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FEASIBILITY DEMONSTRATION
OF
A 90mm HYDROSTATIC RADIAL BEARING
FOR A HYDROGEN TURBOPUMP APPLICATION
(PHASE II)

NOVEMBER 1969 THROUGH JANUARY 1970

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AEROJET NUCLEAR SYSTEMS COMPANY

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FOREWORD

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I. SUMMARY

This report presents the results of the follow-on phase for a 90mm liquid hydrogen journal bearing program to demonstrate the feasibility of such a system for a hydrogen turbopump application. The previously used tester was utilized to demonstrate the effect of design improvements of the hydrostatic test bearing, specifically in terms of load capacity. The principal bearing design changes consisted of increasing the size of the inlet orifices in order to increase the pressure ratio from 0.2 to 0.5, doubling the areas of the pockets (recesses) and providing a 15° chamfer around each pocket to serve as a smooth flow transition between the bearing pockets and the bearing lands. The bearing design was analytically investigated, again using the available computer program, assuming perfect parallelism between the journal and the bearing. Based on the previous experience, the calculated results were adjusted to account for angularity of the loaded shaft in the tester which simulates in part a typical turbopump of relatively short bearing span with overhanging load. On this basis, a load capacity of 1200 to 1500 lbs was predicted for operation at 0.6 eccentricity ratio (radial deflection equal to 60% clearance). This was considered an encouraging improvement over the test results with the previous bearing design which supported 550 lbs at 0.6 eccentricity ratio and 1230 lbs at an eccentricity ratio of 1.0. The actual load capacity of the improved test bearing, as demonstrated by testing with LH₂ at 12,000 rpm and 700 psi ΔP , was 1180 lbs at 0.6 eccentricity ratio. The load capacity at an eccentricity ratio of 1.0, (just prior to rubbing metal-to-metal contact), was 1960 lbs. Five test runs at various loads were made and a total of 1473 seconds operating time was accumulated. The LH₂ flow rate was approximately 0.5 lb/sec, which was slightly higher than the "uncorrected"

calculated value of 0.38 lb/sec. The actual pressure ratio, as calculated from measurements of pocket, inlet, and exit pressures, was in complete agreement with the analytical prediction of 0.5. Prior to the rotating tests, several non-rotating off-design and transient mode experiments were conducted. No instability was noted in any of the tests.

In the last of the five test runs, a speed increase to 24,000 rpm was attempted. During the gradual speed ramp at no load, shaft seizure occurred at about 23,000 rpm, terminating the test. Upon disassembly, it was found that the failure was caused by separation of the tungsten carbide flame plating of the journal. Inspection of the damaged part revealed a structural weakness of the bond which failed when the speed and the corresponding stresses acting on the interface were increased. It is felt that had the plating failure not occurred, the performance of this bearing at 24000 rpm would have been at least as successful as the performance at 12,000 rpm, and at 24000 rpm in the earlier hydrostatic bearing work (Reference 1).

The demonstrated results indicate the feasibility of hydrostatic bearings for LH₂ NERVA turbopump applications. A full-scale program directed toward development/qualification of an operational turbopump shaft support system is recommended. It is concluded from these results that such a program should concentrate on developing the operational capability of the system (start/shutdown and transient behavior) for the turbopump duty cycle requirements.

II. INTRODUCTION

The work reported herein was preceded by an IR&D funded LH₂ Hydrostatic Journal Bearing Feasibility Demonstration Program (Reference 1). In the initial effort (Reference 1), a 90mm diameter hydrostatic journal bearing

was tested using liquid hydrogen as the pressure fluid and utilizing a modified available ball and roller bearing tester for the test equipment. The bearing arrangement in this tester simulated that which exists in a turbopump having a relatively short span between bearings and an overhanging load, a condition conducive to shaft conical whirl. The resulting shaft angularity effect, typical for a turbopump application, was not accounted for in the test bearing calculations based on an available "ideal case" analytical technique which assumes perfect parallelism between the journal and the bearing. Thus, a deviation in performance from that predicted by analysis had to be expected.

During the course of this early development program, six steady-state test runs were made, three each at 12000 and 24000 rpm with the following results:

The total accumulated time during which the rotating shaft was supported on the LH₂ fluid film was 24 minutes, 39 seconds. Loads up to 1130 lbs were successfully supported by the hydrostatic test bearing during these experiments, although this was less than the desired design goal of 2000 lbs or the load predicted by analysis for an ideal case. The operation was stable and the LH₂ flow rate was less than 0.30 lb/sec.

