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ROCKETDYNE LOX BEARING TESTER PROGRAM*

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ABSTRACT

Bearings in the low- and high-pressure liquid oxygen turbopumps for the Space Shuttle Main Engine (SSME) Phase II development program have occasionally worn, limiting the life of the bearings. The cause, or causes, for the ball wear were unknown, however, several mechanisms were suspected. Two testers were designed and built for operation in liquid oxygen to empirically gain insight into the problems and iterate solutions in a timely and cost efficient manner independent of engine testing. Schedules and test plans were developed that defined a test matrix consisting of parametric variations of loading, cooling or vapor margin, cage lubrication, material, and geometry studies. Initial test results have indicated that the low pressure pump thrust bearing surface distress is a function of high axial load. Initial high pressure turbopump bearing tests have produced the wear phenomenon observed in the turbopump and identified an inadequate vapor margin problem and a coolant flowrate sensitivity issue. These tests have provided calibration data of analytical model predictions to give high confidence in the positive impact of future turbopump design modification for flight. Various modifications will be evaluated in these testers, since similar turbopump conditions can be produced and the benefit of the modification will be quantified in measured wear life comparisons.

INTRODUCTION

The Space Shuttle Main Engine (SSME) turbomachinery has repeatedly demonstrated its reliability and high performance during launch. The Space Shuttle program emphasis continues to strive for increased design margin and uprated power to support larger payloads. The uprated turbomachinery for the SSME has been tested to extreme limits in demonstrating life margin in the Phase II development program, more so than the original flight configuration. Ball bearings in the high pressure oxidizer turbopump (HPOTP) and low pressure oxidizer turbopump (LPOTP) have not, in general, met program life requirements, leading to maintance overhaul of the turbopumps for bearing replacement. Rocketdyne is conducting a component test program to improve bearing life. This paper describes the test program and results from recent tests.

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The SSME HPOTP is shown in Fig. 1 and provides the high speed, power, and performance shown in Table 1. The HPOTP rotor is supported by two duplex pair of preloaded angular contact ball bearings. For nomenclature, the ball bearings are numbered one through four from the preburner impeller toward the turbine. The bearings are cooled by liquid oxygen (LOX), which is supplied from sources downstream of the main impeller. During mainstage operation, the bearings support radial loads while axial thrust is carried by the main impeller balance piston. During startup and shutdown, the turbine end bearings also experience transient axial loads when the bearing support cartridge bottoms on travel stops.

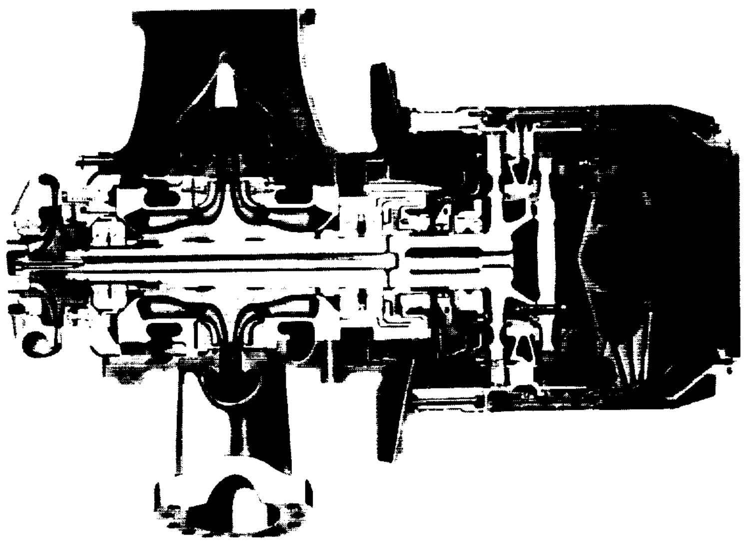

Fig. 1. SSME High-Pressure Oxygen Turbopump

Table 1. SSME HPOTP Performance Data

Parameters	Rated Power Level Main	Rated Power Level Boost	Full Power Level Main	Full Power Level Boost
Pump inlet flow rate (lb/sec)	1070.6	111.6	1157.8	129.4
Pump inlet pressure (psia)	379.3	3985.9	392.4	4403.6
Pump discharge pressure (psia)	4108.7	7106.7	4556.0	7861.2
Pump efficiency	0.684	0.809	0.650	0.800
Turbine flow rate (lb/sec)	61.8		69.0	
Turbine inlet pressure (psia)	5015.3		5660.8	
Turbine inlet temperature (R)	1407.2		1596.3	
Turbine pressure ratio	1.506		1.550	
Turbine efficiency	0.749		0.755	
Turbine speed (rpm)	27,102		29,675	
Turbine horsepower	22,902		29,174	

Wear on several No. 2 and No. 4 bearings has been experienced in Phase II HPOTP testing. The condition of bearings resulting from this testing is summarized in Fig. 2 and 3. Wear of the No. 1 and No. 3

