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**EXPERIMENTAL EVALUATION OF 150-MILLIMETER BORE BALL BEARING
TO 3-MILLION DN USING EITHER SOLID OR DRILLED BALLS**

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EXPERIMENTAL EVALUATION OF 150-MILLIMETER BORE BALL BEARINGS
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ABSTRACT

Seven 150-mm bore ball bearings were run under 8900 Newton (2000 lb) thrust load at speeds from 6670 to 20 000 rpm (1 to 3 million DN). Four of the bearings had conventional solid balls and three bearings had drilled (cylindrically hollow) balls with 50 percent mass reduction. The bearings were under-race cooled and slot-lubricated with Type II ester oil at flow rates from 4.35 to 5.80 liters per minute (1.15 to 1.57 gal/min). Friction torque and temperatures were measured on all bearings. While there was considerable spread in the temperature data, the drilled ball bearings tended to run slightly cooler at higher speeds. No significant difference in torque was noted, between the solid and drilled ball bearings. One bearing of each type was rerun at 17 800 Newtons (4000 lb) thrust load. The solid ball bearings performed satisfactorily at 3 million DN. However, at about 2 million DN the drilled ball bearing experienced a broken ball and cracks appeared in two other balls as the result of flexure fatigue. Metallurgical examination of the cracked balls indicated a brittle structure in the bore of the drilled balls.

INTRODUCTION

Recent trends in gas turbine design and development have been toward engines with higher thrust-to-weight ratios and increased power output,

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which result in a requirement for higher shaft speeds and larger shaft diameters.^[1] Bearings in current production aircraft turbine engines operate in the range from 1.5 to 2 million DN (bearing bore in millimeters times shaft speed in rpm). Engine designers anticipate that turbine bearing DN values will have to increase to 2.5 to 3 million by 1980.

When ball bearings are operated at DN values above 1.5 million, centrifugal forces produced by the balls can become significant. The resulting increase in Hertz stresses at the outer-race ball contacts may seriously shorten bearing fatigue life. It is therefore logical to consider methods for reducing the factors that contribute to ball centrifugal loading, such as ball mass, ball orbital speed, and orbital radius. Theory indicates that reductions in ball mass can be quite effective in extending bearing fatigue life at high speeds.^[2]

Both thin wall spherically hollow and drilled balls have been evaluated in short-time high speed bearing experiments.^[3-7] The drilled ball concept, wherein ball mass is reduced as much as 50 percent by machining an accurate concentric hole through the ball, shows particular promise for high speed applications.

Drilled balls, as compared with welded hollow balls, have several advantages, (1) fabrication is accomplished by standard ball processes, (2) they can be easily inspected for flaws, (3) hole concentricity can be maintained very accurately which alleviates problems of ball unbalance at high speed, and (4) a smooth surface finish can be achieved, without the irregularities present in the weld area of a spherically hollow ball. One disadvantage of drilled balls is that special cages are required, with ball alignment restraints, to prevent the edge of the hole from damaging

the race grooves during bearing start-up.

The objectives of this investigation were (1) to experimentally evaluate 150-mm bore ball bearings having either solid conventional or drilled balls using under-race cooling and slot lubrication techniques at speeds to 3 million DN, (2) to compare torque and temperature data of the drilled and solid ball bearings over the same operating conditions, and (3) to compare the experimental torque and temperature data with that that were obtained analytically from a computer program that predicts performance for both the drilled and solid ball bearings.

APPARATUS AND INSTRUMENTATION

Bearing Test Rig

Dynamic testing of the 150-millimeter bore ball bearing was conducted on a three bearing test spindle as shown schematically in figure 1. The spindle was belt driven by an AC motor through an eddy current clutch. The output speed of the clutch was increased through a combination of pulleys, belts, and a gear box by a factor of nearly 15 times. A wide, thin-section flat belt connected the output of the gear box to the test spindle. An adjustable idler wheel maintained tension on the final drive belt at high speeds. Figure 1 shows the spindle with the test bearing cantilever mounted. The bearing was mounted on the shaft with a 0.0635 mm (0.0025 in.) interference fit. Thrust load was applied to the test bearing by a hydraulic cylinder, connected to the bearing housing by a steel chain. This method of loading minimized possibilities of misalignment and produced only slight torque tare. The force produced by the hydraulic cylinder was calculated from pressure gage readings. Thrust load measurements were accurate to within ± 1 percent of the gage readings. Reaction

to the thrust load was provided by a 216-R (80 mm bore) ball support bearing. The 216-R ball bearing and a 116-KR (80 mm bore) ball bearing provided radial support for the shaft. A preload spring, plus a sliding housing fit for the 116-KR bearing (see fig. 1) maintained axial load on the support bearings at all times, thus assuring radial rigidity.

The lubrication system included a common oil sump with electric heating elements, a supply pump and filter. Separate oil supply lines were used for the support bearings and test bearing. The oil supply to the test bearing was cooled as necessary to maintain a constant oil inlet temperature of 367 K (200° F). Support bearing oil had an additional cooler to provide lower inlet oil temperature to the support bearings. All lubricant and coolant for the test bearing entered the test head from the center of the hollow shaft as shown in figure 1. Oil reached the outer diameter of the shaft near the unloaded half of the inner ring, then passed under the inner ring through twelve equally spaced axial passages, each 2.38 mm (0.0938 in.) deep by 4.78 mm (0.188 in.) wide. As the oil reached the juncture of the two inner ring halves a portion was forced, by centrifugal force, into the bearing through the radial passages at the interface. The remaining oil passed on to the end of the axial passages and was discharged separately. During actual testing most of the oil entered the bearing. Multiple oil-out holes were provided on each side of the bearing housing for lubricant scavenge. Multiple holes were also provided for cooling oil scavenge. The several oil-out passages in each area were manifolded, then pumped back to the sump. Oil-in and oil-out flow rates were measured by volumetric type flow meters. Oil-out flow meters were installed in suction lines to the scavenge pump. A Type II

ester oil meeting MIL-L 23699 specification was used with a viscosity of 5.5×10^{-3} N/sec-m² (0.8×10^{-6} lb-sec/in.²) at 367 K (200° F). Oil flows ranged from 4.35 to 5.80 liters per minute (1.15 to 1.57 gpm).

