PREPRINT NO. 85-AM-4E-3
ASLE PREPRINT

Performance Simulation of a Solid-Lubricated Ball Bearing©

HOWARD E. BANDOW (Member, ASLE)
Air Force Wright Aeronautical Laboratories
Wright-Patterson Air Force Base, Ohio 45433

STANLEY E. GRAY
Mechanical Technology Incorporated, Latham, New York 12110

PRADEEP K. GUPTA
PKG Incorporated, Clifton Park, New York 12065

Fundamental friction and wear experiments are undertaken for the prescribed materials of a solid-lubricated ball bearing for a high-speed turbine engine. Based on the experimental friction data, appropriate traction models for the ball/race, ball/cage, and cage/race interactions are derived. These traction models are input to the bearing dynamics computer program, ADORE, to obtain the dynamic performance simulation of the bearing under the expected conditions of operation. The computer simulations indicate excessive ball/cage and cage/race interactions, leading to a possible instability and an ultimate cage failure. The predicted failure mode agrees with the experimental observations of the bearing performance under similar conditions of operation.

INTRODUCTION

Solid-lubricated rolling-element bearings are being developed for many applications, including those where temperatures are too high for liquid lubricants to function. In these applications, novel bearing materials, notably ceramics, are also employed to increase the temperature capability beyond that which metals can sustain. This paper describes a study of the performance of a bearing developed for application in turbine engines. The bearing consisted of steel races, ceramic balls, and a cage constructed of woven graphite fibers in a polyimide matrix with the solid lubricant blended in the matrix material.

The objective of this effort was to obtain an understanding of how the tribological properties of the solid lubricant and the materials used for the various bearing elements influence the overall bearing performance. Models describing the traction forces in sliding contact between surfaces were formulated from experimental data gathered in the program and input to a computer program designed to simulate the dynamic performance of rolling-element bearings. The computer predictions are then compared to the experimentally observed performance of test bearings to demonstrate the value of the combination of experimental materials tests and analytical simulation of bearing performance for development of future bearings.

The performance of rolling-element bearings is greatly influenced by the friction levels between interacting bearing elements. The contacts between the balls and races are often subjected to very high loads and rolling speeds while the sliding speeds, although a great deal smaller than the rolling speeds, vary significantly as a function of the ball/cage and cage/race collisions. The contacts between the balls and the cage and between the cage and the race are intermittent in nature with forces being nominally zero but possibly very high during contact. Also, since these elements interact in a pure sliding mode, the sliding velocities are generally very high.

The present study is directed toward a developmental solid-lubricated ball bearing for high-temperature turbine engine applications, where the material of the races is M-50 tool steel, the balls are ceramic (NC-132, hot-pressed silicon nitride), and the cage is made of a polyimide composite material impregnated with the solid lubricant. Thus, the lubricant at the ball/race interface is in the form of a transfer film formed by the solid lubricant released from the cage as a function of the ball/cage collisions. Since the friction forces, generated as a function of the complex thermomechanical interaction between the various bearing elements, would have a profound effect on the bearing performance, the development of a realistic model for the frictional interactions at the ball/race, ball/cage, and cage/race contacts is crucial.

The approach taken in the present study consists of three steps: (1) experimental studies to understand the tribological behavior of individual contacts in the bearing, (2) input the experimentally derived friction behavior to computer models and obtain analytical simulation of bearing performance, and (3) evaluate the computer predictions in the light of experimentally observed bearing performance. For the selected bearing materials, the fundamental friction and

Presented at the 40th Annual Meeting
in Las Vegas, Nevada
May 6–9, 1985
Final manuscript approved February 28, 1985

wear experiments were first conducted over the expected range of operating conditions, and the friction or traction behavior at the ball/race, ball/cage, and cage/race interfaces in the bearing modeled. These traction models were then used as input to the bearing dynamics computer program ADORE (*1*) (Advanced Dynamics Of Rolling Elements). The bearing performance simulations were then generated over the expected conditions of operation. The computer predictions so obtained were compared to actual bearing performance data to establish the validity of the modeling procedure and prove the overall feasibility of the developmental approach.

EXPERIMENTAL

Fundamental friction and wear experiments were undertaken to model the behavior at the ball/race, ball/cage, and cage/race interactions. The behavior at the ball/race contact is simulated by a ceramic (NC-132 hot-pressed silicon nitride) ball sliding against the circumference of a disk of M-50 tool steel, where the lubricant transfer is provided by rubbing the cage material against the disk. The ball/cage interaction is modeled by a high-speed sliding experiment where the cage material slides against a NC-132 ceramic disk. Similarly, the cage/race contact is simulated by an experiment where the cage material slider rubs against a M-50 disk. The description of tests of additional material combinations completed as part of the program is omitted here for brevity but described elsewhere (*2*).

Test Materials and Specimens

The materials chosen for the study are expected to be suitable for high-temperature rolling-element bearings. Disk specimens of M-50 tool steel were prepared to represent the bearing races, while a set of NC-132 hot-pressed silicon nitride, as manufactured for the test bearing, were obtained for use as ball specimens. For the high-speed ball/cage sliding tests, the NC-132 disk specimens were made from the same material from which the bearing balls were constructed. Some typical physical and thermal properties of these materials are given in Table 1. The balls were 5.56 mm (7/32 in) diameter and ground to precision bearing tolerances and finishes. The disks were 50.8 mm (2 in) diameter by 12.7 mm (0.5 in) wide with a 15.24 mm (0.6 in) diameter bore for mounting purposes; roundness, squareness, concentricity and flatness tolerances were within 5.0 um (0.0002 in) TIR and the outer diameter was within 0.1 um (4 min) surface finish.

Two composite materials were chosen to be representative of the cage material of the bearing (*4*). The first of these materials was designated HAC-1/T50 F1 and consisted of 50.1 percent woven graphite fiber and 41.3 percent polyimide matrix material. Blended with the matrix material were a 2.2 percent gallium, indium, tungsten diselinide mixture and 1.4 percent (NH_4) $2HPO_4$. The material contained 4 percent by volume of voids. The second material, designated HAC-1A/T300 F1 was of similar composition but contained approximately double the volume of lubricant material.

Test Setup

The experimental device used to conduct the tests is illustrated in Fig. 1. It is based upon a horizontally mounted precision spindle supported the oil-lubricated ball bearings which can be driven at the remote end by an integral air turbine for high-speed operation or by an electric motor for low-speed operation. The front end of the spindle is water cooled to protect the support bearings from excessive operating temperatures. A clam-shell-type electric furnace containing an array of quartz heater rods was placed around the test section to provide ambient temperatures to 700°C (1300°F).