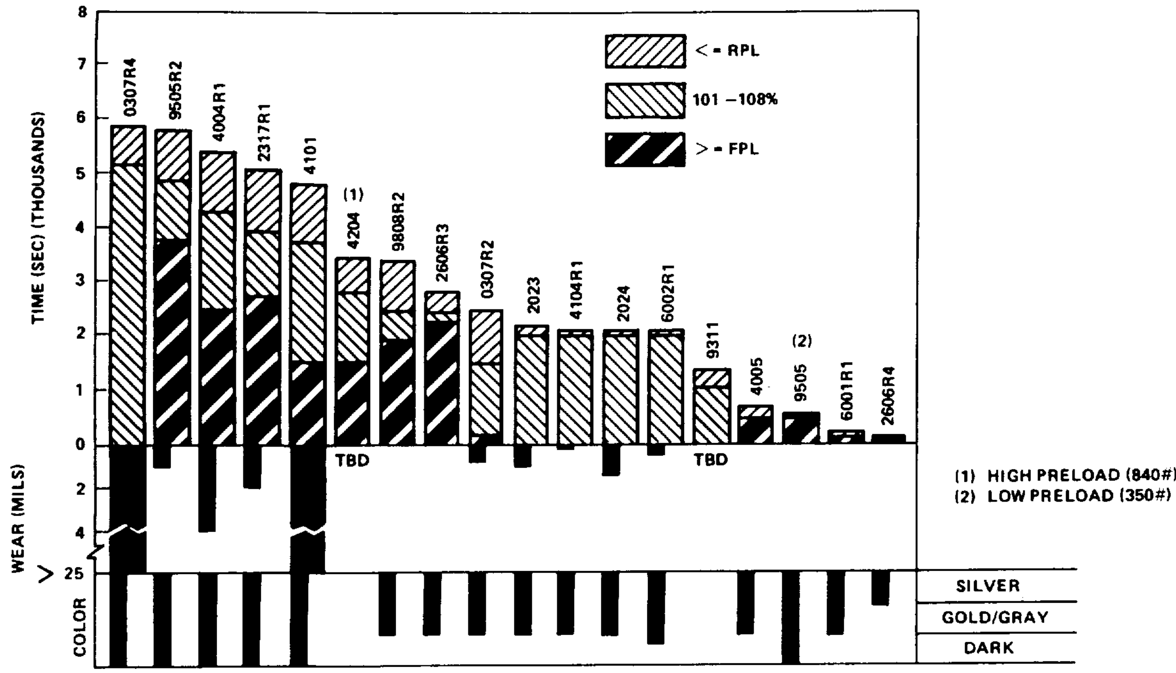

Fig. 2. HPOTP Phase II Pump End Bearing No. 2 Wear History

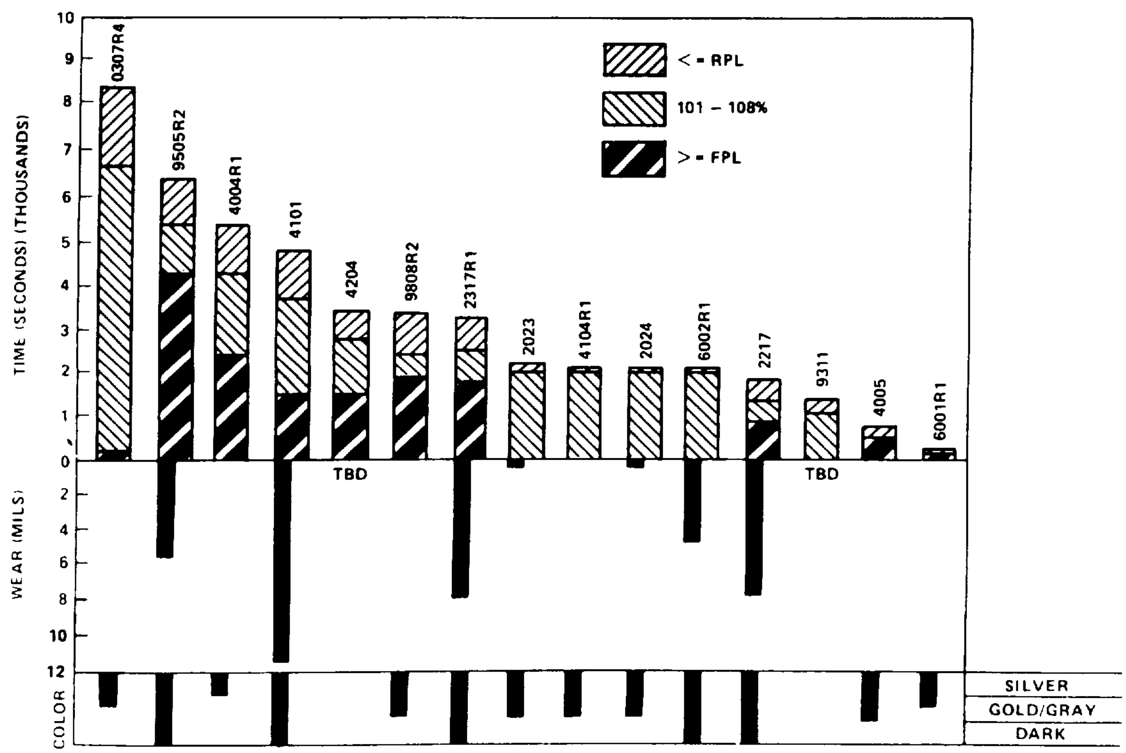

Fig. 3. HPOTP Phase II Turbine End Bearing No. 4 Wear History

bearings has only been observed in conjunction with severe wear of the No. 2 and No. 4 bearings.

The SSME LPOTP is shown in Fig. 4 and provides the performance data shown in Table 2. The LPOTP rotor is supported by an 85 mm thrust bearing at the inducer end, which carries the unidirectional axial load, and a preloaded 55 mm angular contact ball bearing on the turbine end. Radial load in the LPOTP is low, evidenced by axisymmetric raceway tracks on used bearings. Thrust bearing wear has been experienced in some LPOTPs, as summarized in Fig. 5. Typically, the balls show discoloration and patches of shallow surface distress.

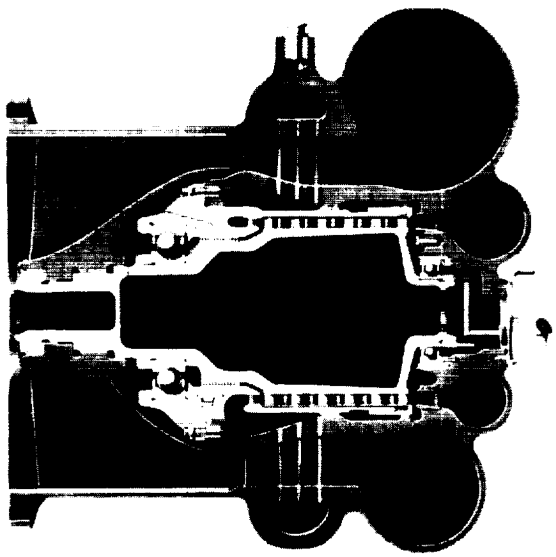

Fig. 4. SSME Low Pressure Oxygen Turbopump

Table 2. SSME LPOTP Performance Data

Key Performance Data at 109% Thrust (Full Power Level)	
Pump inlet flowrate (lb/sec)	972
Pump inlet pressure (psia)	100
Pump discharge pressure (psia)	432
Pump efficiency	0.67
Turbine flowrate (lb/sec)	188
Turbine inlet pressure (psia)	4369
Turbine inlet temperature (R)	195
Turbine pressure ratio	10.1
Turbine efficiency	0.65
Turbine speed (rpm)	5300
Turbine horsepower	1770

Suspected Causes

The potential for improvements in bearing performance has been demonstrated in component tests and by some turbopumps that have operated without bearing wear for periods considerably longer than average. Cooling and loading are the primary variables in the turbopumps, which can result in differences in bearing performance. Because of its low viscosity and poor lubricant properties, only cooling is provided by LOX. Lubrication is provided by a transfer film of teflon from the Armalon cage to the balls and by a moly-disulfide coating on the balls and raceways, hence is not believed responsible for variations in bearing performance.