Temperatures were measured by chromel-alumel thermocouples at the following locations:

- (1) Oil-in center of hollow shaft
- (2) Test bearing outer ring
- (3) Oil-out, inboard side of test housing
- (4) Oil-out, outboard side of test housing
- (5) Oil-out, test bearing underrace cooling oil
- (6) Oil sump
- (7) Oil at input flow meter to test bearing
- (8) Outer rings of both support bearings

Test bearing torque was measured by a strain gage mounted on the restraining arm. The strain gage force produced a signal that was amplified and recorded on a millivolt recorder. This force times the torque arm length constituted the torque.

Test Bearings

The test bearing specifications are listed in table I. The bearings were 150-millimeter bore, split inner-ring, angular-contact ball bearings with 22.2 mm (0.875 in.) diameter balls. The one-piece machined cages were outer-race located. Photographs of the bearings with one-half of the inner ring removed are shown in figure 2. In each mating face of the inner ring halves there were twelve radial slots of 1.25 mm (0.050 in.) radius extending from the bore to the raceway. The specification for the conventional solid ball and drilled ball bearings were identical except

for the drilled balls and a modification made to the drilled ball cage.

Drilled ball design. - A section view of the drilled ball detail is shown in figure 3. A matched set of 23 Grade 10 balls were selected for each drilled ball bearing. Parallel flats 16.84 mm (0.663 in.) apart were ground on each ball. A 13.39 mm (0.527 in.) diameter concentric hole was electric discharge machined (EDM) through each ball. The hole was then finish ground to 13.54 mm (0.533 in.) diameter, maintaining concentricity with the ball outer diameter to within 0.020 mm (0.0008 in.). The hole diameter of 13.54 mm (0.533 in.) resulted in a weight reduction of approximately 50 percent from that of a solid ball.

Drilled ball cage design. - The cage design for the drilled ball bearings was the same one-piece design as that used for the solid ball bearings with one exception. The drilled ball cage had integral ribs machined into opposite sides of each ball pocket as illustrated in figure 4. These ribs restricted ball twisting movement to about 37° and prevented the edge of the hole from riding on the race groove during bearing operation. The ribs also restricted ball spin about a vertical axis. After silver plating all cages were balanced to within 30 gram-millimeters (0.042 oz-in.).

PROCEDURE

Test Procedure

Each bearing was installed in the test housing so that the puller groove half of the inner race was on the load mechanism side of the rig. Lubricating oil was preheated and pumped through the rig. Approximately 4450 newtons (1000 lbs) thrust load was applied to the stationary test bearing.

The test rig was started and allowed to run at 3000 rpm while the drive load, and lubrication systems and instrumentation were checked. When no irregularities were noted, the load was increased to the test value, either 8900 to 17 800 newtons (2000 or 4000 lbs), the speed was increased to 6670 rpm. The flow meter for the inlet oil to the test bearing was set at 30 percent capacity.

The test rig was run at 6670 rpm and scheduled load until the oil-in temperature was stabilized at 367 ± 1.7 K ($200^{\circ} \pm 3^{\circ}$ F). The rig was then run for a period of at least ten minutes during which the difference in temperature between oil-in and bearing outer race temperature did not vary more than 1.1 K (2° F). Instrument readings were recorded during this time.

This procedure was repeated at speeds of 10 000, 13 330, 15 000, 16 670, 18 330, and 20 000 rpm. During the first three tests for the solid ball test bearings, including the check-out bearing test, the oil-in flow was increased to 40 percent of the flow meter capacity after running a 15 000 rpm datum point. In the remaining tests oil flow rate was increased to 40 percent after completion of the 13 330 rpm datum point.

Analytical Procedure

A computer program^[2] capable of evaluating the thermal and kinematic performance of high speed ball bearings subjected to thrust loading was used to calculate the bearing temperatures, torque, heat generation, maximum Hertz stresses, and fatigue life of the 150 mm bore ball bearings used in this investigation, under elastohydrodynamic (EHD) lubrication conditions. Both solid and drilled ball bearing configurations (table I)

were computed over the range of operating speeds to 3 million DN at thrust loads of 8900 and 17 800 newtons (2000 and 4000 lb) with oil flow rates of 4.35 and 5.80 liters per minute (1.15 and 1.57 gpm). The computer output for bearing temperature and torque at both thrust loads was plotted and compared on the experimental data curves.

The bearing oil inlet temperature, used for the computer input at each speed, differed from the experimental constant value of 367 K (200° F) to account for heat gain by the oil for each speed increment. Therefore for each 3330 rpm (0.5 million DN) increment above 6670 rpm the oil temperature was increased 5.5 K (10° F), so that at the maximum shaft speed of 20 000 rpm (3 million DN) the oil inlet temperature value used was 389 K (240° F).

The bearing thermal model used in the computer program was based on the frictional heat generated within the bearing, the amount of lubricant within the bearing cavity, and the flow rate of lubricant through the bearing. The major sources of bearing heat generation were from (1) the sliding motion at the ball-raceway contacts, (2) viscous drag caused by the balls plowing through the lubricant, and (3) viscous drag of the cage rails sliding over the piloted race lands. Heat was dissipated from the bearing-shaft-housing system through conduction, convection, and radiation heat transfer modes. For each bearing operating condition 31 temperatures were printed out in a specified distribution array for the bearing-shaft-housing system.