In the last of the six test runs, rubbing contact between bearing and journal was deliberately induced at a shaft speed of 24000 rpm by gradually increasing the bearing load. This established a 1230 lb load capacity corresponding to an eccentricity ratio of 1.0. The momentary high speed rubbing contact caused seizure and severe damage to the test bearing and journal. No other parts of the tester or facility were damaged.

Steps for the replacement of the damaged bearing were initiated together with other preparations to continue this initial feasibility demonstration effort. The scope and the results of this follow-on effort are the subject for this report.

III. TECHNICAL DISCUSSION

A. Technical Objective

The principle objective of this (follow-on) program was to extend the initial experience of demonstrating the feasibility of a 90mm diameter prototype size LH₂ journal bearing for turbopump application in the NERVA size pressure class. Fiscal constraints dictated limitation of this effort to no more than two steady-state tests, possibly to include some "rubbing contact mode" and transient operation experiments.

The previously used tester was to be utilized, after repair of the journal and replacement of the bearing that were damaged in previous testing when rubbing contact between the journal and the bearing was induced (Reference 1). While it was recognized that the lack of perfect parallelism between the journal and the hydrostatic bearing was probably a major contributor to the lack of correlation between actual load capacity and the predicted capacity, it was strongly felt that some degree of load capacity improvement was possible in the same tester setup, in spite of the noted angularity and misalignment effects. Therefore, it was proposed to increase the inlet orifices in order to increase the pressure ratio from 0.2 to 0.5 at the zero eccentricity position of the shaft, and to enlarge the area of the pockets. Increased pressure ratio results in increased pocket pressure which acting over the larger surface area of the larger pockets provides increased load capacity. Pulse line connections were incorporated for measurement of the

pocket pressures during testing. Another test measurement improvement was to measure shaft lateral displacement in the LH₂ cavity of the tester in order to avoid the temperature variation problems that were encountered with the displacement transducer during the previous test runs. Also, minor facility improvements were made to assure fluid supply temperature less than 58°R, continuous measurement of LH₂ flow rate, and fluid temperature measurement within the bearing inlet cavity.

The desirable load carrying capacity was still 2000 lbs, the same goal that was set at the outset of the Hydrostatic Bearing Feasibility Demonstration Program. The steady-state operating parameters remained as initially specified as follows:

Fluid	Liquid Hydrogen
Bearing Supply Pressure	900 psig
Bearing Exit Pressure	200 psig
LH ₂ Fluid Temperature	59°R max.
Shaft Speed	24000 rpm

B. Test Bearing Analysis and Design

The design of the test bearing was modified to improve its load carrying capacity while retaining the same envelope size, the same operating parameters, and the same arrangement within the test equipment. The design changes of the test bearing consisted of:

1. increasing the diameter of the primary (inlet) orifices from 0.037" to 0.058" in order to effect a change in pressure ratio from 0.2 to 0.5. Pressure ratio is defined by pocket pressure minus exit pressure divided by supply pressure minus exit pressure.

2. doubling the area of each of the 8 pockets (recesses), making the new pocket size 0.98 x 0.98 inch, while retaining the original pocket depth of 0.009 inch.

3. providing a 15° chamfer around each pocket to serve as a smooth fluid flow transition between bearing pockets and bearing lands.

4. providing pressure pickup holes (.007 inch dia) on four (90° apart) of the eight pockets, including suitable pulse line connections for hookup to pressure transducers.

The mechanical design of the new test bearing is delineated in Figure 1. A comparison of parameters between the original and the modified test bearing design is tabulated below:

	Original	Modified
Diameter, D, inch	3.54 (90mm)	3.54 (90mm)
Length, L, inch	2.00	2.00
L/D	0.565	0.565
Number of pockets	8	8
Pocket Width, inch	0.60	0.98
Pocket Length, inch	0.80	0.98
Pocket Depth, inch	0.009	0.009
Diameter of Orifices, inch	0.037	0.058
Operating Radial Clearance, C, inch	0.0015	0.0015
Supply Pressure, psig	900	900
Exit Pressure, psig	200	200
Pressure Ratio, β	0.3 (0.2 actual)	0.5

As in the original design, the bearing and the journal material were Inconel 718 and the critical surfaces were flame plated with tungsten carbide and polished.

This modified test bearing design was analytically investigated, again using the available computer program (Reference 2), assuming perfect parallelism between the journal and the bearing. The results of these calculations provided performance predictions for radial load capacity, radial stiffness, and LH₂ flow rate, as presented by the curves in Figures 2 and 3.

