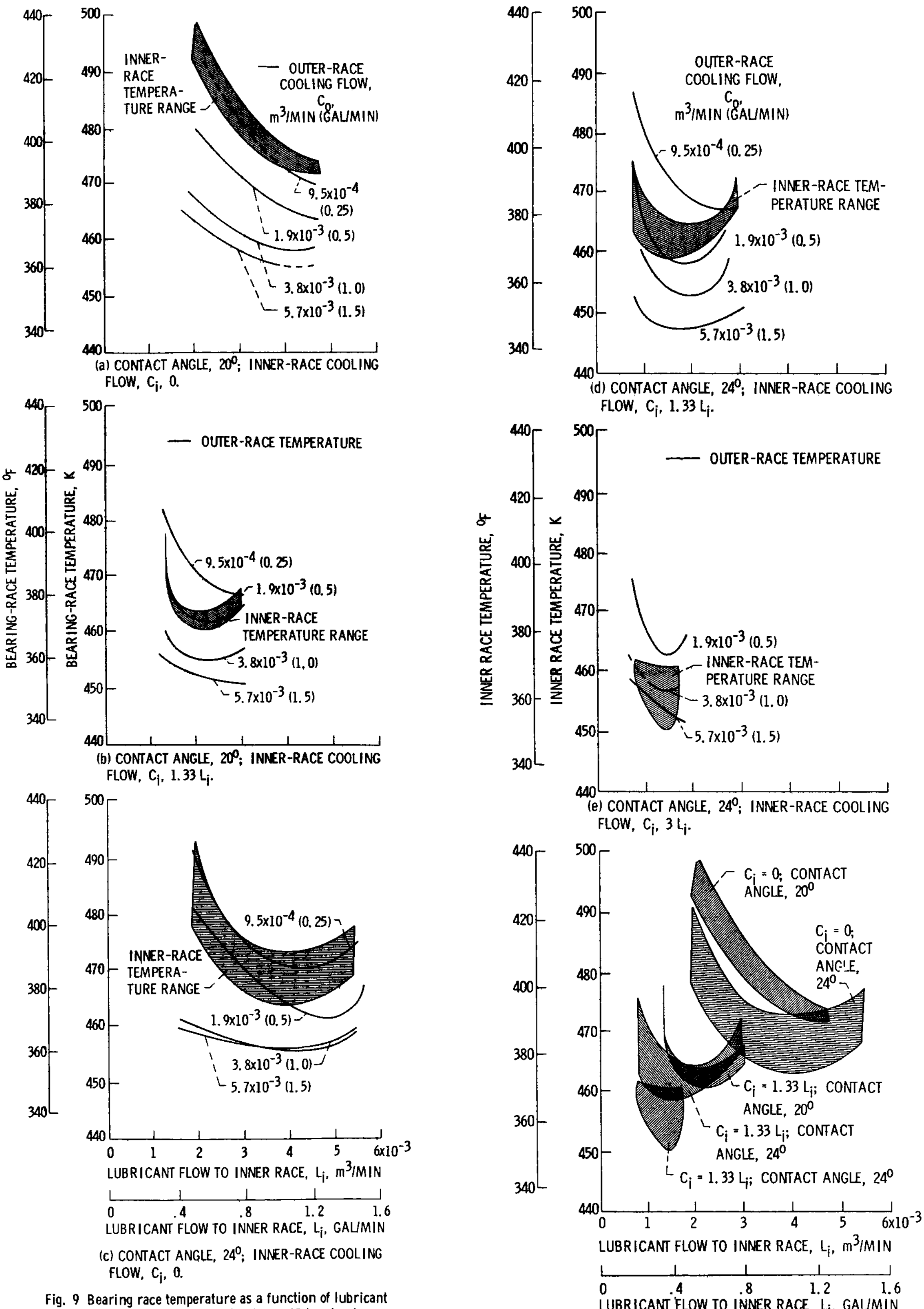

(a) CONTACT ANGLE, 20°; INNER-RACE COOLING FLOW, C_i, 0.

(b) CONTACT ANGLE, 20°; INNER-RACE COOLING FLOW, C_i, 1.33 L_i.

(c) CONTACT ANGLE, 24°; INNER-RACE COOLING FLOW, C_i, 0.

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

(d) CONTACT ANGLE, 24°; INNER-RACE COOLING FLOW, C_i, 1.33 L_i.

(e) CONTACT ANGLE, 24°; INNER-RACE COOLING FLOW, C_i, 3 L_i.

(f) SUMMARY OF INNER-RACE TEMPERATURES.

Fig. 9 Concluded.

