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PAST PERFORMANCE ANALYSIS OF HPOTP BEARINGS

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Materials and Processes Laboratory

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16. ABSTRACT

The past performance analysis conducted on three High Pressure Oxygen Turbopump (HPOTP) bearings from the Space Shuttle Main Engine (SSME) is presented. Metallurgical analysis of failed bearing balls and races, and wear track and crack configuration analyses were carried out. In addition, one bearing was tested in laboratory at very high axial loads. The results showed that the cracks were surface initiated and propagated into subsurface locations at relatively small angles. Subsurface cracks were much more extensive than was apparent on the surface. The location of major cracks in the races corresponded to high radial loads rather than high axial loads. There was evidence to suggest that the inner races were heated to elevated temperatures.

A failure scenario was developed based on the above findings. According to this scenario the HPOTP bearings are heated by a combination of high loads and high coefficient of friction (poor lubrication). Very high internal radial loads can be generated by loss of bearing internal clearance resulting from localized heating. These internal radial loads are apparently responsible for the bearing failures. Different methods of extending the HPOTP bearing life are also discussed. These include reduction of axial loads, improvements in bearing design, lubrication and cooling, and use of improved bearing materials.

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CREDITS

The metallurgy/metallography and associated efforts were contributed by B. N. Bhat, Metallurgy Research Branch, Metallic Materials Division; the bearing analysis and turbopump implications were contributed by F. J. Dolan, Lubrication and Surface Physics Branch, Engineering Physics Division, both of Materials and Processes Laboratory, MSFC.

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INTRODUCTION

At present the service life of the Space Shuttle Main Engine (SSME) high pressure oxygen turbopump (HPOTP) bearings appears to be limited by premature cracking and spalling. Considerable effort has been expended to understand and alleviate this problem. For example, detailed analyses of spalled bearings have been made by Battelle [1]. It appears that the HPOTP bearings are sometimes subjected to extremely high axial loads during start-up and shut-down. Transient axial loads as high as 3636 kg (8,000 lb) have been measured; estimated values are even higher, approximately 4545 kg (10,000 lb). While attempts are continually being made to reduce the high transient axial loads, detailed failure analyses of spalled balls and races were conducted for the purpose of improving our understanding of the spalling problem so that suitable solutions to the problem may be identified. In addition, a HPOTP bearing was tested in the Materials and Processes (M&P) laboratory, Marshall Space Flight Center (MSFC), by deliberately subjecting it to very high axial loads for the purpose of comparison with bearings tested in the HPOTP. The details of these investigations are presented in this report. The fracture analyses of these HPOTP bearing balls have previously been reported [2]; fracture analysis of HPOTP races are presented in this report. Furthermore, HPOTP bearing failure mechanisms are discussed in the light of the findings of the fracture analyses. Finally, different approaches to bearing life extension are presented.

TEST PROCEDURE

Three HPOTP bearings were selected for this investigation. Details of these bearings such as serial number, test history, and as-received condition are given in Table 1. Pertinent remarks on each bearing are given in the last column of Table 1. The detailed test history of these bearings is given in Tables 2, 3, and 4. Bearing S/N 8517929, which was in relatively good condition, was further tested at very high axial loads in the M&P Laboratory. The test arrangement is shown in Figure 1. The bearing elements were cleaned before assembly. They were placed in a fixture duplicating the turbopump bearing fits, installed in a testing machine, and loaded in increments of 455 kg (1000 lb). During the test, the bearing was rotated manually at approximately 10 rpm, using the lever arrangement shown in Figure 1. The load versus time curve for this test is given in Figure 2. The first noticeable sign of yielding was observed at 10,000 kg (22,000 lb) axial load. The axial load was increased to a maximum of 12,727 kg (28,000 lb) and held, but it gradually decreased to 11,364 kg (25,500 lb) because of yielding. The load stabilized at 11,364 kg (25,500 lb) and no further yielding occurred. The test was stopped after about 300 revolutions. The load was removed and the bearing was disassembled and examined for damage.

TABLE 1. SOME DETAILS OF THE HPOTP BEARINGS INVESTIGATED IN THIS REPORT

Bearing	Identification	Test History	As-Received Condition	Comments
1	S/N 8549531 Position No. 3 HPOTP 9008 Engine 2004	Total run time 2406 sec, 1090 sec at FPL, 7 starts. (Table 2)	Heavily spalled balls, inner and outer races. Prominent wear tracks. Shoulder damage on inner race.	Examined by Battelle. Both balls and races were examined.
2	S/N 8517824 Position No. 4 HPOTP 2007 Engine 0009	Used on MPTA testing. Total run time 6091 sec, 17 starts (Table 3)	Prominent wear tracks in balls, inner and outer races. Spalls on inner race and two balls.	Only inner race was examined.
3	S/N 8517929 Position No. 4 HPOTP 0103 Engine 2003	Total run time 2311 sec, 16 starts (Table 4)	Wear track clearly visible in balls and races. No visible spalls.	This bearing was used for high axial load test at M&P Laboratory.

TABLE 2. OPERATING CONDITIONS FOR HPOTP 9008, ENGINE NO. 2004 (BEARING S/N 8549531)

Test No.	Duration (sec)	Mainstage (sec)	Power Level Durations (Percent)					
			90	100	103	105	106	109
902-187	100.00	95.90		95.90				
902-188	110.00	105.86	17.0	45.86	30.00		13.00	
902-189	125.06	120.92	11.0	83.92		16.00		10.00
902-190	241.02	236.84		235.84		1.00		
902-191	610.00	605.83	7.10	230.63		8.00		360.10
902-192	610.00	605.83	7.10	230.63		8.00		360.10
902-193	610.00	605.83	7.10	230.63		8.00		360.10

TABLE 3. OPERATING CONDITIONS FOR HPOTP 2007, ENGINE NO. 0009 (BEARING S/N 8517824)

Test No.	Duration (sec)	Mainstage (sec)	Power Level Durations (Percent)					
			65	70	30	90	100	102
901-266	1.50	0.00						
901-267	39.62	35.76					35.76	
901-268	520.00	516.20				92.50	423.70	
901-269	2.85	0.00						
901-270	520.00	515.86				92.50	423.70	
901-271	823.00	818.88				301.60	517.88	
901-272	665.03	660.85				43.50	617.35	
901-273	519.90	515.84				92.50	423.30	
901-274	520.00	515.91				92.50		423.01
901-275	520.00	515.83				92.50	423.30	
901-276	520.00	515.82				92.50	423.32	
901-277	10.00	5.88					5.88	
901-278	10.00	5.87					5.87	
901-279	300.00	295.86				50.00	245.86	
901-280	520.00	515.92				92.50	423.42	
750-061	300.00	296.12	36.0	101.00	13.00		146.12	
750-062	300.01	296.12	6.50		3.50		286.12	

TABLE 4. OPERATING CONDITIONS FOR HPOTP 0103, ENGINE NO. 2003 (BEARING S/N 8517929)

Test No.	Duration (sec)	Mainstage (sec)	Power Level Durations (Percent)			
			70	80	90	100
901-226	1.40	0.00				
901-227	4.35	0.00				
901-228	15.35	11.19			11.19	
901-229	23.00	18.90			11.40	7.50
901-230	18.34	13.96			13.96	
901-231	60.00	56.08				56.08
901-232	520.00	516.10			89.00	427.10
SF005A2001	1.47	0.00				
SF005001	55.68	51.76				51.80
SF006001	19.36	15.49				15.50
SF006003	10.35	6.49				6.50
SF006004	508.70	504.79	58.70	65.00	45.00	336.00
SF007001	5.42	1.50				1.50
SF007002	523.43	519.49	97.20	93.00		329.30
SF008001	539.48	535.64	121.70	70.00		344.26
SF009001	4.72	0.50				0.50

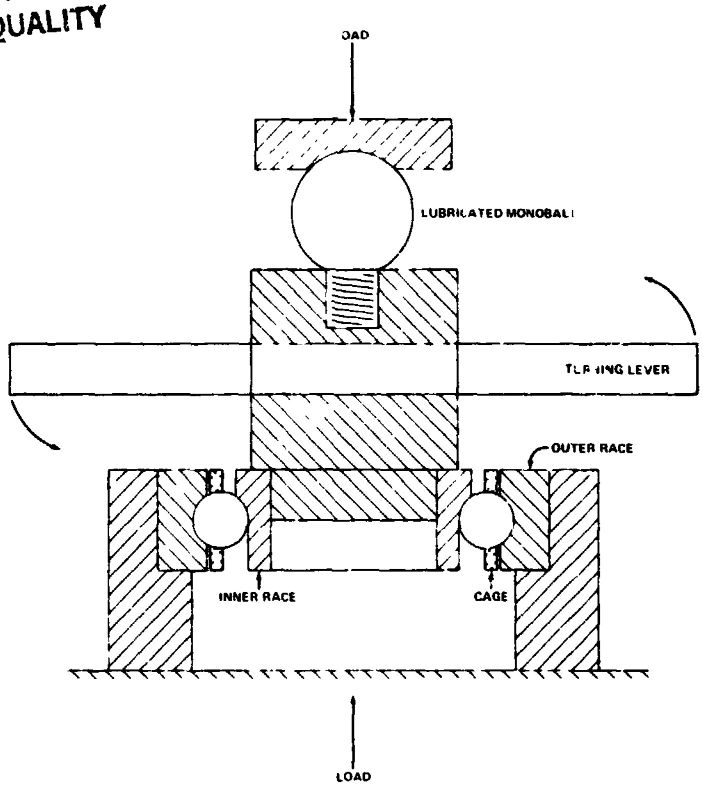

Figure 1. Arrangement for testing HPOTP bearing.

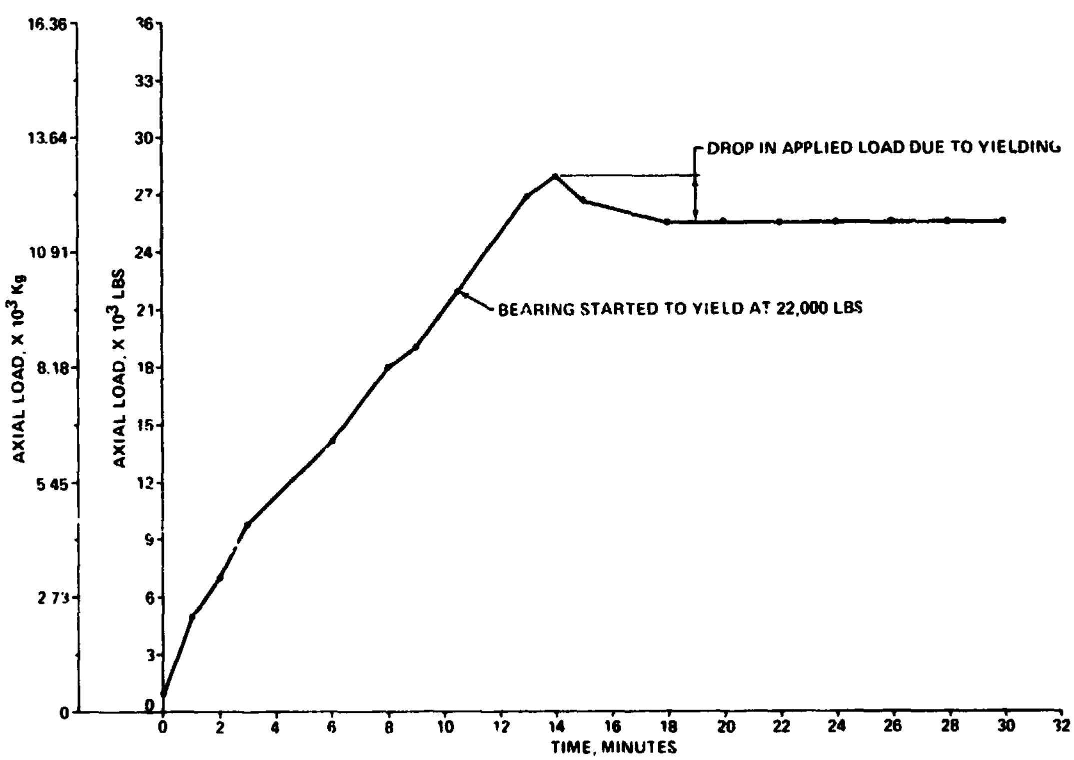

Figure 2. Plot of axial load versus time in laboratory testing of HPOTP bearing S/N 8517929.