The cause of short HPOTP bearing life is suspected to be either loading, lack of adequate cooling, or a combination of both. A loading problem could arise from excessive radial loads, or high or low axial

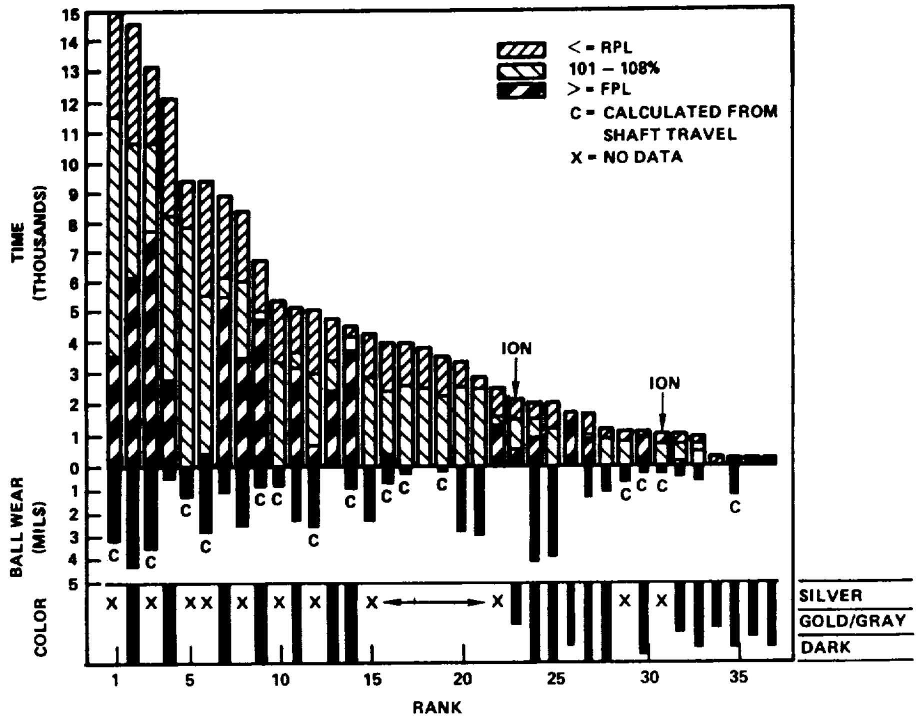

Fig. 5. LPOTP Thrust Bearing Wear History

loads caused by outer ring binding during axial shaft excursions. Inadequate cooling stems from vaporization due to insufficient vapor margin caused by a high coolant supply temperature, low pressure, or inadequate flow. Proposed bearing wear scenarios are not consistent with all evidence. For example, in the case of the cooling issue, the No. 4 bearing has better coolant vapor margin than the No. 3 bearing because it is upstream, but it experiences the wear.

Several causes of LPOTP thrust bearing wear have been postulated: (1) high axial loading, (2) unloading, (3) cage/retainer nut interference, and (4) support misalignment. Again, no single cause is supported by all evidence.

Test Program Plan

The overall objectives of the LOX bearing test program are to improve bearing life in the HPOTP and LPOTP by obtaining a better understanding of bearing behavior in a flood cooled LOX environment and to develop and evaluate improvements in operating conditions, bearing geometry, and materials through component testing. Two types of testing are being conducted: parametric tests, where fluid conditions and load are varied over a wide range; and comparison tests, where bearing life at constant operating conditions is determined for various modified bearings, and compared to that of the baseline bearing.

The program test schedule is presented in Fig. 6 and a test description summary in Tables 3 and 4. Separate testers are used for HPOTP turbine end bearing and low pressure (LP) thrust bearing tests. The