It was decided to limit the nominal steady-state operation eccentricity ratio (radial displacement ÷ radial clearance) to 0.6 to maintain what may be considered a good margin of safety for minimum allowable fluid film thickness (rub margin). For 0.6 eccentricity ratio, the performance predictions from the above listed curves were:

Radial Load Capacity, $F_B = 1850.1\text{b}$

Radial Stiffness, $K = 1.65 \times 10^6 \text{ lb/inch}$

LH₂ Flow Rate, $Q = 38 \text{ gpm}$

Again, these analytical predictions are based on the ideal case assuming perfect parallelism between the journal and the bearing. In conjunction with the analysis, it was estimated (Reference 2) that the actual load capacity at 0.6 eccentricity ratio would be predicted by the load of the ideal case at 0.4 to 0.5 eccentricity ratio. Accordingly, the load capacity to be expected from this bearing at 0.6 eccentricity ratio was 1200 to 1500 lbs. While this load was still below the desired 2000 lbs objective, this projected capacity was considerably above the 550 lbs obtained in tests with the previous bearing design (Reference 1). Therefore, it was felt that testing of this design modification was warranted.

C. Test Equipment

The same test equipment used for the tests of Reference 1 (a modified electric motor driven 50mm ball and roller bearing tester) was also utilized for this program. The new test bearing of revised design was installed in the bearing housing by a leak-tight, shrink-fit assembly, followed by finish grinding of the bearing bore. The repaired journal (replated with tungsten carbide) was also finish-ground after shrink fit assembly on the shaft. Changes to improve instrumentation as stated were incorporated in the tester hardware. The tester is illustrated in Figure 4, Drawing No. 113600-2 (2 sheets). Significant instrumentation improvements were pressure measurement provisions (90° apart) in four of eight pockets in the bearing, and arrangement of a displacement transducer adjacent to the test bearing in the 200 psig bearing exit cavity where a fairly constant LH₂ temperature control could be maintained. In a related separate program, it was established that this specific displacement transducer, a product by the KAMAN Nuclear Corporation, can be operated in liquid hydrogen (no test experience on temperature cycle/life limit) and calibration at liquid hydrogen temperature was accomplished. The specific displacement transducer that was used in this test had a linear calibration of 40 mv/mil in the temperature range of -402°F to -413°F. The millivolt/displacement plot had the same slope at +68°F (ambient temperature) as at cryogenic temperature. However, this does not imply that the transducer is not temperature sensitive in the intermediate range; in fact, each transducer tested was found to have its own characteristic calibration curve, although quite linear in the range of interest.

The tester configuration is considered adequate for the initial feasibility demonstration of a prototype size hydrostatic bearing. The tester deviates from the "ideal" case, generally assumed for the hydrostatic bearing analysis computer program, in that there is some shaft angularity and misalignment in contrast to the analytical model which assumes that journal and bearing surfaces are parallel for all eccentricity ratios. In this tester design, the shaft is, in effect, pivoted at the ball (slave) bearing which places the hydrostatic bearing and the journal in an angular relationship with each other rather than parallel. For large bearing spans, this effect could be neglected, but for the comparatively short (6.035 inches) bearing span of this tester, a deviation of actual bearing performance from that predicted for the "ideal" case was to be expected. While this condition does not allow direct correlation between experimental results and analytical prediction, this tester shaft angularity may in fact more closely simulate the actual typical turbopump case than the ideal aligned case assumed for the mathematical model. It was felt that if hydrostatic bearing performance is sensitive to rotor conical whirl, then this condition may as well be part of the feasibility demonstration.

Loading of the test bearing is provided by a unidirectional radial pneumatic (gaseous hydrogen) radial loader. The span between the hydrostatic test bearing and the ball "slave" bearing is 6.035 inches, and the centerline distance between the slave bearing and the pneumatic loader is 9.157 inches.

The distance between front edge of the 2" long hydrostatic test bearing and the "slave" ball bearing is 7.035 inches, and the span between the shaft displacement transducer and the "slave" bearing is 4.177 inches.

Figure 5 shows a photograph of the hydrostatic test bearing assembled in the tester housing, and Figure 6 shows the journal as part of the tester shaft assembly.

D. Testing and Results

The overall testing effort was conducted in three phases:

(1) checkout of tester and facility, (2) non-rotating tests, and (3) rotating tests.