METALLURGICAL ANALYSIS AND RESULTS

The bearing elements (viz, inner race, outer race, and balls) were initially examined under a stereo viewer. Any spalls, cracks, or surface distress were noted. The races were sectioned both parallel and perpendicular to the wear track. The sections were mounted, polished, and etched in Vilella's reagent and examined in a metallograph. The results of these examinations are summarized in Table 5 for the three bearings studied in this work. Photomicrographs and macrographs of cracks and spalls were taken and are presented in Figures 3 through 13. These figures are self-explanatory and they illustrate the cracking patterns present in the bearing elements. Photomicrographs of spalled balls were omitted because they have been reported previously in References 1 and 2.

TABLE 5. EXAMINATION RESULTS FOR HPOTP BEARINGS

Bearing	Inner Race	Outer Race	Balls
S/N 8549531	Extensive spalling, shoulder damage. Major spalls located away from the shoulder. Numerous cracks present. Cracks originate at surface, propagate inside at relatively small angles, 15 to 30 degrees. Angles often increase to 45 degrees or more. Crack branching observed. Branches are often at 90 degrees to each other and 45 degrees to the surface. Large cracks frequently run substantially parallel to the surface. Maximum crack depth is approximately 0.25 mm (0.010 in.). Oxide film is present in the cracks at depths of 0.08 to 0.25 mm (0.003 to 0.010 in.). Thickness of oxide film is variable, 1 to 5 m. No oxide film present on the surface (illustrated in Figs. 3, 4, and 5 and macrograph in Fig. 8).	Spalling present, reported in Reference 1. Also contained shallow cracks (Fig. 6), up to 0.025 mm (0.001 in.) deep, originating at the surface, and propagating inside at small angles.	Cracks and spalls, reported in References 1 and 2.
S/N 8517824	Crack pattern very similar to that of S/N 8549531 described above. Oxide film in cracks is thicker, up to 25 m (Fig. 7). Rounded metal particles present in the oxide.	No spalls present. Not examined metallographically.	Not examined metallographically.
S/N 8517929	Severe shoulder damage. Prominent wear track located at the shoulder. Numerous cracks and small spalls present in the wear track. The cracks appear to originate at the surface, starting at small angles and propagating inwards at increasing angles. Deformed metal is evident up to a depth of 0.1 mm (0.004 in.). Deepest crack approximately 0.03 mm (0.012 in.) deep. No crack branching was observed. (Figs 9, 10, and 11, macrograph in Fig. 8)	No visible spalls. Showed very shallow, up to 0.025 mm (0.001 in.), surface cracks starting at the surface at small angles. Angles increase with depth (Fig. [illegible])	No visible spalls. Showed very shallow, up to 0.008 mm (0.0003 in.), surface cracks, starting at small angles (Fig. 13).

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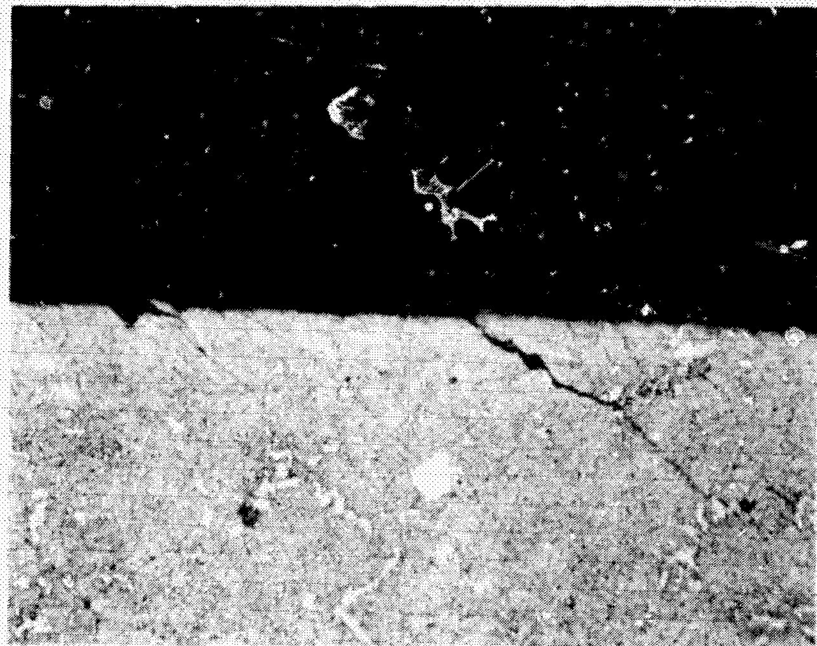
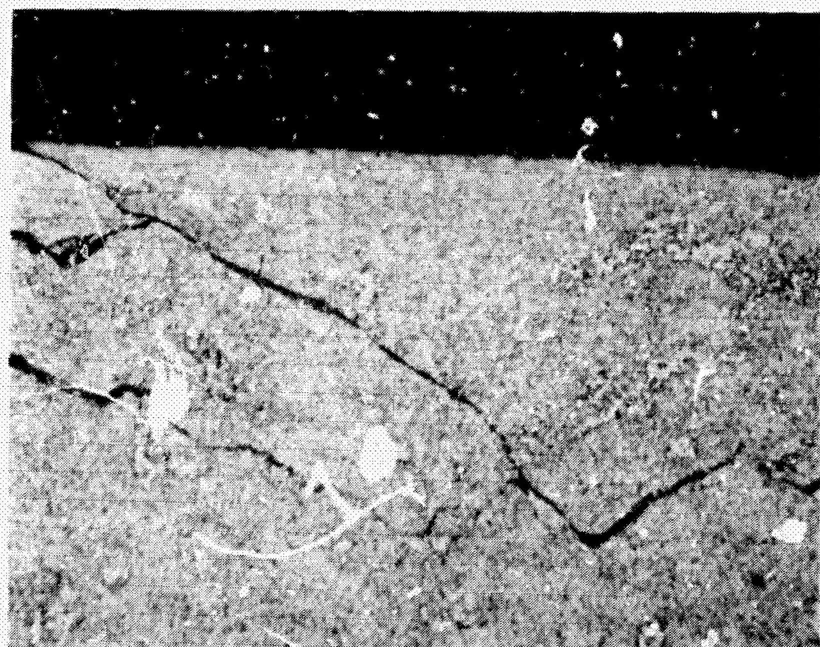
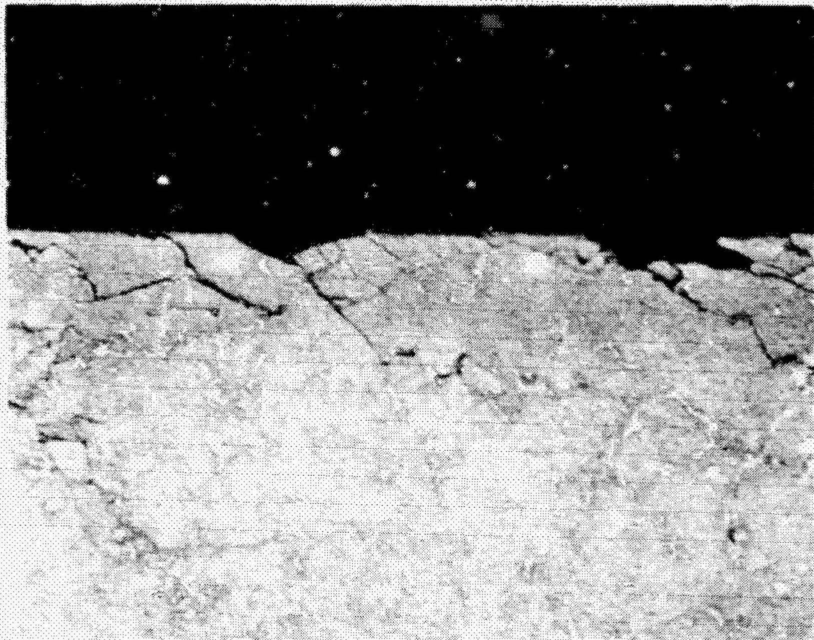


A. HPOTP bearing S/N 8549531, original mag. 400X.

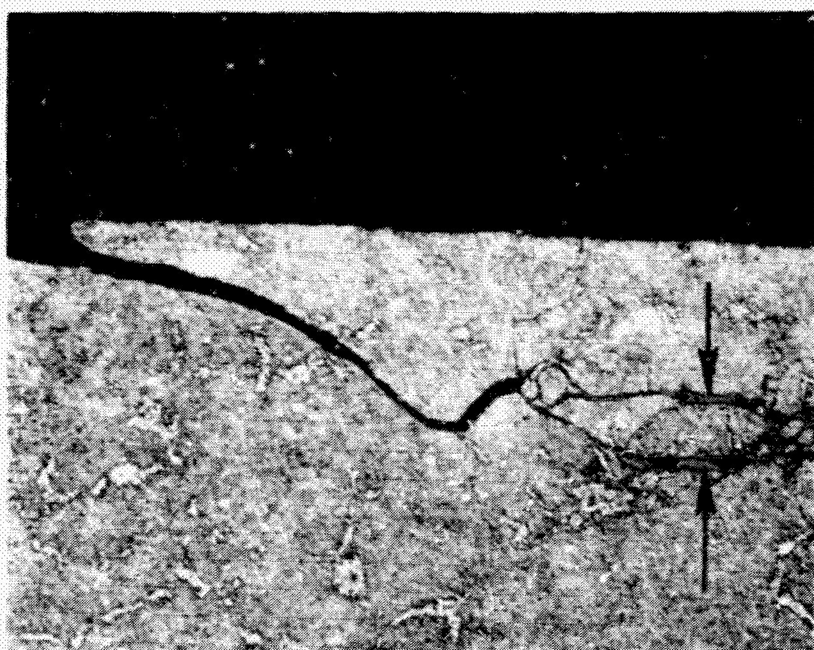


B. HPOTP bearing S/N 8549531, original mag. 400X.

Figure 5. Photomicrographs of inner race sectioned parallel to the wear track. Vilella's reagent. Note the crack origin at surface (A), and also note branching at 90 degrees (B).