RESULTS AND DISCUSSION

Test Bearing Results

8900 newton (2000 lb) thrust load tests. - The conventional solid bearing tests were run with four ball bearings (numbers 162A, 433A, 384A, and 398A) at 8900 newton (2000 lb) thrust load at speeds from 6670 to 20 000 rpm (1 to 3 million DN). The shaft speed, running time, inlet oil temperature and flow, scavenge oil temperature and flow (from both the loaded and unloaded side of the inner race), outer race temperature, and bearing torque for each shaft speed were recorded during each test.

Bearing 162A was used to check out the test rig and to establish the adequacy of the inlet oil flow rates. In preliminary rig shakedown tests oil flow rates from 4 to 6 liters per minute had been determined as sufficient to lubricate and cool the bearing at 8900 newton (2000 lb) thrust load at speeds to 19 500 rpm. Therefore, an oil flow rate of 4.35 liters per minute (1.15 gpm) at 367 K (200° F) inlet temperature was used at speeds to 15 000 rpm and a flow rate of 5.80 liters per minute (1.57 gpm) was used at speeds above 15 000 rpm for bearing 162A. These two oil flows proved to be sufficient and were used for the remaining bearings in the test program.

Bearing 384A ran significantly hotter than did bearing 162A. The scavenge oil temperature on the oil-out side (which is the loaded half of the inner race) of the bearing was considerably higher than on the oil-in side; in fact this temperature difference at 18 330 rpm was 76 K (138° F). This bearing was the only one for which underrace discharge oil was measured, occurring only at the 16 670 and 18 330 rpm shaft speeds.

Although bearing 398A ran hottest of any bearing tested, it nevertheless operated satisfactorily throughout the speed range. The scavenge oil temperature from the oil-out side of the bearing was again much higher than the oil-in side. Measured scavenge oil flow from the oil-in (unloaded inner race) side of the bearing was much greater than the oil flow from the loaded side, indicating that oil distribution within the bearing was irregular.

Bearing 433A ran considerably cooler than either bearings 384A and 398A and slightly warmer than did bearing 162A. The scavenge oil temperatures measured from the oil-inlet and oil-outlet sides of bearing 433A indicated higher oil-outlet side temperatures. This temperature increase varied from 5 K (8° F) at 6670 rpm to as much as 24 K (43° F) at 20 000 rpm (see fig. 1). Additionally, scavenge oil temperatures on the oil-outlet side measured more than 10 K (18° F) higher than bearing outer race temperature at the two highest shaft speeds. Maximum torque measured for bearing 433A was 9.49 newton-meters (84 in.-lbs) at 20 000 rpm.

The three drilled ball bearings (numbers 429A, 377A, and 230A) were run under the same conditions as were the solid ball bearings and all three operated satisfactorily through 20 000 rpm (3 million DN).

Outer race temperatures for solid ball bearing 433A and drilled ball bearings 429A and 377A are plotted as a function of speed in figure 5. The drilled ball bearings had slightly lower temperatures than did bearing 433A throughout the speed range.

Scavenge oil temperatures for the three drilled ball bearings indicated the oil-out side (loaded half of the inner-race) were from 12 to

25 K (21° to 45° F) higher than oil-in side temperatures at speeds from 15 000 to 20 000 rpm. The oil-out temperatures were also hotter than the bearing outer race temperatures by approximately 11 K (20° F) at 20 000 rpm. These oil temperature differences were similar to those for bearing 433A, the only solid ball bearing that operated normally throughout the speed range.

Bearing torque as a function of shaft speed for bearings 433A, 429A, and 377A is shown in figure 6. While the torque values of bearing 433A (solid ball bearing) are somewhat lower than those for bearing 377A, they are not significantly different from those indicated for bearing 429A. Bearing 433A and 429A show comparatively lower torque (fig. 6) and higher temperature values (fig. 5) whereas bearing 377A has higher torque and lower operating temperatures.

17 800 newton (4000 lb) thrust load tests. - Solid ball bearing 433A and drilled ball bearing 377A were selected for testing at the 17 800 newton (4000 lb) thrust load. Bearing temperatures and torque as a function of shaft speed are presented in figure 7. The same oil flows as in the 8900 newton (2000 lb) load tests were used. Bearing 433A ran successfully throughout the speed range. Bearing torque and temperature were significantly higher than in the lower load test. Drilled ball bearing 377A ran successfully at the two lower speed settings, but experienced a broken ball after running for 6 minutes at 13 000 rpm. Failure was indicated by a high and erratic torque accompanied by a rapidly increasing temperature. Seizure did not occur and the bearing was operated a short time at reduced speed and load while the operating problem was diagnosed.

Bearing Post-Test Inspection

Solid ball bearings. - Visual examination of bearings 162A, 384A, and 433A after the 8900 newton (2000 lb) load showed them similar in appearance. The races varied from unchanged to a light straw color on the load side. The balls were straw colored. The outer diameter of the cage had two lightly polished rings where contact was made with the outer race. Cage ball pocket polishing was normal with a shiny band in the silver around the periphery of each pocket. Bearing 384A also had shiny bands around each ball and the outer race had a shiny ball track at its center as a result of a high speed no load condition.

Bearing 398A had run hottest of the solid ball bearings. Races and cage were brown colored and the balls had a bluish tinge. The cage outer diameter show two polished bands in the silver plating where contact was made with the outer race, with heavier polishing on the inner-race load side. Ball pocket polishing was similar to the other three bearings.

Because of the unequal scavenge oil temperatures and measured unequal discharge oil flows from either side of bearings 384A and 398A, the test rig was disassembled after running bearing 398A and the individual parts were carefully examined. Examination revealed that excessive stock on a chamfer of the oil-in side clamp plate prevented the plate from contacting the inner race. A small gap was evident between the face of the plate and the end of the inner race and permitted oil to flow into the housing cavity and escape into the scavenge system without providing sufficient lubricating and cooling benefit to the bearing. The oil-in (unloaded) half of the inner race was always unloaded so there was no tendency to close the gap. These same basic flow and scavenge oil tem-

perature irregularities appeared to have also happened with bearing 384A.

