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VOLUME III OF VI

TRW REPORT No. ER-6041

AEC CATEGORY C-92B, M-3679 (43rd EDITION)

MERCURY RANKINE PROGRAM (SNAP 2) TOPICAL REPORT No. 28

MERCURY RANKINE PROGRAM (SNAP 2)
DEVELOPMENT OF LIQUID-MERCURY-LUBRICATED BEARINGS
VOLUME III
TILTING-PAD BEARING EXPERIMENTAL RESULTS
(TITLE UNCLASSIFIED)

AEC RESEARCH AND DEVELOPMENT REPORT

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FOREWORD

This development of Liquid-Mercury-Lubricated Bearings Topical Report defines the analytical and experimental results obtained with fluid film bearings lubricated by liquid mercury. This report includes the design approach, analytical techniques, procedures, test results, and experimental activity performed by TRW Inc. as a facet of the Mercury Rankine Power Conversion Program during the period of January 1961 through June 1963.

This report consists of the following six volumes:

Volume I	Analytical Design Approach and Status of Bearing Systems
(With Appendixes)
Volume II	Plain Bearing Experimental Results
Volume III	Tilting-Pad Bearing Experimental Results
Volume IV	Three-Sector Bearing Experimental Results
Volume V	Three-Pad Bearing Experimental Results
Volume VI	Spiral-Groove Thrust Bearing Experimental Results

For convenience of the reader, citations of reference works are the same in each volume; e. g., Reference 6 cited in Volume I and Reference 6 cited in Volume II are the same publication.

Volume I contains the Nomenclature and the References for the study. All other Appendixes are also included in Volume I.

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ABSTRACT

This volume presents the results of the activity associated with the development of tilting-pad journal bearings.

On the basis of preliminary laminar analysis with turbulent corrections the three-shoe tilting-pad is selected, sized, and tested. The excellent stability characteristics of this bearing type is not influenced by the turbulent flow conditions in the clearance space. A Reynolds number correction factor is established to reflect the increased load capacity over laminar prediction when operating in the region between $1000 < Re < 8000$.

The tilting-pad bearing meets the major CRU requirements. However, seal erosion, additional seal power loss, possible pivot point wear, and reduced load capacity are the disadvantages associated with the use of this bearing type.

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I. INTRODUCTION AND SUMMARY

The tilting-pad journal bearing was selected for evaluation to overcome early CRU development problems associated with unstable bearing operation and severe thermal gradients causing unacceptable degrees of misalignment.

Preliminary analysis using laminar flow equations predicted adequate load carrying capacity, stability freedom from critical speed problems, and satisfactory flow-pressure requirements. Based on these analyses, three-pad tilting-pad bearings were designed and tested to establish performance characteristics including parametric data, endurance testing, and shock and vibration evaluation.

The basic bearing was 0.625 inches in diameter and contained three shoes or pads pivoted approximately 5/9ths from the leading edge. Major dimensional variables were axial length and pivot point and machined-in clearances, i. e., pre-load factor. A series of parametric, endurance, and shock and vibration tests established performance and characteristics of the bearing, the pivot configuration, and the end seals used to maintain a flooded bearing cavity.

In the process of determining basic capability, the analytical prediction techniques were improved to include the influence of the turbulent or superlaminar flow regime.

The development tests verified the stability characteristics, freedom from critical speed, and satisfactory flow-pressure requirement of the tilting-pad bearing. However, in addition to establishing that the basic capability was satisfactory for CRU requirement, the tests revealed a seal cavitation-erosion problem, verified the additional power loss associated with the seals, and demonstrated the susceptibility of pivots, unless properly designed, to suffer Brinnelling and wear. Because of these characteristics, the development of stable, high-load capacity hybrid bearings and the improvement in CRU design assuming good high temperature alignment, further development of the self-aligning bearing was postponed.

II. TILTING-PAD BEARING

As it became evident in the CRU development program that an alternate bearing capable of tolerating misalignment and clearance changes was highly desirable, greater emphasis was placed on the tilting-pad bearing. The predicted stability of the bearing and good zero-load fluid-film stiffness also appeared attractive. The major disadvantages appeared to be the need for an optimum load orientation and the possibility of fretting due to rapid pad oscillation under dynamic loading. Other disadvantages were greater multiplicity of pieces, more difficult assembly, and the need for fluid dynamic seals to maintain flooded conditions in the bearing.

Figure 1 shows the tilting-pad bearing consisting of three pads, each mounted on a ball that permits axial and circumferential tilting. The tilting capability makes the bearing self-aligning and retains bearing alignment if housing distortions occur. Mercury lubricant enters the bearing between the pads and flows circumferentially between the journal and pads due to journal rotation. The lubricant then spirals axially out of the bearing through the shaft seals which provide the controlling restrictions for bearing flow and assure flooded operation. Figure 2 shows the assembled bearing with the end plate removed.

The tilting-pad bearing derives its load-carrying capacity from hydrodynamic action. Being purely hydrodynamic, all three pads must be completely flooded with lubricant. This is accomplished by providing a large radius at the leading edge of each pad to ease the entrance conditions and providing a screw seal to restrict the outflow of lubricant from the cavity containing the pads.

A tilting-pad bearing with satisfactory load capacity, stability, flow-pressure requirements and fluid-film stiffness was designed, analyzed, and tested. The tilting-pad bearing configuration selected for analysis consisted of three pivoted pads (shoes). The pivots were located 120 degrees apart and 5/9 of the pad length from the leading edge. Since the solution requires an iterative force balance and pad inclination to establish the running position of the journal within the bearing, the dimensions of the bearing to be analyzed were specified as follows:

Shaft diameter	0.625 inch
Aspect ratio of pad (B/L)	0.50 and 0.75
Arc length of pad	118 degrees
Diametral clearance	0.0015, 0.00175, and 0.0020 inch at pivot

The laminar analysis without pre-load predicted the load-deflection characteristics at 40,000 rpm shown in Figure 3. Table I is a summary of the performance at speeds from 10,000 to 30,000 rpm.

Figure 1. Tilting-Pad Bearing Schematic

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Figure 2. Assembled Tilting-Pad Bearing with End Plate Removed

[illegible]

Figure 3. Laminar Load Carrying Capacity vs Eccentricity Ratio for a Tilting-Pad Bearing at 40,000 RPM

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TABLE I

LOAD CARRYING CAPACITY FROM 10,000 TO 30,000 RPM

ϵ	Load Carrying Capacity (Net) at 30,000 rpm					
	B/L = 0.5			B/L = 0.75		
	mD = 0.0015	mD = 0.00175	mD = 0.002	mD = 0.0015	mD = 0.00175	mD = 0.002
0	0	0	0	0	0	0
0.1	0.9	0.654	0.504	2.415	1.785	1.35
0.2	1.725	1.26	0.969	4.38	3.21	2.46
0.3	2.67	1.95	1.5	6.54	4.8	3.69
0.4	3.33	2.445	1.875	8.7	6.39	4.89
0.5	4.23	3.12	2.37	11.34	8.1	6.18
0.6	5.22	3.84	2.715	13.5	9.9	7.59
0.7	6.39	4.71	3.6	16.65	12.24	9.39
0.8	7.8	5.76	4.41	20.4	14.94	11.4
0.9	9.6	7.05	5.4	25.05	18.45	14.1

ϵ	Load Carrying Capacity (Net) at 20,000 rpm					
	B/L = 0.5			B/L = 0.75		
	mD = 0.0015	mD = 0.00175	mD = 0.002	mD = 0.0015	mD = 0.00175	mD = 0.002
0	0	0	0	0	0	0
0.1	0.595	0.436	0.335	1.61	1.19	0.9
0.2	1.15	0.84	0.645	2.91	2.14	1.64
0.3	1.78	1.31	1.0	4.36	3.2	2.45
0.4	2.22	1.63	1.25	5.8	4.25	3.25
0.5	2.81	2.07	1.58	7.35	5.4	4.11
0.6	3.48	2.56	1.81	9.0	6.6	5.05
0.7	4.26	3.13	2.4	11.1	8.15	6.25
0.8	5.2	3.83	2.93	13.6	9.95	7.6
0.9	6.4	4.7	3.6	16.7	12.3	9.4

ϵ	Load Carrying Capacity (Net) at 10,000 rpm					
	B/L = 0.5			B/L = 0.75		
	mD = 0.0015	mD = 0.00175	mD = 0.002	mD = 0.0015	mD = 0.00175	mD = 0.002
0	0	0	0	0	0	0
0.1	0.3	0.218	0.168	0.805	0.595	0.45
0.2	0.575	0.42	0.323	1.46	1.07	0.82
0.3	0.89	0.65	0.5	2.18	1.6	1.23
0.4	1.11	0.815	0.625	2.9	2.13	1.63
0.5	1.41	1.04	0.79	3.78	2.7	2.06
0.6	1.74	1.28	0.905	4.5	3.3	2.53
0.7	2.13	1.57	1.2	5.55	4.08	3.13
0.8	2.6	1.92	1.47	6.8	4.98	3.8
0.9	3.2	2.35	1.8	8.35	6.15	4.7

NOTE:

$$mD = \frac{C_D}{D} \cdot D = C_D$$

m = clearance modulus $\frac{C_D}{D}$

D = diameter

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Appendix C presents a laminar analytical technique developed at TRW to include the influence of pre-load on the load capacity of tilting-pad bearings. This analysis was extended to include the influence of pre-load on power loss and fluid film stiffness. Further extension of the analysis resulted in dimensionless design curves of load capacity, power loss, and bearing fluid film stiffness as shown in Figures 4, 5 and 6. These design curves were obtained from the developed analytical technique for the following tilting-pad bearing configuration:

- 1) Three equal shoes
- 2) Each shoe with a bearing arc of 98°
- 3) Pivot location $(x/B) = 0.575$

The analytical technique is general and will permit development of similar design curves for a tilting-pad bearing with any number of shoes, i. e. different arc length, different pivot locations, and for any desirable pre-load. This technique of accurately predicting laminar performance is essential to the establishment of turbulent flow factors and the correlation scheme being developed at TRW. This scheme is described in Sections B and C.

The influence of pre-load on the load-deflection characteristics are shown in Figure 7 and compared to the prediction which neglects pre-load. The influence of the "lobing" caused by pre-loading is seen to result in a lower load capacity-to-eccentricity ratio of 0.5 and higher capacities at $\epsilon > 0.5$ as the increased convergence from the leading edge to the pivot point begins to dominate.

Appendix C is expanded to include the influence of radial clearance on the predicted performance of a three-pad tilting-pad journal bearing.

A series of tests were conducted on the tilting-pad bearings with different clearances and for two different axial lengths. In addition, the pivot configuration and the screw seals received attention during the parametric endurance and vibration tests. Table 2 presents a summary of the test series on tilting-pad bearings. Paragraphs A through G review the highlights of these tests.

A. MDBTF I-1 and -1A Test Series

The primary object of this initial test was to conduct a preliminary evaluation of tilting-pad bearings to determine their feasibility for application in CRU's. The two separate tests constituting this series were conducted on bearings having the following parameters:

$D = 0.625$ inch

L (axial) = 0.322 inch

Pivot Point Clearance = 0.0018 to 0.0022 inch and 0.003 to 0.0032 inch

Figure 4. Dimensionless Load Capacity ($\bar{W}_T$) vs Eccentricity Ratio (ϵ) for a Three-Pad Tilting-Pad Journal Bearing

Figure 5. Dimensionless Power Loss ($\bar{P}_T$) vs Eccentricity Ratio (ϵ) for a Three-Pad Tilting-Pad Journal Bearing

Figure 6. Dimensionless Bearing Stiffness ($\bar{K}$) vs Eccentricity Ratio (ϵ) for a Three-Pad Tilting-Pad Journal Bearing

Figure 7. Load Eccentricity for Tilting-Pad Bearing at 40,000 RPM

TABLE 2

TILTING-PAD BEARING TEST SUMMARY

Test Series	Bearings and Pivots					Remarks	Reports
	Diameter	Length	(Cp Inch)	Pivot Config	Seals		
MDBTF I-1 & -1A	0.625	0.322	0.0018 to 0.0033	Ball 3/32" on flat	Screw seal on shaft	Feasibility and parametric	
MDBTF II-1, -1A & and -1B	0.625	0.322	0.0016 to 0.002	Ball 3/32" on ground conical seat	Screw seal on shaft	Parametric and influence of rotating load	Ref TRW report ER-5010
MDBTF II-2, -2A	0.625	0.322	0.0023	Ball 3/32" on ground conical seat	Screw seals in housings	Parametric, influence of clearance, pre- load unbalance and screw seal relocation	Ref TRW report ER-5099
MDBTF II-2B, -2C	0.625	0.322	0.0023	Ball 3/32" on ground conical seat	Screw seals in housing	200 hour endurance with 20 lb rotating load per bearing and vibration test	Ref TRW report ER-5494
MDBTF III-1, -2	0.625	0.407	0.0015 and 0.0025	Six different pivot config- urations and 3/32" and 5/32" balls	Screw Seals in housing	1) Parametric tests 2) Verification of turbulent analysis 3) Modification to include (L/D)	Ref TRW report ER-5488
MDBTF III-3	0.625	0.407	0.0025	Six different pivot config- urations and 3/32" and 5/32" balls	Screw seals in housing	500 hour and endurance with rotating load of 20 lb/bearing	Ref TRW report ER-5804

Machined-in clearance $\approx$ 0.004 inch

Pivot location - 5/9 from leading edge (desired)

Pivot configuration = 3/32 inch dia ball with a flat in the shoe and elliptical seat in housing

Materials - 17-4 PH stainless, Rc 30-40

A schematic of the bearing assembly and a photograph of the actual bearing are shown in Figures 8 and 2 respectively.