A disk-shaped test specimen is mounted on the support spindle, as shown in Fig. 2. Some special features are incorporated in the design to allow for the extreme temperatures in the test area and the differences in thermal expansion coefficient of the disk materials: (1) by using materials of progressively lower coefficients of thermal expansion near the end of the shaft, (2) by use of a tie bolt to absorb thermal stresses in the axial direction, (3) by selecting the bore of the test specimens to give the best fit over the temperature range of the tests, and (4) by design of a special sleeve with stress relieving slots over which the test specimens were mounted; a mounting was achieved which minimized loss of concentricity at high speeds and with which no cracking of the ceramic test disks due to thermal stress was experienced.

The static test specimen, mounted on the end of the loader arm, makes sliding contact with the periphery of the disk. The other end of the loader arm is mounted on a simple universal joint utilizing instrument-size ball bearings to minimize friction forces. The slider is loaded against the periphery of the disk by dead weight and the friction is measured with a strain gauge load cell attached to the arm by a rigid, self-aligning linkage.

TABLE 1—TYPICAL PHYSICAL AND THERMAL PROPERTIES OF BALL AND DISK MATERIALS (*3*)

Material	Hardness at 20°C Rc	Elastic Modulus at 20°C GPa (10^6 PSI)	Thermal Conductivity Cal/s m °C 20°C	Thermal Conductivity Cal/s m °C 800°C	Coefficient of Thermal Expansion 10^{-6}/°C 0–800°C
M-50 Tool Steel	64	190 (28)	13.4	—	12.3 (300°C)
Silicon Nitride (NC-132)	78	310 (45)	7.3	4.7	2.9

650 °C BALL-ON-DISC FRICTION AND WEAR TESTER

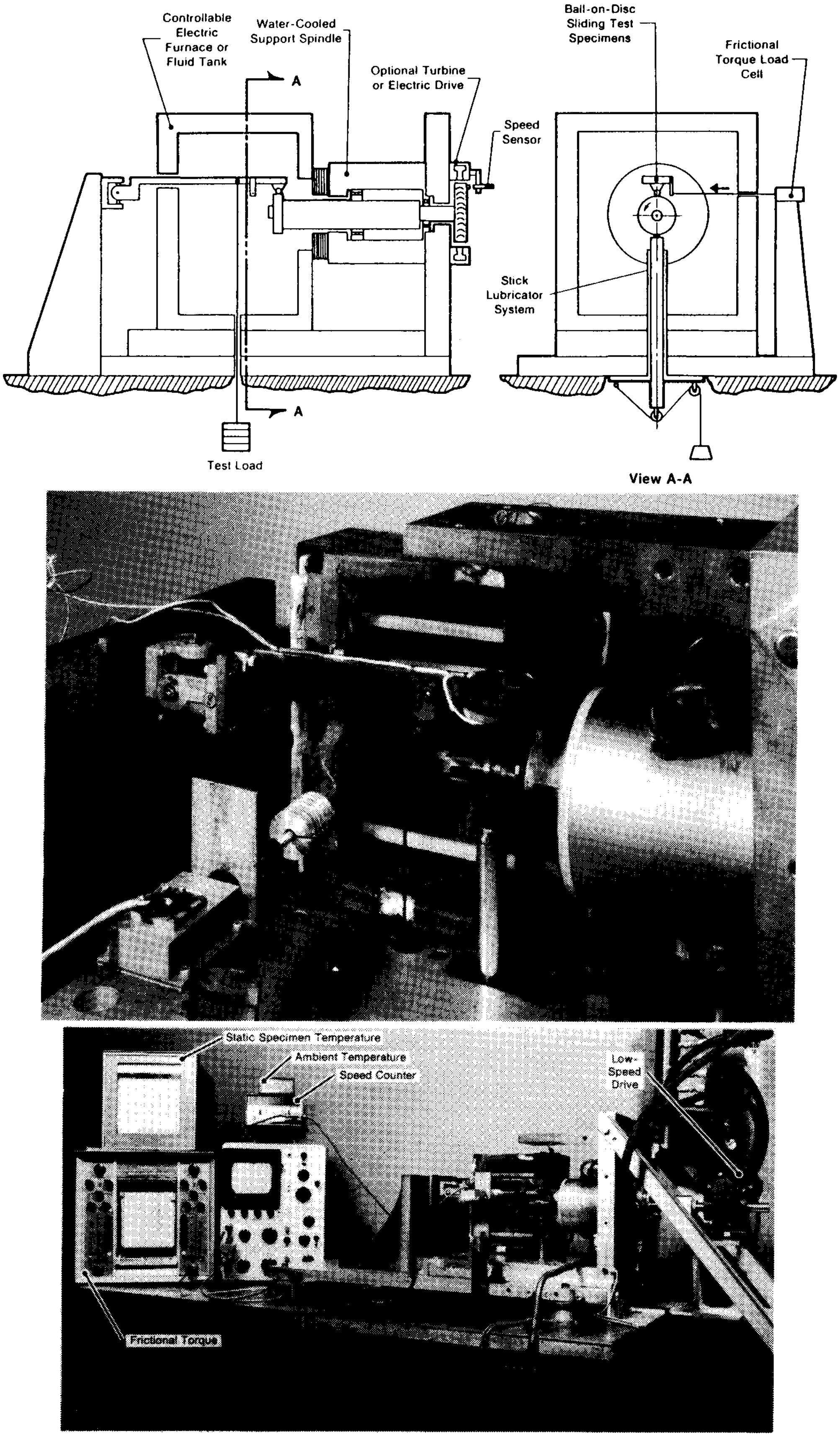


Fig. 1—Continuous sliding ball on disk material tester
(a) Schematic diagram
(b) Photograph of test section
(c) Photograph of rig and instrumentation

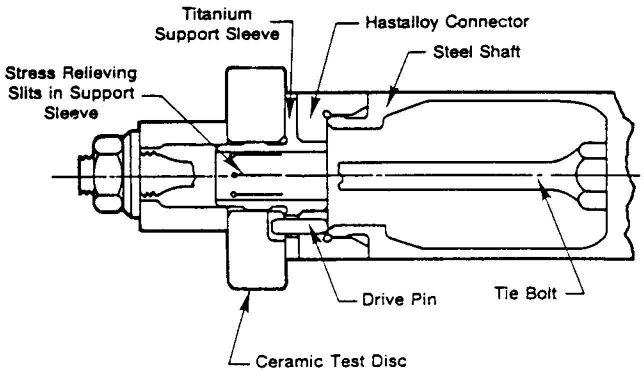


Fig. 2—Mounting of ceramic disk on metal shaft

Simple brackets were designed to hold either ball-shaped sliders or prismatic sliders of rectangular cross section, of the cage material, on the end of the loader arm. When balls were used as the sliders, a block of the solid lubricant containing cage material was loaded against the disk rim surface to provide a source of lubricant for transfer to the contact between the disk and slider.