Bearing 433A was run after the clamp plate had been modified and contact with the inner race was assured. This bearing ran considerably cooler with more equal oil discharge temperatures than did either bearings 384A or 398A. The scavenge oil flow from either side of bearing 433A was also more equal.

Bearing 433A was examined after the 17 800 newton (4000 lb) load test and showed all components a darker straw color with ball tracks on the races better delineated than in the lower load test. Figure 8 is a photograph of bearing 433A after the 17 800 newton (4000 lb) load test with the inner race removed. The ball pocket and outer locating surface cage wear can be seen.

Drilled ball bearings. - The three drilled ball bearings were similar in appearance to solid ball bearing 433A after the 8900 newton (2000 lb) load test. Cage ball pockets indicated light contact between the edges of the hole in the ball and the ends of the machined ribs on both sides of each pocket. It appears that contact occurred during bearing start up, when the balls had oriented themselves. Drilled ball contact with cylindrical surfaces of the pockets formed a different wear-band pattern than that created by conventional solid balls. Contact between the conventional balls and their pockets was uniform around the periphery of the cage pocket, whereas the drilled ball contact on the pockets was heaviest at the fore and aft positions because of the restricted ball spin arc. This wear phenomenon resulted in an edge effect at the ends of the wear band with ridges being impressed into the silver plating. Bearing 377A after testing is shown in figure 9. Cage ball pocket and

locating surface wear and a fractured drilled ball, that had broken into two pieces, can be seen. All cage pockets showed heavy contact between the edges of the balls and the machined ribs in the cage pockets. It was assumed that some bearing instability occurred after the ball failure and that the heavy contact resulted. However, in no case was the silver plating ruptured. The cage outer surface and ball pocket contact areas looked about the same as after the lower load test as shown in the closeup view of figure 10. The loaded half of the inner race showed a more pronounced ball track than was evident after the lower load test, plus a nick and scratches. The outer race was somewhat scored by the broken ball. Ball showed minor surface distress as a result of running over the roughened outer race. Two balls, in addition to the one that was broken, showed cracks in the bores. The bores of all other balls appeared normal.

Metallurgical examination of fractured drilled balls. - Examination of the broken ball revealed an area of shallow pits in the bore which resulted from inadequate clean up of, or, from too great radial penetration by the electric discharge machining. It appeared possible that the fracture might have initiated from a pit, but abrasion had removed evidence. However, cracks in the two unbroken balls were, with one exception, not associated with any pit. In the one exception the crack started at the edge, passed through the pit and axially on into the ball. All cracks which were observed started at the edge of the ball.

Metallurgical examination of several cracks revealed a penetration of approximately 0.05 millimeter (0.020 in.) into the matrix of the ball. Figure 11 shows one of the cracks under 1000 X magnification, perpendicular to the ball's axis. The subsurface microstructure of the broken

AISI M-50 drilled ball near the bore of the hole was examined at 1000X magnification. A brittle, untempered martensitic structure was observed; apparently this area had overheated and had rehardened during the EDM machining operation. The areas of the ball, not adjacent to the hole, revealed a more normally tempered martensitic structure.

Ball fractures also occurred in the drilled ball bearings reported in [7] wherein 125-mm bore ball bearings were run under conditions similar to those in this investigation. In one bearing a drilled ball had fractured during a run at 2.8 million DN at 13 350 newton (3000 lb) thrust load. In another bearing running at 2.6 million DN and 22 250 newton (5000 lb) thrust load, eleven out of 21 drilled balls had failed. The ball fractures were similar to that shown for bearing 377A in figure 9. Electron fractographic analysis of the broken balls in [7] revealed that a fatigue crack had originated near the hole bore (ID ball surface) and had propagated to the ball outer surface.

A finite element analysis [8] was utilized to determine the type and magnitude of the stresses in the drilled balls of [7]. The analysis showed that a large reversing stress occurs in the bore of the ball as the ball rotates through one revolution under Hertzian loading at three million DN. The maximum tangential stress at the bore ranged from $470 \times 10^6 \text{ N/m}^2$ (68 000 psi) in tension to about $153 \times 10^6 \text{ N/m}^2$ (22 000 psi) in compression, which is a stress change of $623 \times 10^6 \text{ N/m}^2$ (90 000 psi), and is more than adequate to produce flexure failures in the drilled ball. The balls in this investigation were subjected to the same type of cyclic stress of a lower magnitude and it is assumed that they also failed in flexure fatigue.

Comparison of Bearing Experimental and Analytical Results

Comparison of the computed results between the solid and drilled ball bearings indicated that the values for outer race temperature, oil-out temperature, and torque were approximately the same at each speed, for each of the two loads.

Comparison between the computed and experimental bearing outer race and oil out temperatures indicated that the computed values were slightly lower for both solid and drilled ball bearings at all speeds, except at 20 000 rpm where the computed temperatures were a few degrees higher. Computed outer race temperatures for both solid and drilled bearings at 8900 newtons (2000 lb) load are plotted on figure 5 (dashed line) for comparison with solid ball bearing 433A and drilled ball bearings 377A and 429A. Similar results for the computed temperatures at 17 800 newton (4000 lb) load for bearings 433A and 377A can be seen in figure 7(a).

The computed bearing torque values were less than one-half the experimental values at both thrust loads. Computed torque values at 8900 newton (2000 lb) load are plotted on figure 6 and compared with these for bearings 433A, 377A, and 429A. Computed torque values at 17 800 newton (4000 lb) load for bearings 433A and 377A are plotted and compared with experimental values in figure 7(b). Although the torque values did not agree with experimental data for these bearings, the computed temperature data showed good correlation with actual bearing temperatures. It was concluded therefore that the computer program was a useful tool for predicting operating temperatures of 150 mm bore ball bearings over a range of operating conditions.