A. HPOTP bearing S/N 8549531, original mag. 200X.



B. HPOTP bearing S/N 8549531, original mag. 200X.

Figure 4. Photomicrographs of inner race sectioned parallel to the wear track. Vilella's reagent. Note the cracking and spalling (A) and oxide in the cracks (B).

HPOTP bearing S/N 8549531, original mag. 100X.

Figure 5. Photomicrograph of crack pattern in the inner race of bearing, S/N 8549531.

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HPOTP bearing S/N 8549531, original mag. 400X.

Figure 6. Photomicrograph of outer race, sectioned parallel to the wear track. Note surface crack initiation. Crack depth approximately 0.025 mm (0.001 in.).

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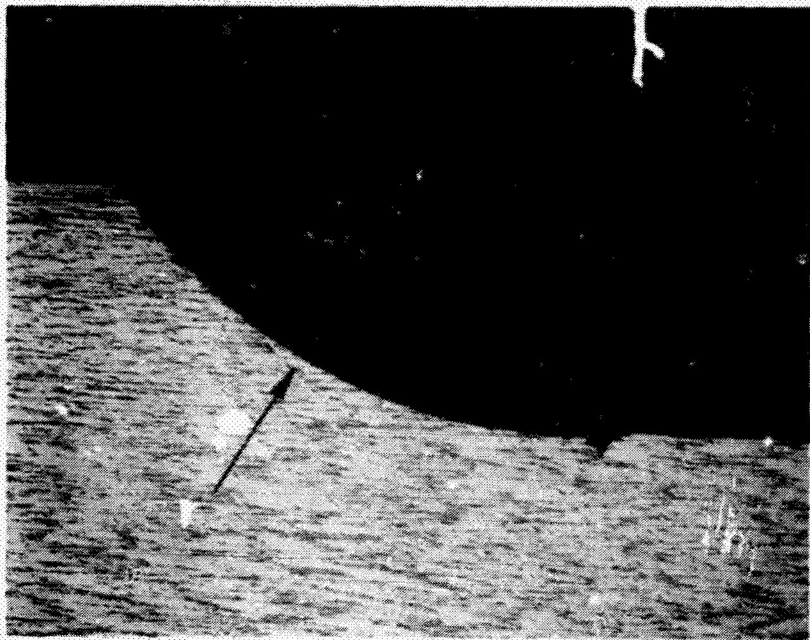

A. HPOTP bearing S/N 8517824, 10X.

B. HPOTP bearing S/N 8517824, 400X.

Figure 7. Photomacrograph (A) and photomicrograph (B) of inner race sectioned perpendicular to the wear track. No visible shoulder damage. Location of major spalls shown by arrow in A. Note the thick oxide layer filling the crack in B.

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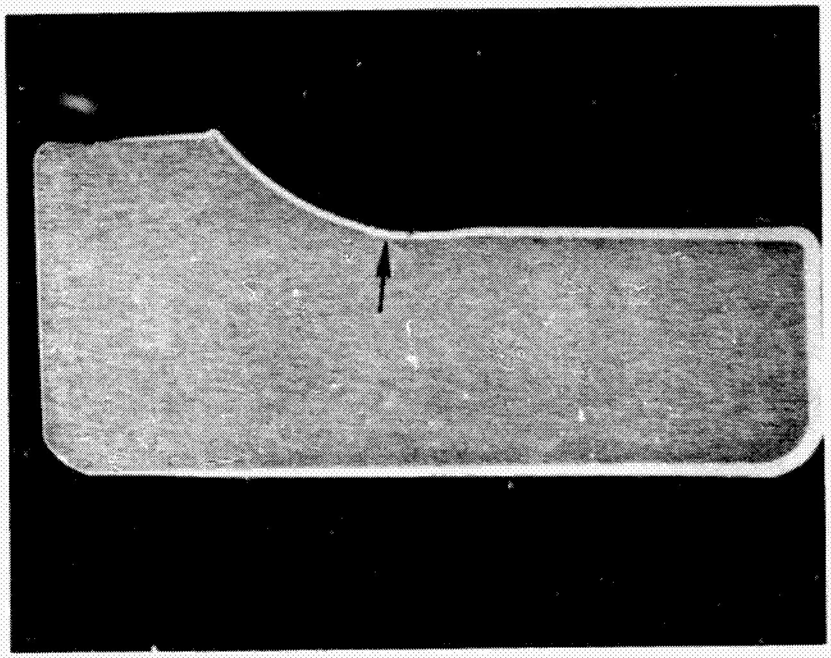
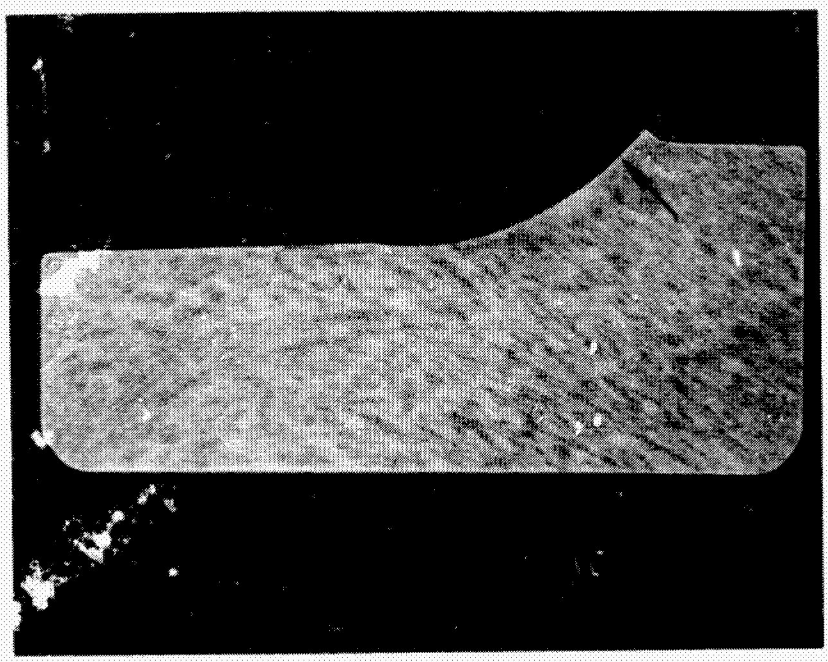


A. Bearing S/N 8549531, original mag. 5X.



B. Bearing S/N 8517929, original mag. 5X.

Figure 8. Photomacrographs of inner race sectioned perpendicular to the wear track. (A) HPOTP 9008, engine 2004 nickel plated, (B) laboratory test. Note the shoulder damage due to high axial loads. Arrows show the areas where major cracking occurred.

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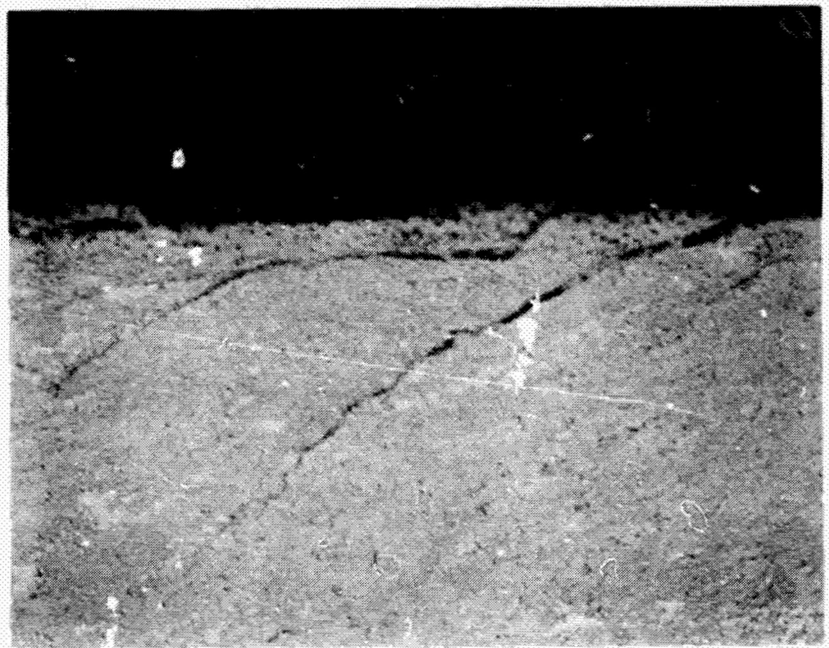
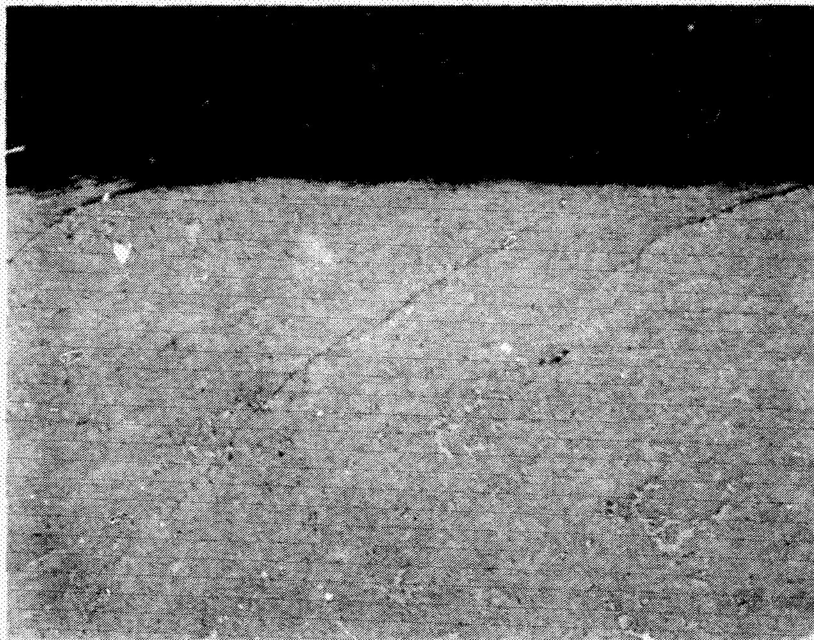


A. Bearing S/N 8517929, original mag. 400X.

B. Bearing S/N 8517929, original mag. 400X.

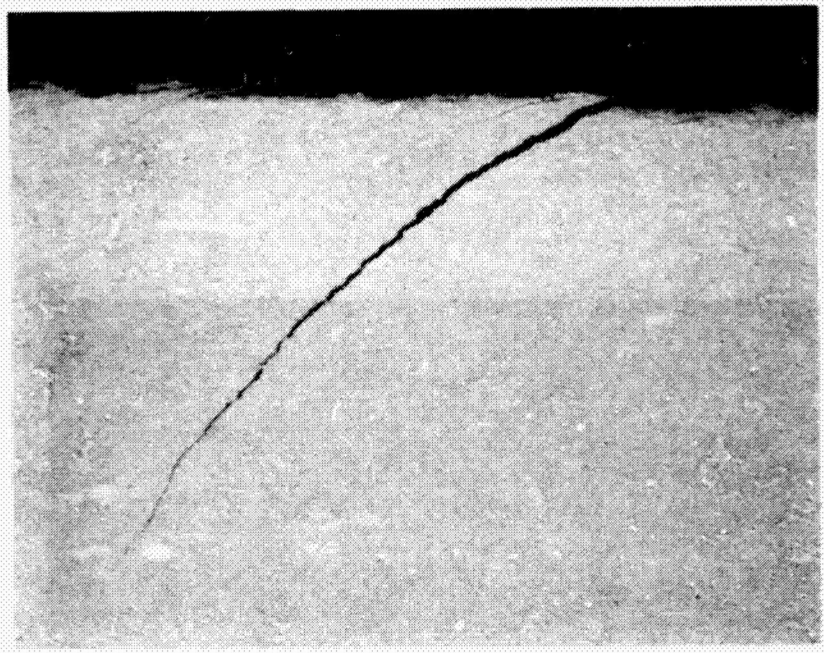
Figure 9. Photomicrographs of inner race sectioned parallel to the wear track. Vilella's reagent. Note the crack origin at the surface. The cracks start at small angles to the surface and propagate inward at increasing angles.