The initial tests demonstrated that the tilting-pad bearing was feasible and worthy of further development. The bearing was free from half-frequency whirl at all loads up to maximum tested speed (45,000 rpm), demonstrating that turbulence in the clearance space did not destroy the inherent stability of the self-adjusting bearing. Loads up to 16 lb were successfully supported at 40,000 rpm.

With an initial pivot point clearance of 0.0018 to 0.0022 inch, the bearings in the MDBTF I-1 test performed satisfactorily until the leading edge of the loaded shoe appeared to contact the shaft at 40,000 rpm and with a unidirectional load of 16.4 lb.

In the MDBTF I-1A test with bearings lapped to a diametral clearance of 0.003 (same machined-in clearance), some form of vibration (not half-frequency whirl) was encountered at 40,000 rpm and loads below 12 lb. Amplitude of vibration was 0.0006 inch maximum.

Evaluation of these phenomena revealed the following:

- 1) The leading edge contact occurred because of an incorrect pivot location. The pivot point was in the center of shoe due to tolerance stackup; consequently, it theoretically had zero laminar load capacity.
- 2) Instability was due to an unpre-loaded condition of the bearing when the pivot clearance was increased. Brinnelling at the pivot, due to soft material properties and the seat configuration, aggravated the encountered vibration.

The load capacity at 40,000 rpm is superimposed on Figure 7. Turbulence tends to increase load capacity, but increasing the clearance did not influence load capacity or fluid film stiffness appreciably, although the larger clearance bearing appeared weaker at $\epsilon > 0.5$. Fluid film stiffness was 19,400 lb/in. and 19,200 lb/in. for the I-1 and I-1A tests respectively at a shaft weight of 2.5 lb.

Values of C_R Tested:

<u>"A" End</u>	<u>"B" End</u>
MDBTF I 0.0009 inch	0.0011 inch
MDBTF I-1 0.0015 inch	0.0015 - 0.0016 inch

Figure 8. Schematic of Tilting-Pad Shoe and Ball

Effectiveness of the screw seals located on the shaft was disappointing. A flow of 5 lb/min required as little as 0 to 16 psia at 40,000 rpm, indicating that additional work is required to improve seal effectiveness.

B. MDBTF II-1 Test Series

To overcome the problem areas encountered with the MDBTF I design and to obtain parametric data, the MDBTF II-1 test series was conducted. Two 0.625-inch diameter by 0.322-inch long tilting-pad journal bearings with the diametral clearance at the pivot point set at 0.0016 to 0.0020 inch were used.

Materials used were as follows:

Shoes	- BG 41 (modified 440C), Rc 55 to 60
Sleeves	- Bluechip 18-4-1 Tool Steel, Rc 55 to 60
Balls (Pivot)	- AISI 52100, Rc 60 to 67
Pivot configuration	- Spherical in shoes

All parts were hardened to alleviate the Brinnelling problem at the pivot point encountered in the I-1 test series.

The conical seat contained a lapped spherical seat to increase the contact area, thereby reducing the Hertz stresses. This seat was of the same radius as the pivot ball, but the cylindrical seat in the housing was ground 0.001 inch larger than the ball. The result of lapping the ball into the seat may create a fretting or wear problem at the expense of alleviating the Brinnelling problem. For zero wear and consequently increased freedom from fretting, the relative motion between shoe and ball and ball and housing should be pure rolling with no sliding or slipping. With the present pivot configuration, sliding predominates; this may require some modification later. To stabilize the unloaded shoe, all shoes were preloaded 0.001 inch to insure a converging wedge in the unloaded shoe at eccentricity ratios approaching one.

The clearance between the shaft and shoes is influenced by the stackup of several parts. The difficulty in measuring the complex shapes to accuracies better than 50 micro-inches and the problems of inspecting the loose assembly prevented determination of the exact pivot point clearance. Therefore, the operating clearance for this test series had to be defined as a range of clearance resulting from the limits set up by four different inspections. This range was from 0.0016 to 0.0020 inch.

The MDBTF II-1 test, conducted to determine the influence of changes in material, pre-load and pivot location, and the effect of rotating load on tilting-pad bearing performance, consisted of three tests which differed only in degree to which the shaft-rotor assembly balance was varied. The balance condition and rotating load are tabulated as follows:

<u>Test</u>	<u>Degree of Unbalance</u>	<u>Centrifugal Force at 40,000 rpm (i. e. , Rotating Load)</u>
MDBTF II-1	0.0005 in.- oz	< 1 lb/bearing or considered negligible
II-1A	0.007 in.- oz	10 lb/bearing
II-1B	0.014 in.- oz	20 lb/bearing

Figures 1 and 2 show the tilting-pad assembly arrangement.

The performance characteristics of interest in tilting-pad bearings, as in other bearings, are:

- 1) Load-carrying capacity, including unbalance
- 2) Attitude-eccentricity locus
- 3) Stability
- 4) Fluid film stiffness and critical speed
- 5) Flow-pressure requirements
- 6) Power loss
- 7) Wear

These characteristics are covered in the following sections for this series of tests, and in other parametric tests on different tilting-pad bearing sizes.

1. Load-carry Capacity

Figures 9, 10, and 11 show a composite of load capacity over a 20,000 to 40,000 rpm speed range. The eccentricity ratio for this figure is based on a pivot point radial clearance of 0.0008 inch ($C_{Dp} = 0.0016$). Since the measured clearance varied between 0.0016 and 0.002 inch, the curves should be adjusted for the 0.002 inch case. Consequently, the data represent a band of load performance for the above clearance range and must be used with caution because of the doubt about the exact minimum film thickness; i. e., for an eccentricity ratio of 0.5, the minimum thickness would be 0.0005 inch for the 0.002 inch clearance and 0.0004 inch for the 0.0016 inch clearance. The adjustment for actual clearance is made from the following equation of minimum film thickness

$$h_{min} = C_R (1 - \epsilon) \quad \dots 1$$

Figure 9. Load vs Eccentricity Ratio at 20,000 RPM

Figure 10. Load vs Eccentricity Ratio at 30,000 RPM

Figure 11. Load vs Eccentricity Ratio at 40,000 RPM

Although some scatter exists in the load-deflection data for this series of tests, no definite trends were evident as a function of rotating load and direction of unidirectional loading.

MDBTF I-1 results (Figure 7) are superimposed in Figure 11. Based on these data it appears that the pivot clearance during MDBTF II-1 test series was 0.0018/0.002 inch.

2. Attitude-Eccentricity Locus

Figure 12 shows a composite of the attitude-eccentricity data for the three separate tests. The scatter in data has been attributed primarily to instrumentation reading error caused by the interpretation of the circle center as seen on the oscilloscope, and the width of the scope beam as opposed to variation in bearing performance. Figure 13 illustrates the difficulty in establishing the actual eccentricity ratio as the amplitude of the shaft locus increases in size with increasing rotating load. Figure 12 shows that the attitude-eccentricity locus is relatively independent of speed and mass center location, i. e., unbalance, and that displacement of the shaft is almost directly in line with the applied load. As a result, the attitude angle remains small for all values of eccentricity ratio from 0 to 1.0. It is this feature that provides the tilting-pad bearing with its inherent stability.

3. Stability

The bearings were completely free from half-frequency whirl under all conditions tested in this series. The synchronous whirl component at shaft frequency caused by the purposely imposed unbalance was the only form of vibration encountered except at speeds above 30,000 rpm when the bearing lube supply was permitted to fall below 5 psig. Under these conditions, random and non-repeatable vibrations were noted. This was attributed to pad starvation caused by the inability to maintain a full bearing cavity and the presence of noncondensables within the package. The shoes may consequently have fluttered due to a varying pressure gradient.

4. Fluid Film Stiffness and Critical Speed

Evaluation of the data revealed that the shaft assembly passed through its first resonance at a very low speed, causing it to rotate about its center of mass. If a shaft is rotating about its center of mass, no centrifugal forces due to unbalance are generated. This was shown in the test by a constant amplitude of shaft displacement (relative to the bearing) with varying speed. Had the shaft not been rotating about its mass center, centrifugal effects would have caused the shaft displacement to increase with the speed squared. The displacement noted during this test series was a measure of the vector sum of the rotor-journal eccentricity and the mass center-journal eccentricity.

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MDBTF II-1, 1A, 1B
SHAFT CENTER LOCUS

Bearing -

Tilting Pad

D 0.625 inch

L 0.322 inch

$C_{Dp} = 0.0018$ inch

Figure 12. MDBTF II-1, 1A, 1B Shaft Center Locus

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Bearing:

Tilting Pad

$D = 0.025$ inch

$L = 0.322$ inch

$C_{Dp} = 0.0018$ inch

Figure 13. Oscilloscope Screen of Typical Run During MDBTF II-1, -1A, -1B

Figure 14. Rigid Shaft Critical Speed vs Bearing Stiffness MDBTF II-1

The bearing operating above critical at the speeds tested is shown in Figure 14 which shows both shaft critical and bearing stiffness as a function of speed. The bearing stiffness was calculated from the slope of the load deflection curve, dW/de , Figures 9, 10, and 11. At the test speeds (20,000 to 40,000 rpm), the bearing stiffness curve lies above the shaft critical curve. Extrapolation of the stiffness curve from 20,000 rpm to 0 rpm would indicate that a resonance should have occurred at about 13,000 rpm. This, however, was not noted. No clear-cut resonance point was defined at any speed. Several factors could have contributed to this:

- 1) The shaft critical speed curve is based upon the following relationship at low speeds:

$$f_{cr} = \frac{60}{2\pi} \sqrt{\frac{2K}{M}} \quad . . . 2$$

Where:

f_{cr} = critical speed

K = single bearing stiffness

M = shaft mass

This relationship is valid, however, only for an infinitely stiff shaft with one degree of freedom. The actual shaft critical speed will fall somewhat below that indicated by the above idealized case.

- 2) The bearing stiffness below 20,000 rpm is unknown. It is known that the stiffness at zero speed is zero, and, therefore, it is entirely possible that the shaft operates at all times above the first mode critical.
- 3) The instrumentation and speed control is not adequate to detect small amplitude and phase changes at low speeds. Because of the relatively high degree of damping the small amplitude changes encountered on passing through critical speed which occur at low frequencies are extremely difficult to pick up during a test. However, these large damping characteristics permit operation at, close to, or through a critical speed without encountering large amplitude changes for reasonably well balanced rotors where the distance between geometric and mass centers is within 10% of the bearing clearance.

5. Pressure Requirements

Since the bearing is purely hydrodynamic, the supply pressure does not affect its load-carrying ability if the bearing cavity is full of liquid. This was verified during the test by varying lube flow (and pressure) at various bearing loads and recording the shaft center position. It was demonstrated that pressure changes from 5 to 30 psig had no effect on shaft position. Variations beyond this range were not attempted due to instrumentation limitations.

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Figure 15. View of Journal After Test - Alternator End

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The performance of the screw seal used to keep the bearing cavity flooded was disappointing. With the flow maintained at approximately 5 pounds/minute the supply pressures increased from about 5 psi to approximately 25 psi as the shaft speed was increased from zero to near 30,000 rpm. At speeds above 30,000 rpm the supply pressure rapidly fell to near zero at 40,000 rpm. It was necessary then to increase the flow to about 8 pounds/minute to maintain pressures above 5 psig. This peculiar pressure performance with speeds above 30,000 rpm was attributed to the centrifugal force field throwing the mercury from the threads, thus rendering them ineffective as a seal. This checks with centrifugal seal results obtained as part of the CRU IV alternator seal study.

6. Wear

Figures 15, 16, and 17 are photographs of the bearing surfaces following the MDBTF II-1B test. They show the excellent surface condition of the bearings after some 50 hours of operation. The slight wear noted on the journal surfaces has been attributed to start-stop damage (approximately 50 were made during the test series), foreign particles, and metal-to-metal contact when the bearing was overloaded at 40,000 rpm. Only slight scratches can be seen on one of the shoes (B-1, shown in Figure 17.

Inspection of the pivot point revealed no wear or Brinnelling of any of the parts. However, traces of rust were found on the ball pivots where contact was made with the shoes and sleeve. Some of this can be attributed to water which inadvertently got into the test fixture early in the test series and some to fretting damage at the contact points. The fretting was compounded by the presence of water and oxygen.