Instrumentation was provided to measure temperatures, sliding speed, and frictional force. Chromel-alumel thermocouples were mounted on the slider and in the furnace chamber with outputs displayed on a strip chart recorder and on digital readouts. A fiber-optical sensor was directed at a black and white segmented target on the turbine wheel or motor shaft and connected to a frequency counter with digital readout to indicate the sliding speed. The output from a strain gauge load cell was recorded on a strip chart recorder to provide a record of the frictional force.

In order for test results to provide legitimate data for the formulation of traction models to simulate bearing performance, the test conditions were selected to be representative of conditions in the ball/race, the ball/cage, and the cage/race contacts. Because conditions are so different in these three types of contact, separate tests were conducted to simulate each type of contact. The ball/race contact is characterized by high Hertzian stresses of a continuous nature and relatively low sliding velocities which may vary greatly in magnitude with time. The ball/cage and cage/race contacts are intermittent in nature with the stresses being nominally zero but becoming relatively high during brief transients when collisions between bearing elements occur. Because of the different material pairs involved, different tests were conducted to generate data for each of those types of contact. Table 2 shows test conditions for the tests discussed here.

Prior to performing friction and wear tests, the disk specimen was mounted onto the spindle sleeve and checked to assure that radial runout was less than 7.6 um (0.0003 in) TIR. The spindle was then run up to speed at room temperature and the runout and tiebolt tightness were rechecked. The procedure was then repeated at the maximum test temperature. Following this, the load arm with the slider to be used in the test was mounted and carefully adjusted so that the slider and the disk were properly aligned. The torque measuring system was statically calibrated before and after each test by lifting it just clear of the disk on a thin thread and then applying a series of known loads while the output of the load cell was recorded on a strip chart recorder, creating a stepped calibration plot.

Sliders of the cage material were dimensionally checked using a precision micrometer before and after each test to determine wear. The diameter of the wear scar on ball sliders was measured after each test using an optical micrometer attachment on a microscope to determine wear and the contact stress at the conclusion of the test.

Due to limited availability of ball and disk test specimens and especially cage material, the test procedures were planned to make maximum utility of these materials. Continued usage of the same specimen pairs during a series of load or speed changes was sometimes made. This was made possible by realigning the specimens so that a new wear track was utilized or sometimes refinishing specimen surfaces.

Test Conditions

The initial series of tests were performed to simulate the cage/race contacts and consisted of a slider made of the cage material loaded against the M-50 disk. A series of constant speed tests were made at 5000, 10 000, 20 000, and 30 000

Table 2—Summary of Test Conditions

Test Parameter		Material Test—Pairs		
		M-50 Steel vs HAC-1*	Silicon Nitride NC-132 vs HAC-1*	Silicon Nitride NC-132 vs M-50 Steel†
SPEED	rpm cm/s	0–30 000 0–7980	0–30 000 0–7980	0–50 0–13.3
AMBIENT TEMPERATURE	°C °F	204 R.T., 400‡	204, 316 R.T., 400, 600	204 R.T., 600
CONTACT STRESS	MPa Kpsi	0.69 to 20.7 0.1 to 3	0.69 to 20.7 0.1 to 3	690 to 1725 100 to 250§

NOTES: * HAC-1/T50 Fl and HAC-1A/T300 Fl.
† Lubricated by stick lubricant transfer.
‡ Temperatures limited because of bond to metal strength of HAC materials.
§ Initial contact stresses.

rpm at each of several constant loads and at room temperature. Termination of operation at each speed level was based on torque smoothness, self-generated temperature rise and wear rate. Typical test times were in the range of 4 to 8 minutes at each speed level. Based on experience with room temperature tests, an appropriate loading was selected for elevated temperature tests. For M-50 tests, the temperature was restricted to 204°C (400°F) because of concern with the strength of the epoxy bond between the lubricant material and its mounting block. At elevated temperatures, a continuous, but slow, sweep was made through the speed range to 30 000 rpm over a few minutes period with friction and temperature data taken during a brief dwell interval at selected speed values. The ball/cage interaction was simulated by testing with cage material sliding on a disk made of silicon nitride was performed in a similar manner, with the exception that elevated temperature testing was performed both at 204°C and 316°C (600°F).

To approximate conditions in the ball/race contacts, tests were conducted in which a ball-shaped slider was loaded against the disk. Lubrication was provided to the contact by transfer from a stick of the lubricant-impregnated cage material which was loaded against the rim of the disk with a load of 13 N (3 lbf), which represents a typical ball/cage collision force. Sliding velocity was varied through a range of 0–50 rpm under a series of constant loads at room temperature and at 316°C. The test was allowed to run for approximately one minute at each speed before taking data to allow conditions to stablize. Before beginning to test at a new temperature level, the load arm was adjusted so that the slider would be in a new axial position on the disk and the disk was prelubricated by stick transfer for one hour.

Test Results

Slider on Disk

The results of room temperature friction tests performed with a slider of HAC-1/T50 F1 cage material against a M-50 tool steel disk at four speeds and with a series of increasing loads are shown in Fig. 3. If specimen temperatures became excessive, or if drive spindle speed or frictional torque became erratic at any speed, no higher load was applied at that speed. The data indicate that under these conditions the loading which could be sustained by the cage material slider was limited to 3.4 MPa (500 psi) in the 5000 to 10 000 rpm range and less than 0.34 MPa (50 psi) at 30 000 rpm. The magnitude of the friction coefficient shows an increase with speed from a typical value of 0.3 at 5000 rpm to 0.6 or 0.8 at 30 000 rpm. With increasing speed, the torque became increasingly erratic. With the increasing friction forces at higher speeds, the temperature measured on the slider mounting block became higher, reaching 166°C (330°F) at the end of the 30 000-rpm test.