The effect of DN on the theoretical (computed) fatigue life of the

150-mm bore bearing can be seen in figure 12 for both thrust loads. For the solid ball bearing (solid lines) an increase in speed from a DN value of 1.5 million to 3 million results in a reduction of life by a factor of 20 for the 8900 newton (2000 lb) load and by ten times for the 17 800 newton (4000 lb) load. A 50 percent reduction in ball mass can be quite effective in extending bearing fatigue life. This improvement can be illustrated by comparing the drilled ball fatigue curves (dashed lines) with those for the solid balls (solid lines). At 3 million DN and 17 800 newtons (4000 lb) thrust load, for example, fatigue life is improved by a factor of 1.5.

The maximum Hertz compressive stresses at 20 000 rpm (3 million DN) for the 8900 newton (2000 lb) thrust load were $1138 \times 10^6 \text{ N/m}^2$ (165 000 psi) and $1413 \times 10^6 \text{ N/m}^2$ (205 000 psi) and the inner and outer races, respectively, for the solid ball bearing. At the 17 800 newton (4000 lb) load these stresses increased to $1331 \times 10^6 \text{ N/m}^2$ (193 000 psi) at the inner race contact and to $1579 \times 10^6 \text{ N/m}^2$ (229 000 psi) at the outer race contact. For the drilled ball bearing the inner-race stress remained essentially the same for both loads. However, the stress at the outer-race contact was reduced about 7 percent from the values with the solid balls.

SUMMARY OF RESULTS

Tests were conducted with 150 millimeter bore, angular contact split-inner ring ball bearings using either solid balls or drilled balls with a 50 percent mass reduction. Bearings were operated at thrust loads of 8900 and 17 800 newtons (2000 and 4000 lb) at speeds from 6670 to 20 000 rpm (1 to 3 million DN) using a type II ester oil as lubricant. Friction torque and outer race and scavenge oil temperatures were measured on all

bearings. A digital computer program was used to determine, analytically, the performance of the solid and drilled ball bearing designs over the same operating conditions. The investigation provided the following significant results:

1. Both solid and drilled ball bearings were run successfully over the speed range at the 8900 newton (2000 lb) load. The three drilled ball bearings tended to run a few degrees cooler than the solid ball bearings especially at the higher speeds. No significant difference in torque data was noted, however.

2. The solid ball bearing, rerun at 17 800 newton (4000 lb) load, performed satisfactorily at 3 million DN. At about 2 million DN the drilled ball bearing experienced a broken ball and cracks were evident in two other balls as the result of flexure fatigue failures. Metallurgical examination of these balls revealed a brittle structure in the hole bore. These ball failures were similar to those reported by another investigator wherein 125-mm bore ball bearings with 50 percent drilled balls were run under thrust loads to 3 million DN.

3. Comparison of the experimental temperature and torque data of the solid and drilled ball bearings with that determined analytically from a computer program for predicting high speed ball bearing performance, indicated good temperature correlation at all speeds. The measured bearing torques were more than twice the computed values.

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TABLE I. - BEARING SPECIFICATIONS

[Bearings were made to ABEC Grade 5 tolerances.]

Rings and Balls:

Material	Consumable electrode vacuum melt M-50 steel
Hardness	Rockwell C 60-63
Bore	150 mm (5.9053 in.)
o.d.	225 mm (8.8583 in.)
Width	35 mm (1.3780 in.)
Number of balls	23
Ball o.d.	22.225 mm (0.875 in.)
Pitch diameter	186.89 mm (7.3578 in.) nominal
Contact angle	0.5236 rad (30°) nominal
Radial clearance	0.107-0.142 mm (0.0042-0.0056 in.) under 146.8 newton (33-lb) load
Inner race radius	52% ball diam
Outer race radius	51% ball diam

Cage:

Material	SAE 4340 steel, silver plated 0.025 - 0.051 mm (0.001-0.002 in.) thick per AMS-2412
Hardness	Rockwell C 28-32
Cage width	28.14 mm (1.108 in.) maximum
Ball pocket clearance	0.635-0.838 mm (0.025-0.033 in.) diametral
Cage land clearance	1.01 mm (0.040 in.) diametral

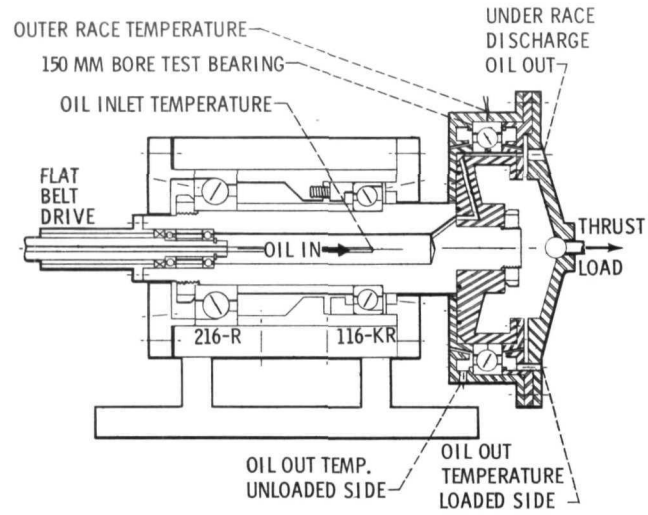
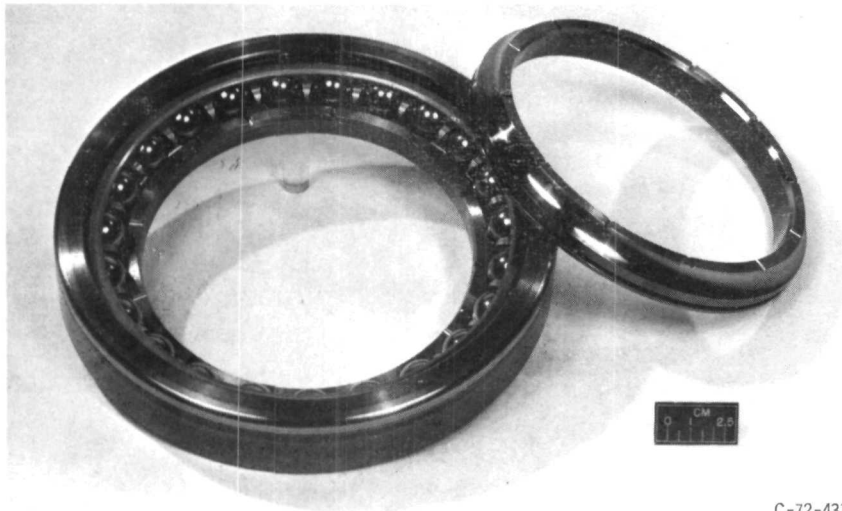


Figure 1. - Test spindle schematic.