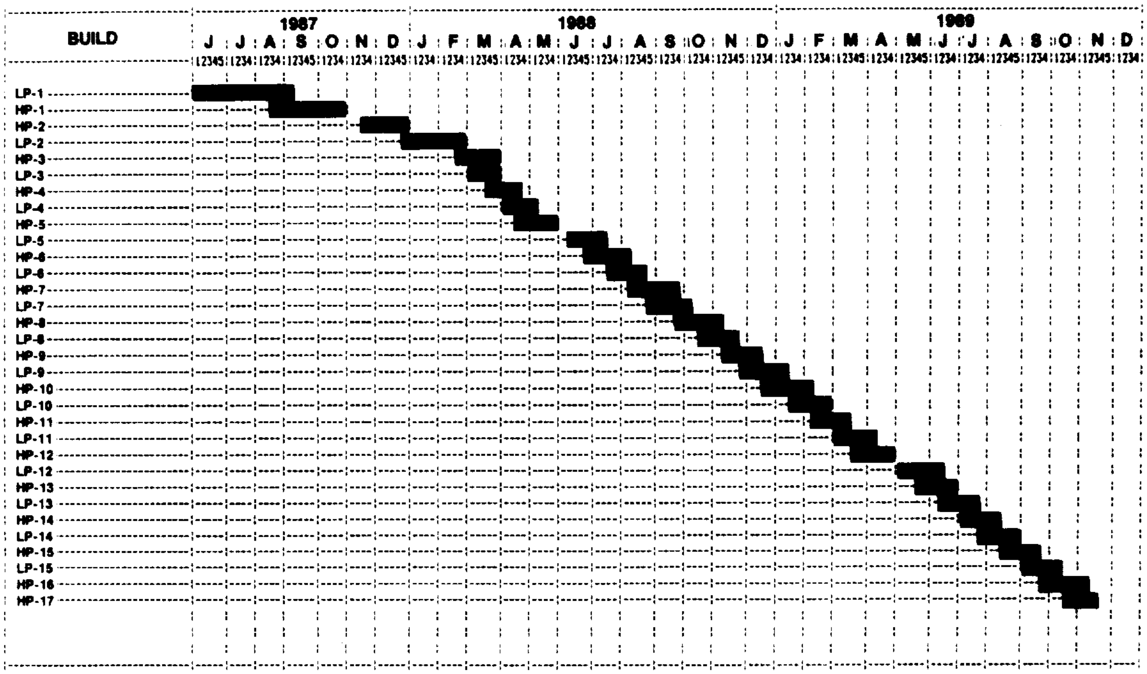

Fig. 6. High and Low Pressure Bearing Test Schedule

Table 3. High Pressure Bearing Test Plan

Build	Title	Description
HP-1	Phase I (para)	Parametric characterization of operating condition effects
HP-2	Phase II (para)	Parametric characterization of operating condition effects
HP-3	Low vapor margin (baseline)	Baseline duration test of Phase II bearing to define wear rates
HP-4	Phase I	Baseline duration test; wear rate
HP-	Low shoulder	Cooling performance of low shoulder outer ring
HP-	Ion implant	TiC balls/CrN raceway; wear rate performance
HP-	SiN balls/ion raceways	Wear rate performance
HP-	TFE coated cage	Wear rate performance; teflon transferability
HP-	Salox cage	Wear rate performance; teflon transferability; ball scuffing
HP-	BFT coated cage	Wear rate performance; teflon transferability
HP-	TDC	Wear performance of thin dense chrome at high speed in LOX environment
HP-5	Gold plate	Wear rate performance
HP-	Radial load	Cooling performance and pressure differential
HP-	Crowned outer race	Radial load capability; baseline testing
HP-	Duplex and roller bearing	Cooling performance and pressure differential
HP-	Large IRC bearing	Phase I bearing with increased clearance
HP-	CRB-7	Material evaluation

Table 4. Low Pressure Bearing Test

Build	Title	Description
LP-1	LN_2 checkout	Tester check with LN_2
LP-2	Parametric (baseline)	Parametric characterization of operating condition effects
LP-3	Life at 3000 lb axial load	Load effects on wear performance
LP-6	Misalignment	Radial load simulation vis bearing misalignment
LP-4	Life at 4500 lb axial load	Effects of loading on wear rate and bearing life
LP-	Flow effects	Effects of coolant flow rate on cooling performance
LP-	Ion implanted	TiC balls; CrN raceways; wear performance
LP-	SiN balls/ion race	Wear rate performance
LP-	Geometry (curvature)	Effects of raceway curvature on heat generation and wear rate
LP-	Light series	Effects of reduced bearing cross section on radial load - wear rate
LP-	Salox cage	Wear performance; teflon transferability
LP-	CRB-7	Material evaluation
LP-5	Increased contact angle	Standard bearing with ball diameter reduced 0.001 in.

interchangably into the Facility Test Cell and are tested alternately, providing efficient usage of test and assembly resources.

This paper presents results from the first two high pressure (HP) tester builds, which evaluated the influence of bearing coolant conditions on bearing performance, and of Builds 2 and 3 of the LP thrust bearing tester that addressed axial load and cooling issues.

TEST FACILITY

Both high- and low-pressure bearing testers use the same facility, with minor modifications to accommodate different flow and instrumentation requirements. Facility capabilities are summarized in Table 5. In the design of the facility, a major effort was spent developing a system that could operate the testers at known and repeatable conditions. to ensure consistency, feedback controlled servovalves were used to maintain constant coolant flowrate, coolant pressure, and bearing loads. Data on tester control parameters are reduced in real time to permit adjustment of servo-setpoints where test parameters cannot be directly controlled. For example, bearing loads, which in the LP tester are controlled by the piston pressure differential, are measured using strain gaged supports. The strains are converted to actual load using a scaling and averaging algorithm operating in real time on an IBM-PC. The piston pressure differential setpoint is manually adjusted as required to obtain the desired load.

Table 5. Test Facility Capability Using Liquid Oxygen

Flowrate (lbm/sec)	3 to 18
Supply pressure (psig)	100 to 800
Supply temperature (F)	-260 to -285
Drive speed (rpm)	3,000 to 32,000
Test duration (sec)	>6000
LOX storage tank (gal)	15,000

The facility is capable of continuous testing for durations in excess of 6000 sec. Continuous high pressure LOX is supplied by a pair of tanks that are alternately pressurized and refilled under computer control. A recent modification to this system, which controls the minimum pressure of LOX during the refill portion of the cycle, permits control of LOX temperature into the tester within $\pm 2^{o}F$ over the range of inlet temperature from -260 to -285^{o}F.

Additional capabilities of the test facility include:

- An evaporator system for measuring two-phase drain flow
- A LOX recycle system for returning flow to the supply system
- Real-time data reduction
- An integrated computer facility control, redline and data acquisition system.

LOW PRESSURE BEARING TESTER DESIGN

The LP thrust bearing tester is shown in Fig. 7. The shaft is supported by two 85 mm bearings with the outer races fitted to the tester housing bore with a 7.5 in. span. The shaft is electric motor driven through a quill shaft splined to the inlet end of the tester shaft. A gaseous nitrogen (GN_2) purged buffer seal prevents gaseous oxygen (GOX) leakage to atmosphere. The drive end (upstream) bearing is loaded axially by the shaft against a strain gaged ring, used to measure axial load. This ring seats against a shim that may be tapered to permit testing at controlled amounts of outer race misalignment.