Examination of the slider and disk surfaces, shown in Fig. 4, at the conclusion of the test showed evidence that a great deal of wear had taken place on the slider. Gridlike lines, formed by graphite fibers which have been exposed at the surface due to the depth of wear, may be seen on the wear surface of the slider. The circumferential lines seen on the surface of the disk appeared to be cage material and pro-

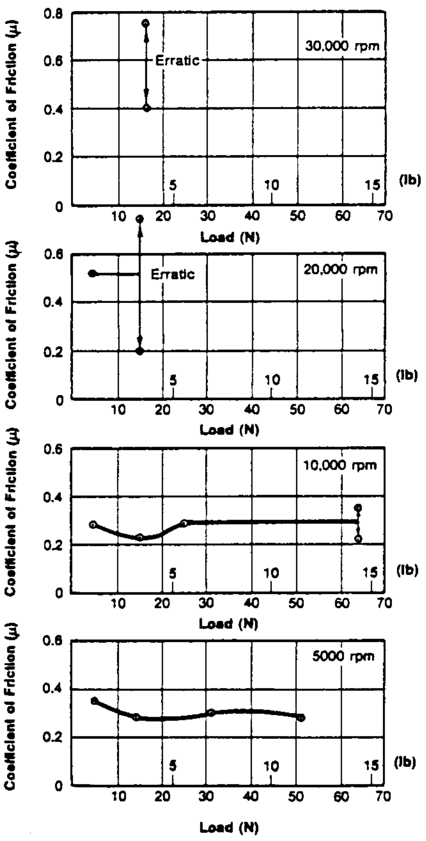


Fig. 3—High-speed data, HAC-1/T50 F1 vs M-50, room temperature

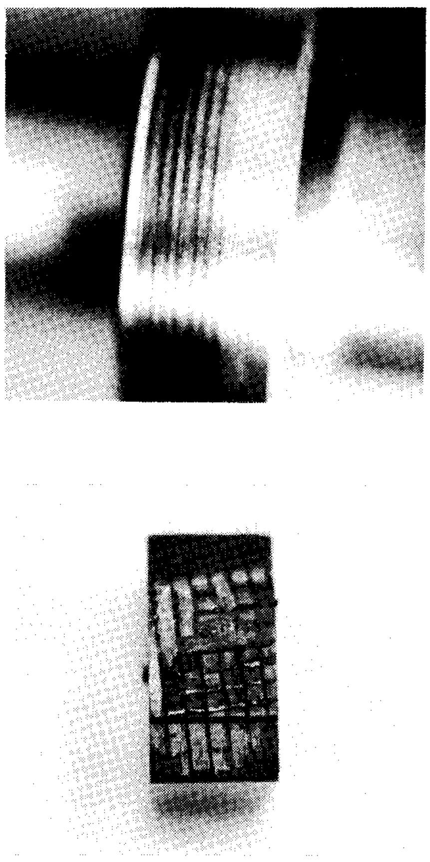


Fig. 4—M-50 disk and HAC-1/T50 F1 specimen after 30 000 rpm test at room temperature.

vided evidence of lubricant transfer to the disk surface. During the accumulated 23 minutes of operation in these tests, the average wear rate of the slider was calculated to be 0.492 mm^3/min (3×10^{-5} in^3/min).

A test at elevated temperature was performed with a different slider of HAC-1/T50 F1 and the same disk of M-50 tool steel, which was first cleaned by wiping with carbon tetrachloride fluid. During this test, a load of 5.1 N (1.15 lb) was applied to the slider. Speed was gradually increased, reaching 30 000 rpm over the six-minute duration of the test. The test was performed at an ambient temperature of 204°C (400°F). The friction coefficient and slider temperature were measured at selected speed values and are shown, labeled as original data, in Fig. 5. During the test, the slider temperature increased with increasing speed, reaching a maximum of 427°C (800°F) at 30 000 rpm. The friction coefficient reached a maximum of 1.1 at 10 000 rpm and declined with further increases in speed to a value of 0.6 at 30 000 rpm.

Post test inspection of the disk surface found it to be clean and smooth with no evidence of material transfer from the slider. The slider was severely worn and structural fibers were exposed and delaminated. The wear of the slider was sufficient to have significantly changed its effective contact area. The contact stress was calculated to have decreased from about 0.69 MPa (100 psi) at initial contact to 0.076 MPa (11 psi) at the end of the test.

Results from a series of tests similar to the previous room temperature tests are shown in Fig. 6 for a slider of HAC-1/T50 F1 against a disk made of NC-132 silicon carbide. The results are similar except that the friction coefficients are significantly lower and the data suggest that higher loading levels can be sustained before self-generated temperatures become excessive. At 20 000 rpm, an excessive load was applied causing torque to become erratic and the slider

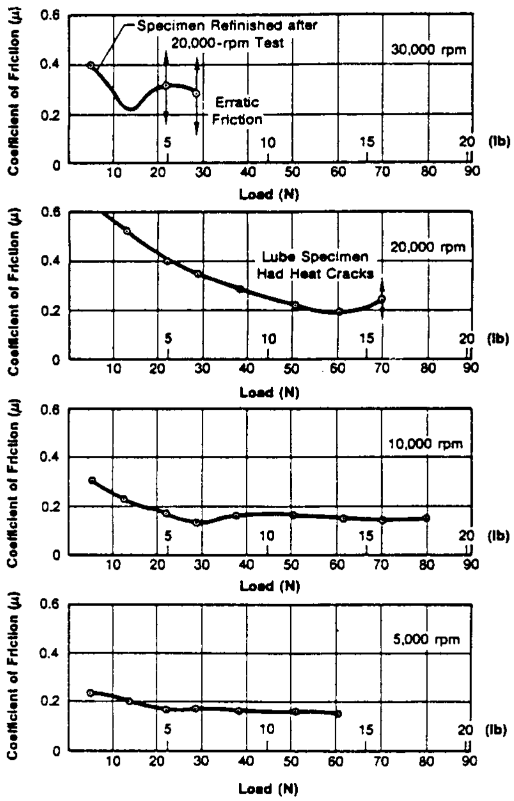


Fig. 6—High-speed data, HAC-1A/T300 F1 vs NC-132, room temperature

temperature to reach a temperature of 174°C (345°F). The test was immediately stopped and the disk surface temperature was measured with a surface contact pyrometer to be 293°C (560°F). Post test examination of the test specimen surfaces, shown in Fig. 7 showed thermal cracking on the surface of the slider while the disk surface was found to be clean and smooth. The slider surface was refinished, the

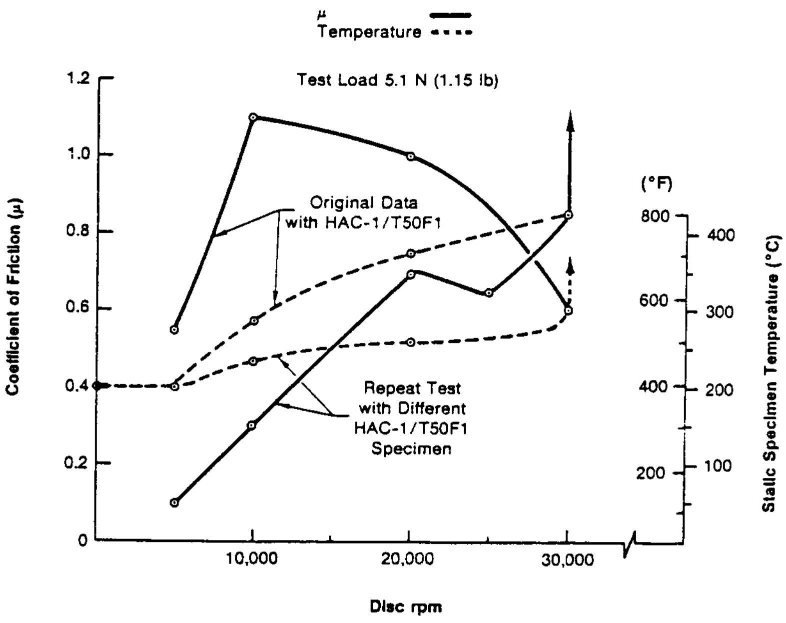


Fig. 5—High-speed data, HAC-1/T50 F1 vs M-50 disk at 204°C ambient

disk surface cleaned, and the mating pair run in for 5 minutes at 10 000 rpm before proceeding to 30 000 rpm to complete this series of tests. At 30 000 rpm, the test was immediately stopped when the torque became erratic; post test inspection showed the specimen surfaces to be in good condition.