C-72-4370

(a) SOLID BALL BEARING.



C-72-4371

(b) DRILLED BALL BEARING.

Figure 2. - Test bearing. Type, angular contact split-inner ring; material, AISI M-50 tool steel; cage type, one-piece machined outer race located.

Figure 2. - Concluded.

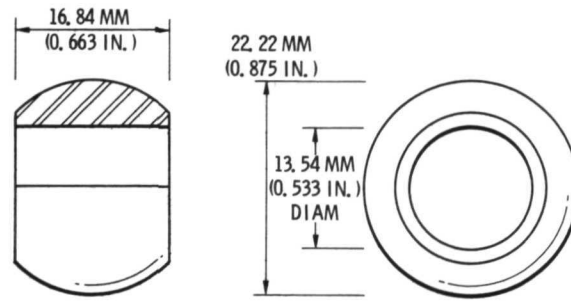


Figure 3. - Details of drilled ball design.

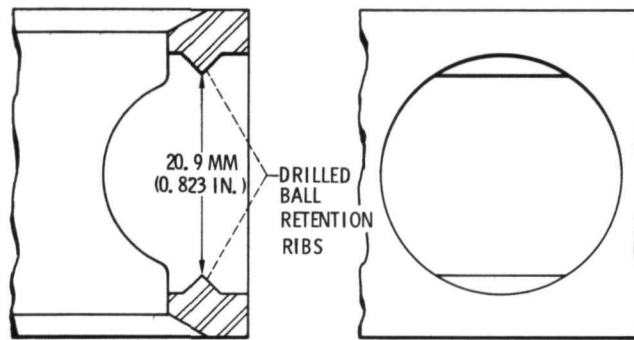


Figure 4. - Detail of drilled ball cage modification.

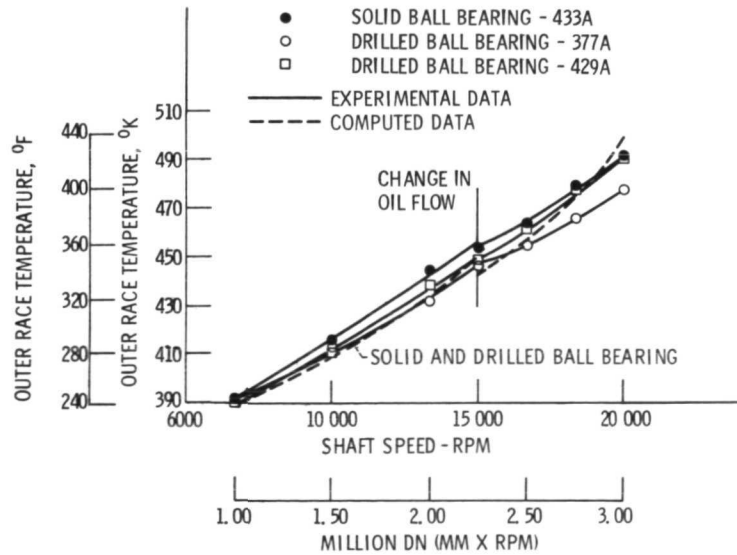


Figure 5. - Comparison of solid and drilled ball bearing temperatures as a function of shaft speed from 1 to 3 million DN; thrust load, 8900 newtons (2000 lb); lubricant MIL-L 23699 at 367 K (200° F) oil inlet temperature. Oil flow 4.35 and 5.80 liters/min (1.15 and 1.57 gpm).