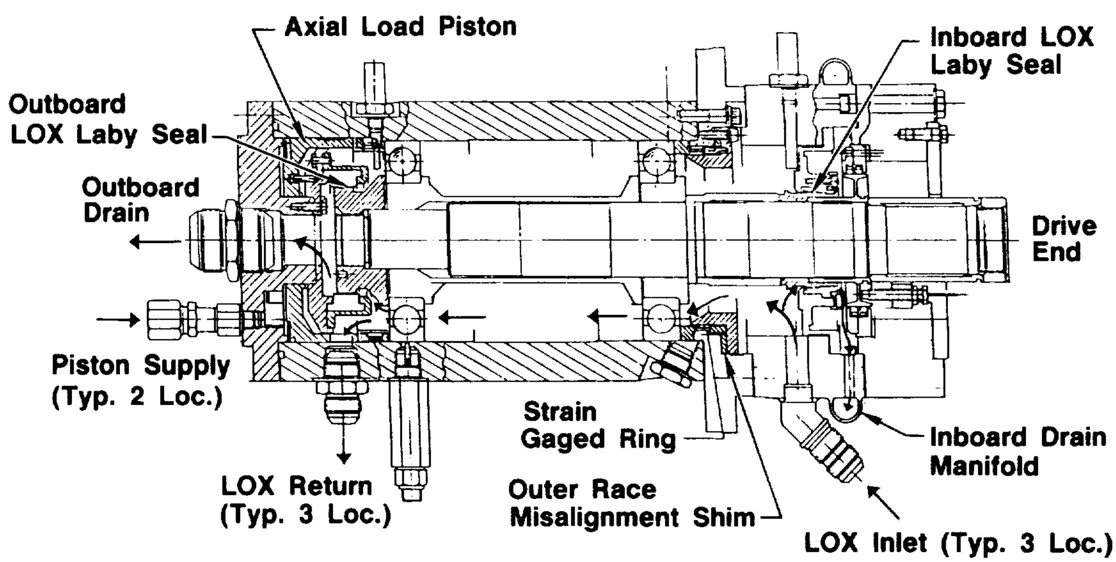

Fig. 7. SSME Low Pressure LOX Thrust Bearing Tester

Liquid oxygen enters the tester radially at three locations in the the mid-housing, flows through both bearings in series, and exits through three radial holes near the outboard end of the main housing. A modified HPOTP primary LOX labyrinth seal restricts LOX leakage along the shaft at the drive end. The leakage combines with GN_2 leakage from the buffer seal in the annular cavity between the two seals and is carried away through the inboard drain line. It passes through an evaporator and is then measured. The measured inlet flow is corrected for this seal leakage to obtain the bearing coolant flowrate.

At the outboard end of the shaft, a second labyrinth seal restricts LOX leakage out of the tester end. Leakage is carried away by the outboard drain line. The outboard drain cavity pressure is controlled to balance the axial pressure loading on the shaft, which develops from the differential pressure on the shaft ends and from pressure drop through the bearings.

An axial load is applied to the outer race of the outboard bearing by a piston contained within the outboard end of the main housing. The piston is pressurized by LOX fed through two ports in the end cover plate. Leakage past the piston is minimized by small radial clearances between the piston and housing. Axial load is controlled by the pressure difference between the piston inlet and the bearing coolant pressure at the exit of the downstream bearing. A Belleville spring preloads the piston to maintain a minimum load of 500 lb. The piston loads the downstream bearing through a strain gaged ring similar to that at the upstream bearing. The piston load is carried through the downstream bearing to the shaft, which transmits the load to the upstream bearing where it is reacted through the inboard strain gaged ring to the housing. Axial loads from 500 to 10,000 lb can be applied to the bearings. There is no provision for applying a radial load to the bearings, as experience with the LPOTP indicates radial loading is small.

HIGH PRESSURE BEARING TESTER DESIGN

The HP bearing tester is shown in Fig. 8. The tester is very similar to the LP tester except in the manner in which the bearings, 57 mm HPOTP turbine end bearings, are mounted (7 in. apart) and loaded. The upstream bearing has a sliding fit in its support housing and is loaded axially against a strain gaged wave spring that is used to measure axial load. The outer ring of the downstream bearing fits snuggly into an antirotated cylindrical cartridge that can slide axially. Both bearings are loaded through the cartridge by a pair of Belleville springs that maintain a constant axial load of 700 to 800 lb over a 0.070-in. operating range.

Load on the inboard bearing can be adjusted from 300 to 2000 lb by varying the outboard drain cavity pressure, which acts on the end of the shaft. Load on the outboard bearing does not change due to pressure loading and is always equal to the Belleville spring load. A piston is not used because of the lower axial loads required for preload simulation of the HP bearings.

Because of the higher operating speed of the HP tester, heat addition to the bearing coolant from fluid churning is significant. By

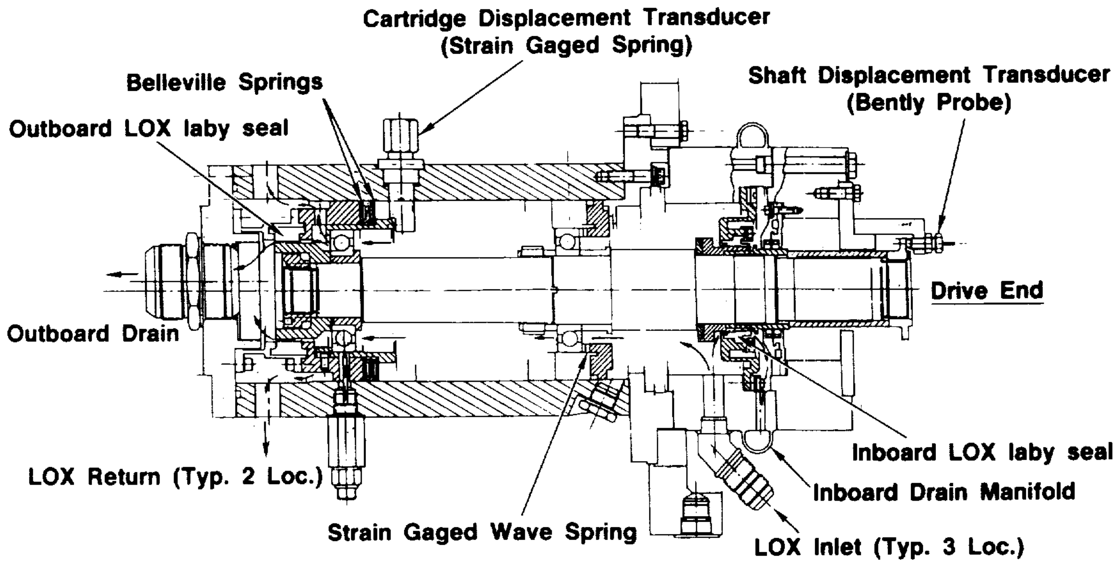

Fig. 8. SSME High Pressure LOX Bearing Tester

installing a slinger at the inboard labyrinth seal, coolant temperature into the upstream bearing can be increased by 20°F, for tests requiring warmer coolant temperature.