7. Correlation with Analysis

The analytical approach to the tilting-pad bearing design consisted of three steps:

- 1) The preliminary design which made use of the laminar, flat slider bearing theory.
- 2) The inclusion of the preload in the laminar analysis. This changes the predicted performance characteristics as shown in Figure 7. The analysis utilized Abramovitz's crowned slider bearing theory (Ref 18) to account for the preload. This method should be more precise than the previous analysis.
- 3) The analysis of the turbulent slider bearing according to the method of Chou and Saibel (Ref 7) which makes use of Prandtl's mixing length theory and contains an empirical constant, but neglects the effect of preload.

Figure 16. View of Journal After Test - Turbine End

Figure 17. View of Pads After Test (A-Alternator End, B-Turbine End)

Figure 18. Experimental and Theoretical Load vs Eccentricity Ratio
 MDBTF II-1 Test Series

The preliminary design analysis was employed in the determination of tilting-pad bearing feasibility for the SNAP 2 CRU. The bearing size, clearance, and pivot point were chosen to meet CRU requirements. No correlation was made with experimental performance except to point out that the analysis was based on laminar flow and that the predicted performance for load-carrying capacity was conservative. The conservatism was established experimentally. The prediction of the bearing pad rocking critical frequency showing it to be far above the operating speed was verified. It was also predicted and demonstrated experimentally that the operating speed is always above the natural frequency of the shaft with the tilting-pad bearings although the increased stiffness measured for the turbulent case resulted in "cross over" at very low speeds.

Having generated considerable data during the MDBTF II-1 test series, TRW embarked on a correlation scheme to include the influence of turbulence in predicting tilting-pad bearing performance.

The predicted load-deflection curves are compared with the experimental curves in Figure 18. As can be seen, the experimental load capacities are in excess of those predicted by this analysis. This is due to the turbulence in the mercury lubricant film. A turbulence factor, K, was defined as

$$K = W_T / W_L \quad . . . 3$$

Where:

W_T = experimental load capacity

W_L = theoretical load capacity based on laminar theory

This factor was plotted versus eccentricity ratio for shaft speeds of 20,000; 30,000; and 40,000 rpm in Figure 19. It increases with increasing speed and decreases with increasing eccentricity ratio, indicating a correlation with Reynolds number if eccentricity ratio is included. The Reynolds number, therefore, is based on the loaded pad leading edge film thickness in the untilted position, h_1 .

$$Re_{h_1} = \frac{\rho U h_1}{\mu} \quad . . . 4$$

Figure 20 shows this turbulence factor K as a function of Reynolds number, Re_{h_1} .

The function is expressed as

$$K = 0.041 Re_{h_1}^{0.446} \quad . . . 5$$

Bearing

$D = 0.625$ inch

$L = 0.322$ inch

$C_{Dp} = 0.0018$ inch

Figure 19. Overall Turbulent Factor for a Tilting-Pad Journal Bearing
MDBTF II-1 Test Series

Figure 20. Overall Turbulent Factor vs Re_{h1} for a Tilting-Pad Bearing
 MDBTF II-1 Test Series

Figure 21. Prandtl's Mixing Length Constant vs Reynolds Number
 For Tilting-Pad Journal Bearing

MDBTR II-1 Test Series

The factor K is equal to 1 when Re_{h1} , the Reynolds number based on leading edge film thickness, is equal to 1350. For a bearing operating with $Re_h < 1350$, laminar theory should predict performance best; above this, the laminar load capacity should be modified by the factor K.

Further analysis was centered around Chou and Saibel's turbulent theory of the slider bearing. Details are presented in Volume I, Appendix D. Preload, however, was neglected here. In this analysis, load capacity is a function of an empirical constant k (Prandtl's mixing length constant). Since this is a rather lengthy analysis, it was programmed on a computer. Load capacity was computed for various values of k and compared to the experimental to obtain a representative value of k for the involved bearing parameters. For purposes of correlation, k was assumed to be a function of

$$k = k(U, \rho, \nu, C_R) \quad . . . 6$$

Dimensional analysis of this functional relationship results in the following:

$$k = C \sqrt{\rho} Re^\alpha \quad . . . 7$$

where $Re = \rho U C_R / \mu$

In Figure 21 the values of $k/\sqrt{\rho}$ obtained as discussed above, are shown plotted versus Re. The equation of the curve is

$$k = 3.25 \times 10^{-6} \sqrt{\rho} Re^{0.46} \quad . . . 8$$

when $Re > 5500$. Below this Reynolds number, this analysis cannot predict load capacity. When operating conditions are such that $1350 < Re < 5500$ the over-all turbulence factor K must be used.

It is perhaps premature to draw definite conclusions on the turbulent analysis of tilting-pad bearings based on the results of one set of tests. Nevertheless, a useful tool has been derived whereby the effects of turbulence can be included by an empirical equation which was established experimentally.

The results of future tests on tilting-pad bearings will be fed into this analysis so that the constants can be refined if necessary and the limits, where they apply, can be clearly established. As the curves show, additional test data are required.

C. MDBTF II-2 Test Series

The MDBTF II-2 test series was undertaken to test an improved bearing screw seal and to further investigate the effect of changed bearing clearance, unbalance, and preload. MDBTF II-2 was the initial test with a balanced rotor. MDBTF II-2A was the second test with a 180-microinch unbalance, equivalent to 20 lb of rotating load at 40,000 rpm.

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Note: All dimensions in inches

Figure 22. Tilting-Pad Test Bearing

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$D = 0.625$ inch
 $L = 0.322$ inch $\mu = 2.0 \times 10^{-7}$ Reyns
 $C_{Dp} = 0.0023$ inch $N = 20,000$ rpm
 $C_{MD} = 0.0041$ inch

Figure 23. Load vs Eccentricity Ratio of Tilting-Pad Journal Bearing at 20,000 RPM, MDBTF II-2, Liquid Mercury at 100°F

$D = 0.625$ inch $\mu = 1.90 \times 10^{-7}$ Reyns
 $L = 0.322$ inch
 $C_{Dp} = 0.0023$ inch $N = 30,000$ rpm
 $C_{MD} = 0.0041$ inch

Figure 24. Load vs Eccentricity Ratio of Tilting-Pad Journal Bearing at 30,000 RPM, MDBTF II-2, Liquid Mercury at 125°F

Figure 25. Load vs Eccentricity Ratio of Tilting-Pad Journal Bearing at 36,000 RPM, MDBTF II-2, Liquid Mercury at 150°F

Figure 26. Load vs Eccentricity Ratio of Tilting-Pad Journal Bearing at 10,000 RPM, MDBTF II-2, Liquid Mercury at 180°F

<u>Load Direction</u>	<u>Shaft Speed (rpm)</u>	<u>Load Capacity (lb)</u>
Down	20,000	10.8
	30,000	16.9
	36,000	20.3
	40,000	22.7
Up	20,000	6.1
	30,000	10.6
	36,000	12.8
	40,000	13.7

Although the load capacities in the two directions are compared on the basis of eccentricity ratio, higher eccentricity ratios are possible for loading upward because the shoes tilt to permit larger displacements as previously explained.

Compared on the basis of minimum film thickness or total load-carrying capacity, loading between the shoes will support a greater load than loading into a pivot. However, because of the larger displacement for a given load when loading between shoes, the stiffness in that direction is considerably less than when loading into a pivot.

The theoretical load capacities were calculated based on the model in Appendix C, and are shown as dashed curves in Figures 23, 24, 25, and 26. The experimental values greatly exceed the theoretically predicted values. This increase is the result of the highly turbulent conditions in the lubricant film.

After the test runs with the balanced shaft, the shaft was unbalanced to an equivalent of 20 pounds rotating load at 40,000 rpm, to 180 microinches of unbalance. The load-eccentricity ratio was determined at 40,000 rpm. This did not change from that observed with the balanced shaft. The only change was in the size of the orbit about the equilibrium position as reported in the MDBTF II-1 test series (see Paragraph B. 2. The radius of this orbit was 172 microinches. Compared to the 180-microinch unbalance, this indicates that operation was above the first-mode critical speed (see Paragraph C. 4.) Since the orbit is larger with the unbalanced shaft, smaller unidirectional loads are permissible to maintain adequate film thickness. A reduction of total unidirectional load capacity by about 5.0 pounds is the penalty for the rotating load.

2. Attitude-Eccentricity Locus

The attitude-eccentricity locus is shown in Figure 27. It can be seen that the locus is practically a vertical straight line with a slight deviation from this with upward loading. Since no explanation has been given for this deviation to date, and because an assumption of linear locus was made for the theoretical model, the horizontal displacements are neglected in the determination of the eccentricity ratio. The maximum error this assumption can introduce is $< 2\%$. This is well within the accuracy of the instrumentation. Then the definition of eccentricity ratio ϵ is

$$\epsilon = \frac{\text{vertical displacement}}{\text{radial pivot point clearance}} \quad . . . 9$$

Based on this definition it is possible to obtain eccentricity ratios > 1 when loading upward since the shaft can move outside the pivot point clearance circle as the pads tilt away (see Figure 27).

Figure 27 shows the extent of the shaft locus loading up and down or between two pivots or shoes and into a pivot or shoe, respectively. It follows that if zero eccentricity exists at the center of the circle made by the three pivot points, then the motion of the shaft away from the center point can be twice as great loading up as loading down for the indicated pivot point arrangement. Or, if loading down from center to just touching represents $0 < \epsilon < 1.0$, loading up will represent $0 < \epsilon < 2.0$. The tilting of the shoes permits shaft displacement within the triangle indicated in Figure 27. Attitude angle is relatively independent of speed, magnitude of rotating load, and flow-pressure characteristics.

3. Stability

The bearings were free from half-frequency whirl instability at all speeds to 40,000 rpm, under all conditions.

4. Fluid Film Stiffness and Critical Speed

The fluid film stiffness (K) is obtained from the plot of W versus ϵ , Figures 23 through 26, as follows:

$$K = \frac{dW}{d\epsilon} \cdot \frac{2}{C_{Dp}} \quad . . . 10$$

A rotating shaft will spin about its geometric axis as long as its rotational speed is less than the lowest natural frequency of the spring mass system. Above this speed the shaft will shift or invert and rotate about its center of gravity. Two major "springs" or stiffnesses involved are the shaft spring stiffness and the spring stiffness formed by the lubricant in the bearing. Since the shaft spring constant is much greater than the bearing spring constant in this case, the former contributes little to the lower critical speed and the spring mass system is then essentially composed of the shaft acting as a

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Figure 27. Attitude-Eccentricity Locus of Tilting-Pad Journal Bearing

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Figure 28. Rigid Shaft Critical Speed vs Bearing Stiffness, MDBTF II-2 Test Series, Loading Into Pivot

concentrated mass in conjunction with the spring constant of the bearing. The critical frequency of this system is then given by equation 2.

$$f_{cr} = \frac{60}{2\pi} \sqrt{\frac{K}{M}} \quad . . . 2$$

Where K = the bearing spring constant or fluid film stiffness

M = the mass associated with one bearing, which is 2.5 pounds

The bearing stiffness is shown in Figure 28 as a function of speed for constant eccentricity ratios and for constant loads. Superimposed on this is the rigid shaft critical speed curve, f_{cr} . It can be seen that under all conditions of loading and for all speeds evaluated the bearing stiffness is below the required stiffness for operation under the critical speed. Therefore, the shaft is always operating above the first critical speed. This was borne out experimentally by the 20 pound rotating load test where the distance between the geometric center and the center of mass is 1.8×10^{-4} inches, corresponding to the observed orbit radius of 1.72×10^{-4} inches.

It is difficult to establish whether the rotor system is always above or passes through this first-mode critical speed at a very low frequency for the following reasons:

- 1) It is difficult to obtain any load-deflection data at speeds below 20,000 rpm because the tilting-pad bearings are purely hydrodynamic.
- 2) Extrapolating existing data, as from Figure 28, can result in considerable error and sufficient doubt about the cross-over point.
- 3) The absence of an increase in amplitude because of large damping in the bearings prevents concise identification at low speeds.

However, even if cross-over does occur (or a resonant frequency is passed), it is of little consequence, since it has no undesirable effect and constitutes no problems. In fact, operation above the critical has the advantage of limiting the increase in the synchronous whirl component due to unbalance.

Figure 28 is a significant plot since it shows a line of limited operation where $\epsilon \rightarrow 1.0$. As long as load and speed combinations do not fall to the left of the line $\epsilon \rightarrow 1.0$, operation should be successful. If the applied loads exceed the limiting line at a particular speed, the bearings are overloaded and will fail. The region where a possible first-mode critical speed will be encountered for the correct combination of load and speed is also shown. Observe that the region is very small and does not extend above 15,000 rpm for any possible operating point.

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5. Flow-Pressure Requirements

To assure lubricant at the inlet to each shoe, flooded operation of the tilting-pad bearing was specified. Screw seals (viscosity pumps) pumping inward against the direction of side flow were selected to assure restrictions at the outlet and therefore to accomplish the desired flooded condition.