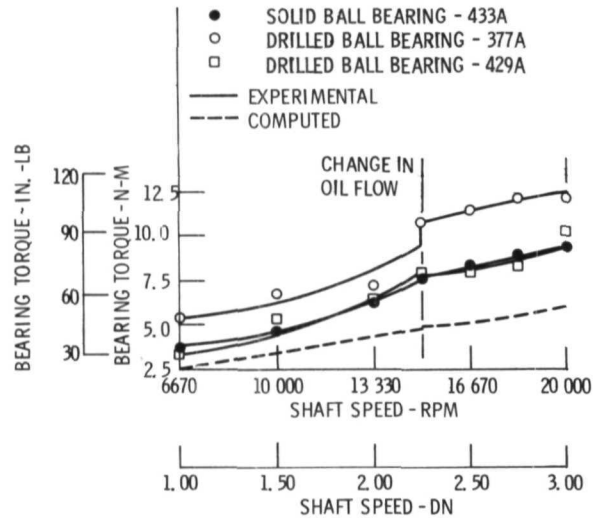
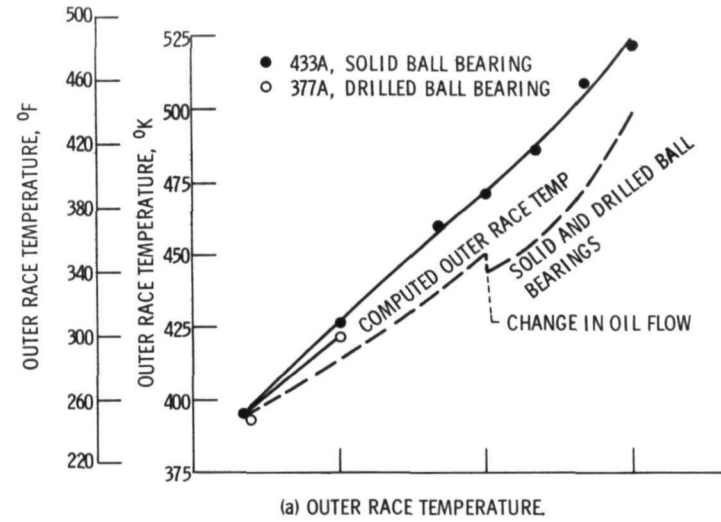
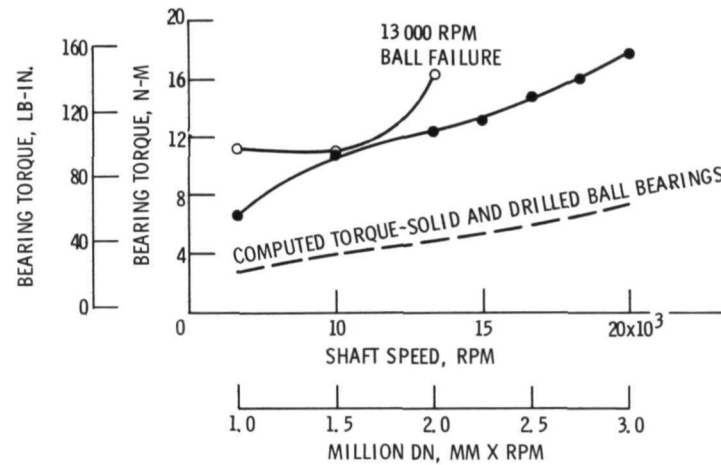


Figure 6. - Bearing torque vs shaft speed - 1 to 3 million DN. Comparison of solid and drilled ball bearing torque as a function of shaft speed from 1 to 3 million DN; thrust load, 8900 newtons (2000 lb); lubricant MIL-L 23699 at 367 K (200° F) oil inlet temperature. Oil flow 4, 35 and 5.80 liters/min (1, 15 and 1, 57 gpm).

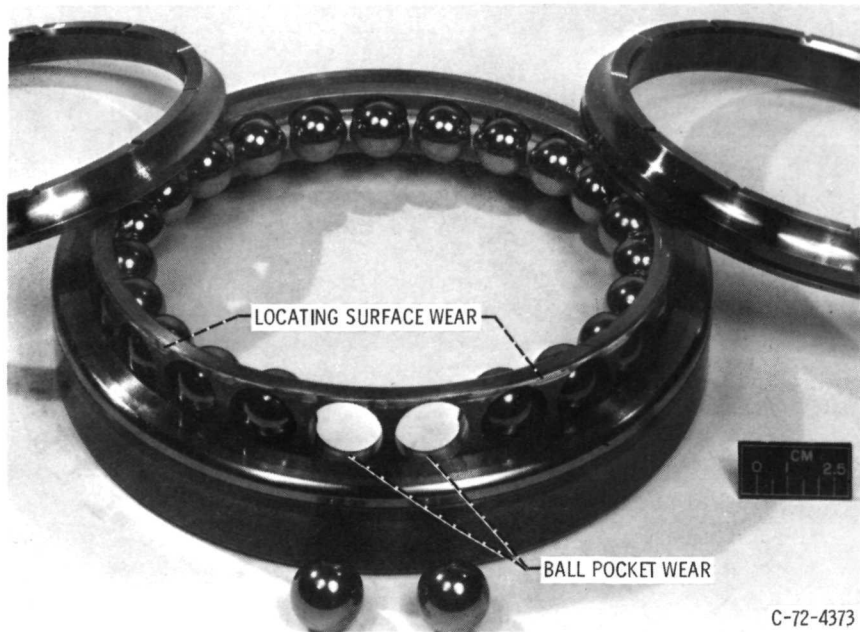


(a) OUTER RACE TEMPERATURE.



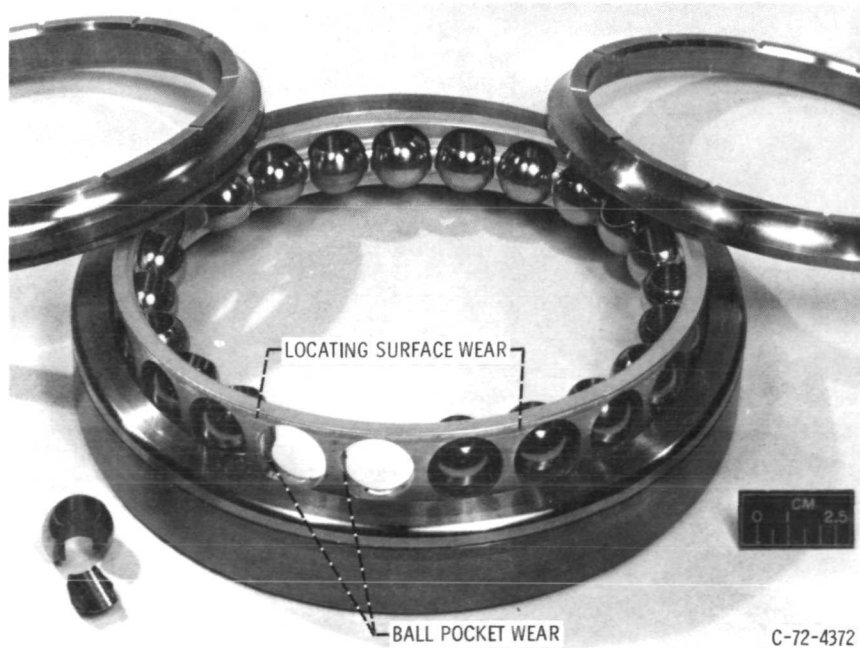
(b) BEARING TORQUE.