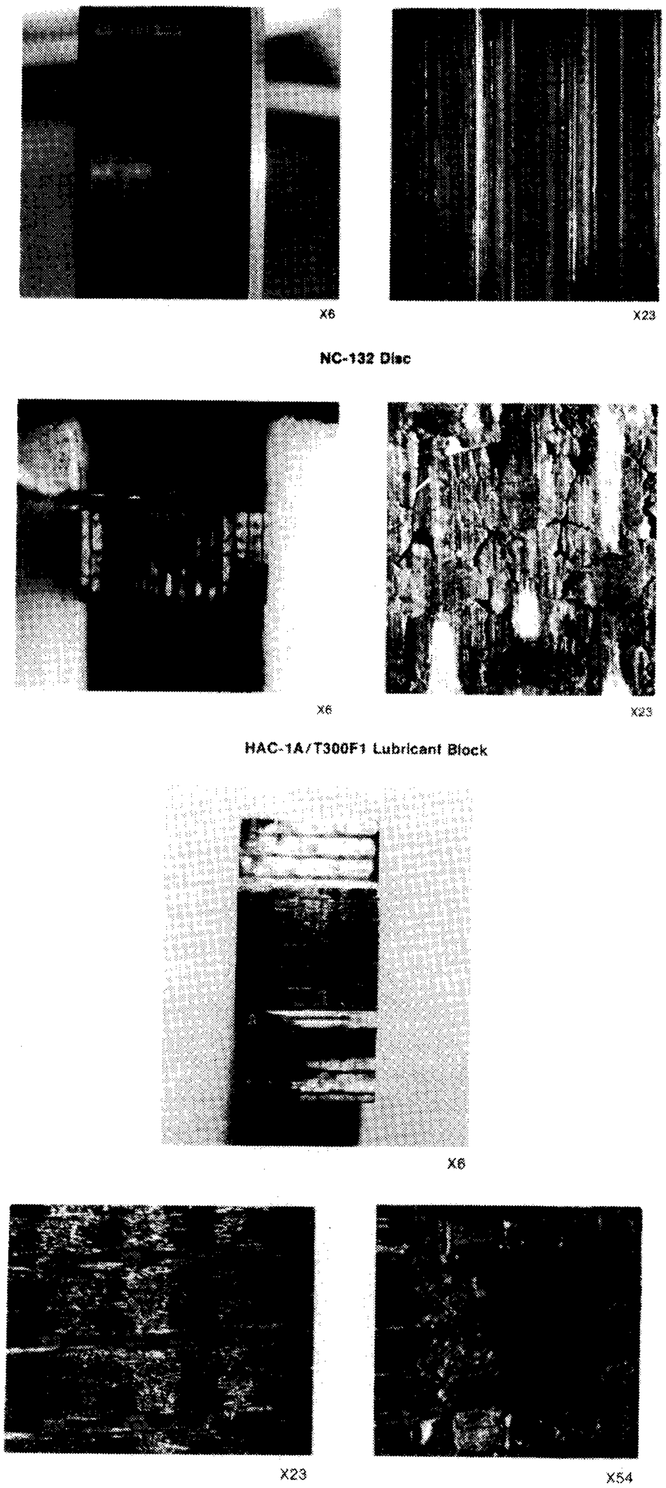


Fig. 7—Condition of HAC-1A/T300 F1 and NC-132 specimens after 20 000 rpm test at room temperature.
(a) Disk before cleaning
(b) Lubricant block before refinishing
(c) Lubricant block after refinishing and testing at 30 000 rpm

The specimens used in the previous test were cleaned or refinished and used for a test at 204°C, similar to the test performed at the same temperature with the cage material sliding on a disk of M-50 steel. In contrast to the case for M-50, the specimen temperature in this test remained stable at approximately ambient, as shown in Fig. 8. The friction coefficient was significantly lower and showed little variation with speed. The condition of the surface of the slider after testing at 30 000 rpm is shown in Fig. 7(c).

Because of the success at 204°C, the decision was made to test this material combination at 316°C (600°F) ambient temperature. Friction coefficient over a similar speed range for a series of loads is shown in Fig. 8 and the results are very similar, with the friction coefficient showing neither strong load nor speed dependence. The temperature data are also similar, showing only a slight rise above ambient at 30 000 rpm. Post test inspection showed the slider surface to be in good condition and the disk surface to be clean and without evidence of material transfer.

To evaluate differences in lubricant materials, the test at 204°C ambient temperature was repeated, with a slider of HAC-1/T50 F1 cage material substituted for the HAC-1A/T300 F1 slider. The results, shown in Fig. 8, are very similar, indicating no significant apparent differences with the two slider materials on NC-132 disks.

After the poor performance of HAC-1/T50 F1 on M-50 had been compared to the very good performance of the cage materials on NC-132, a decision was made to repeat the previous test at 204°C on M-50. The results, shown in Fig. 5, are different in magnitude but similar in that friction levels are much higher than on NC-132 and self-generated temperatures in the specimen are much higher than ambient.

Ball on Disk

To simulate conditions in the ball/race contact, tests were conducted in which a ball of NC-132 silicon nitride was loaded in sliding contact on a disk of M-50 tool steel at room temperature and at 316°C. The disk was prelubricated by running the transfer lubricant stick against the disk at the proposed test temperature for one hour at about 100 rpm. Because of program limitations, the same ball and disk contact zones were used during each series of tests at a given temperature level with the contact loading starting at the minimum value and then increasing in steps. The dwell time to get data at any speed and load point was approximately one minute.