HIGH PRESSURE TESTING RESULTS

The initial parametric testing to define tester behavior was conducted with Build 1 and produced sensitivity data that established key relationships. Nominal operating test conditions were established to match high power level turbopump operation. Nominal conditions were: 30,000 rpm shaft speed, 325 psig cavity pressure, 4.75 lbm/sec coolant flowrate, and 1100 lb. preload on the inboard bearing. During initial testing on HP tester Build 1, a negative pressure differential across the inboard bearing was observed that increased as shaft speed increased; these data are shown in Fig. 9. Most of this measured negative pressure differential is the result of the downstream pressure tap being at a larger radius than the upstream tap and measuring a difference in vortex strength at the two locations. A small portion of the negative pressure differential was calculated to be the result of bearing pumping. As the bearing cavity pressure was reduced below 325 psig, the inboard bearing pressure differential would increase (become less negative) while fluid temperature was constant. This response was the result of a LOX phase change in the bearing, which reduced the effective fluid density downstream of the bearing. At nominal conditions and a 225 psig cavity pressure, vaporization in the inboard occurred above 25,000 rpm as shown in Fig. 10. As the cavity pressure was further reduced, the amount of fluid vaporization increased as shown in Fig. 11. This effect of cavity pressure data shown in Fig. 11 indicates formation of significant vapor below 325 psig for the nominal speed, flow, and load condition.

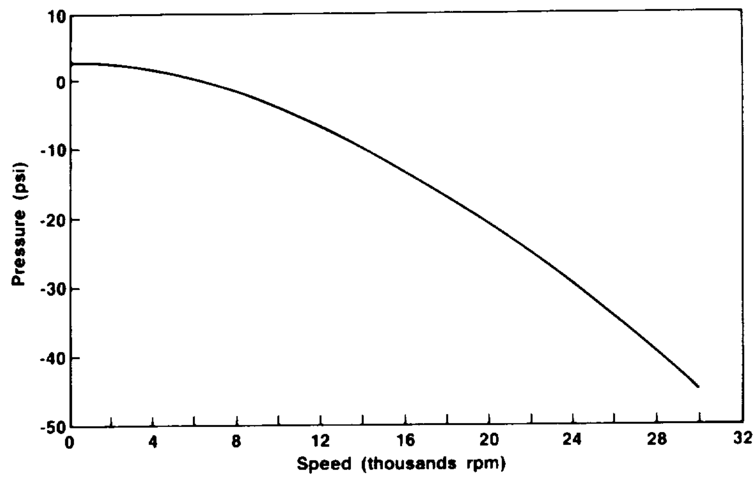

Fig. 9. Effect of Shaft Speed on Inboard Bearing ΔP (At Nominal Flow and Pressure)

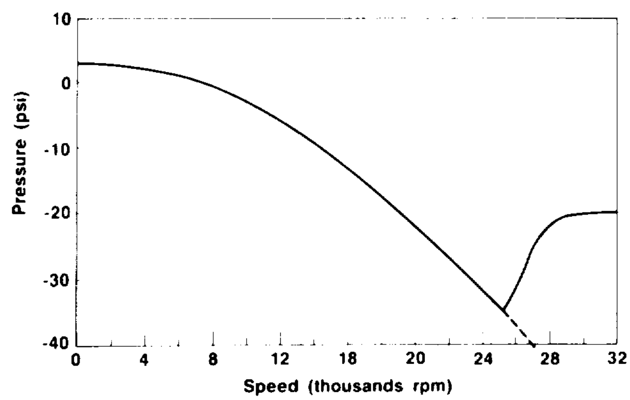

Fig. 10. Effect of Coolant Vaporization on Inboard Bearing ΔP

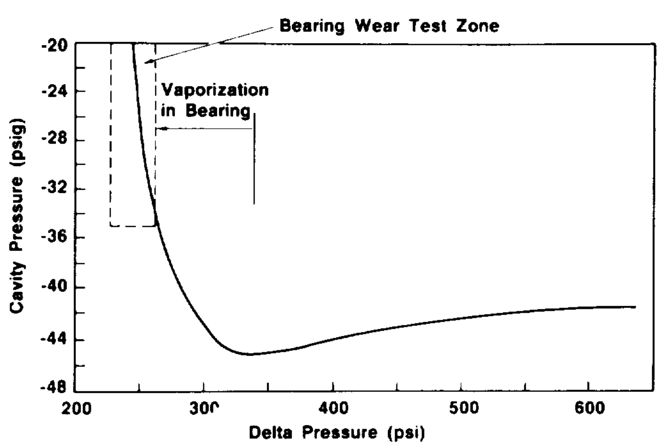

Fig. 11. Effect of Pressure on Inboard Bearing ΔP (At Nominal Speed, Flow, and Load)

A significant coolant flowrate influence on the inboard bearing pressure differential was also measured. As the coolant mass flowrate was reduced, the amount of fluid vaporization in the bearing increased at nominal speed, pressure, and load. The measured relationship is shown in Fig. 12 where a significant change occurs below about 5 lbm/sec. The transition point is a function of cavity pressure and shifts higher in flowrate at higher cavity pressures. The effect is due to rising fluid temperature caused by fluid churning losses in the bearing area, which do not decrease in proportion to flowrate. Axial load variations had no significant effect on coolant temperature rise or bearing pressure differential.