The screw seals in the MDBTF II-1 test did not exhibit very good sealing properties. With the mercury flow maintained at 5 lb/min the supply pressure increased from about 5 psi to approximately 25 psi as the shaft speed was increased from zero to about 30,000 rpm. At speeds above 30,000 rpm, the supply pressure rapidly dropped to near zero at 40,000 rpm. It was necessary then to increase the flow rate to about 8 lb/min to maintain pressures above 5 psig. This peculiar pressure performance with speeds above 30,000 rpm was attributed to the centrifugal force field throwing the mercury from the threads, thus rendering them rather ineffective as seals or a screw pump. To eliminate this problem in the MDBTF II-2 and to obtain better sealing, the pumping threads were machined into the housing (see Figure 22).

The flow characteristics of this improved screw seal with zero bearing load and with shaft weight are shown in Figure 29 for the turbine end, and in Figure 30 for the alternator end bearings. The difference in flow characteristics between the turbine and alternator end bearings is assumed to be the result of different seal clearances, although both were specified as between 0.007 and 0.008 inch.

Considering the theory of viscosity pumps, of which this screw seal is one, the variation in clearance need only be a few ten-thousandths of an inch to have a large effect. The supply pressure required for a flow of 6 lb/min at 40,000 rpm was 26 psig for the turbine end bearings and 34 psig for the alternator end bearings. This should insure a flooded bearing at the operating conditions, thereby giving safe hydrodynamic operation.

Increasing bearing load in either direction tends to increase the pressure drop across the seal. This effect is shown in Figures 31, 32 and 33 for the speeds tested. As the bearing load is increased from zero to 20 pounds at 40,000 rpm, the increase in required supply pressure is about 7 psi to keep the flow constant at either of three flow rates, i. e., 4, 6, and 8 lb/min.

This seal configuration is considered far from optimum for the length. Varying depth of threads, pitch, and clearance can result in increased seal effectiveness if this becomes necessary.

6. Power Loss

Bearing power loss was measured by the method of speed decay. According to this method the power loss is given by

$$H = 1.49 \times 10^{-2} I \left(\frac{\Delta N}{\Delta t} \cdot N \right) \text{ watts} \quad \dots 11$$

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Figure 29. Tilting-Pad Bearing Journal Bearing Flow Characteristics, MDBTF II-2, Turbine End

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Figure 30. Tilting-Pad Journal Bearing Flow Characteristics, MDBTF II-2, Alternator End

Figure 31. Load vs Supply Pressure at a Constant Flow Rate of 4 lb/min - Turbine End, MDBTF II-2

Figure 32. Load vs Supply Pressure at a Constant Flow Rate of 6 lb/min - Turbine End, MDBTF II-2

Figure 33. Load vs Supply Pressure at a Constant Flow Rate of 8 lb/min - Turbine End, MDBTF II-2

Where: I = moment of inertia of the shaft in ft-lb-sec²

N = shaft speed in rpm

$\Delta N/\Delta T$ = rate of speed decay at the point in question with
the drive power turned off

The moment of inertia was measured experimentally. The rate of speed decay was taken from the continuous plot of speed versus time. Tilting-pad bearing power loss measurements are given in Figure 34 after correction for rotor windage loss and thrust bearing power loss. For the present, the bearing power loss will not be separated from the seal power loss since they must be used together.

The power loss of one bearing with the necessary seals is 305 watts at 40,000 rpm. The power loss of a plain bearing of the same diameter, length, and clearance of 0.002 inch is 180 watts. The difference can most likely be attributed to the seals and the higher turbulence of the larger-clearance tilting pad. Based on screw seal tests of similar size seals, power loss attributed to the double seal (on both sides of the bearing) is 80 watts. This would indicate an increase of 25 watts due to larger clearance. Since laminar theory predicts lower power loss with increasing clearance, this power loss increase must be due to increasing turbulence.

7. Bearing Materials and Wear

For the MDBTF II-2 and II-2A tests the selection of bearing materials was as follows

Shoes	- Bluechip 18-4-1 Tool Steel
Sleeves	- Bluechip 18-4-1 Tool Steel
Balls (Pivot)	- AISI 52100 for MDBTF II-2, 440C for MDBTF II-2A

440-C (stainless steel) pivot balls were to be used, but they were not available until the unbalanced rotor test, MDBTF II-2A. The performance of these materials can be seen from the photographs that were taken after the MDBTF II-2 test, which ran 28 hours and 31 minutes at 20,000 to 46,000 rpm. Figures 35 and 36 are photographs of the turbine and alternator journals, respectively. Beyond the slight scratches which are attributed to stops and starts, no wear is evident, and the journals are in excellent condition. The backside of the pads with the pivot balls is shown in Figure 37. The pivot balls are shown separately in Figure 38. All of the balls except 1A were found to be stuck in the pad. The center of the ball was 0.0035 inch inside the pad, which may explain the tendency to stick. It appears that the conical seat is more susceptible to sticking than the hemispherical seat. Ball 1A was stuck in the housing. The wear marks on the balls in the direction of shaft rotation are the result of pad tilt.

Figure 34. Tilting-Pad Journal Bearing Power Loss as a Function of Speed

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Figure 35. Journal, Turbine End, After MDBTF II-2 Tests

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Figure 36. Journal, Alternator End, After MDBTF II-2 Test

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2

3

Figure 37. Reverse Side of Pad with Pivot Balls After MDBTF II-2 Tests
(A - Alternator End, B-Turbine End)

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Figure 38. Pivot Balls After MDBTF II-2A Test

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The balls were macroscopically examined at 20 magnification. Two observations were made:

- 1) A corrosive attack was evident.
- 2) Wear bands were present on the ball surface.

The positive indication that the wear band resulted after the corrosion attack is significant. It suggests that a corrosive attack may have occurred prior to operation, and probably was oxidation rather than mercury corrosion. Metallographic examination substantiates this postulate of oxidation corrosion. Pitting and oxidation layers were present in the photomicrograph. These photographs and the report are attached as Appendix E in Volume I. No evidence of material removal by wear was noted. Therefore, the oxidation which occurred prior to operation obscured any existing wear damage other than what is evident in the photographs. Oxidation may have occurred as a result of the selection of 52100 steel balls and because the unit was left standing without cover gas for a two-week vacation period after initial operation. Previous experience indicates that rusting does occur on surfaces exposed to mercury and air for extended periods.

For test MDBTF II-2A, the balls were replaced by 440-C balls hardened to Rc55-60. The shaft assembly was unbalanced by 180 microinches to represent a rotating load of 20 lb/bearing at 40,000 rpm. During the cleaning and prior to reassembly, the shaft was submerged in an Oakite 33 cleaning solution. For some unknown reason the constituents of the solution etched the Bluechip (18-4-1 tool steel) journals. Figures 39 and 40 show the turbine and alternator end journals prior to the start of test MDBTF II-2A. A surface check revealed no general loss in material although some crystals at the surface certainly were removed. The entire surface appeared to have been sand blasted. Since an estimate indicated that as much as 0.0004 inch had to be removed to clean up all the marks, a decision was made to proceed with the test with the journal condition shown in Figures 39 and 40, thereby retaining the original diametral clearance.

Upon completion of test MDBTF II-2A, which lasted some 10 hours and was primarily a check of the parametric data taken with a balanced shaft, the parts were again inspected. No noteworthy change was observed on the bearing balls, pivot seat, shoes or journals; therefore, no pictures were taken. The absence of fretting-corrosion damage on the balls and in the pivot seats was particularly significant. No rust was found on the balls this time, which strengthened the conclusions that the previous corrosion damage was the result of the material and the unit standing unattended for several days with mercury in it and without cover gas. The surface condition of the journals seemed to have no effect on the bearing performance.

Figure 39. Journal, Turbine End, Prior to Start of MDBTF II-2A Test

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Figure 40. Journal, Alternator End, Prior to Start of MDBTF II-2A Test

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$D = 0.625$ inch
 $L = 0.322$ inch
 $C_{Dp} = 0.0023$ inch
 $C_{MD} = 0.0041$ inch
 $X/B = 0.425$ inch

Figure 41. Theoretical Geometric Parameters of Tilting-Pad Journal Bearing, Loading into Pivot Point (MDBTF II-2)

8. Correlation with Analysis

The analytical procedure of establishing turbulent load factors based on actual performance compared to careful laminar predictions was expanded by using the data generated in this test series. The theoretical bearing load capacities were calculated based on the model in Appendix C (Volume I); these are shown as broken curved lines in Figures 23, 24, 25, and 26. The experimental values greatly exceed the theoretically predicted values, the increase being the result of the highly turbulent conditions in the lubricant film. Here a turbulent factor is defined as before in equation 3

$$K = W_T/W_L \quad \dots 3$$

where W_T = experimental load capacity

W_L = theoretically predicted laminar load capacity

Closer investigation of the figures showed that K is a function of speed, eccentricity ratio, and viscosity. If eccentricity ratio is expressed by a film thickness, the dimensionless grouping of these variables is the Reynolds number. To show the effect of eccentricity ratio on the various film thickness of the pads, the theoretically obtained values are plotted in Figures 41 and 42 for loading down and up, respectively. The leading edge film thickness h_1 , pivot point film thickness h_p , and trailing edge film thickness h_2 are all decreasing with an increase in eccentricity ratio if the loaded pad is considered. The rate of decrease, however, is quite different.

Because of the inability to determine which film thickness governs turbulence in the lubricant film, an attempt was made to correlate K with Re using all three film thicknesses. The best correlation was obtained with the Reynolds number based on the trailing edge film thickness where

$$Re_{h_2} = \frac{\rho U h_2}{\mu} \quad \dots 12$$

This correlation is shown in Figure 43. Points taken from MDBTF II-1 when only the vertical displacement is considered are also plotted on this curve. The theoretical values were computed for lube temperatures of 100°F, 125°F and 185°F which corresponded to the experimental values of 20,000; 30,000; and 40,000 rpm, respectively. Since these points also fall on the curve of Figure 43 there is increased confidence in the value of this correlation. The turbulence factor is then given by the equation of the curve

$$K = 0.545 Re_{h_2,1}^{0.055} \quad \dots 13$$

$D = 0.625$ inch
 $L = 0.322$ inch
 $C_{Dp} = 0.0023$ inch
 $C_{MD} = 0.0041$ inch
 $X/B = 0.425$ inch

Figure 42. Theoretical Geometric Parameters of Tilting-Pad Journal Bearing, Loading Away From Pivot Point (MDBTF II-2)

Figure 43. Tilting-Pad Journal Bearing Turbulent Load Factor as a Function of Reynolds Number

where $h_{2,1}$ is the trailing edge film thickness of the loaded pad, i. e., of pad 1 for loading down and of pad 2 for loading up. Examination of Figure 43 shows that K has a value of 1 when the Reynolds number is about 200. This is below the usually expected critical value of Reynolds number. This is probably because once turbulence is established at the leading edge of the pad (where the film thickness is larger) it is difficult to reestablish laminar flow further downstream because of the high density and inertia of the fluid. This results in a considerable overshoot before laminar flow can be reestablished.

The trailing edge film thickness is actually a good variable to use in predicting bearing performance because load capacity is largely dependent on the conditions around the pivot point. The trailing edge film thickness is the conservative approach for values of $\epsilon < 0.5$, since it is slightly greater than the pivot point film thickness (see Figure 41). In any case, even for $\epsilon > 0.5$, when the pivot point film thickness is slightly less than the trailing edge film thickness, the Reynolds number based on the trailing edge will still be the governing parameter.

D. MDBTF II-2B and -2C Test Series

The objectives of the MDBTF II-2B and -2C tests were:

- 1) To obtain 200 hours of endurance operation at design specification but with ambient temperature mercury
- 2) To subject the tilting-pad bearings to a vibration test on a non-rotating rotor assembly corresponding to system requirements

These tests were expected to give a better understanding of pivot point wear or fretting under simulated space loads.

1. Endurance Test (MDBTF II-2B)

The 200 hour endurance test was conducted on the reassembled MDBTF II-2A unit. Test conditions were as follows:

Speed	40,000 rpm
Rotating load	180 microinches unbalance or 20 lb/bearing at 40,000 rpm
Unidirectional load	2.5 lb/bearing (shaft weight)
Flow rate	7 lb/min alternator bearing 10 lb/min turbine bearing 6.5 lb/min thrust bearing

The dimensions of the test bearing, screw seals and bearing materials are presented below:

Bearing Dimensions

Diameter	0.625 inch
Length	0.322 inch
Pivot point diametral clearance	0.0023 inch
Machined-in diametral clearance	0.0041 inch
Pad circumferential length	0.535 inch
Pivot point position	0.425 of pad length from trailing edge
Pivot balls diameter	3/32 inch
Ball seat in housing	cylindrical, 0.047 inch radius
Ball seat in pad	60° cone

Screw Seal Dimensions

Length	0.187 inch
Screw	60° V-Type - 5/8-32-0.027 inch deep
Clearance between shaft and thread	0.007 + 0.008 inch

Bearing Materials

Balls	440 C Stainless
Pads	18-4-1 Tool Steel
Sleeves	BG 41

Test Results

The results of the endurance run are best shown by the inspection photographs of pertinent parts before and after the test, and the bearing performance during the test.