Friction coefficient is shown as a function of speed for three levels of load at room temperature and at 316°C in Fig. 9. The limited sensitivity to speed occurred mainly below 5 cm/s sliding speed. The friction coefficient was sensitive to load at room temperature but considerably less so at 316°C, where beneficial oxides may be formed or the lubricant may perform better. Contact stress levels were considerably diminished during the term of a test due to wear on the ball. Initial Hertz stress was 1.32 GPa, while the final stress varied from 53.1 MPa at room temperature to 193.1 MPa at 316°C. Post test inspection revealed a brownish red debris around the edge of the contact zone at the end of room temperature testing and two sharp scratches on the disk. At the end of the elevated temperature tests, a light surface band was evident on the disk.

Computer Modeling

The analytical effort in the present program consisted of modeling the friction behavior at the ball/race, ball/cage, and cage/race interfaces, the computer simulation of bearing performance, and then comparison of the computer predictions with actual bearing tests.

In order to demonstrate the significance of the friction experiments undertaken during the present investigation, a main shaft bearing in a small gas turbine engine was selected as the candidate bearing and the influence of the observed friction behavior on the overall bearing performance was investigated. The geometry of the bearing and operating conditions are given in Table 3. The operating speed of 63 500 rpm, thrust load of 450 N and a rotating radial load of 225 N selected are typical of operation.

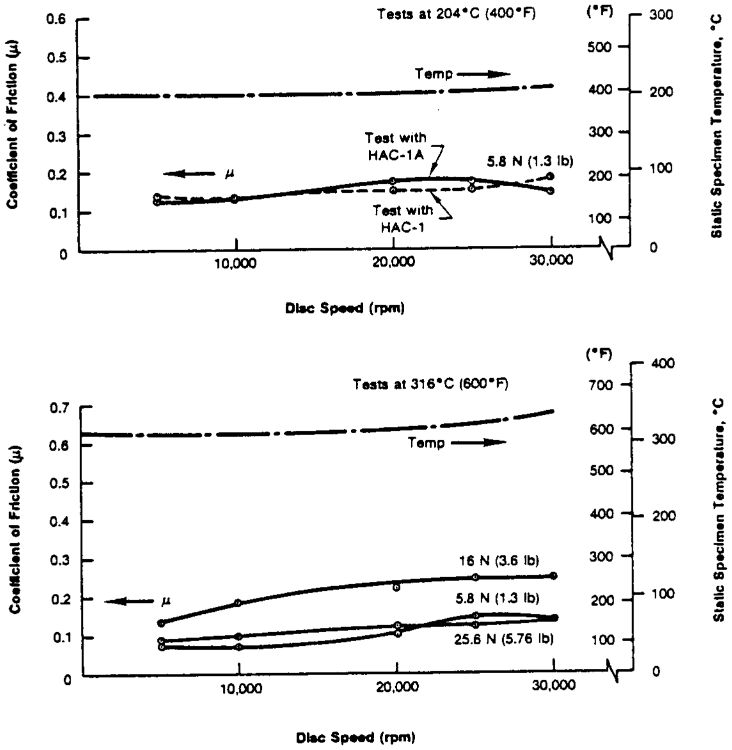
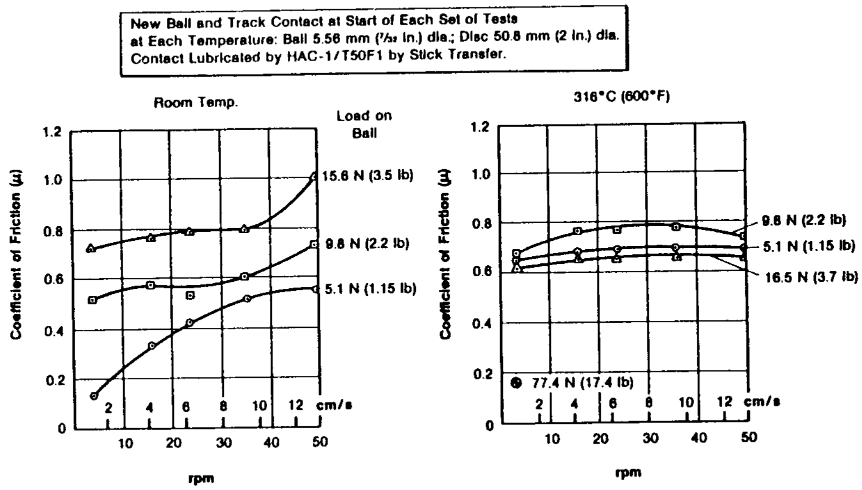


Fig. 8—High-speed tests, HAC-1A/T300 F1 vs NC-132 at 204°C and 316°C ambient.



Total Test Time, min. Final Ball Wear Dia., mm (in.) Final Stress, MPa (lb/in.²)	20 0.61 (0.024) 53.1 (7700)		18 0.33 (0.013) 193.1 (28,000)	10 0.56 (0.022) 315.8 (45,800) — Continuation of Test for 77.4 N Load Point

Fig. 9—Slow-speed sliding tests, NC-132 ball vs M-50 disk

TABLE 3—BEARING GEOMETRY AND MATEIRAL PROPERTIES

PARAMETER	VALUE	UNITS
Bore	29.54	mm
Outside Diameter	47.00	mm
Ball Diameter	5.56	mm
Number of Balls	17	
Pitch Diameter	38.51	mm
Contact Angle	18.6	degrees
Curvature, Outer Race	0.52	
Curvature, Inner Race	0.54	
Cage Outer Diameter	41.22	mm
Cage Inner Diameter	37.39	mm
Cage Width	8.19	mm
Diametral Cage/Race Clearance	0.25	mm
Diametral Ball Pocket Clearance	0.25	mm
Density-Race	7750.	Kg/m^3
Elastic Modulus-Race	200.	GPa
Poisson's Ratio-Race	0.25	
Coeff. of Thermal Expansion-Race	11.7×10^{-6}	$°K^{-1}$
Hardness-Race	7.85	GPa
Wear Coefficient-Race	5.0×10^{-6}	
Density-Ball	3200.	Kg/m^3
Elastic Modulus-Ball	310.	GPa
Poisson's Ratio-Ball	0.26	
Coeff. of Thermal Expansion-Ball	2.9×10^{-6}	$°K^{-1}$
Hardness-Ball	10.0	GPa
Wear Coefficient-Ball	2.0×10^{-6}	
Density-Cage	1500.	Kg/m^3
Elastic Modulus-Cage	1.73	GPa
Poisson's Ratio-Cage	0.30	
Coeff. of Thermal Expansion-Cage	3.0×10^{-6}	$°K^{-1}$
Hardness-Cage	1.0	GPa
Wear Coefficient-Cage	0.1×10^{-6}	

Models of the relationship between slip speed and traction coefficient were based on experimental data discussed above. For the ball/race interface, as shown in Fig. 9 for SiN versus M-50 tests, the traction coefficient is assumed to be constant at 0.60 at slip velocities above 2.5 cm/s. In view of the uncertainties in the experimental data at very low sliding speed, the traction coefficient is assumed to be linear with slip at velocities below 2.5 cm/s. The quantitative accurancy of such behavior has been reported earlier (*5*)(*6*).