1. Checkout of Tester and Facility

The tester was assembled and installed in the facility following a preplanned procedure, (Reference 3). Talyrond equipment was used for checking the critical dimensions of the test bearing, including uniformity of pockets, prior to assembly of the tester.

Excellent repeatability of the displacement transducer readout at LH_2 temperature was established by lateral movement of the shaft within the limits of the journal and bearing clearance. Actual diametral clearance measurement by this method at LH_2 temperature agreed closely with 0.003 inch, the dimension for which the bearing was designed. The shaft movement for these tests was accomplished with a small load applied by the radial loader. When the load for displacing the shaft was increased, a discrepancy of the above measurements was indicated. The cause was determined by analysis to be due to shaft bending deflection in addition to the shaft lateral movement. This distorted the apparent straight line proportionality between shaft lateral movement at the transducer location and the bearing. The technique of correlating the indication by the displacement transducer to the variation of gap between journal and bearing was adjusted accordingly.

A flow calibration through the hydrostatic bearing inlet orifices was conducted with liquid hydrogen. The shaft assembly was removed from the tester for this purpose, making the bearing inlet orifices (eight in parallel) the only significant resistance in the flow path. The flow rates measured compared fairly well (within 12%) with the calculated values, confirming that the orifice coefficient of 0.7 used in sizing this particular orifice geometry (Figure 1) was appropriate.

In association with the above tests, the adequacy of the pocket pressure taps was confirmed. In the tester design small pulse lines connecting the pressure taps to the pressure transducers were used to prevent change to the pocket volume and associated potential fluid instability problems. With careful purging of the lines, the arrangement worked well. Purging became extremely important because the small passages were very sensitive to clogging by minute amounts of moisture turning to ice when the tester was being chilled to LH_2 temperature.

The incorporated design improvements in the LH_2 inlet line and measurement of supply fluid pressure, temperature, and flow rates checked out satisfactorily.

2. Non-Rotating Tests

Non-rotating tests served the purpose of evaluating the response of the hydrostatic test bearing to various typical steady-state and transient load applications without the risk of possible journal and bearing damage due to metal-to-metal contact at high speed. Significant non-rotating experiments and results were as follows:

a. Pressure Ratio Check

The LH₂ supply pressure was set at 900 psig and the pressure at the hydrostatic bearing exit was held at 200 psig. No load was applied by the pneumatic loader; the only load acting on the bearing was that due to the shaft assembly static weight, which for the purpose of this experiment was considered negligible, and the journal was assumed to be in the zero eccentricity position. For this condition the pressures in the upper and in the lower pocket (bearing recess), as indicated by the pressure transducers were exactly 550 psi. Applying this to the formula by which the pressure ratio, β , is defined;

$$\begin{aligned}\beta &= \frac{P_P - P_E}{P_S - P_E} \\ &= \frac{550 - 200}{900 - 200} \\ \beta &= 0.5\end{aligned}$$

Therefore, this test measurement demonstrated complete agreement with the design value and adds confidence to the analytical technique that was applied in sizing this test bearing for the 0.5 pressure ratio.

b. Measurement of Variation in Pocket Pressures with Bearing Load

The LH₂ supply pressure was again set at 900 psig and the hydrostatic bearing exit pressure at 200 psig. The pneumatic loader was actuated and load was applied to the test bearing in increments of 200 lb up to 2400 lb. For each bearing load increment the pressure in the two opposing pockets aligned with the unidirectional load was measured. This test was

repeated four times to check for repeatability. These results are graphically presented in Figure 7. It is significant to note that in the range of about 1850 lb bearing load the slope of the curves level off and the pocket pressures remained constant regardless of increase in load. This indicated that metal-to-metal contact of the journal and bearing occurred at about 1850 lb load, thus establishing the load capacity of this bearing at an eccentricity ratio of 1.0 (zero clearance).

c. Effect of Variation of Supply Pressure at Constant Bearing Load

The LH₂ supply pressure was set at 300 psig, and the hydrostatic bearing exit pressure was controlled at 200 psig. A 640 lb unidirectional bearing load was applied by the pneumatic loader. While the bearing load and the exit pressure were kept constant, the inlet pressure was gradually increased up to the nominal inlet pressure design value of 900 psig. The corresponding pocket pressures were measured for the respective inlet pressure increments, and the results are tabulated as follows:

Supply Pressure P _S psig	Exit Pressure P _E psig	Bearing Load F _B lb	Pocket Upper PP ₁ psig	Pressures Lower PP ₃ psig	LH ₂ Flow Rate G.P.M.
300	200	640	285	255	19
350	200	640	330	270	23
400	200	640	280	290	27
450	200	640	425	315	30
500	200	640	445	325	34
600	200	640	500	370	39
700	200	640	550	420	43
800	200	640	600	475	47
900	200	640	645	520	51

Within the scope of the initial feasibility investigation, this experiment provided some information regarding the bearing behavior at supply pressures below 900 psi (the steady-state design value). No instability (pneumatic hammer type chatter) was apparent.

d. Other Transient Mode Simulation Experiments

In one experiment, the supply pressure to the hydrostatic test bearing was gradually increased from 0 to 900 psig, while the exit pressure was simultaneously raised from 0 to 200 psig. During the simultaneous increase of inlet and exit pressures, the bearing load was gradually increased from 0 to 400 lbs. It is significant that in this experiment the pressurizing fluid changed from the gaseous to the liquid phase; no instability was noted in the bearing behavior.

In another experiment the test bearing was exposed to quick loading and unloading while the supply and exit pressures were kept constant at 900 psig and 200 psig, respectively. Using quick-loading and unloading valves, the load variations were as follows:

Quick loading of bearing, 0 to 400 lbs

Quick unloading of bearing, 400 to 0 lbs

Quick loading of bearing, 0 to 800 lbs

Quick unloading of bearing, 800 to 0 lbs

Quick loading of bearing, 0 to 1600 lbs

Quick unloading of bearing, 1600 to 0 lbs

No adverse effects were noted in any of the above experiments, and the bearing journal appeared to be completely supported on the LH_2 fluid film during the sudden various load applications. Observation of distance transducer output did not reveal any significant shaft excursions.

These tests served the purpose of an initial cursory probing of the various facets of concern without getting involved with details that could not possibly be handled in sufficient depth within the specified program restraints. For this reason, no high frequency dynamic response data was gathered. Pocket pressure measurements were not recorded because the lag time associated with these measurements would not permit valid conclusions for this type experiment. The instrumentation provided for pocket pressure measurements was not intended to furnish high frequency response data.

3. Rotating Tests

Evaluation of the hydrostatic test bearing under dynamic conditions was performed at nominal (design) supply and exit pressures of 900 and 200 psig respectively, and shaft speeds of 6000 and 12000 rpm. The maximum load that the bearing was able to support on the LH₂ fluid film, just prior to light metal-to-metal contact (eccentricity ratio = 1.0) , was 1960 lbs which was slightly higher (6%) than the load capacity established in the non-rotating test for an eccentricity ratio of 1.0. At 0.6 eccentricity ratio, the bearing supported a steady-state load of 1180 lbs. Five test runs at various loads were made, and a total of 1473 seconds of operating time was accumulated.

In the last of the five runs, a speed increase to 24,000 rpm was attempted. During the gradual speed ramp at no load, shaft seizure occurred at about 23,000 rpm, terminating the test. Upon disassembly of the tester, it was found that the failure was caused by separation of the tungsten carbide flame plating of the journal, causing consequent damage to the adjacent tungsten carbide flame plated surface of the hydrostatic bearing. The degree of damage to the journal and bearing is shown in the photographs of Figures 8 and 9. Since the failure occurred as a consequence of speed increase,

the separation of the plating may be associated with the increased stresses acting on the plating bond at the correspondingly larger centrifugal force. Specifically, one or more of the following may have contributed to this weakness at the plating interface:

a. The tungsten carbide plating may have been too thick (.0045") for the specific use. The plating on the journal used in the preceding program at 24,000 rpm (Reference 1) was only .002" thick. The added thickness was an economy measure to repair and utilize the old part for this program.

b. The tungsten carbide plating was not of the same quality as that on the journal in the previous program (Reference 1) and was more sensitive to thermal cycling.

c. Possibly, light blistering (unbonding) of the journal plating may have been induced during the deliberate momentary rub at 6000 rpm to establish metal-to-metal contact, $e = 1.0$. However, this initial blistering of the plating at 6000 rpm was not very extensive because the hydrostatic bearing was immediately thereafter successfully operated at 12,000 rpm while supporting a steady-state load of 1180 lbs (0.6 eccentricity ratio) for 381 seconds. The subsequent increase in speed (near 24000 rpm) and the much higher centrifugal stresses may have caused the unbonded area of plating to lift off and jam between the journal and bearing.