The variation of bearing lubricant flow, supply pressure, drain pressure, and lubricant drain temperature during the run is shown in Figures 44 and 45. On the whole, all of the parameters remained constant and fluctuations were less than 10%, except in the drain pressure. After 40 hours, the drain pressure dropped from 4 to 1.6 psig. This, however, was the direct result of a change in seal cover gas pressure brought on by better sealing requirements for the capacitance proximity probes.

The shaft orbit traces, as displayed on the cathode ray oscilloscope, were periodically photographed. Examples are shown in Figures 46 and 47. The size of the orbit is the result of the 180 micro-inch unbalance. The difference between the vertical and horizontal dimensions of the orbit is the result of combined unidirectional and rotating loads. The shift of the trace from the left side of the screen to the right side after forty hours of operation occurred when the cover gas pressure was changed to obtain better sealing. Only the horizontal probe was affected, the vertical probe is usually not affected by fine mercury spray because it is higher up. Figures 46 and 47 show that the size of the shaft orbit in the bearing clearance space and the stability of this orbit did not materially change during the span of the endurance run. Although some change can be observed in the vertical position of the shaft (measured by orbit center), which is attributed to instrumentation drift, it can be taken as relatively constant and unaffected during the 200 hours of endurance. This indicates that film thickness also remained relatively constant, so that bearing performance was not noticeably affected.

Photographs of pertinent bearing parts taken before and after the endurance run are shown in Figures 48 to 59. The journals were the same ones used in the -2A test. Their condition was discussed in Paragraph C and is shown in Figures 39 and 40. The roughened condition on the journals may have contributed to or accelerated the erosion damage seen at the conclusion of the 200-hour endurance test.

Upon completion of the 200 hour test run the unit was disassembled and the bearings photographed. Figures 48 and 49 show the appearance of the journals. Cavitation-erosion type marks can be seen in two rings around both the turbine and alternator journals. These rings occurred directly under the axial edges of the pads and the inlet to the screw seals. Figure 50 shows a right-end view and Figure 51 a left-end view of the pads. A view of the bearing surfaces of the pads is shown in Figure 52. These photographs show that the bearing surfaces are free of wear marks. Cavitation-erosion damage can be observed on the corners of the leading edges of pads B-1, B-2, and A-3. This type of damage is attributed to the flow pattern in the flooded bearing cavity. An analysis of this flow pattern indicates that a large amount of mercury is recirculated, resulting in low pressure and highly turbulent mercury by the pad inlets. This is further churned around by the leading screw in the screw seal. This combination then gives localized low pressure turbulent regions where cavitation and resulting damage occur. Damage is worst on the journals since the inertia forces tend to lower the pressure on the journal surface even further. This cause of cavitation damage can be reduced or completely eliminated by redesigning the mercury inlet and flow paths so that mercury is under high pressure everywhere.

Figure 41 Bearing Flow and Supply Pressure During Endurance Run (MDBTF II-2B)

Figure 45. Bearing Drain Pressure and Temperature During Endurance Run (MDBTF II-2B)

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0 HOURS

21 HOURS
29 HOURS

48 HOURS

65 HOURS
65 HOURS

Figure 46. Traces of Shaft Position During Endurance Run at Various Times (0 to 65 Hours)

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60 HOURS
100 HOURS

112 HOURS
149 HOURS

157 HOURS
157 HOURS

181 HOURS
181 HOURS

Figure 47. Traces of Shaft Position During Endurance Run at Various Times (60 to 181 Hours)

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Figure 48. Journal, Turbine End, After MDBTF II-2B Endurance Test

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Figure 49. Journal, Alternator End, After MDBTF II-2B Endurance Test

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A-2

A-3

A-1

B-1

B-2

B-3

Figure 50. Right End View of Pads After MDBTF II-2B Endurance Test
(A - Alternator End, B - Turbine End)

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A-1

A-2

A-3

B-1

B-2

B-3

Figure 51. Left End View of Pads After MDBTF II-2B Endurance Test
(A - Alternator End, B - Turbine End)

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A-1

A-2

A-3

B-1

B-2

B-3

Figure 52. Bearing Surface of Pads After MDBTF II-2B Endurance Test
(A - Alternator End, B - Turbine End)

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A

B

Figure 53. Pivot Balls After MDBTF II-2B Endurance Test
(A - Alternator End, B - Turbine End)

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Figure 54. Cavitation-Erosion Type Damage to Screw Seal After
MDBTF II-2B Endurance Run

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Figure 55. Instrumented Vibration Test Package (MDBTF II-2C)
Clamped to Shaker Table

The pivot balls are shown in Figure 53 as they appeared after the endurance test. Slight wear marks and traces of oxidation can be observed on them. It should be noted that the seats had products of corrosion on them from previous tests. No major damage or dimension change, however, could be observed. Figure 54 shows the cavitation-erosion type damage at the inlet to the screw seal. For a more detailed explanation of these phenomena, Appendix F (Volume I) presents the report of the materials analysis performed on the parts after the endurance and shake tests.

2. Vibration Test (MDBTF II-2C)

Upon completion of the endurance run, the pivot balls were replaced and the test fixture (now called MDBTF II-2C) was subjected to the vibration test. The housing support was modified to fit the MB C-10 shaker. A control accelerometer was located on top of the housing near the plane containing the center of gravity. A monitor accelerometer was located on top of each bearing housing. Another monitor was located near the center of the mounting plates. The mounting stud of this accelerometer was threaded into a mounting bolt head, thus monitoring g-level at the top of the shaker table. The instrumented package is shown clamped to the shaker table in Figure 55.

The orientation of the shake axes is shown below.

The vibration and frequency levels are intended to be as follows in all three planes.

0.25g	5 to 20 cps
5.0g	20 to 400 cps
7.5g	400 to 1500 cps
15g	1500 to 3000 cps

The test was started with a vibration level of $1/2g$, a five-minute sweep from 20 to 3000 cps, to determine the maximum permissible loading without overloading the shaker.

A 25-minute sweep was then performed consisting of 0.25g from 5 to 20 cps, 5g from 20 to 400 cps, and 7.5g from 400 to 2800 cps. This was followed by a sweep of 12g from 1500 to 2800 cps. Between successive frequency ranges the g level was adjusted to the next higher level.

After the 25-minute sweep the bearing monitors were relocated to indicate cross-axis acceleration (y and z) and a 5-minute, 20 to 2000 cps sweep made at 1-g input. Based on a combined specimen plus the moving element weight of 63 plus 17, or 80 lb, the test plan was limited to an amplitude of 12g and to a frequency of 2,800 cps. The results of the sweeps were as follows:

Sweep 5 - 2800 cps

800 cps: input - 7.5g
 resonance alternator bearing (A) - peak at 20g

810 cps: input - 7.5g
 resonance turbine bearing (B) - peak at 16g

930-970 cps: two modes of resonances are apparent at both bearings
 input - 7.5g
 resonance levels - 12 to 19g

1760 cps: input - 7.5g
 resonance at alternator bearing - 45g
 input - 12g
 resonance at alternator bearing - 80g

1770 cps: input - 7.5g
 resonance at turbine bearing - 24g
 input - 12g
 resonance at turbine bearing - 42.5g

Cross Accelerations: Input at 1.0g

z (longitudinal axis) = axis

900 cps = 1.0g

1100 cps = 1.5g

1950 cps = 3.0g

2400 cps = 2.0g

y (lateral) = axis

1020 cps = 1.2g

2400 cps = 0.7g

Shake table monitor plots are shown in Figures 56 through 60. For a discussion of the effect of the shake test on the bearing parts, see Appendix F (Volume I).

The completion of MDBTF II-2C represented the end of the test phase of the 0.322-inch long (axial) tilting-pad bearing. As indicated in the initial analysis, the load capacity of this "short," purely hydrodynamic bearing, although adequate from the requirement point of view, was considerably less than the capacity of hybrid bearings discussed later. To increase load capacity and, thereby, provide a wider margin of operation, the initial analysis included a longer tilting-pad bearing with a $B/L \approx 0.75$, or axial length (L) of 0.407 inch. The major conclusions from the 0.322-inch bearing test series are presented before summarizing the tests conducted on the "longer" tilting-pad bearing.

The results of the MDBTF I and MDBTF II test series on the 0.625-inch diameter, 0.322-inch long tilting-pad bearing with a pivot point diametral clearance from 0.0016 to 0.003 inch confirmed many of the design objectives and the performance trends predicted by theory. These test results made the following conclusions possible:

- 1) Freedom from half-frequency whirl instability is possible at all speeds up to 40,000 rpm at all load (or zero g) tested.
- 2) Operation is generally above the first mode critical speed and below the second for a rotor system similar to the bearing fixture rotor (which does not have an overhung or cantilevered mass), or above the first and second mode with a heavy overhung mass.
- 3) The rate of increase of bearing stiffness with speed results in the shaft passing through the first mode critical speed (inversion speed) below 15,000 rpm, depending on selected clearance and pre-load.

CRU-V VIBRATION TEST SPECIFICATION X (VERTICAL) AXIS

Figure 56. CRU-V Vibration Test Specification X (Vertical) Axis

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Figure 57. MDBTF II-2C Resonance Search (C-10 Table Monitor 1/2g Input)

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Figure 58. MDBTF II-2C Vibration Test (C-10 Table Monitor Recorded,
 0.25 D A 5 to 20 cps, 7.56g Control, 400 to 2800 cps,
 Run No. 2, 10-8-62)

[illegible]

Figure 60. MDBTF II-2C Vibration Test (Cross Axis)
 (Control Accel Recorded, 1g Input, Run No. 4, 10-8-62)

- 4) The ultimate load capacity of the bearing exceeds 20 lb unidirectional at 40,000 rpm with a well balanced shaft. With a rotating load equivalent to 20 lb/bearing at 40,000 rpm, the maximum unidirectional load is reduced to 15 lb/bearing.
- 5) Unbalancing the mass center from the geometric center by 180 micro-inches, or equivalent to a rotating load of 20 lb/bearing at 40,000 rpm, does not affect the load-eccentricity relationship. The magnitude of the shaft orbit remains constant with speed for a given unbalance since operation is above first mode critical. As the unbalance is increased the initial amplitude or synchronous whirl component increases to correspond to the distance between geometric and gravity centers.
- 6) Varying the load direction up and down results in only small changes in attitude angle, which are generally $< 10^\circ$.
- 7) Random instabilities, mainly low frequency vibrations, occur when the preload is inadequate to maintain a convergent-divergent wedge at the shoe's leading edge, when flow starvation occurs, or when the friction at the pivot exceeds the generated hydrodynamic pressure, tending to pivot the shoe and prevent it from tilting.
- 8) The screw seal located on the rotating member (the shaft) is inadequate to keep the bearing flooded over its required speed and flow range.
- 9) Placing the screw in the housing improves the screw performance and its effectiveness in keeping the bearing flooded. The required supply pressure for a flow rate of 6 lb/min at 40,000 rpm is 26 to 34 psig.
- 10) Lubricant flow rate between 4 and 8 lb/min does not affect load capacity.
- 11) The power loss of one bearing with necessary screw seals is 305 watts at 40,000 rpm.
- 12) The pivots (hardened balls) and properly contoured seats do not suffer Brinnelling, fretting, or wear under conditions of rotating load. The AISI 52100 steel pivot balls showed no signs of fretting, but they were badly corroded due to contact with water, oxygen, and mercury. Changing to 440C stainless steel hardened to Rc 56-60 appeared to eliminate corrosion.
- 13) The bearing operated satisfactorily during the 200 hours of endurance test with ambient mercury.
- 14) No significant change was observed in the bearing clearance.

- 15) Fretting wear of the balls was negligible.
- 16) The shaft suffered cavitation-erosion damage directly under the leading screws of the two screw seals. The corners of the pad leading edges also showed signs of erosion but to a much smaller degree. This damage is attributed to the flow path inside the bearing cavity, the analysis of which indicates large amounts of mercury recirculation resulting in a low-pressure turbulent region along the pad leading edges. Combination of this with the highly turbulent flow at the screw seal inlet results in pockets of severe turbulence accompanied by low pressure at the leading edge corners.
- 17) The cavitating region at the pad leading edge corners also results in a slight pad flutter which in turn results in a slight shaft flutter. This gives a blurred shaft trace on the oscilloscope.
- 18) Resonances were found during non-operating vibration tests between the frequencies of 800 and 1770 cps. Amplification factors were 1.5 to 8.
- 19) No significant damage to the bearing parts was observed due to the vibration test.
- 20) The turbulent load factor is a function of the Reynolds number. At the pad's trailing edge it is expressed by equation 13.

$$K = \frac{0.545}{0.055 \text{ Re}_{h_2, 1}} \quad \dots 13$$

From the analysis performed and the experimental data available, it appears that the conditions at the leading edge of the pad determine the onset of turbulence in the lubricant film. Once the lubricant is in turbulence, laminar conditions may not be reestablished, resulting in a low value of critical Reynolds number at the trailing edge. The critical value is

$$(\text{Re}_{h_2})_{cr} \approx 200$$

The preceding test series demonstrated the feasibility of the tilting-pad bearing for application to a combined rotating unit. Its inherent stability, misalignment capability, and low critical inversion speed are important advantages. Two areas required further development to permit this type of bearing to meet the life objective of the SNAP 2 system. These were:

- 1) Erosion due to the screw seals

- 2) Pivot configuration which resulted in zero or negligible change during the required life cycle

To evaluate these areas further and at the same time provide a greater operating margin in load carrying capacity, the next series of tests was conducted.