To approximate the behavior shown in Fig. 8, the traction coefficient at the ball/cage interface SiN versus HAC-1A/T300) was assumed to have a constant value of 0.1; this is justified because the sliding velocity at the ball/cage interface is high enough that the experimental friction data show no significant speed dependence. Similarly, the friction coefficient at the cage/race interface is assumed to be constant at 0.60; this is based on the experimental data presented in Fig. 5.

Using the computer program ADORE, the performance of the bearing is simulated over more than one revolution of the shaft. Variations in the bearing motion come from several sources. The rotating load causes a sinusoidal variation in the contact loading of the balls, the contact angle, and, to some degree, the orbital velocity. Collisions between the balls and the cage, either as a result of these variations or as a result of variations in cage motion due to collision with the race, cause larger variations in the ball orbital velocity and large spikes in ball slip velocity, as shown in Fig. 10.

Following a ball/cage collision, the slip velocity quickly returns to its nominal motion, indicating an inherent stability in the bearing motion, but impact loads at the time of the collision are predicted to be as high as 80 N, as shown for several balls in Fig. 11. Such collisions are predicted to be in a direction such that the forces act against the webs

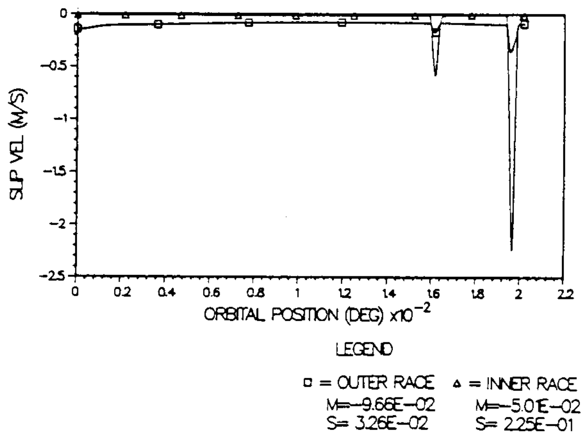


Fig. 10—The variation in slip velocity at the ball/race contact

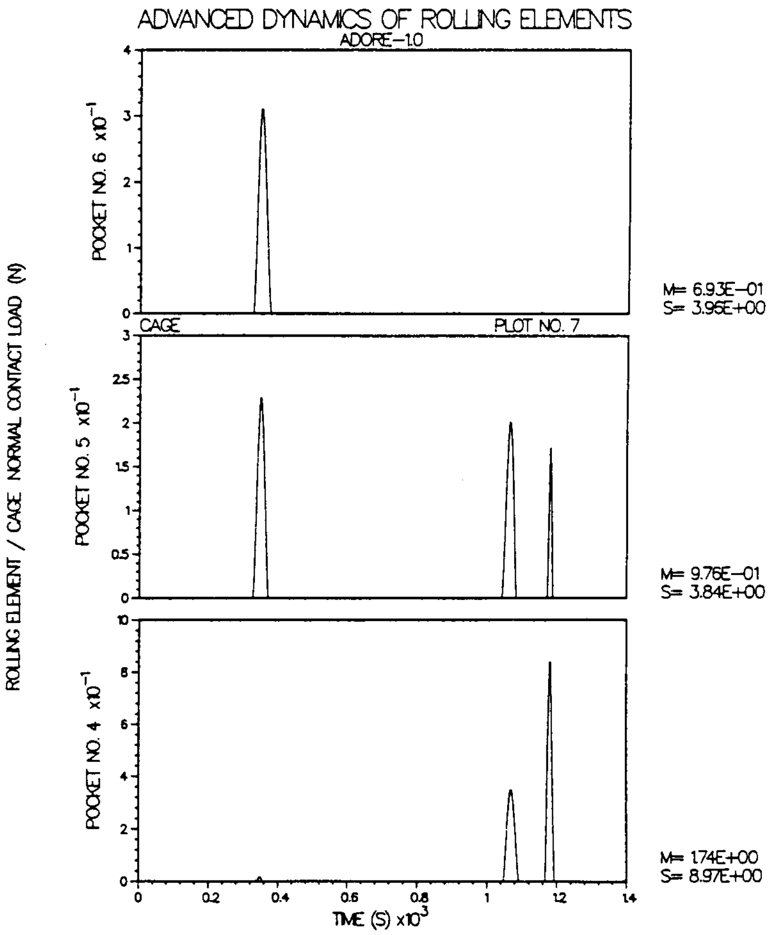


Fig. 11—Ball/cage collision forces in adjacent ball pockets

between ball pockets. The simulation also predicts that usually several adjacent balls will collide with the cage at the same time, which may set up shear forces sufficient to fracture the webs between ball pockets.

Forces acting at the cage/race interface may also become quite large, as shown in Fig. 12. The increasing magnitude of both the force and frequency of collisions at the cage/race interface indicates some instability in the cage motion. This is also indicated by the variation in the power loss, as shown in Fig. 13.

Experimental data on the performance of the bearing under consideration were recently obtained by Meeks and Eusepi (7). The duration of tests in the bearing program was quite limited and dominated by cage failures. In some cases, the cage was completely destroyed while, in other cases, the two rims of the cage were separated due to fractured pocket walls. A typical failed cage [from Ref. (4)] is shown in Fig. 14. The balls and races were typically free of any significant damage. These results are in good agreement with the computer predictions discussed above.

The agreement between the failure mode observed in laboratory tests of a bearing and that which is simulated by the computer modeling of bearing performance, based on the experimentally determined frictional behavior between the interacting elements, substantiates the overall validity of the approach taken in this study. It must be emphasized that the fundamental experiments to characterize the friction behavior of the materials involved under conditions