Pertinent measurements of temperatures, pressures, flow rate, drive motor current, shaft speed, LH₂ flow rate, and shaft lateral displacement were taken at measuring point locations indicated in the flow schematic diagram (Figure 10). In Figures 11 through 15 respectively, the following test parameters are plotted as a function of time for each test run:

Bearing Supply Fluid Temperature	T_s ($^{\circ}\text{F}$)
Bearing Exit Fluid Temperature	T_e ($^{\circ}\text{F}$)
Bearing Supply Fluid Pressure	P_s (psi)
Bearing Exit Fluid Pressure	P_E (psi)
Drive Motor Current	I_{ϕ_B} (amp)
Load on Hydrostatic Bearing	F_B (lb)
Bearing Pocket Pressure (Upper)	PP_1
Bearing Pocket Pressure (Lower)	PP_3
LH ₂ Flow Rate	$\dot{w}$ (lb/sec)
Eccentricity Ratio = $\frac{\text{Displacement}}{\text{Clearance}}$,	(inch/inch)

The values for eccentricity ratios were derived by first plotting the shaft position measurements obtained from the distance transducer during the rotating tests and the corresponding hydrostatic bearing load imposed by the pneumatic loader. Then the corresponding eccentricity ratios were found by applying a linear relationship (a simplifying assumption) between the values established by test for eccentricity ratios of zero (no load) and 1.0 (onset of metal-to-metal contact), respectively. This data is presented in the plots on Figure 16 and 17.

Highlights of the five tests, in the order in which they were run, are summarized as follows:

Run No. 1188-01-OW-007-1

265 seconds run duration at 6000 rpm; no load applied during the first 125 seconds, then gradual load ramp (115 seconds) to 1220 lbs and holding at 1220 lbs steady load during the remaining 25 seconds of this run. The test was smooth and no abnormalities were noted, (Reference Figure 11).

Run No. 1188-01-OW-007-2

442 seconds run duration at 6000 rpm; no load applied during the first 60 seconds, then quick loading to 1020 lbs followed by gradual load ramp to 1715 lbs bearing load at which time shutdown was dictated by LH₂ run tank depletion. The test was smooth and no abnormalities were noted (reference Figure 12).

Run No. 1188-01-OW-007-3

267 seconds run duration 6000 rpm; no load applied during the first 74 seconds, then gradual load increase until onset of light metal-to-metal contact between the journal and the bearing was effected, which occurred at 1960 lbs bearing load. Onset of metal-to-metal contact was indicated by sharp rise of motor current, whereupon the bearing load was released and the test was shut down. Prior to the momentary rubbing, the test run was smooth (reference Figure 13).

Run No. 1188-01-OW-007-4

431 seconds run duration at 12,000 rpm; no load applied during the first 54 seconds, then gradual (30 seconds) load ramp to 1145 lbs, holding at this load and speed for 294 seconds. The load was then removed and the rotor rotation continued without externally applied bearing load for the remaining 43 seconds of the run, when the test had to be shut down because of LH₂ depletion. The test was smooth throughout the duration of the run. Note that for 294 seconds at the 1145 lbs steady-state bearing load the shaft was approximately in the 0.6 eccentricity ratio position (reference Figure 14).

Run No. 1188-001-OW-008-1

The objective of this test was to repeat the steady-state load of Run 1188-01-OW-007-4 described above, except that the shaft speed was to be increased to 24,000 rpm. Without externally applied bearing load, the test was started by a 12 second ramp to 12,000 rpm, holding steady-state speed for 37 seconds at no load, followed by an attempted gradual no load speed ramp to 24,000 rpm. At about 23,500 rpm, trouble was indicated by an abrupt rise in drive motor current, requiring immediate shutdown of the test after 68 seconds of run time. Posttest inspection showed that rubbing contact had occurred and caused severe seizure of the bearing and the journal. From inspection of the damaged hardware, it is reasonable to assume that rubbing contact was initiated by separation of the tungsten carbide flame plating from the journal. Since the bearing performed satisfactorily at lower speeds, it was assumed that this failure was associated with the increased stresses that were acting on the plating bond at the higher speeds. Three possible causes for this material weakness were already mentioned (reference Figure 15).

IV. GENERAL CONCLUSIONS

The table on the following page presents a comparison of technical objectives, analytical predictions, and test results for some pertinent technical parameters of this 90mm LH₂ hydrostatic test bearing.