E. MDBTF III-1 and -2 Test Series

The object of this third test series was to test a bearing with improved load capacity, which was marginal with the previous shorter bearing, and at the same time to obtain the influence of length-to-diameter ratio on the turbulent load factor established in the two previous test series. Consequently, a longer bearing was used. The comparison between predicted and actual performance was then expected to provide information on the generated analytical techniques. A longer range objective of the program was to obtain analytical and empirical design procedures for turbulent film fixed and self-aligning hydrodynamic bearings that could be applied to various bearing diameters, lengths, clearances, and preloads. This test series represents one of the necessary experimental steps in accomplishing that objective.

Another important feature in this third series of tests was the inclusion of six different pivot point configurations. In a short-term parametric test of this type, these different configurations do not influence performance. Therefore, the performance of the various pivot configurations will be evaluated during a long-term endurance run with hot mercury. Then the effect of fretting, wear, and Brinnelling will be established so that the best configuration can be selected for incorporation into the final design. A slightly modified screw seal configuration was also evaluated for performance. Influence of erosion on the long-term performance of the seal was investigated in the subsequent endurance run.

The following sections report the results of this third series of tests. To investigate the effect of clearance on bearing performance and to include more variables in the turbulent load factor correlation, the test was divided into two parts. In the first part of the test the bearing had a pivot point diametral clearance of 0.0015 inch, while in the second part of the test it was 0.0025 inch. In the second part of the test, the critical speed was investigated by using a dummy NaK pump rotor mounted on the shaft.

1. Analytical Prediction

Prior to the test, the predicted performance of the longer (0.427-inch) tilting-pad bearing was established. The prediction was based on the laminar analysis reported in Appendix C (Volume I) and the turbulent load factor established during the MDBTF II-2 test series (Section C). A summary of the analytical results are presented below for

Figure 61. Predicted Load Capacity of 0.427-Inch Long Tilting-Pad Journal Bearing

the bearing with the following dimensions:

D - diameter	5/8 inch
L - length	0.427 inch
C _{Dp} - diametral clearance	0.0015 to 0.0018 inch
C _{MD} - machined-in clearance	0.0034 to 0.0035 inch
B - pad circumferential length	0.535 inch

To obtain the average performance of this bearing, a C_{Dp} of 0.0016 inch and a C_{MD} of 0.0035 inch were chosen for the clearances.

The load capacities, laminar and turbulent, in contrast to eccentricity ratio are shown in Figure 61. The turbulent load factors are also shown in Figure 61. The load capacity of this bearing at 40,000 rpm with a conservative eccentricity ratio of 0.6 is 44.0 lb.

Bearing Critical Speed

The fluid film stiffness of the 0.427-inch long bearing is obtained from Figure 61 where

$$K = \frac{\Delta W}{\Delta \epsilon} \frac{2}{C_D} \text{ lb/in.} \quad \dots 14$$

This stiffness is plotted versus shaft speed in Figure 62 for constant eccentricity ratios and for constant loads. Superimposed on this plot is the stiff shaft critical which is defined by equation 2

$$f_{cr} = \frac{60}{2\pi} \sqrt{\frac{K}{M}} \quad \dots 2$$

where M is the mass associated with one bearing in/lb. In this case M = 2.5 lb. Figure 62 indicates that operation of the 0.427-inch tilting-pad journal bearing will be at times below and at times above the inversion point critical speed, depending on the bearing load and eccentricity. The bearing fluid film stiffness with only shaft weight applied to the bearings is shown separately in Figure 63. Then the inversion point critical speed will occur at 12,500 rpm. Operation, however, will be far below the second mode without an overhung weight (such as NaK pump rotor).

Power Loss

Since no turbulent power loss factor had been obtained so far, an estimate of the longer bearing power loss is as follows:

Figure 62. Predicted Fluid Film Stiffness vs Shaft Speed for 0.427-Inch Long Tilting-Pad Journal Bearing

Figure 63. Predicted Fluid Film Stiffness vs Shaft Speed for 0.427-Inch Long Tilting-Pad Bearing With Shaft Weight Only

Figure 64. Tilting-Pad Test Bearing MDBTF III Test Series

$$H_{0.427} = H_{0.322} \exp x \frac{0.427}{0.322} \quad . . . 15$$

$$= 170 \text{ watts} \times \frac{0.427}{0.322} \quad (\text{at } 40,000 \text{ rpm})$$

$$= 225 \text{ watts plus screw seals}$$

This assumption is based on the laminar relationship that power loss (H) is directly proportional to the axial length of the bearing (L), i.e., $H = f(L)$.

Since Reynolds number, and consequently turbulence, is independent of bearing length, the above assumption is reasonable. It may be somewhat low since the end leakage (axial flow) for the longer bearing is less and this will have an effect on power loss.

Power loss for the combination of bearing and screw seals is

$$\begin{aligned} H_{(\text{brg} + \text{SS})} &= 225 + 2(60) \\ &= 345 \text{ watts/1 brg and 2 screw seals} \end{aligned}$$

2. Test Results

The test bearing is shown in Figure 64. Actual dimensions of the bearing are as follows:

$$D = 0.622 \text{ inch}$$

$$L = 0.407 \text{ inch}$$

$$C_D = \left. \begin{array}{l} 0.0015 \text{ inch for MDBTF III-1} \\ 0.0025 \text{ inch for MDBTF III-2} \end{array} \right\} \text{Preload} \left\{ \begin{array}{l} 0.57 \\ 0.286 \end{array} \right.$$

$$C_{MD} = 0.0035 \text{ inch}$$

$$B = 0.535 \text{ inch}$$

$$\bar{X}/B = 0.575$$

Lubricant is fed into two annular grooves at the ends of the pads. The screw seal configuration is as follows:

$$L = 0.187 \text{ inch}$$

$$D = 0.622 \text{ inch}$$

Figure 65. Loading Up vs Eccentricity Ratio of MDBTF III-2 Tilting-Pad Bearing

Clearance = 0.007 to 0.008 inch (Diametral)

Thread included angle = $33^{\circ}53'$

Lead angle = 10°

Figure 64 shows the three different pivot point configurations. The two bearings have different size balls, one being $3/32$ inch in diameter and the other $5/32$ inch. This way six different pivot points can be evaluated. The pivot configurations were evaluated during the test series reported in the next section.

This series of tests was conducted with ambient temperature mercury inlet to the bearings. Load-displacement measurements were made at speeds of 10,000; 20,000; 30,000; 36,000; and 40,000 rpm with a completely balanced shaft. Flow-pressure characteristics were also determined at the same speeds. A critical speed search was made from about 5 to 45 thousand rpm. Power loss was obtained by the method mentioned above. Then an overhung weight simulating the NaK pump was put on the shaft and the critical speed search repeated.

At the conclusion of the test with the 0.0015-inch clearance bearing, it was found that the thrust bearing spacer was highly eccentric and acted as an additional journal bearing, making it almost impossible to interpret the data. This condition was corrected, but only the 0.0025-inch clearance test could be continued. Good data were obtained in the second part. Consequently, the discussion refers only to the test with the 0.0025 inch clearance.

Load Capacity

Figures 65 and 66 show the load-deflection characteristics if this bearing for shaft speed of 10,000; 20,000; 30,000; 36,000; and 40,000 rpm. As before in the MDBTF II-2 series, the limiting eccentricity ratio (ϵ) loading down into a pivot point is 1.0, while loading up between pivot points is 2.0.

The load capacity of this bearing, based on the 0.0002-inch pivot point film thickness, for loading downward only, is tabulated below:

<u>Speed</u>	<u>Actual Load Capacity (for $C_{Dp} = 0.0023$)</u>	<u>Predicted Load Capacity (based on $C_{Dp} = 0.0016$ and $L = 0.427$ in.)</u>
10,000	10 lb	7.5 lb
20,000	23.5 lb	22.0 lb
30,000	33 lb	40.0 lb
36,000	42 lb	52.0 lb
40,000	52 lb	60.0 lb

Figure 66. Loading Down vs Eccentricity Ratio of
MDBTF III-2 Tilting-Pad Bearing

The theoretical laminar load curves are shown by broken lines in Figures 65 and 66. These were calculated based on the theoretical model reported in Appendix C (Volume I). As can be seen, the experimental values are much higher than the laminar predictions. This increase is attributed to the turbulence in the lubricant.

Direct comparison between predicted turbulent and actual turbulent is not possible because of variation in pivot point clearance. A new set of predicted curves is necessary for the clearance of 0.0023 inch. The theoretical model was based on the vectorial summation of the forces exerted by all the pads.

Attitude Angle

Since the pads tilt until the resultant force is through the pivot point and normal to the shaft, the resultant shaft locus is always in line with the load line. The attitude-eccentricity locus of the test bearing is shown in Figure 67. It can be seen that it is very near the theoretically predicted line. This is remarkably good agreement when all of the possible experimental errors are considered. Turbulence in the fluid film does not alter this locus. This type of locus leads to the conclusion that this bearing will be free from half-frequency whirl since no tangential forces are generated that would sustain such whirling motion. No half-frequency whirl was encountered experimentally.

Fluid Film Stiffness and Critical Speed

The slope of the load-deflection curves at any point represents the fluid film stiffness. For this bearing, this is also the radial stiffness since the shaft displacement is in the direction of the applied load. The slope, or stiffness, was taken from Figures 65 and 66 for the point of shaft weight acting on the bearings and plotted as a function of speed in Figure 68. Values from this plot can be used to evaluate shaft-bearings system critical speeds. The predicted stiffness relationship with speed based on the turbulent analysis for a pivot point clearance of 0.0016 inch is also shown in Figure 68. As expected, the softer 0.0025-inch clearance bearing tested has considerably lower stiffness values. As reported previously, the 0.407-inch long bearing tested here operated above the first mode critical speed at all times. This conclusion is substantiated graphically in Figure 68 where shaft speed is plotted versus bearing stiffness with shaft weight acting on the bearings. Superimposed on this is the shaft critical speed as a function of bearing stiffness obtained with a computer program. The required bearing stiffness for encountering the first mode critical is always greater than that of the bearing. Consequently, above 10,000 rpm, operation is above the critical speed. No amplitude change could be detected at speeds below 10,000 rpm, making it impossible to pinpoint the critical speed. Additional evidence that operation was above the first mode critical was the relatively constant shaft displacement as a function of speed. Phase angle changes, however, could not be determined because of interference of runout due to machining.

With the dummy NaK pump mounted on the shaft, somewhat different results were obtained. Shaft deflection and phase angle measurements for this run are shown in Figure 69. It can be seen that the displacement amplitude and the phase angle

$D = 0.622$ inch

$L = 0.407$ inch

$C_{DP} = 0.0025$ inch

$C_{MD} = 0.0035$ inch

$B = 0.535$ inch

$m = 0.286$ inch

Figure 67. Attitude - Eccentricity Locus of Tilting-Pad Bearing

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Figure 68. Comparison of Critical Speed with Bearing Stiffness
MDBTF III-2 Test Series

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Figure 69. Deflection and Phase Angle of Shaft with NaK Rotor,
MDBTF III-2 Test Series

increased with speed. Although the phase angle measurements were somewhat uncertain, it is believed that the second mode critical speed was approached. The effect of the NaK pump (overhung mass) is to lower the second mode critical speed shown in Figure 68 without an overhung mass. At 44,000 rpm, operation appeared to be close to the critical speed as indicated by the 85° phase angle change. Operation at this speed was satisfactory, and no ill effects were observed except for the increased shaft deflection.