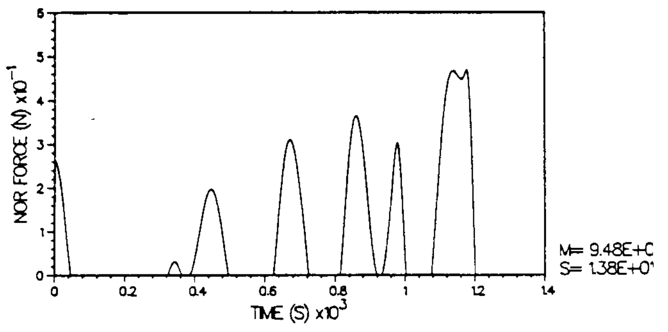


Fig. 12—Normal force at the cage/race interface

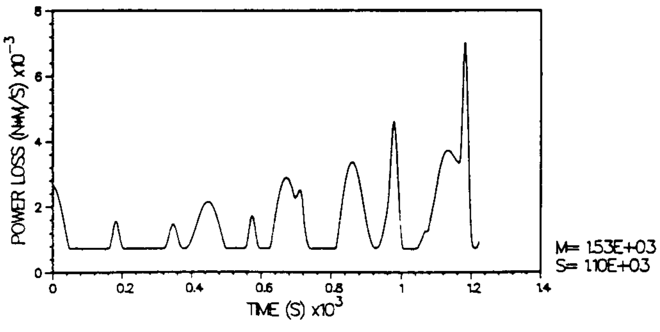


Fig. 13—Total power loss in the bearing as a function of time

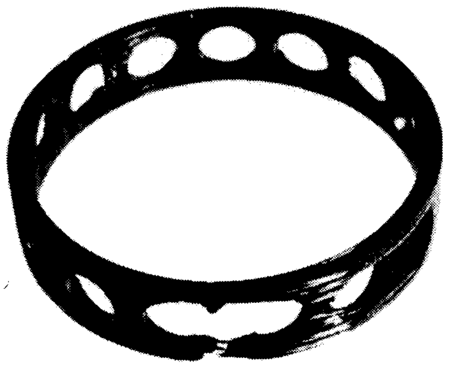


Fig. 14—Self-lubricating composite cage after full bearing testing

experienced in the contacts between bearing elements are a significant part of the modeling process and realistic friction or traction models are significant inputs to the computer simulations of overall bearing performance. Thus, the experimental evaluation of the frictional behavior of potential solid lubricants is a key step in the overall development of a solid-lubricated bearing. The computer modeling of bearing behavior, on the other hand, cannot only provide a realistic simulation of bearing performance, but a parametric evaluation of performance of the bearing as a function of the frictional behavior of the materials may result in substantial guidance for material development for a prescribed performance specification of the bearing.

CONCLUSIONS

The computer simulation of a high-temperature engine ball bearing operating with solid lubrication was successfully completed. To achieve this, the study used experimental friction and wear data which was modeled and used as input to the advanced bearing dynamics computer program ADORE.

In the first series of experimental tests, the friction, wear, and lubricating behavior of HAC-1 sliders against M-50 and NC-132 disks were determined at sliding contact speeds up to 30 000 rpm, and ambient temperatures to 316°C and over a range of loads; these conditions are representative of the bearing operation. The test results showed that HAC-1 sliding on NC-132 had significantly lower friction forces, lower wear rates, more stable operating temperatures, and higher load capacity than it did when sliding on M-50.

In the second series of experimental tests, the low-speed sliding velocities occurring between balls and races during normal operation were simulated by sliding a ball of NC-132 against a disk of M-50 which received lubrication by rubbing a stick of HAC-1 material against the disk. The tests were conducted with sliding speeds from 0 to 15 cm/s and at temperatures to 427°C. The initial Hertzian contact stress between the ball and the disk (typically 1.55 GPa) was reduced rapidly due to wear. The friction measurements from these two series of tests were used to derive appropriate traction models for input to the ADORE computer program to simulate bearing performance.

The results of the computer simulations demonstrated that large forces would be transmitted to the walls between cage ball pockets during collisions between balls and the cage. These happen because of variations in the ball orbital velocities due to the radial component of the load and because of the large friction forces acting between the balls and races and between the cage and the race. The magnitudes of the forces transmitted to the cage is sufficient to cause rapid wear of the cage material and even to cause structural failure to the cage. These results agree with tests of an actual bearing carried out earlier, where cage failures from failed webs were typically observed.

Recommendations

The study has demonstrated the viability of a novel approach toward understanding bearing behavior. Development of better performing bearings could result if some questions raised by the study were properly addressed.

Significantly different friction forces were observed under the same operational conditions when cage material was sliding against NC-132 than when the same cage material was sliding against M-50. This leads one to question whether other material combinations could lead to even better results and whether there might be ways to modify the surface of the steel, say by coating it in some way with silicon nitride to improve its friction characteristics.

In some cases, there was little visual evidence of transfer of lubricant to the contacting surfaces which leads to a series of questions. Could one improve the transfer process by changes in such parameters as rubbing force? Are there more reliable ways of supplying the lubricant, such as plating it on the bearing surfaces themselves or blowing it into the contracts in form of a powder? Has the fact that tests simulating the ball/race contact were conducted without rolling between the contacting surfaces affected the results observed? Are there other lubricant materials which would lead to better performance?

A program to answer these and many similar questions was beyond the scope of the current study. Such a program would provide enhanced understanding and significant benefits. Its cost and effort would be reduced greatly by taking an approach similar to that used in this study involving fundamental experimental studies to characterize the tribological behavior of materials and analytical simulation of bearing performance using appropriately derived traction models to simulate the effects of material properties on bearing performance.

ACKNOWLEDGMENTS

The technical effort described was funded by DARPA under contract number F33615-83-C-5037 administered by Air Force Wright Aeronautical Laboratories, Materials Laboratory, Non-Metallic Materials Division, Wright-Patterson Air Force Base, Ohio. The principal investigator was S. Gray of Mechanical Technology Incorporated. The experimental work was performed at Mechancial Technology Incorporated. Pradeep Gupta performed the analytical studies. The cage materials used in the study were developed by Hughes Aircraft Co. under contract number F33615-78-C-5196, funded by DARPA and administered by Air Force Wright Aeronautical Laboratories.

REFERENCES

(1) Gupta, P. K., *Advanced Dyanmics of Rolling Elements*, Springer-Verlag (1984).

(2) Gray, S. and Gupta, P. K., "Friction and Wear of Solid Lubricated Contact in Gas Turbine Engine Bearings," AFWAL-TR-84-4143, November 1984.

(3) Bhushan, B. and Sibley, L. B., "Silicon Nitride Rolling Bearings for Extreme Operating Conditions," *ASLE Trans.*, **25,** 4, pp 417–428 (1982).

(4) Gardos, M. N., "Solid Lubricated Rolling Element Bearings Final Report," AFWAL-TR-83-4129, February 1984.

(5) Gupta, P. K., "Some Dynamic Effects in High-Speed Solid-Lubricated Ball Bearings," *ASLE Trans.*, **26,** 3, pp 393–400 (1983).

(6) Meeks, Crawford R. and Eusepi, Marin W., "Solid Lubricated Rolling Element Bearings—Part V: Development of High Speed, High Temperature Bearings for Turbine Engines," *Proc. Third Int. Conf. on Solid Lubricants,* Denver, CO, August 5, 1984, pp 285–295.