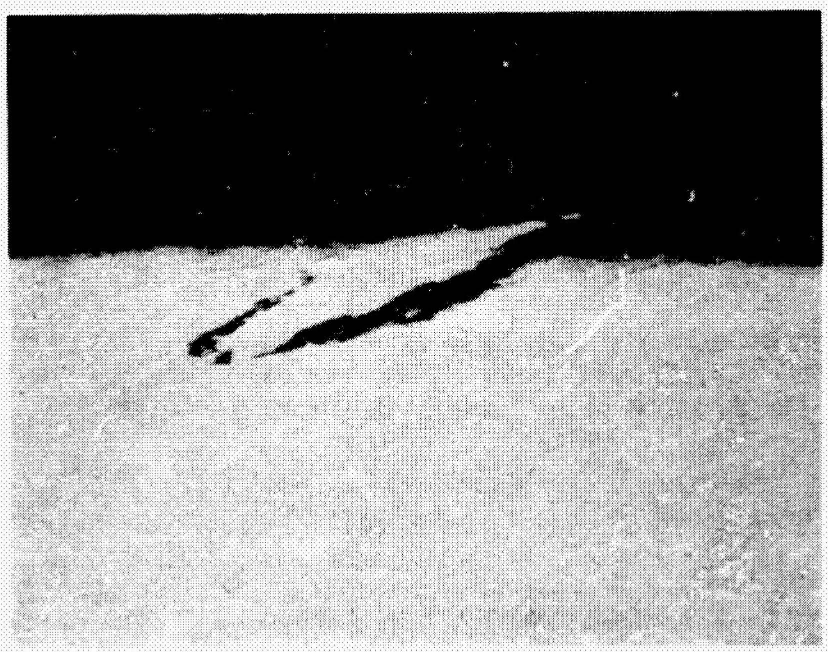
A. Bearing S/N 8517929, original mag. 200X.

B. Bearing S/N 8517929, original mag. 200X.

Figure 10. Photomicrographs of inner race sectioned parallel to the wear track. Note the low angles of crack initiation. Deformed metal is evident in the first few thousanths of an inch.



A. Bearing S/N 8517929, original mag. 200X.



B. Bearing S/N 8517929, original mag. 200X.

Figure 11. Photomicrograph of a deep crack (A) and an incipient spall (B) on inner race.

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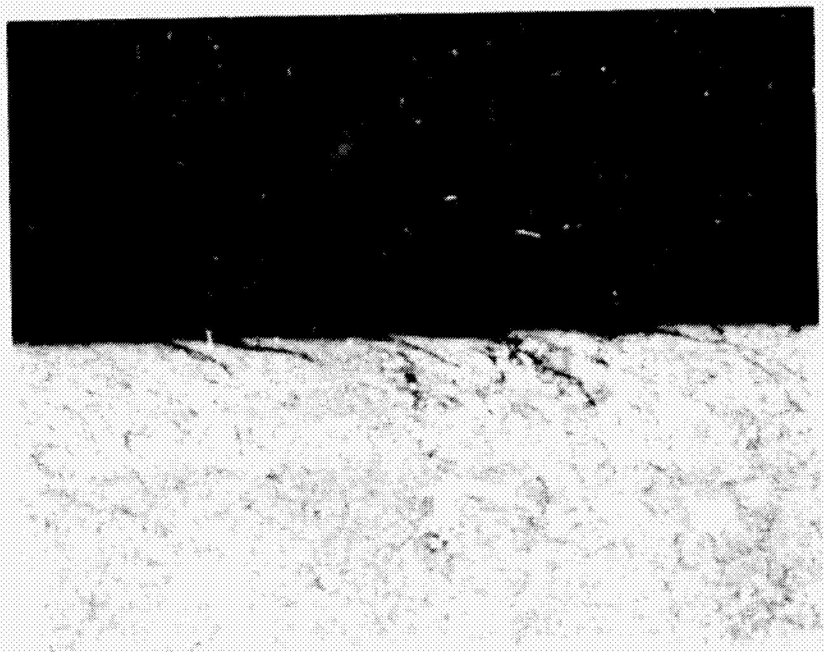

Bearing S/N 8517929, original mag. 400X.

Figure 12. Photomicrograph of outer race sectioned parallel to wear track. Tested at high axial loads. Note surface crack initiation. Crack depth approximately 0.025 mm (0.001 in.).

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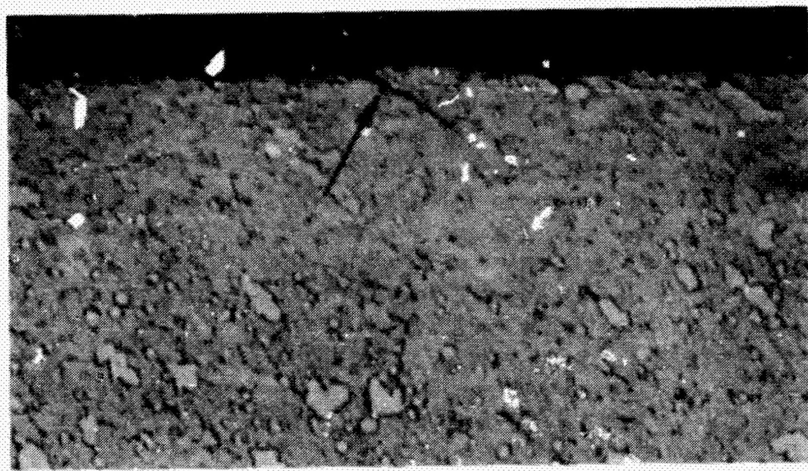
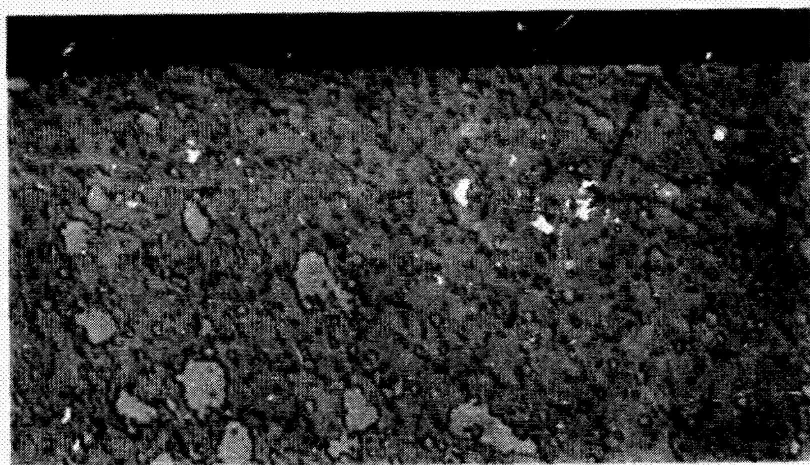
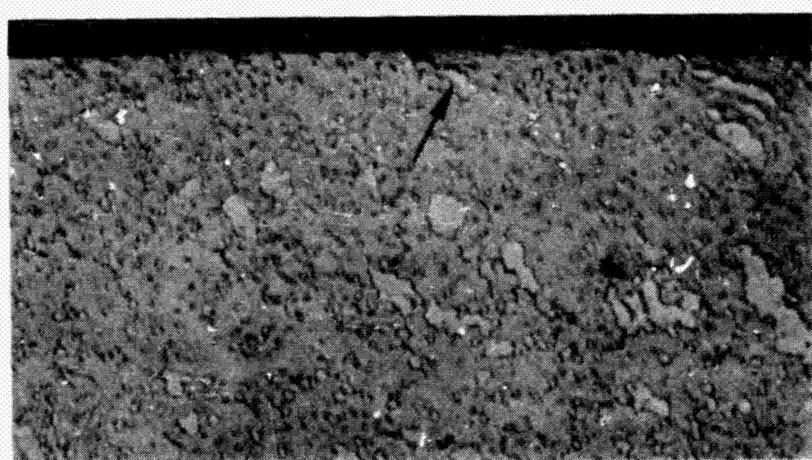


A. Bearing S/N 8517929, mag. 800X.



B. Bearing S/N 8517929, mag. 800X.



C. Bearing S/N 8517929, mag. 800X.

Figure 13. Photomicrographs of cracks in bearing ball. Note the surface cracks and carbides associated with crack origin (arrows). Crack depth approximately 0.008 mm (0.0003 in.).

DISCUSSION

A. Crack Configuration

Crack initiation propagation, depth, etc., for inner races of HPOTP bearings are compared with those of HPOTP balls in Table 6. It is significant that the cracks appear to be largely surface initiated in both cases and propagate inward at relatively small angles. Crack branching is common to both, and angles associated with branching are similar. However, there are some significant differences. Cracks sometimes propagate very deep in the balls, but relatively shallowly in the races, usually at a depth less than 0.25 mm (0.010 in.). Major cracks in the inner races are substantially parallel to the surface whereas a major crack in a ball from HPOTP 9008 was 2.29 mm (0.090 in.) deep. This particular crack started at a shallow angle but, after some depth, was found to propagate at nearly 90 degrees to the surface [2]. In general, subsurface damage in the inner races appears to be much more severe and widespread than is evident on the surface. This result may be significant because the reusability of bearings is based on a simple visual examination, which may be inadequate for screening bad races.

Crack branching seems to occur at 90 degrees and at 45 degrees to the surface. This crack pattern may be the result of a high, concentrated load [2]. Surface initiation of cracks suggests that they originated under conditions of poor lubrication or high coefficient of friction [3]. Since the service lives of these races have been relatively short, Hertzian stresses in the races must have been very high. It is difficult to estimate the stresses accurately, however. Battelle has attempted to predict the maximum Hertzian stresses and relate them to depth of cracks assuming that the major cracks are associated with the maximum Hertzian stress [1]. On the basis of this analysis, crack depths of 0.25 mm (0.010 in.) correspond to Hertzian stresses in excess of 4.23 GPA (600 ksi) which is greater than the yield strength of 440C steel [approximately 1.83 GPA (260 ksi)]. Obviously, the raceway would plastically deform under these high stresses. It should be noted that the Battelle report refers to cracks in the inner race as subsurface cracks. This statement appears to be in error, because Battelle evaluated a transverse section of the race taken at 90 degrees to the wear track. In this case, surface initiated cracks could appear as subsurface cracks and erroneously be interpreted as such. Therefore, it appears that bearing races (and balls) should be sectioned parallel to the wear track for the purpose of crack analysis.