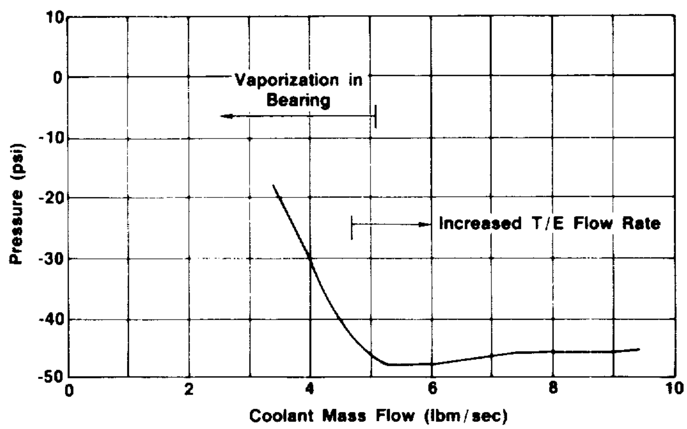

Fig. 12. Effect of Coolant Flow Rate on Inboard Bearing ΔP
(At Nominal Speed, Pressure, and Load)

Build 1 testing showed that the presence of vapor alone does not result in immediate rapid bearing wear. Both upstream and downstream bearings remained in good condition following 2800 sec of operation at 30,000 rpm, including 800 sec with vaporization at the upstream bearing. Rapid wear of the downstream bearing was initiated when coolant flow and pressure were simultaneously reduced, resulting in substantially more vaporization as indicated by the dashed box portion of Fig. 11.

The downstream bearing was found to be worn similar to a pump end 45 mm No. 2 bearing with darkened balls, average ball wear of 0.0046 in diametral, a concentric and wide inner race track with a high inner race contact angle, and a wide outer race track with a negative contact angle. A typical ball from the bearing is shown in Fig. 13. The upstream bearing remained in good condition, inspite of operating at a higher load. The bearing spring strain gage measurement provided cage speed frequency versus time data shown in Fig. 14, which aided in determining wear initiation time.

Testing of Build 2 of the HP bearing tester was conducted with lower coolant temperature to investigate bearing performance with good coolant conditions. Coolant pressure and flowrate were reduced to the levels of Build 1 tests without producing evidence of vaporization in

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Fig. 13. High Pressure Build 1 Downstream Bearing Ball (Posttest)

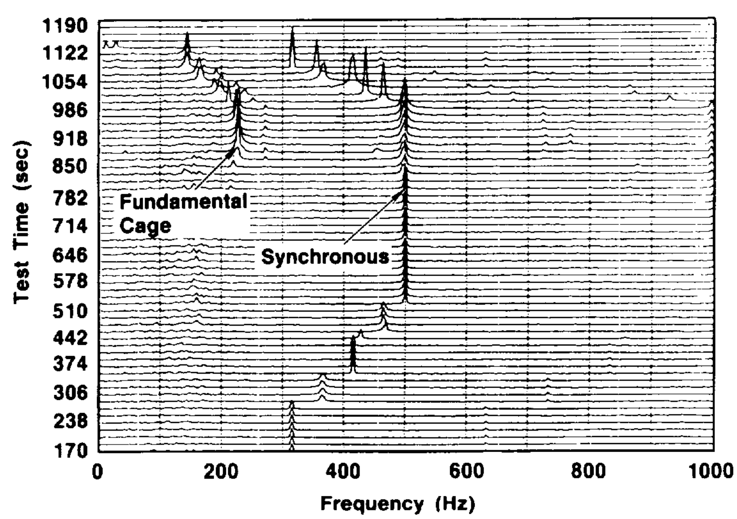

Fig. 14. Bearing Cage Frequencies Generated with Wear and Erosion

the bearings. Following 6900 sec at 30,000 rpm, the downstream bearing balls remained shiny with no wear, thus demonstrating a significant improvement in bearing life as a result of increased vapor margin.

During the Build 2 tests, the upstream bearing was subjected to over 1000 sec of high speed, low axial load operation (30,000 rpm, 300 lbf), and 1000 sec at 12,000 rpm with an axial load less than 100 lbf. This was done to determine the effect of low bearing preload and unloading of the bearing during the shutdown transients. Although the upstream bearing wore slightly (average ball wear was 0.00036 in. diameter), it performed much better than predicted. It is concluded from this test that, with good cooling, the brief unloading experienced by the No. 4 bearing during the shutdown transient should not affect bearing performance.

Application to HPOTP Modifications

Build 1 and 2 tests focused attention on the cooling adequacy of the HPOTP pump and turbine end bearings. Bearing coolant capacity relative to the bearing heat load is presented in Fig. 15. The coolant upstream enthalpy margin is the quantity of heat the coolant can absorb before vaporization would begin to occur at the bearing exit, that is, the difference in coolant upstream enthalpy and liquid saturation enthalpy at the bearing exit pressure. Vaporization will occur when the enthalpy margin times the flowrate is less than the bearing heat load. Vaporization limits for bearing heat rates of 20, 30, and 40 Btu/sec are shown in the Fig. 15; tester data are shown by circles. The closed (dark) circles are test points where fluid vaporization was experienced, evidenced by upstream bearing pressure differential measurements, hence, a maximum bearing heat generation rate of 40 Btu/sec is indicated in the tester upstream bearing. Results of Build 2 tests, without vaporization, yield a heat generation rate of 30 Btu/sec based on the coolant temperature rise across the upstream bearing.