Figure 7. - Bearing outer-race temperature and torque as a function of shaft speed at 17 800 newton (4000 lb) thrust load. Type II ester oil MIL-L-23699 at 367° K (200° F) oil inlet temperature. Oil flow 4, 35 and 5.80 liters/min (1, 15 and 1, 57 gpm).



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Figure 8. - Solid ball bearing S/N 433A after 17 800 newton (4000 pounds) load run at speeds from 6670 to 20 000 rpm (1 to 3 million DN) showing ball pocket and outer locating surface cage wear.



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Figure 9. - Drilled ball bearing S/N 377A after 17 800 newton (4000 pound) load run at speeds from 6670 to 13 000 rpm (1 to 2 million DN) showing ball pocket and locating surface cage wear. Ball fractured at 13 000 rpm.

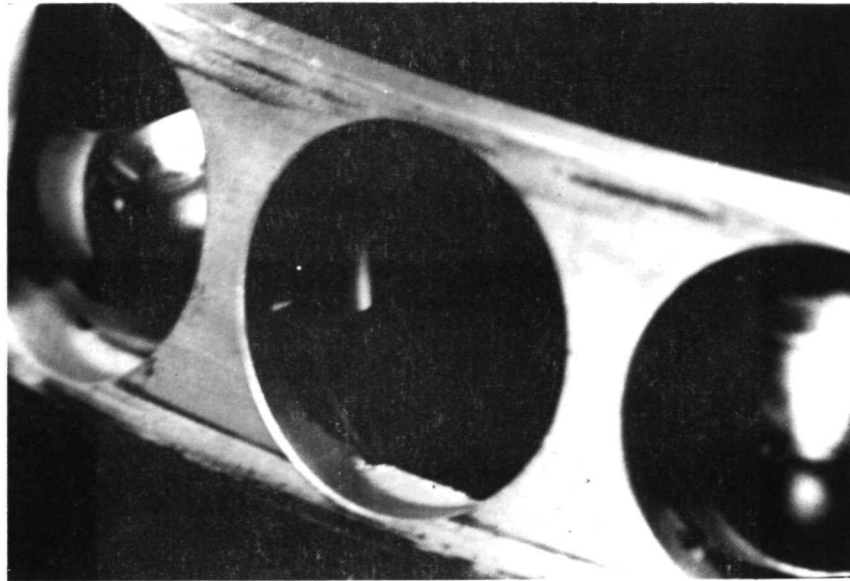


Figure 10. - Portion of cage from drilled ball bearing S/N 377A after 17 800 newtons (4000 pounds) load test.

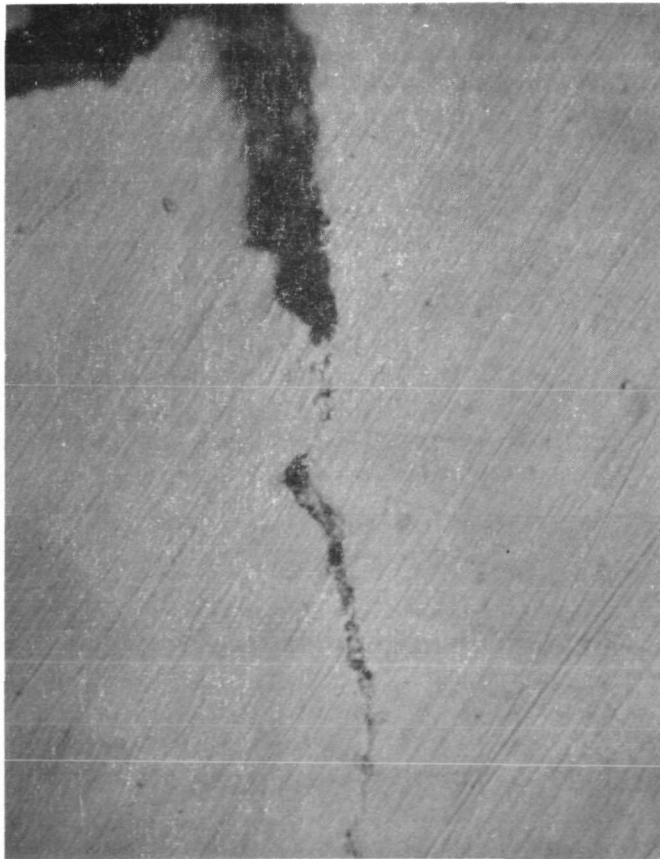


Figure 11. - Crack in ball from drilled ball bearing S/N 377A, perpendicular to axis - X1000. (Bore of drilled ball is near top of photo.)

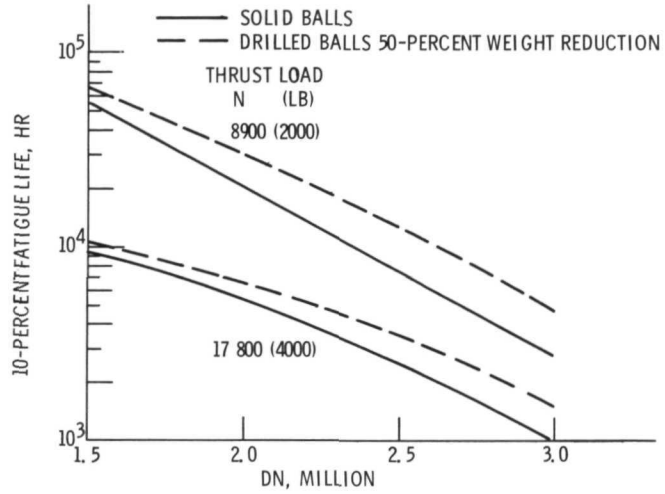


Figure 12. - Theoretical fatigue life of 150-mm bore ball bearing with solid and with 50-percent drilled balls for two thrust loads. (Based on analysis of ref. 2.)