A comparison of crack configuration of the HPOTP bearing tested under heavy axial loads is also given in Table 6. Main differences are (1) lack of branching, (2) presence of a deformed metal layer, and (3) absence of an oxide film in cracks. No major spalls were observed. The reasons for these differences are not well understood and can only be speculative. One possible reason is that the HPOTP bearings were run at a high speed (about 30,000 rpm versus 10 rpm in the lab test). Other factors could be the total number of cycles of stress repetition and the temperature. In addition, 440C steel is likely to be more brittle at cryogenic temperatures than at room temperature. Still another possibility is the transformation of retained austenite. Under stress and thermal cycles of HPOTP operation, retained austenite could transform into brittle, untempered martensite or bainite which could fracture in a brittle manner.

Another interesting difference between the laboratory test and HPOTP test is the fact that the crack propagation direction changed in the HPOTP test bearings after some depth. The major cracks appear to run essentially parallel to the raceway

TABLE 6. COMPARISON OF CRACKS IN HPOTP BALLS AND INNER RACE

Item	Inner Race	Balls (Ref. 2)	Inner Race, Tested at High Axial Load
Crack Initiation	Surface initiated. No subsurface initiated crack observed so far.	Mostly surface initiated, occasionally subsurface initiated.	Surface initiated.
Crack Propagation	Cracks propagate inside at relatively small angles. Angles generally increase with depth, up to about 45 degrees to the surface. Major cracks substantially parallel to the surface.	Cracks propagate inside at relatively small angles. Angles generally increase with depth, up to 90 degrees to the surface.	Cracks propagate inside at relatively small angles. Angles generally increase with depth, up to as high as 80 degrees. Major cracks at 45 and 60 degrees to the surface.
Crack Depth	Up to 0.25 mm (0.010 in.).	Up to 2.29 mm (0.090 in.).	Up to 0.30 mm (0.012 in.).
Branching	Numerous branching, usually at 90 degrees, and at 45 degrees to the surface.	Numerous branching, usually at 90 degrees and at 45 degrees to the surface.	No branching.
Spalling	Numerous spalls located on the wear track.	A few spalls located on the wear track.	No major spalls, but incipient spalls present.
Oxide Film	Oxide film is present, starting at about 0.08 mm (0.003 in.) deep and all the way up to 0.25 mm (0.010 in.) deep.	No oxide film present.	No oxide film present.
Plastic Deformation	Not evident.	Not evident.	Deformed layer of metal present to a depth of about 0.05 mm (0.002 in.).

surface or wear track (Fig. 5). This behavior is not observed in the laboratory test bearing and is in contrast to the propagation pattern in the balls in which cracks seem to propagate at increasing angles. This apparent crack stabilization in the races is not well understood but is very fortunate, because it suggests that the inner race is not likely to fail catastrophically by transverse cracking.

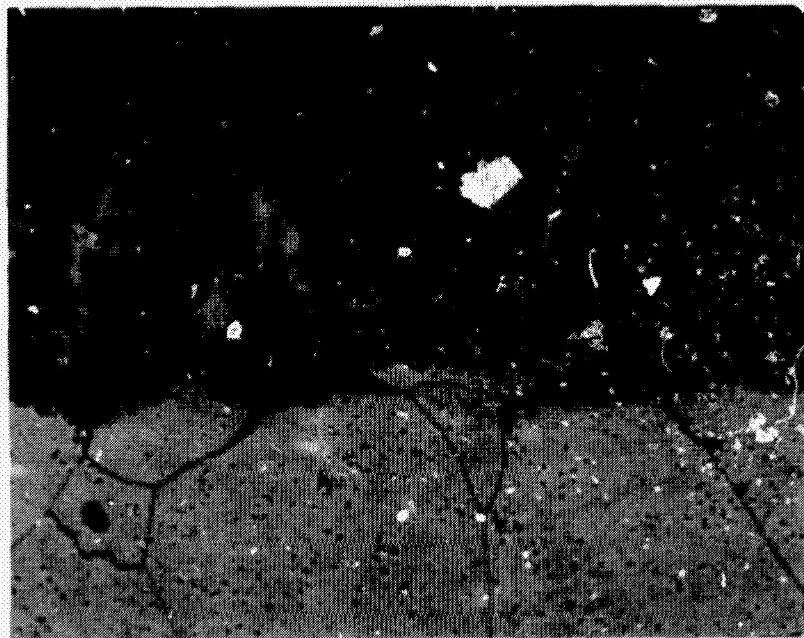
There were very few cracks present in the balls or the outer race in the laboratory test bearing. The cracks present were very shallow, less than 0.025 mm (0.001 in.) deep. Many of these cracks seemed to originate at carbides (Fig. 13). Overall, it appears that under conditions of very high axial loads and practically zero radial loads, the inner race is damaged much more than either the balls or the outer race. This conclusion is also supported by the general experience of the bearing industry where inner races are usually observed to fail first by fatigue [4]. In HPOTP bearings used in engine tests, however, it is claimed that the balls spall before the inner race [5]. The result of the high axial load test conducted at M&P Laboratory does not support this claim.

Another important result of the laboratory test is that the bearing did not fail catastrophically even under axial loads in excess of 25,000 lb. It is r uring that the bearing can withstand such occasional very high axial loads witho astrophic failure.

B. Oxide in Cracks

Another interesting feature of cracks in the HPOTP bearing raceway is the presence of oxide. For example. Figure 5 shows oxide films, 1 to 25 μm thick, usually present below the surface, starting at a depth of 0.076 mm (0.003 in.). There is no oxide film present on or near the surface, however. Furthermore, no such oxide film was observed in cracks in HPOTP bearing balls. The presence of an oxide film is indicative of elevated temperatures. Therefore, experiments were conducted to determine the temperature at which oxide layers, similar to the ones observed in the HPOTP bearing races, might form. A piece of 440C steel was heated to 482°C (900°F) and 816°C (1500°F) in air (20 percent O_2, 79 percent N_2) for 1 hr each. The oxide film formed under these conditions was extremely thin, less than 1 μm. Only surface blackening occurred. At 1316°C (2400°F), however, a very thick, 0.635 mm (0.025 in.) oxide film was obtained in 30 min. The oxide film cracked and flaked off when cooling to room temperature. Figure 14 is a photomicrograph of oxide formed on the metal. The oxide is porous, and the metal-oxide interface is rough. By comparison, the oxide film in the HPOTP inner race does not contain pores (Fig. 7) and has a very smooth interface with the metal. The metal particles in the oxide appear round as opposed to irregularly shaped pieces in Figure 14. These results suggest that the oxide in the bearing races were probably formed at an elevated temperature high enough to melt the oxide. Once molten, the oxide could readily flow and fill up the cracks. An energy dispersive X-ray analysis of the oxide layer showed that the oxide had about the same ratio of iron to chromium (the two major components of 440C steel) as the parent metal. Temperatures in excess of 1400°C (2552°F) would be required to melt $FeO-Cr_2O_3$ at approximately 80 percent FeO [6]. However, the nature of the oxides formed at high oxygen pressures present in the turbopump is not known. Chromium is known to form high oxides (e.g., CrO_2, CrO_3) at low temperatures and high oxygen pressures [7], but such information is lacking for iron. Handbook of Physics and Chemistry reports a melting point of 300°C

ORIGINAL PAGE
BLACK AND WHITE PHOTOGRAPH



440C Steel 400X

Figure 14. Photomicrograph of 440C steel held at 1316°C (2400°F) in air, showing the oxide layer. Note the porosity of oxide and roughness of metal-oxide interface.

for CrO_2 and 196°C for CrO_3, but these values are not supported by other workers in the field [7,8]. Binary phase diagrams of FeO-CrO_3 of FeO-CrO_2 are not available; hence, it is difficult to determine the melting temperature of the oxide found in the HPOTP bearing races operating in a high pressure oxygen environment. At the present time, however, available data strongly suggests that the HPOTP bearing raceways became locally very hot. There is no evidence of softening or tempering of martensite next to the oxide [1]. Figure 15 shows loss of hardness of martensite at 482°C (900°F) as a function of time for 440C steel. It takes approximately 12 hr at 482°C (900°F) to lose one point of hardness on the Rockwell C scale. Time taken for softening at elevated temperatures is not known but is likely to be shorter. At 1400°C (2552°F) significant softening may be expected in a matter of minutes. These considerations suggest that high temperatures, if attained, must have been localized and of short duration. Such localized heating could have resulted from plastic deformation under extremely high Hertzian stresses or from frictional heat generated by rubbing of fractured pieces of metal. Since the austenitizing temperature for 440C steel is approximately 1925°F, temperatures higher than this which were later quenched by lox may not show any loss of hardness.

As noted earlier, no oxide film was observed on the raceway or immediately below it. The oxide appears to be located deeper inside, starting at 0.076 mm (0.003 in.). This result can be explained by the fact that the raceway is well cooled by liquid oxygen flowing at the surface but not deeper inside. Heat generated below the surface must be dissipated to the surface by conduction. This condition could result in subsurface areas running hotter than the surface during operation of the HPOTP. The heated region appears to be 0.076 to 0.25 mm (0.003 to 0.010 in.) deep.

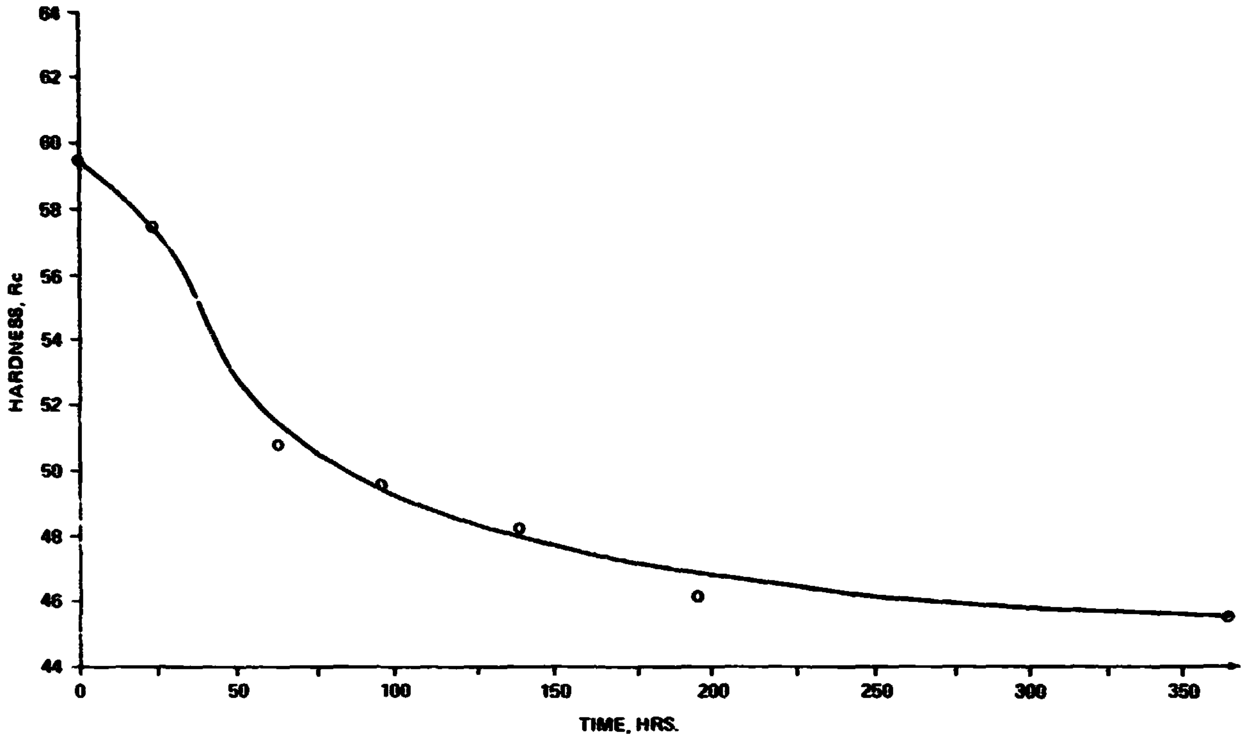


Figure 15. Hardness versus time at 482°C (900°F) for 440C steel.