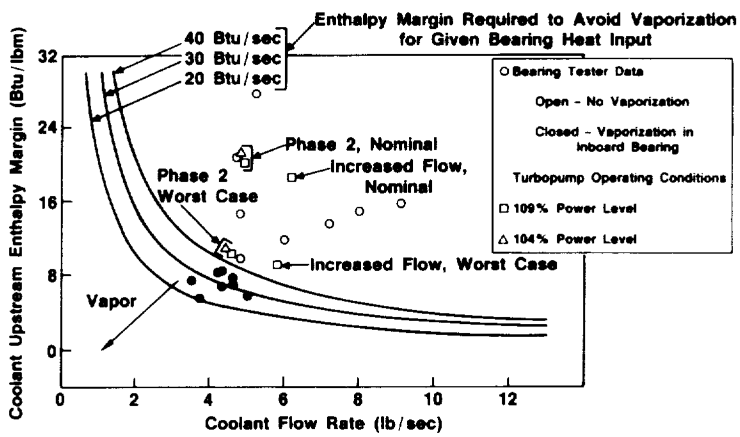

Fig. 15. HPOTP Turbine End Bearing Comparison of Tester Data With Operating Coolant Conditions

Also shown in Fig. 15 are predicted coolant conditions for the HPOTP turbine end bearings under turbopump operating conditions. In the worst-case condition, during engine operation with the LOX tank vented to minimum pressure, the bearings are expected to experience some coolant vaporization because duplex bearings have a higher heat generation than the single tester bearing. A HPOTP design modification to increase coolant flow shows that the vapor margin will be increased slightly.

In Figure 16, the tester data are presented along with the pump end bearing set operating points. The pump end bearing, in the worst case, operates with no vapor margin. A HPOTP design modification incorporates a back pressure seal downstream of the bearings to raise coolant pressure. This will significantly improve bearing coolant conditions at the pump end for all conditions as shown in Fig. 16.

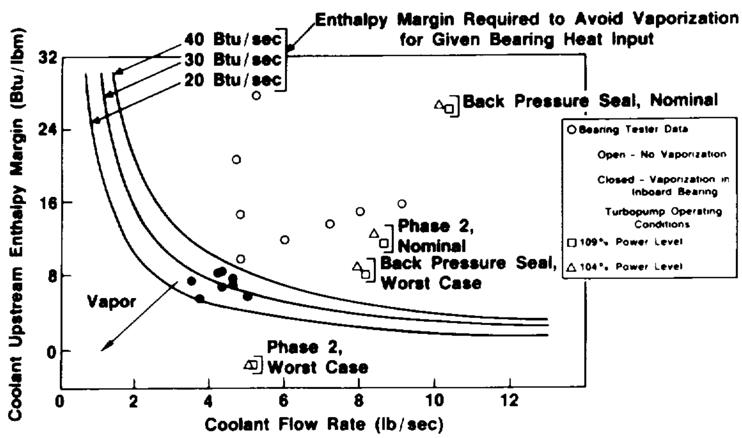

Fig. 16. HPOTP Pump End Bearing Comparison of Tester Data With Operating Coolant Conditions

LOW PRESSURE TESTING RESULTS

Build 1 testing was used to check out the test facility and tester using liquid nitrogen (LN_2), and LOX was used for subsequent testing.

For Build 2, parametric tests were conducted where coolant flowrate was varied from 4 to 16 lbm/sec (LPOTP flow is approximately 10 lbm/sec) at 290 and 180 psig. No bearing wear or indications of fluid vaporization were obtained indicating that bulk vaporization of coolant does not occur in the LPOTP. Bearing heat generation rate was found to be strongly influenced by applied load, however the bulk temperature rise across the bearing was only 1 or 2^{o}F.

After accumulating 6600 sec at loads generally less than 2000 lbf, rapid bearing wear occurred in less than 200 sec when load was increased to 6000 lbf. The worn bearing was similar in appearance to worn LPOTP bearings; an example is shown in Fig. 17.

Build 3 testing consisted of operating at 3000 lbf axial load at coolant conditions slightly worse than predicted in the LPOTP. Moderate bearing wear (0.0005 in. ball diameter) was obtained after 18,000 sec, demonstrating that bearing life in excess of current LPOTP life could be obtained at 3000 lbf load, even with poorer coolant conditions. The condition of the bearings was similar to used LPOTP bearings.

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Fig. 17. Used Thrust Bearing Ball From LPOTP Unit 009
(Showing Typical Surface Distress Condition)

Application of Results to LPOTP

Build 2 and 3 testing suggests that premature turbopump wear may be caused by axial loads between 3000 and 6000 lbf. Reduction of the peak thrust load in the turbopump by changing the thrust balance may not be feasible because unloading of the bearing may occur at some operating conditions. Various approaches to extend bearing life at high load are being pursued.

The testing has demonstrated that bulk vaporization of coolant is not likely to occur in the LPOTP, hence an increase in coolant flowrate is not likely to be beneficial.

CONCLUSIONS

To date, the HP tester results have added significant experimental evidence that support proposed fixes to the HPOTP. It has been shown that:

1. Bearing wear, similar in appearance to that of the HPOTP No. 2 bearing, is initiated by poor coolant conditions.

2. Significant improvements in bearing life are obtainable by improving coolant vapor margin.

3. Current turbopump coolant flowrates and pressure levels are marginal for preventing coolant vaporization within the bearing.

Additionally, measurement of fluid pressures upstream and downstream of the bearing have helped quantify vortex strength and bearing heat generation data has been obtained.

Results of LP thrust bearing testing strongly suggest that the turbopump experiences sustained loads under some operating conditions in the 3000 to 6000 lbf range and that improvements in bearing geometry and/or materials, are required to extend bearing life.

The test facility and bearing tester have demonstrated the ability to subject the test bearings to known, repeatable operating conditions that can individually be varied over a wide range to evaluate separate influences. Moreover, they are providing an efficient and rapid means of evaluating new bearing geometries and materials at test conditions representative of the turbopumps.

CURRENT STATUS AND FUTURE PLANS

High pressure tester Build 3 and 4 have been completed, which compared Phase II and Phase I bearing life under similar conditions of poor vapor margin. Approximately 40% longer life was obtained from the Phase II bearing, which has larger internal clearance and increased inner race curvature. Test data are presently being reviewed. As indicated in Table 3, near term plans are to evaluate several material changes. Longer terms plans are to modify the upstream bearing position in the tester to accept a duplex bearing pair to obtain data on the flow field between the bearings. Also planned is the addition of a radial loading device to evaluate the effect of combined radial and axial loads.

The LP tester is currently completing a life test at 4500 lbf axial load, which will serve as a baseline for evaluating improvements to bearing geometry and materials. Testing of a high contact angle (for increased thrust capacity) bearing is planned next. A test to evaluate the influence of outer race misalignment at low load is also planned for the near future.