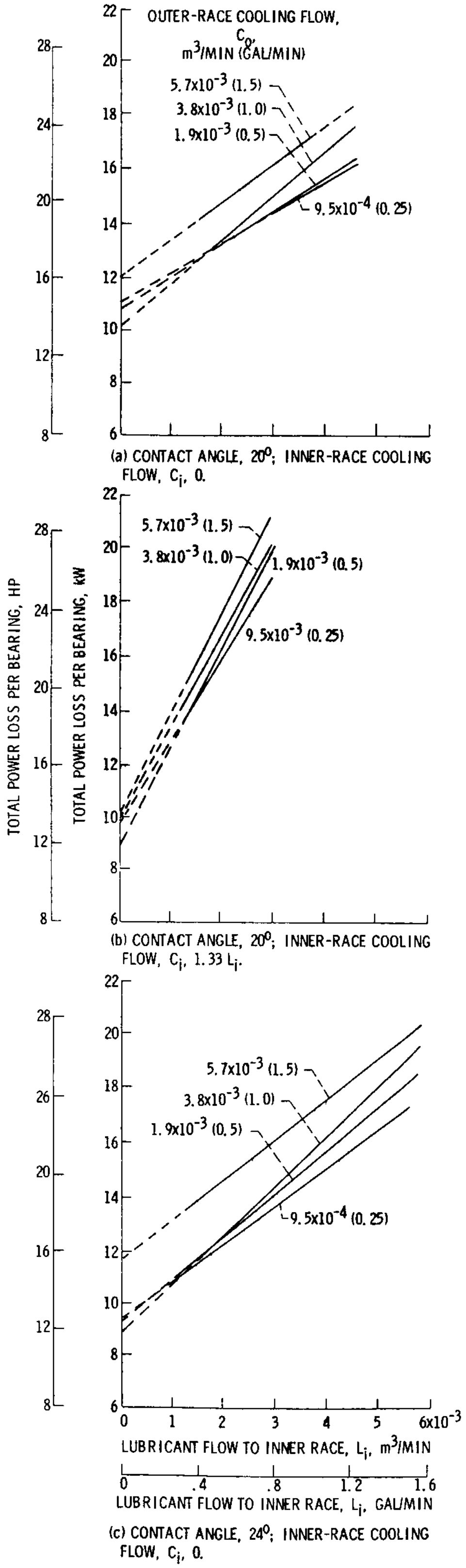

Fig. 10 Bearing power loss as a function of lubricant flow into bearing, L_i, for varying inner (C_i) and outer (C_o) race cooling rates. Bearing type, 120-mm bore angular-contact ball bearing; bearing thrust load, 22 241 N (5000 lb); speed, 25 000 rpm (3×10^6 DN); oil inlet temperature, 394 K (250°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

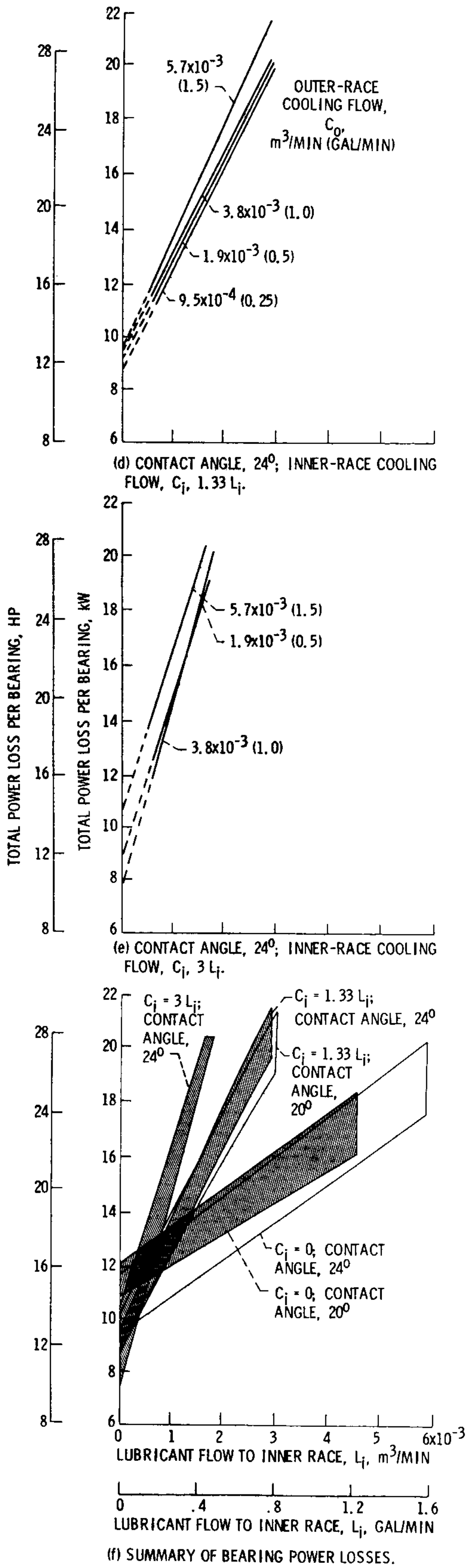

Fig. 10 Concluded.