Hertzian stresses and the radial loads associated with these crack depths are extremely high. A crack depth of 0.25 mm (0.010 in.) means Hertzian stresses in excess of 4.23 GPA (600 ksi) [1]. For HPOTP bearings these stresses translate into either high axial loads greater than 5455 kg (12,000 lb) or radial loads greater than 1364 kg (3,000 lb) [1]. Subsurface heating will also cause the subsurface layer of metal to expand and generate hoop stresses in the surface layer. These hoop stresses will add to the existing tensile stresses at the race surface and contribute to crack initiation.

It should be noted that no thick oxide film was found in the cracks in the HPOTP bearing balls. This result suggests that the balls are cooled better than the inner race and do not become hot enough to form a thick oxide film. Surface blackening of the balls, however, indicates a very thin oxide layer and suggests that ball surface temperatures could have reached values as high as 250° to 400°C (482° to 752°F).

C. Location of Cracks and Spalls

The most interesting feature of the inner race failure is the location of cracks and spalls. In the bearing tested under high axial loads (bearing S/N 8517929) areas of major cracking are located at the shoulder (Fig. 8, B). The shoulder is deformed under high axial loads. In the HPOTP bearing S/N 8549521, however, areas of major cracks and spalls are located away from the shoulder (Fig. 8, A), close to the lowest point on the raceway. In addition, there is evidence of shoulder damage. In the case of bearing S/N 8517824 (Fig. 7), a[illegible] of major cracks and spalls are about halfway in between the shoulder and the [illegible]ttom of the raceway. There is no evidence of shoulder damage on this race suggesting that this [illegible] h[illegible]d not experienced axial loads as high as the other HPOTP bearings. Nevertheless, depth and configuration of the cracks in this bearing are similar to those found in HPOTP bearing S/N 8549531. This evidence further suggests that spalling of the inner races in HPOTP bearings may not be associated with high axial loads as commonly believed. The location and configuration of cracks are consistent with high radial loads. This point is further discussed in the next section.

HPOTP BEARING FAILURE ANALYSIS

A number of factors appear to contribute to the premature failure of HPOTP bearings. For instance, high axial loads have been considered as a major cause of bearing failures. As a result of the present investigation, additional factors contributing to reduced bearing life have been identified, e.g., lack of good lubrication and thermal factors. All three factors are discussed below, and a HPOTP bearing failure scenario is developed. For convenience the analysis is started with a description of HPOTP operation.

A. Operation of HPOTP

Figure 16 is a cutaway view of the HPOTP. It incorporates a balance piston to counteract residual unbalanced axial forces during normal pump operating power levels (50 to 109 percent). The balance piston performs its function through a system of

FIGURE 16. HIGH PRESSURE OXYGEN TURBOPUMP

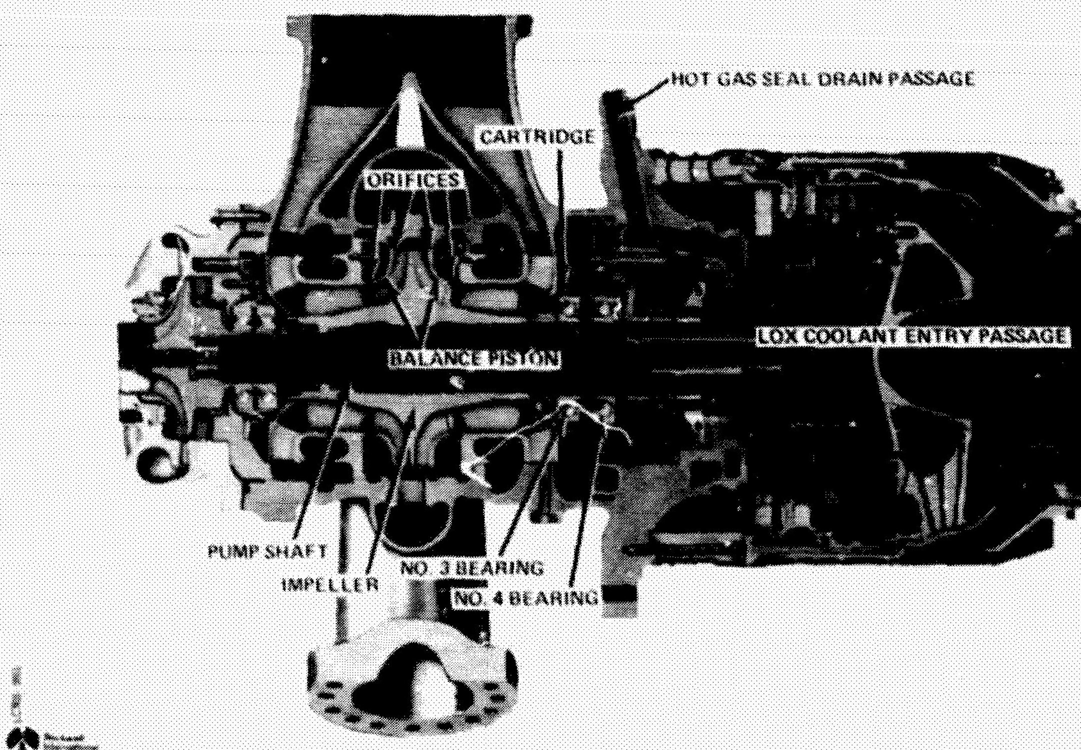


Figure 16. High pressure oxygen turbopump.

orifices on the impeller and in the housing. The pump shaft, impeller, bearings, and bearing cartridge are designed to slide axially, approximately ±1 mm (±0.040 in.) about the null position to bring about the desired orifice adjustments which produce the pressure (and forces) necessary for balancing action. Positive stops utilizing the turbine end bearings are incorporated at each end of the allowable axial travel. Any overtravel of the shaft causes additional axial loads (above and beyond the preload) in these bearings. In addition, any hangup of the bearings and cartridge will result in balance piston forces being applied through the turbine end bearings (Nos. 3 and 4) as additional axial load. It should be noted that the balance piston force required to counteract net pump axial force varies with pump speed. Balance piston force can be as high as 19,500 kg (43,000 lb) at Rated Power Level (RPL) (100 percent power level). Therefore, an axially free and dynamic shaft and balance piston is required at all operating pump speeds.

Clearances in the HPOTP turbine bearing and cartridge area are shown in Figure 17. The total diametral clearance between the turbine bearing outer races and the cartridge is 0.063 to 0.089 mm (0.0025 to 0.0035 in.). The total diametral clearance between the bearing cartridge and the support housing is 0.041 to 0.081 mm (0.0016 to 0.0032 in.). An axial hangup can result if the clearances listed above are reduced to zero by some means. Actually three different types of hangup can occur (1) one or both bearings can hang up in the cartridge, (2) the cartridge can hang up in the support housing, or (3) both types of hangup can occur simultaneously. This situation could develop because of the following reasons:

1) The main housing contains an internal primary lox seal drain passage, operating at about -173°C (-279°F) on one side of the housing while approximately 160 degrees to this there is a turbine primary hot gas seal drain internal passage

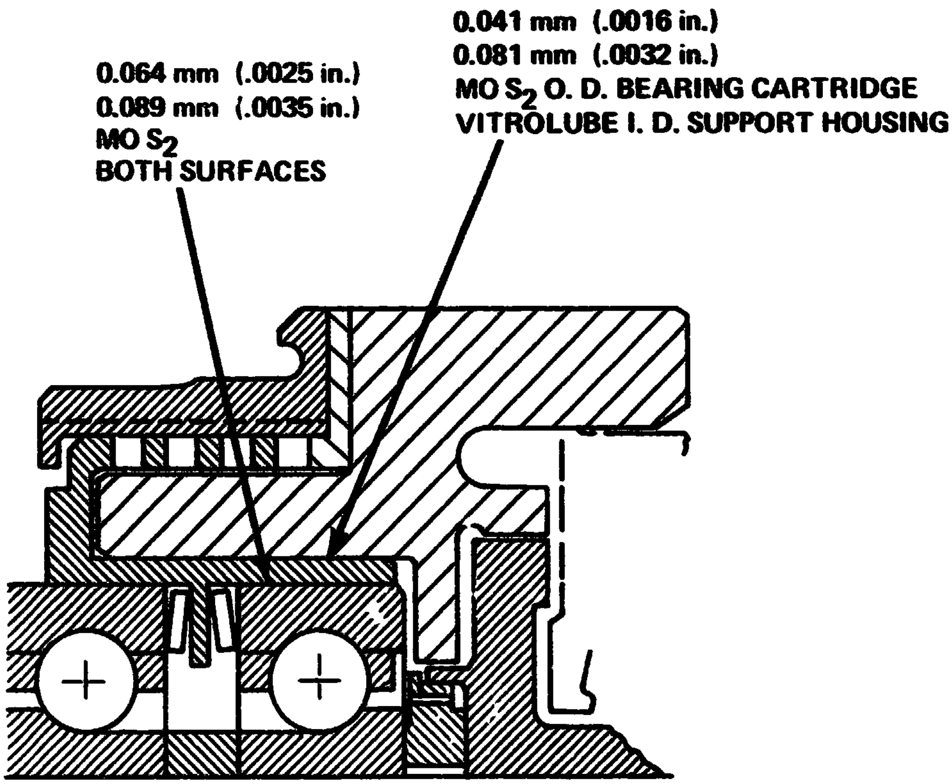


Figure 17. Bearing clearances in HPOTP bearings No. 3 and 4.

operating at about 500°C (932°F). These temperature differences cause thermal stresses which can result in distortion of the housing. The housing tends to become egg shaped and cartridge clearance tends to be reduced.

2) The bearing outer race outside diameter becomes enlarged as a result of hoop stresses resulting from bearing preload and centrifugal ball loads and as a result, outer race clearances are expected to be reduced.

3) Ball and race diameters could increase because of frictional heat generation. This heat generation, W_t (in Watts), is given by the following expression [9]:

$$W_t = 0.00514 \, \mu_f \, Fdn \quad .$$

where

μ_f - coefficient of friction

F = bearing load, kg

d = bearing bore, cm

n = bearing speed, rpm.