Power Loss

Friction power loss was measured by the speed decay method (in watts) as indicated in equation 11:

$$H = 1.49 \times 10^{-2} NI \frac{\partial N}{\partial T} \quad \dots 11$$

Where:

N = shaft speed, rpm

I = moment of inertia of the shaft, ft-lb-sec²

$\partial N/\partial T$ = rate of speed decay at the point in question with the
drive power turned off

The moment of inertia was measured experimentally while the rate of speed decay was taken from a continuous plot of speed versus time. The power loss of one thrust bearing and two journal bearings including seals, the correction for thrust bearing, and the correction for screw seals are all shown in Figure 70 as obtained from the speed decay. The power loss of one bearing is 350 watts without seals at 40,000 rpm, or 470 watts with seals. Screw seal power loss was obtained from previous tests at TRW. Power loss in some cases was also given by turbine inlet and outlet temperatures and pressures. Turbine power is

Where: $H = Q_a \Delta h \quad \dots 16$

Q_a = air flow rate

Δh = enthalpy change

This can also be written as

$$H = Q_a c_p (T_1 - T_2) \text{ Btu/sec} \quad \dots 17$$

Figure 70. Tilting-Pad Bearing Power Loss vs Shaft Speed,
MDBTF III-2 Test

For a choked nozzle

$$Q_a = \frac{0.532 A P_1}{\sqrt{T_1}} \quad \dots 18$$

For this nozzle

$$Q_a = 0.0148 \frac{P_1}{\sqrt{T_1}} \text{ (lb/sec)} \quad \dots 19$$

Then the power loss in watts becomes

$$H = \frac{7.50 P_1 (T_1 - T_2)}{\sqrt{T_1}} \quad \dots 20$$

A reasonably good agreement was found between these two methods at higher inlet pressures where both were available (see Figure 70). The turbine power method made it possible to obtain some power loss data with applied bearing load where the speed decay was not feasible.

For laminar film tilting-pad journal bearings the theoretical friction power was derived as shown in Appendix C, (Volume I), assuming constant viscosity. It can be written as follows:

$$H = \mu R^2 N^2 L (R/C_{MR}) \times k_1 \times f(\text{geometry}) \quad \dots 21$$

where k_1 is a constant, and the geometry function f is determined by preload, pivot point position, number of pads, length of pads, and eccentricity ratio. This equation can be regrouped in two dimensionless parameters.

$$\left(\frac{H}{N^3 L R^4 \rho} \right) = k_2 f(\text{geometry}) \times \left(\frac{\rho C_{MR} R N}{\mu} \right)^{-1} \quad \dots 22$$

This may not seem dimensionless if H is in watts. However, the shape of a log-log plot of

$$\left(\frac{H}{N^3 L R^4 \rho} \right) \text{ versus } \left(\frac{\rho C_{MR} R N}{\mu} \right)$$

will not change since the conversion constants can all be absorbed in the constant k_2 . This curve is called a "Taylor Plot" (see Ref 21). The term on the right side is the

Figure 71. Taylor Plot of Tilting-Pad Journal Bearing, MDBTF III-2 Test Series

Reynolds number since RN is the surface velocity with the 2π buried in k_2 . There will be a separate curve for each eccentricity ratio. An attempt was made to eliminate this by placing the eccentricity function into the Reynolds number and removing it from the geometry function by basing the Reynolds number on the trailing edge film thickness of the loaded pad instead of the bearing clearance. This checks out reasonably well on the theoretical curve. The theoretical and experimental curves (based on relatively few data points) are both shown in Figure 71. The experimental curve departs from the theoretical laminar one at a Reynolds number of about 300. This correlates with the Reynolds number of 250 obtained from the plot of K vs Reh^2 shown later. After the departure, the curve rises up to a Reynolds number of about 3,000, then seems to decrease again but at a different rate than in the laminar portion. This trend is similar to other turbulent bearing data reported in the literature, e.g., Ref 21. This curve may be used to predict the power loss of a three-shoe tilting-pad journal bearing in the turbulent region. For the laminar region the design curves based on Appendix C (Volume I) are recommended.

Flow-Pressure Characteristics

The flow-pressure characteristics of this bearing design are established by the effectiveness of the end screw seals. Since this bearing is fully hydrodynamic, i. e., self-acting, the only pressure and flow requirements are to keep the bearing cavity full and to have sufficient cooling flow. The MDBTF II-1 test series determined that a minimum supply pressure of 5 psig is required to keep the cavity full. This can be realized readily with the screw seal while still providing the necessary flow. However, in the actual turboalternator package, the screw seal clearances must be maintained at maximum values to prevent interference and reduce power loss. This can best be accomplished by optimizing the screw performance.

Analytical methods for screw seals that would satisfactorily predict performance are not available. An empirical expression, however, is given in Ref 22. It is reproduced below.

$$P = \frac{\mu U L_s C}{H^2} \quad . . . 23$$

Where:

L_s = seal length

H = thread depth

P = maximum attainable sealing pressure

C = coefficient dependent on helix angle and

$H/\frac{1}{2}C_{SD}$ = thread depth to clearance ratio

Figure 72. Flow Data for Tilting-Pad Journal Bearing A - End
MDBTF III-2 Test Series

This expression cannot be used to predict flow-pressure relationship, but it can be used to optimize the screws for pressure generated. The results of optimization yield a thread configuration with a helix (or lead) angle of 10° , and an $H/\frac{1}{2}$ CSD ratio of 7-1/2. Based upon some experimental data at TRW the included angle was determined to be $33^\circ 53'$.

The flow-pressure data obtained with this screw seal are given in Figure 72 for the A-end and in Figure 73 for the B-end. The difference in performance is caused by the larger clearance in one of the seals at the B-end.

Correlation with Analyses - Turbulent Load Factor

In the MDBTF II-2 test series a turbulent load factor K was defined as the ratio between the experimental and theoretical load capacity. Good correlation of this factor was found with Reynolds number when based on the loaded pad trailing edge film thickness as in equation 12

$$Re_{h_2} = \frac{\rho U h_2}{\mu} \quad \dots 12$$

This correlation included factors for different shaft speeds (and therefore surface velocities) loading down and up, two clearances, and many values of eccentricity ratio, all of which influence the trailing edge film thickness, h_2 . The temperature effect due to increased heating of the mercury at higher speeds was also included by employing the correct values of viscosity and density, although it was found to be negligible. The major limitation of the previous correlation was that it represented a single bearing length, one L/B ratio. This third test series provides a turbulent load factor for another bearing length.

The theoretical expression for laminar bearing load capacity is given in Appendix C (Volume I). This expression can be rewritten as follows when the assumption is made that the side leakage factor is a function of L/B ratio only:

$$W = \mu NLR (R/C_{MR})^2 f(\text{geometry}) g(L/B) \quad \dots 24$$

This expression holds for bearings with laminar films. The effect of turbulence is to change the side leakage function $g(L/B)$ and the viscosity μ , which now becomes an apparent turbulent viscosity, μ_t .

$$\mu_t = \mu_t \left(\rho, k, \frac{\partial \bar{u}}{\partial y} \right) \quad \dots 25$$

Where:

k = Prandtl mixing length

$\bar{u}$ = time average local fluid velocity

Figure 73. Flow Data for Tilting-Pad Journal Bearing B - End
 MDBTF III-2 Test Series

The empirical turbulent load factor obtained in these tests, therefore, gives the ratio of the apparent turbulent viscosity to the laminar viscosity, μ_t/μ , for the specific L/B ratio. To establish the function $g(L/B)$, test data from another length bearing was required.

The load deflection data from this test series, shown in Figures 65 and 66, were reduced and plotted in Figure 74. The resultant turbulent load factor for an L/D ratio of 0.655 or L/B ratio of 0.76 was

$$K = 0.0397 \text{ Re}_{h2}^{0.589} \quad . . . 26$$

This indicates the onset of turbulence, i. e., when K is 1.0, occurs at a Reynolds number, Re_{h2} , of about 250. This is lower than the generally accepted value of 1000 for fixed geometry bearings. In the MDBTF II-2 test series it was postulated that the low onset of turbulence in this kind of a bearing is caused by the large pad leading edge film thickness giving a high local Reynolds number, and that it is difficult to reestablish laminar conditions thereafter.

It has already been mentioned that the turbulent load factor calculations are based on slightly improved laminar analytical values over the previous test series. Therefore, before any comparison is made, the previous test data must be modified using the same improved analytical values. The previously established turbulent factor is shown in Figure 74, and agreement is quite good up to Reynolds number of 10,000. Since the laminar analysis correction will result in only minor change in the turbulent factor established for the 0.322-inch long bearing, i. e., $K = 0.055 \text{ Re}_{h2}^{0.545}$, confidence in this approach and capability in predicting performance using these factors has been established and verified for two different lengths of tilting-pad bearings.

At the end of this parametric test, the bearing parts, which include the shoes, pivot balls and journal, were found to be in good condition. The test included 22 stop-starts under shaft weight. No problems were encountered at either start or stop. This was a short test, however. The endurance capability was determined through a long-term test run under simulated operating conditions, and is summarized in Paragraph F, MDBTF III-3 test series.

F. MDBTF III-3 Test Series

Previous test series demonstrated that the tilting-pad bearing load capacity, stability, power loss, and flow-pressure characteristics met the Combined Rotating Unit requirements. The results of a 200-hour cold endurance run were reported previously. These results, however, were inconclusive for pivot point and cavitation-erosion evaluation. To more completely simulate operating conditions, a 500-hour endurance run was scheduled that would use 425°F mercury for the lubricant. Rotating load due to an unbalanced shaft would induce conditions leading to cavitation and erosion. Various pivot point configurations were to be evaluated. This test was conducted under the designation MDBTF III-3, and a summary of the results are presented in this section.

Figure 74. Turbulent Load Factor as a Function of Reynolds Number
MDBTF.III-2 Test Series

The tilting-pad test bearing was the bearing used for the MDBTF III-1 and -2 test series and is shown in Figure 64. The dimensions were as follows:

Bearing Dimensions

Diameter	0.622 inch
Length	0.407 inch
Pivot point diametral clearance	0.0025 inch
Machined-in diametral clearance	0.0035 inch
Pad circumferential length	0.535 inch
Pivot point position	42.5% of pad length from trailing edge

Screw Seal Dimensions

Length	0.187 inch
Screw	33°53' included angle 10° lead angle
Diametral clearance	0.007 to 0.008 inch

Six different pivot configurations were included in the two bearings. Balls of 3/32-inch and 5/32-inch diameter were used for the alternator (A) and turbine end (B) bearings, respectively.

The different configurations are shown in Figure 75. The materials of the various bearing parts are given below:

<u>Part</u>	<u>Material</u>
Sleeves	BG-41
Housings	BG-41
Pads	(A) BG-41 (B) Bluechip
Balls	SST 440C

Alternator (A) - End 3/32 Inch Dia Balls

Turbine (B) - End 5/32 Inch Dia Balls

Figure 75. Pivot Point Configurations
MDBTF III-3 Test Series

The test was run at a shaft speed of 36,000 rpm with an unbalance equivalent to 15 lb/bearing rotating load at this speed. The shaft with these bearings, however, appears to pass through the first mode critical speed below 10,000 rpm, so that the shaft will tend to rotate about its center of gravity. It was felt that this would give the most severe rocking of the shoes and therefore, act as a severe test for pivot point fretting and wear. Unidirectional load on the bearings was due to shaft weight only at the start of the test, 2.5 lb/bearing.

Mercury flow rates to the bearings were held constant at 10 lb/min. The temperature of the mercury was 425° F.

After the MDBTF III-1 and -2 test series, which lasted some fifty hours, the bearings (balls, pads, and journals) were carefully inspected and photographed. In general, the condition of all parts was good. The balls and pivot seats did not indicate any wear or increase in pivot point clearance. The leading edges at the ends of the pads showed the beginning of some cavitation-erosion damage indicating that flow separation occurred at those points. This was the result of a poor transition between the leading edge radius and the beginning of the bearing working surface. The sketch below exaggerates the condition.

The working surfaces of the pads and journals indicated some scratch marks of no measurable magnitude. These, it was assumed, occurred during the repeated start-stop cycles when metal-to-metal contact existed. The surfaces were still in very good condition. The alternator end journal had wear marks on it corresponding to the screw seal labyrinths. During the initial assembly and first spin up of MDBTF III-1, interference between screw seals and shaft occurred because of an eccentric condition. The turbine and journal had a ring of discoloration in the seal area which was due to mercury contamination. Additional details of balls, pivot points, pads, and journals are presented in the following sections, with photographs showing visible conditions.

1. Test Results

No major changes in operation could be detected for the first 520 hours. After this point the trace of the shaft orbit exhibited a vibration of constantly increasing amplitude. The unidirectional loading on the bearings was varied to establish the influence on amplitude and frequency. For the constant flow rates, the supply pressures increased somewhat for both bearings.

Pivot Points

The six different pivot point configurations were quite different in performance as was indicated by the inspection after the test. The diametral clearance as well as the dimension of ball seat in shoe and housing for each pad was determined before and after the test.

The change in these dimensions is tabulated below.

Alternator Bearing (A-end)

<u>Shoe No.</u>	<u>Change in A'</u>	<u>Change in B'</u>
1	0.00185 inch	0.00210 inch
2	0.00025 inch	0.00120 inch
3	0.00135 inch	0.00340 inch

Maximum change in pivot point diametral clearance $\approx$ 0.00475 inch.

Turbine Bearing (B-end)

1	0.00015 inch	0.00020 inch
2	0.00055 inch	0.00030 inch
3	--	0.00035 inch

Maximum change in pivot point diametral clearance $\approx$ 0.00085 inch.

These results indicate that the performance of the turbine-end bearing with the 5/32-inch balls was far superior in dimensional change to that of the alternator end bearing with the 3/32-inch balls. Contact stress calculations tend to predict this trend. These results, however, are not functions merely of contact stresses. Other factors such as sliding motion enter into the problem.