	Objective	Analytical Predictions		Test Results	Comparison To Test Results With a Previous Design (Ref. 1)
		Uncorrected	Corrected For Angularity Effects		
Flow Rate, lb/sec	2.0	0.38	----	0.5	0.3
Pressure Ratio,	0.5	0.5	0.5	0.5	0.2
Load Capacity, lb			1200		
At 0.6 Eccentricity Ratio	2000	1850	to		
At 1.0 Eccentricity Ratio	----	----	1500	1180	550
			----	1960	1230
Bearing Stiffness at 0.6 Eccentricity Ratio, lb/inch	5×10^6	1.65×10^6	----	1.3×10^6	1.2×10^6
Supply Pressure, psig	900	900	900	900	900
Exit Pressure, psig	200	200	200	200	200
Shaft Speed, rpm	24,000	24,000	24,000	12,000	24,000

The performance of this test bearing reflects a significant improvement over the bearing design used in a previous test (Reference 1). Although the load capacity was below the desired objective, the load (1180 lbs) supported on a LH₂ fluid film at 0.6 eccentricity ratio came close to the analytically predicted value when adjusted for angularity effects. In terms of load capacity, the design approach of increasing the size of the supply orifices, increasing the pressure ratio, enlarging the area of the pockets (recesses) and modifying their geometry provided the desired improvement. The measured LH₂ flow rate of 0.5 lb/sec was slightly higher than the predicted (uncorrected) value of

0.38 lb/sec; however, this measured flow rate may include some leakage flow, and the predicted number was not adjusted for angularity effects. In any case, the 0.5 lb/sec flow rate necessary to operate this bearing was well below the design limit of 2.0 lbs/sec that was set in the original objective. While the testing of this bearing was done at only half the speed (12,000 rpm), it is felt that from a fluid mechanics/load capacity point of view, there is no reason why the bearing could not have performed equally well at the 24,000 rpm shaft speed.

The structural failure of the tungsten carbide flame plating points out that there is need for detail work to be done in selection of materials and material coatings suitable for this specific use and requirement. Suitable rub-material combination of the bearing and the journal will be an important factor in making a hydrostatic bearing "live through" start and shutdown transients and/or accidental rubbing contact modes of turbopump operation. One rubbing contact experiment that was accomplished in this program was the momentary and light rubbing at 6000 rpm (Run No. 1188-00-OW-007-3). This experiment was followed immediately by a test at 12,000 rpm supporting 1140 lbs at 0.6 eccentricity ratio. The run was smooth, and if any damage to the plating was done by the preceeding experiment it was not apparent by the hydrostatic bearing performance during this run.

In a series of non-rotating tests that preceded the above discussed runs, preliminary indication of bearing behavior at other than steady-state design and conditions was obtained. No instability was noted in any of these experiments, as indicated by the absence of typical pneumatic hammer chatter. Complete agreement between the analytically predicted and the actual pressure ratio ($P_R = 0.5$) was demonstrated. It was indicated that the load capacity

with shaft rotation was slightly higher than with the non-rotating shaft, but this observation needs to be confirmed and analytically correlated in future work.

The instrumentation improvements that were provided, specifically pocket pressure measurement and shaft lateral movement measurement, were beneficial and the experience gained with using this instrumentation suggested several important refinements that should be incorporated in a more encompassing future program. For this effort, instrumentation development was minimized due to fiscal restraints. However, development improvements in this field are definitely possible and will be an important key for more accurate data and analysis correlation that can lead to improvements of analytical techniques in the future.

The computer program used in the test bearing design pertains to an "ideal case" mathematical model which assumes perfect parallelism between the journal and the hydrostatic bearing. Adjustment of the analytical results to include angularity and misalignment effects, had to be based on judgement and previous program experience. On this basis, the correlation of predicted and experimental performance was reasonably close. However, the analytical technique on which this computer program is based should be updated to include angularity and misalignment effects in this mathematical model and to incorporate the latest state-of-the-art information of more recent publications (References 4 and 5). With these improvements, more "precise" predictions should be possible in the future.

The fabrication and assembly of the hydrostatic tester hardware provided valuable experience that can be directly applicable to the use of hydrostatic bearings in a turbopump. This includes precision ECM processes

and special design and manufacturing considerations affecting fits, clearances, and internal leakage to which hydrostatic bearings are extremely sensitive. Valuable information with respect to instrumentation requirements and techniques was also acquired.