This heat is dissipated by cold liquid oxygen flowing through the bearing. The liquid oxygen enters through bearing No. 4 and exits through bearing No. 3 (Fig. 16). For a HPOTP bearing operating under a nominal 725 kg (1600 lb) total load and 28,000 rpm, approximately 60 kW (56.9 Btu/sec) heat is generated with a favorable coefficient of friction (μf) of 0.10. Under less favorable conditions of $\mu f = 0.22$, the heat generation is calculated to be 131 kW (124 Btu/sec), which is very high. Since heat generation is also a function of bearing load, abnormally high loads can be expected to generate even more heat and could cause serious overheating of the bearings. This will cause the bearings' inner races and balls to grow in size and reduce the bearings' internal clearances. This view is supported by a recent thermal study of HPOTP bearings by Battelle [10].

Bearing internal clearances and forces through the balls can also change with contact angle. Figure 18 is a plot of forces through the balls as a function of bearing contact angle. A loss of bearing internal clearance results in a decrease in contact angle and an increase in reactive force through the balls, which is equivalent to an increase in axial load on the bearing. This, in turn, will cause increased heat generation which serves to make the situation worse. The problem is more pronounced in a 15 degree angular contact bearing than a 25 degree angular contact bearing, as illustrated in Figure 18 where the 15 degree contact angle is closer to the "knee" of the reactive force curve than the 25 degree contact angle. It should be noted that the turbine end bearings (Nos. 3 and 4) have a 15 degree contact angle and all other HPOTP and even HPFTP bearings have a 25 degree contact angle. Therefore, it appears that Nos. 3 and 4 bearings may be more susceptible to high bearing reactive forces and associated thermal problems than the other SSME turbopump bearings.

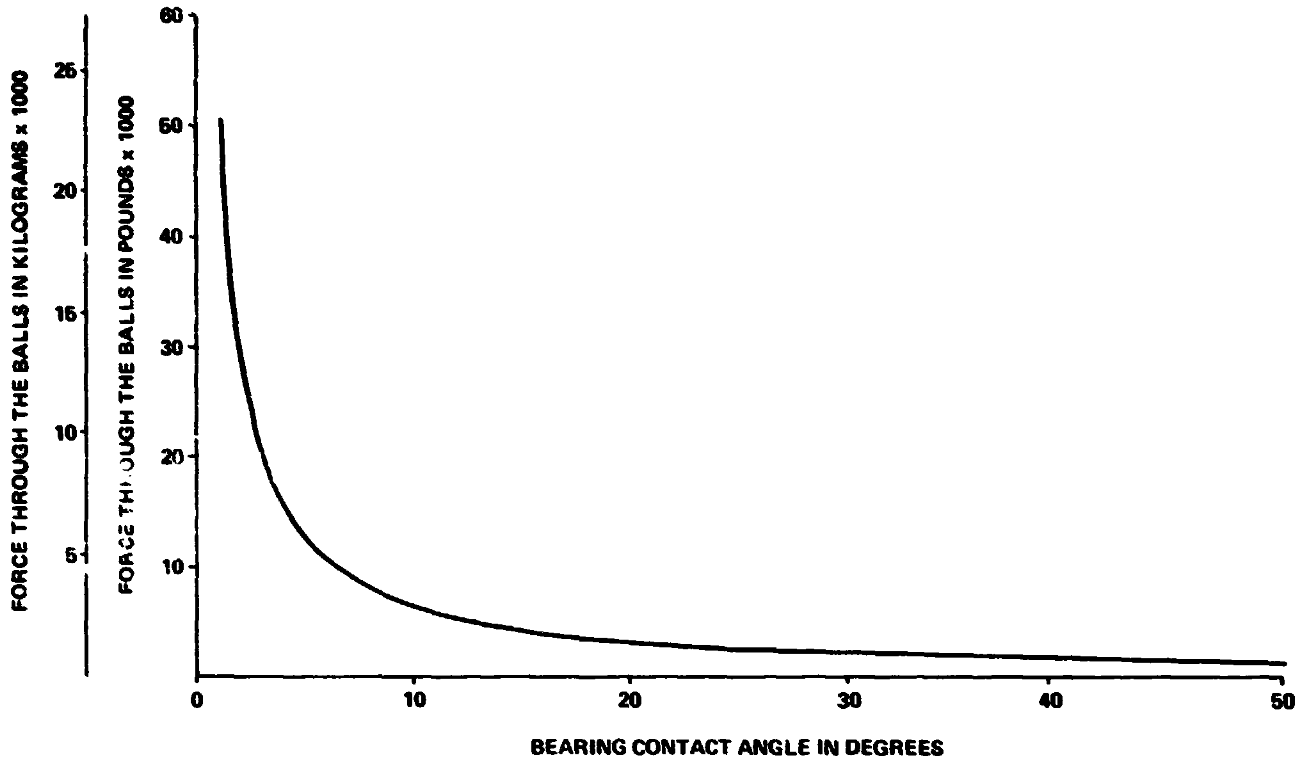


Figure 18. Bearing preload reactive force through the balls as a function of bearing contact angle for a constant 500 kg (1100 lb) preload.

In extreme cases bearing internal clearances may become zero or even negative. As this situation develops during HPOTP operation, the preload reactive force across the balls becomes very high and heat generation goes up causing bearing temperatures to rise. At some point, the temperature of the balls and inner race becomes sufficiently elevated that the lox cooling fluid at their surfaces exceeds its saturation temperature, and gas bubbles are formed at the hot surfaces causing the heat transfer film coefficient to decrease drastically. With a reduced heat transfer film coefficient the bearing temperature will rise further. Higher temperatures are expected to improve the rate of heat transfer until the heat dissipated is again equal to the heat generated; hence, a new stable but elevated temperature operating condition may be attained. Hence, it appears that the HPOTP Nos. 3 and 4 bearings may be capable of operating in a bistable thermal mode wherein under moderate heat generation conditions cooling is accomplished by a liquid (lox) and bearing operating temperatures are generally low, about -160°C (-256°F). Under severe operating conditions high bearing temperatures are believed to cause a transition to gas (gox) cooling. At this point bearing temperatures index upwards [to estimated +250°C (482°F) or more, based on ball surface discoloration] as a result of a drastically reduced heat transfer film coefficient. This high temperature operating mode appears to be at least partially the result of this bearing's 15 degree contact angle and lack of sufficient internal diametral clearance. It should be noted that while the bearing is operating at the high temperature operating mode the bearing and cartridge would be locked up or at least free motion would be severely restricted and any axial movement of the shaft would overload the turbine end bearings.

B. High Loads, Axial and Radial

HPOTP bearings appear to carry very high loads. High transient axial loads, 2273 to 3636 kg (5000 to 8000 lb) have been measured during shutdown of two instrumented HPOTPs. High transient axial loads are believed to be present at start-up as well. The position of balls on the inner race at lox temperatures as a function of axial and radial loads has been determined by Battelle [1]. It has been shown that under very high axial loads the balls could ride on the shoulder of the inner race. Evidence of shoulder damage on the inner race of HPOTP bearing S/N 8549531 (Fig. 8, A) suggests that the axial loads must have been high. In the laboratory test bearing S/N 8517929, applied axial load was 11,591 kg (25,500 lb) and this bearing sustained a higher shoulder damage (Fig. 8, B) than HPOTP bearing S/N 8549531 suggesting that the axial loads experienced by the latter were less than 11,591 kg (25,500 lb). The absence of shoulder damage in S/N 8517824 implies that axial loads in this bearing were not as high as the other two.

It is interesting to note that the wear track in bearing S/N 8517929, tested at high axial loads, is at the shoulder as expected. However, contrary to expectations, the wear track in bearing No. 8549531 is not at the shoulder but away from it. The wear track in bearing S/N 8517824 is somewhere in between. It appears that the wear tracks in these races and the spalls associated with them are not caused by high axial loads but are due to high radial loads. Battelle has also observed that the spalls in S/N 8549531 are consistent with a high radial load of 2000 lb and a relatively low axial load of 850 lb [1]. Apparently, this bearing has experienced both high axial and high radial loads. The major fatigue spalls, however, appear to be associated with high radial loads.

As described in the previous subsection A, high axial loads may be due to two causes (1) an unbalanced (axial load) condition present in the HPOTP at start-up and shutdown and (2) axial hangup of the bearings and/or cartridge in the housing caused by loss of clearances. Under normal running conditions, axial loads are believed to be minimal, approximately 500 kg (1100 lb) preload. Stationary radial loads are believed to be low because the ball tracks in the outer races of these bearings and others evaluated in recent history have not shown significant eccentricity. Rotating radial loads through rotor unbalance could generate high synchronous radial loads. However, in the bearings investigated in recent history (including those in this report) damage around the inner races has been fairly uniform, suggesting the presence of very low synchronous radial loads. Therefore, it appears that radial loads of thermal origin are the likely cause of damage in the HPOTP bearings. This point is further discussed under thermal factors.

C. Lubrication

The HPOTP bearings are lubricated by a Teflon film transferred from the armalon cage material. This method of lubrication is not believed to be very efficient in the SSME HPOTP situation. Hence, it may be assumed that lubrication in HPOTP bearings would be poor at best. This assumption is supported by the mode in which cracks initiate in the raceways. The cracks appear to be surface initiated which is a sign of poor lubrication. Under conditions of good lubrication the cracks which cause spalling initiate at subsurface locations and then propagate to the surface [3]. This does not appear to be the case here. The short service lives of these bearings suggest very high loads. The coefficient of friction between balls and raceways is estimated as 0.22 for a clean bearing and 0.08 for a bearing with a full PTFE film [10]. Heat generation is directly proportional to the coefficient of friction and under conditions of high coefficient of friction and high loads thermal growth can occur and bearing internal clearance may be lost. Further temperature increases could cause excessive thermally induced bearing stresses, increased heating, and rapid failure of the bearing. Therefore, any improvement in lubrication should help to decrease heat buildup and to increase the bearing service life.

D. Thermal Factors

It was shown earlier in subsection A that heat generation in HPOTP bearings caused by friction will be high if (1) the coefficient of friction (μ_f) is high, (2) bearing loads are high, and (3) bearing speeds are high. In HPOTP bearings the coefficient of friction is believed to be moderate in the beginning, but is expected to increase with time as bearing surface finishes begin to deteriorate. The exact relationship between μ_f and time is not known, however. In general, a bearing may be expected to run hot if the coefficient of friction is high. This condition in combination with high bearing speeds, high bearing loads, and lack of adequate cooling (by gox instead of lox) can lead to high operating temperatures for the HPOTP bearing.

High operating temperatures tend to reduce the internal clearance in a bearing through localized thermal expansion. If the temperatures are sufficiently high the clearance could be reduced to zero or even negative values. In the latter case, races and balls will experience internal radial loads. The magnitude of these internal radial loads will depend on temperature, but they could become very high. Under these conditions the bearing could attain a thermal runaway condition because high temperatures lead to higher thermally induced stresses, i.e., higher loads, leading to more heat generation and even higher temperatures.

Heating
↓
Loss of clearance
↓
Thermal stresses
↓
More heating

Loss of bearing internal clearance will tend to move the ball contact path away from the shoulder and towards the lowest point in the inner race (corresponds to largest clearance) in spite of the axial loads that may be present. Actual position of the ball contact path will depend upon the actual operating bearing internal clearance and the relative magnitudes of radial and axial loads.

In the above discussion, heat is generated at the ball-raceway contact and is conducted into the race from the surface. But, most of the heat generated at the surface will be dissipated into the liquid oxygen stream. If loads are extremely high, however, the point of maximum shear stresses will move to locations significantly below the surface. For example, for 1364 kg (3000 lb) radial load, the point of maximum Hertzian stress is 0.15 mm (0.006 in.) below the surface [1]. If the material deforms under stress at these subsurface locations, heat will also be generated at these locations which will not be readily dissipated by conduction to the surface, and hence, hot spots will develop at subsurface locations.

Based on the above arguments the following model is developed to explain cracking of the HPOTP races. The races become warm because of a high coefficient of friction and high radial or axial loads. Bearing clearances begin to decrease and internal thermally induced radial loads start to increase and the bearing approaches a thermal runaway condition. Cooling mode changes from lox to gox. Surface cracks are nucleated at this time as a result of high loads and a high coefficient of friction, and these cracks propagate deeper as the thermally induced radial loads increase. Regions of maximum Hertzian stresses and the plastic deformation associated with them move below the surface causing localized overheating. It is also possible that cracked material rubs against the material below it to produce heat locally. The presence of an oxide layer in the cracks, as has been found in bearings S/N 8549531 and 8517824, strongly suggests that temperatures on the order of 1371°C (2500°F) have been attained locally. It is also possible that molten oxide acts as a lubricant which minimizes further temperature rise.

It should be noted that both axial and radial loads can initiate a thermal runaway condition which subsequently can lead to extremely high radial loads. In the authors' opinion, the transient axial loads being experienced at start-up are not likely to cause thermal runaway conditions in the HPOTP because of their transient nature. It takes time to build up temperature in a bearing when starting from cold conditions and transient axial loads (3 to 4 sec duration or less) are not likely to cause a significant rise in temperature. It may, however, hasten the process of heating up by other loads under normal operating conditions and the axial and radial loads together may produce a synergistic action. Lack of adequate cooling could also hasten the process of thermal runaway. In this regard, HPOTP bearings in Position No. 3 are not likely to be cooled as well as the bearing in Position No. 4 because cooling oxygen flows from Position No. 4 to Position No. 3. Therefore, it is probable that No. 3 bearings run hotter than No. 4 bearings and hence are more likely to reach the thermal runaway condition and fail. Apparently, this has been the case with HPOTP bearings tested so far [11].

E. Correlation Between Bearing Performance and HPOTP Operating Conditions

The test conditions of HPOTP assembly Nos. 9008, 2007, and 0103 are summarized in Tables 2, 3, and 4, respectively. Durations of each test at different power levels are listed. It should be noted that the bearing S/N 8517929 in HPOTP 0103 was in good condition after the test. On comparison with operating conditions there is a good correlation between spalled races and power levels (Table 2) and duration of tests (Table 3). HPOTP 0103 (Table 4) was not tested for long durations at high power levels and the S/N 8517929 bearing from this pump was in good condition. These results are consistent with the thermal runaway model described in the previous sections. Higher power levels generally mean higher pump speeds and higher radial loads and these factors contribute to overheating of bearings. Long running times also contribute to temperature buildup by (1) inadequate transfer film lubrication and accompanying rise in coefficient of friction and (2) subsurface heat buildup with time. Therefore, it appears that spalling of races might be reduced or even prevented by (1) reducing high axial and radial loads and (2) improving the lubrication, or (3) reducing the duration of each run. The last recommendation is further supported by the fact that spalls have not been observed in HPOTP tests run for very short durations (less than 50 sec) [11].

F. Failure Analysis Summary

Evidence and arguments supporting the bistable thermal operation and thermal runaway models of HPOTP bearing (Nos. 3 and 4) failure can be summarized as follows:

1) Inner race damage high on the shoulder – evidence of high axial loads (believed to be transient in nature)

2) Spalls located low or near the bottom of the inner races – evidence of high radial loads

3) No evidence of high external rotating (synchronous) radial loads – no eccentric wear tracks on the inner races

4) No evidence of heavy stationary radial loads – no eccentric ball wear track on the outer races

5) Wear tracks correspond to thermally induced radial loads – heavy, uniform ball wear track low in the inner and outer races, i.e., loads produced internally within bearing when it approaches thermal runaway

6) Presence of oxide in cracks and ball discoloration – evidence of overheating caused by thermally induced radial loads and lack of adequate cooling caused by gox formation

7) Actual bearing damage correlates with HPOTP power level and running time consistent with thermal runaway model.

The following are likely drivers which tend to cause the HPOTP bearings to approach thermal runaway:

1) Poor lubrication – increases heat generation rate

2) Inadequate cooling – increases bearing surface temperature

3) Small bearing contact angle (15 degrees) – too close to the "knee" of the ball force curve. A small loss of clearance tends to produce a big increase in bearing internal load.

4) High bearing loads – generate heat.

No significant external radial loads are expected. Once the bearings lock up or excessive drag is induced in the pump housing axially, the bearings are subject to balance piston axial loads. This situation further aggravates the thermal runaway condition.

HPOTP BEARING LIFE EXTENSION

The failure analysis of HPOTP bearings presented in this report suggests methods of preventing such failures and extending the useful life of these bearings. These methods are reducing bearing loads and improving lubrication, design, and cooling. Use of improved bearing materials is also a possibility. Each method is discussed in the following subsections.

A. Reduction of Axial Loads

The approach of reducing bearing loads has merit since bearing life varies inversely as $(\text{load})^3$ [9]. A reduction in load by a factor of two increases the bearing life by a factor of eight. If the high transient axial loads were the major cause of failure it would appear that decreasing these loads from measured 3636 kg (8000 lb) to 909 kg (2000 lb) should increase the bearing life by a factor of 64, which is probably sufficient to meet the design life of 7½ hr for the SSME. At present, it is not clear whether transient axial loads of less than 909 kg (2000 lb) can be achieved consistently. Furthermore, as discussed in the previous sections, the premise that high axial overloading is the root cause of the problem may be in error. It is highly probable that spalling of the balls and races is not associated with high axial overloads because spalls have been observed in races that do not show evidence of shoulder damage caused by high axial loads (e.g., S/N 8517824). The failure seems to be caused by high internal radial loads of thermal origin. The high axial loads could, however, trigger the process of thermal runaway and be an indirect cause of bearing failure. Therefore, bearing axial loads should be reduced.

B. Bearing Design Change

Several bearing design changes are being considered based on the premise that high axial loads are the major cause of HPOTP bearing failure. For instance, the inner race shoulder can be raised to better accommodate the high axial loads and minimize shoulder damage. This design is being tested by Rocketdyne [5]. One possible disadvantage of this design is the trade off in cooling characteristics because of reduced cross sectional area now available for the coolant flow. Again, if axial loads are not a major factor in the bearing failure this technique may not help to extend the bearing life significantly. Use of a bearing with balls bigger than those used in the present 57 mm bearing might help to reduce the stresses in the bearing and hence extend the bearing life. However, this approach would involve major modifications in the turbopump. Change in the contact angle from 15 to, for instance, 25 degrees would reduce the tendency for thermal runaway. But, this approach may result in some loss of radial stiffness in the bearing and possibly affect pump rotor stability, and should be examined further.

C. Lubrication

Evidence of high temperature deep in the inner race (presence of oxide film) strongly implies that poor lubrication and an associated tendency toward a thermal runaway condition may be the real life limiting factors. Good lubrication alone is expected to increase the life of these bearings significantly; but actual improvement attainable is not known. When the thermal effects associated with poor lubrication are considered, a dramatic increase in bearing life may be realized through improved lubrication. High loads coupled with poor lubrication have a synergistic effect in reducing the bearing life through (1) surface crack nucleation and (2) thermal growth and thermal stresses of potentially very high magnitude. Therefore, improving lubrication appears to be an important approach to bearing life extension. Improved lubrication techniques such as sputter coating of MoS_2 on balls are being tested by Rocketdyne at the present time. It appears that most dry film lubricants will be lost after a relatively short time of turbopump operation. However, advanced techniques such as ion implantation might be used to retain the lubricants for longer times. Work is being done in this area [12].

D. Cooling

Improved cooling is expected to help lower the bearing operating temperature. For best results, cooling by liquid oxygen (lox) should be maintained at all power levels. Occasionally discoloration of balls has been observed in the HPOTP bearings suggesting that bearing temperatures were high, greater than 250°C (482°F), and cooling was not adequate. Internal heating and oxidation of inner races [observed at depths of 0.08 mm (0.003 in.) or more] could be alleviated by increasing the thermal conductivity of the bearing material. However, this is a material limitation and not a cooling problem. It is best to avoid heating at subsurface locations by avoiding the high axial or radial loads. For best results, the maximum Hertzian stress should be limited to the first 0.08 mm (0.003 in.). This depth corresponds to radial or axial loads of less than 455 kg (1000 lb) [1].

E. Improved Bearing Materials

In principle it is possible to extend the life of HPOTP bearings by using materials that have improved load bearing capacity. The choice of 440C steel for the turbopump bearings is largely because of its superior corrosion resistance compared with other premium bearing materials such as 52100 or M-50 which have higher hardness, load bearing capacity, and fatigue life. BG-42 is a relatively new corrosion resistant bearing material that is harder than 440C (hardness Rc58-63 for 440C versus Rc61-64 for BG-42) and is being tested by Rocketdyne. Both 440C and BG-42 contain relatively large carbides that tend to initiate fatigue cracks and limit bearing life. To correct this deficiency, it has been proposed to develop new corrosion resistant bearing steels containing a high percentage of small carbides using powder metallurgy techniques [13]. This development will take some time, however.

SUMMARY AND CONCLUSIONS

1) Subsurface cracks in HPOTP bearing raceways are much more extensive than is apparent on the surface.

2) Cracks formed in the inner race under high axial loads are located at the shoulder of the race.

3) The location of major cracks in HPOTP bearing races corresponds to high radial loads rather than high axial loads.

4) There is evidence to suggest that the inner races were heated to elevated temperatures. The temperatures could have been high enough to form molten iron-chromium oxide in the cracks at subsurface locations.

5) The inner race failures observed on these and other recent bearing failures are consistent with a thermal runaway model. According to this model the HPOTP bearings are heated by a combination of high loads and high coefficient of friction (poor lubrication). Very high internal radial loads can be generated by loss of bearing internal clearance resulting from heating. These radial loads are apparently responsible for the bearing failures.

6) Based on the above mechanism of failure, it appears that the service life of HPOTP bearings can be extended by the following:

a) Reducing bearing loads, e.g., through improved design and start-up procedures

b) Improving lubrication, e.g., through ion implanted MoS_2

c) Increasing bearing clearances, e.g., by using a higher bearing contact angle

d) Increasing the coolant flow rate through the bearing to ensure lox cooling at all pcwer levels

e) Using improved bearing materials, e.g., materials having small carbides (to be developed).

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APPROVAL

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The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

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