V. RECOMMENDATIONS

Based on the demonstrated results, hydrostatic bearings are considered feasible for LH₂ turbopump application and a full scale NERVA turbopump oriented LH₂ hydrostatic bearing development program is recommended. The overall effort should be directed toward development/qualification of an operational turbopump shaft support system. Since in practice, turbopump rotor assemblies are not perfectly rigid, the analytical effort must be extended considerably beyond the mathematically convenient and classically ideal case of perfect parallelism. A concept for a more versatile tester, utilizing much of the existing tester hardware, is available and would make possible a considerable improved test development effort.

Included in this program must be the analysis, design, and development of means for rotor support during the transient modes of operation with such possible options as:

- a. pressurizing fluid supply furnished by accumulator during start and shutdown transient;
- b. rubbing contact during transient mode (Boot Strap operation);
- c. hybrid (Hydrostatic/Rolling Element Combination) bearing operations.

The final program phase should include the application and evaluation of the developed hydrostatic radial bearing system in a LH₂ turbopump.

LIST OF REFERENCES

- 1) The Feasibility of a 90mm LH₂ Hydrostatic Journal Bearing, Dec 1968 thru Nov 1969 Summary Report, by E. Roesch and J. B. Accinelli, Report No. 774OR-70-011.
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- 4) Hydrostatic Bearings for Cryogenic Rocket Engine Turbopumps, J. M. Reddecliff and J. H. Vohr, ASME Journal of Lubrication Technology, July 1969.
- 5) Performance of Hydrostatic Bearings for NERVA Class Turbopumps, Dec 1969, Final Report F-C2286, by S. Heller and O. Decker, The Franklin Institute Research Laboratory.

BEARING RADIAL STIFFNESS AND RADIAL LOAD CAPACITY
AS A FUNCTION OF ECCENTRICITY RATIO
(P. S. BUCKMANN, 9-2-69)

1. RADIUS, 1.77 IN.	7. SUPPLY PRESSURE, 900 LB/IN ²
2. LENGTH, 2.00 IN.	8. EXIT PRESSURE, 200 LB/IN ²
3. CIRCUMFERENTIAL LAND, .411 IN.	9. SPEED, 24,000 RPM
4. AXIAL LAND LENGTH, .51 IN.	10. RADIAL CLEARANCE, .0015 IN.
5. NUMBER OF RECESSES, 8.	11. PRESSURE RATIO, β , .5
6. LUBRICANT, LH ₂	

FIGURE 2
774OR-70-012

VOLUME FLOW RATE AS A FUNCTION OF PRESSURE RATIO
AND RADIAL CLEARANCE
(P. S. BUCKMANN, 9-2-69)

1. RADIUS, 1.77 IN.	6. LUBRICANT, LH_2
2. LENGTH, 2.00 IN.	7. SUPPLY PRESSURE, 900. LB/IN ²
3. CIRCUMFERENTIAL LAND, .41 IN.	8. EXIT PRESSURE, 200. LB/IN ²
4. AXIAL LAND LENGTH, .51 IN.	9. SPEED, 24000 RPM.
5. NUMBER OF RECESSES, 8.	10. ECCENTRICITY RATIO, $\epsilon = 0.0$

FIGURE 3
7740R-70-012

HYDROSTATIC BEARING TESTER, DRAWING NO. 1136000-2

HOUSING SUBASSEMBLY, HYDROSTATIC BEARING

FIGURE 5

7740R-70-012

BEARING JOURNAL

SHAFT ASSEMBLY
HYDROSTATIC BRG
P/N 1136071
PRE B/U # 5

SHAFT/JOURNAL ASSEMBLY, HYDROSTATIC BEARING

FIGURE 6

7740R-70-012

POCKET PRESSURE AS A FUNCTION OF BEARING LOAD -
NON-ROTATING

FIGURE 7
7740R-70-012

PHOTOGRAPH SHOWING DAMAGE TO JOURNAL

FIGURE 8

7740R-70-012

PHOTOGRAPH SHOWING DAMAGE TO HYDROSTATIC BEARING

FIGURE 9

7740R-70-012

* ELECTRICAL INTERLOCK EXISTS SO THAT WHEN SHUTOFF VALVE IS CLOSED, BYPASS VALVE COMES OPEN.

FIGURE 10 FLOW SCHEMATIC DIAGRAM OF TESTER IN THE FACILITY

FIGURE 11

FIGURE 12 7740R-70-012

FIGURE 13

FIGURE 14

7740R-70-012

BEARING LOAD AS A FUNCTION OF SHAFT MOTION

BEARING LOAD AS A FUNCTION OF ECCENTRICITY RATIO

FIGURE 17
7740R-70-012