In the case pictured above, for instance, the sliding motion at point P is a direct function of $r \sin \phi$ for the same magnitude of tilt, which is independent of pivot point configuration. The pivot point dimension change, which is a result of wear and plastic deformation, is a complicated function of contact stress, sliding motion at the contact areas, and materials. We will not derive and correlate this function here, but we will show that in general no correlation with contact stress is to be expected. This is the case with the 5/32-inch pivot balls. For the smaller 3/32-inch balls, however, the contact stresses are so high that they predominate over the other effects, and the dimension change is a function of contact stress only (see Figure 76). The magnitude of the dimension change with the small balls is prohibitive. For the larger balls, the least change was obtained with the spherical radius seat. This was 0.00015 inch and 0.00020 inch in the housing and shoe, respectively. No wear was detected in the housing 90° conical seat with the large ball.

The condition of the pivot points in the shoes and the condition of the balls before the test are shown in Figures 77 and 78. It can be seen that none of the seats actually had point or line contact, but that the seating occurred over some finite area, generally a lapped seat. The pivot balls exhibited some corrosion deposits. This is the carry-over from the previous parametric test that was run on these sets of pivoted pad bearings.

A view of the pivot points and the balls after the test can be seen in Figures 79 and 80. The contact surfaces in the pads with the 3/32 inch balls have moved forward in the direction of rotation due to the Brinnelling effect. This could not be observed with the larger 5/32-inch balls. No damage was evident on any of the pivot balls. A check on their diameter and out of roundness substantiated this.

Erosion

Serious erosion damage was observed on the journals at the conclusion of the test. The shoes also suffered additional damage on the leading edge corners. The condition of the journals and shoes before and after the test is shown in Figures 81 to 87. Figures 81 and 82 show that the journals and the shoes were in good condition before the test. However, as mentioned previously, the shoes showed beginnings of cavitation-erosion at the leading edge corners.

Figure 76. Pivot Point Dimension Change vs Contact Stress
MDBTF III-3 Test Series

Figure 77. Pivot Points in the Shoes Before Endurance Test,
MDBTF III-3 Test Series

1

2

3

Figure 78. Pivot Balls Before Endurance Test,
MDBTF III-3 Test Series

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Figure 79. Pivot Points in the Shoes After Endurance Test,
MDBTF III-3 Test Series

CONFIDENTIAL

0405587030

DECLASSIFIED

CONFIDENTIAL

A

B

1

2

3

Figure 80. Pivot Balls After Endurance Test,
MDBTF III-3 Test Series

DECLASSIFIED

CONFIDENTIAL

CONFIDENTIAL

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Figure 81. Journal Before Test
MDBTF III-3 Test Series

CONFIDENTIAL

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1

2

3

Figure 82. Shoes Before Test
MDBTF III-3 Test Series

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Figure 83. Journal, Alternator End, After Test - (Bottom)
MDBTF III-3 Test Series

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Figure 84. Journal, Alternator End, After Test - (Top)
MDBTF III-3 Test Series

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Figure 85. Journal, Turbine End, After Test - (Bottom)
MDBTF III-3 Test Series

CONFIDENTIAL

Figure 86. Journal, Turbine End, After Test - (Top)
MDBTF II-3 Test Series

Figure 87. Cavitation-Erosion Damage to Shoes After Test,
MDBTF III-3 Test Series

The alternator end journal appearance after the test is shown in Figures 83 and 84. While it appears that most of the bearing working surface except the very edges was undamaged, the seal area was badly eroded, particularly under the leading threads of the screw seal. The sharp grooves that are visible in this area are the result of partial contact with the threads of the screw seal. The contact was made possible by the large clearance change in this bearing due to pivot point yielding, as discussed in the previous section. This large clearance change is also postulated to be responsible for the majority of the serious erosion damage in this bearing.

Increased clearance normally increases the intensity of turbulence; in this bearing it also increases the lubricant recirculation creating low pressure regions at the shoe leading edges. When this effect is combined with the vortices at the leading thread of the screw seal, the result is highly turbulent, low pressure regions at the shoes' leading edge corners. In these regions vapor bubbles are formed and collapsed with accompanying erosion of material. This is known as cavitation-erosion. This type of damage is accelerated by the dynamic effect of the shaft on this cavitating film. Under normal operating conditions, the shaft traces out an orbit in the bearing clearance because of unbalance in the shaft assembly. Once every cycle, the shaft tends to squeeze the cavitating film, building up higher pressures and collapsing the cavities at a faster rate. This results in increased erosion.

An additional phenomenon was observed with this bearing. A random excitation caused the shaft-bearing system to vibrate at its lowest natural frequency, the amplitude of which is negligible at normal operating clearances. At the increased clearance that this bearing experienced, however, the vibration amplitude became quite large, and the cavitation-erosion was thereby increased. The score marks or wear tracks at the edges of the bearing journal appear to have been caused by particles coming loose and being ground around before passing out of the bearings.

Figures 85 and 86 show the condition of the turbine end journals after the test. The cavitation-erosion damage on this end is much less severe than on the alternator end. This correlates well with clearance change which was an order of magnitude less than that in the alternator end bearing. As a matter of fact, the erosion that can be seen on the inboard side of this journal occurred under the seal area where the seal clearance was 0.014 inch instead of 0.007 to 0.008 inch due to a machining error. No damage could be observed under the outboard seal where the desired clearance of 0.007 to 0.008 inch was maintained. It consequently appears that the increased seal clearance was the factor leading to the cavitation-erosion in this bearing.

The damage suffered by the shoes was localized around the leading edge corners (see Figure 87). The damage on the alternator end bearing shoes (A) was quite severe on the corners, and beginnings of erosion can be seen on the front third of the shoes' bearing area. The rub marks visible on the alternator shoes could have been the result of operation when the bearing was weakened by the large clearance increase and the high amplitude vibration resulted. However, the particles plucked out of the pads and journals in the area of severe cavitation-erosion are more likely to be the cause for the marks. The turbine end bearing shoes were in good condition after the test except for beginnings of erosion on the leading corners.

In summary, the above results indicate that the seal clearance as well as the bearing clearance should be reduced to reduce cavitation-erosion in this tilting-pad journal bearing. A smooth transition at the leading edge of the shoes should eliminate the cavitation-erosion observed there. With bearing clearances below 0.0025 inch and seal clearances below 0.008 inch, no cavitation-erosion should occur for the first 645 hours of operation. Ambient pressure could be increased to further prolong operating life.

Stability

The shaft-bearing system at the test speed of 36,000 rpm operates well above the first system critical speed and below the second mode by approximately 5000 rpm, based upon the previous test series. This is reproduced in Figure 88. The shaft unbalance of 180 microinches (distance between geometric centerline and center of mass) is equivalent to a rotating load of 15 lb/bearing. This unbalance causes the shaft to trace out a somewhat circular path in the bearing clearance at shaft speed. Above the first critical speed the trace should theoretically be about 180 microinch radius and rather insensitive to speed. The experimental observations agreed with the theoretical predictions inasmuch as the trace orbit was relatively constant at a radius of 180 microinches.

Superimposed over this synchronous whirl, a vibration of much lower and more random frequency was observed. The double amplitude of the vibrations was 0.0001 inch at the start of the endurance test. This increased slowly to 0.00035 inch after 552 hours of running time. At this point, the load on the bearings was increased from shaft weight of 2.5 lb/bearing to 4.0 lb/bearing. The effect was to suppress the amplitude of vibration. But after an additional 120 hours of running, the amplitude increased to 0.0015 inch. Raising the applied load to 5.0 lb/bearing again suppressed the amplitude to 0.0003 inch. In about another 30 hours of operation, the amplitude grew to 0.0027 inch, then the load was increased to 13.0 lb/bearing. After this point the amplitude increase was more rapid so that the test was terminated. The plot of amplitude versus time and the representative shaft orbit traces are shown in Figure 89. During the entire test, the amplitude of the synchronous whirl remained constant.

The proximity probe output was fed into a wave analyzer to determine the relationship between frequency and amplitude of the vibration. Although there was considerable scatter in the data points, the general trend is shown in Figure 90. Increasing load resulted in a decrease in amplitude and a corresponding increase in the vibration frequency. To explain the occurrence of the vibration, the following mechanism is postulated. Vibrating inducing forces are present in the bearing system. The source of these forces could have been the friction in the pivot points or the high turbulence at the pad's leading edges. This hypothesis assumes that the friction at the pivot point combined with the turbulence and the rotating load prevents the bearing-rotor system from operating in a stable manner. Under such conditions the synchronous (fundamental) whirl component, when operating above the threshold of stable operation, will excite the lowest natural

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Figure 88. Critical Speed of Shaft Bearing System (MDBTF III-2)

Figure 89. Vibration Amplitude and Shaft Orbit vs Time
MDBTF III-3 Tilting-Pad Bearing Endurance Run

Figure 90. Vibration Amplitude vs Frequency From Wave Analyzer
MDET III-3 Tilting-Pad Journal Bearings

frequency of the bearing-rotor system, i. e., the first mode critical speed. The vibrational frequency generally occurred at 130 cps, which according to Figure 90 is the approximate crossover point of the bearing stiffness with the bearing-rotor critical. The hypothesis would therefore tend to be verified and explain the source of the vibration.

Further support for this explanation was obtained in the following manner. As the vibrational amplitude increased, a unidirectional load was applied on the shaft. The result was immediate and enlightening. First the amplitude of vibration dropped, then the frequency of vibration increased by a measurable amount. This is to be expected if the hypothesis is correct. As the load is increased the operating eccentricity ratio increases, i. e., smaller film thickness. The result is to have a larger hydrodynamic generated pressure force on the shoes which tends to overcome friction and permits the shoes to tilt better, thereby decreasing vibration amplitude. At the same time the smaller film thickness results in a stiffer bearing which in turn pushes up the first mode bearing-rotor critical speed. As operation continued and the wear and friction increased on the pivot point, the amplitude again increased and the frequency dropped because of the larger bearing clearance resulting in a weaker (less stiff) bearing. This series of events was duplicated each time the load was changed. Figure 89 shows the trend, types of shaft orbits, and how applied loads initially reduced amplitudes. When combined with Figure 90, the increase in vibration frequency, i. e., increase in first bearing rotor system critical speed, is shown as amplitude decreases, i. e., weight load.

A typical wave analyzer trace is shown in Figure 91.

Flow-Pressure

The flow-pressure characteristics for the entire test are shown in Figure 92 for both bearings. It is a time-base plot of supply pressure required to keep the flows constant at 10 lb/min. Although there is a short term variation in supply pressure of ± 3 psi, the long term increase is quite evident in the alternator bearing. The increase in pressure is from 30 psig at the start to about 48 psig at the finish. The turbine bearing pressure remained relatively constant at 25 psig.

The flow-pressure characteristics of these bearings are functions of seal performance; but seal performance does depend on the condition of the entering fluid. In the case of the alternator bearing, where the clearance opened up considerably, the intensity of turbulence in the lubricant was high, resulting in increased pressure drop across the seal. The turbine bearing clearance change and the pressure change were negligible.

The completion of the MDBTF III test series represented the end of the component test on the tilting-pad journal bearing during this report period.

Figure 91. Typical Wave Analyzer Trace, Vertical Proximity Probe, MDBTF III-3

Figure 92. Flow-Pressure Characteristics of Tilting-Pad Bearings During Endurance Run
MDBTF III-3 Test Series

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2. Conclusions

Specific conclusions based on the results of the MDBTF III test series are summarized below:

- 1) Freedom from half-frequency whirl instability was verified with the longer bearing under all test conditions.
- 2) Operation above the first mode and below the second mode was again encountered as predicted by analysis. For the longer bearing, the extrapolated crossover point was between 5,000 and 10,000 rpm. The longer bearing and rotor combination appeared to be approaching the second mode critical speed around 45,000 rpm.
- 3) The load capacity based on a film thickness of 0.0002 inch is approximately 52 lb and 42 lb at 40,000 rpm and 36,000 rpm, respectively, loading into a pivot. Loading between pivots will result in a slightly greater capacity for the same film thickness.
- 4) The near vertical attitude-eccentricity locus measured for the short bearing was not influenced by the longer bearing.
- 5) Random instabilities were traced to coincide with the lowest bearing-rotor critical speed. These occurred at relatively small amplitudes initially but at greater amplitudes as the clearance and turbulence increased.
- 6) The location of screw seals in the housing, the stationary member, did not eliminate erosion damage on the shaft under the seal.
- 7) Lower clearance (radial) screw seals suffer less damage than large clearance seals.
- 8) Improper blending of the pad's inlet radius with the bearing working surface creates cavitation-erosion damage at the leading edge and may be a contributor to pad vibration as the pressure profile fluctuates, consequently contributing to pivot point wear.
- 9) The best pivot point configuration of those tested is the larger 5/32 inch diameter ball (used as the pivot) and the spherical seat. It exhibited no measurable wear.

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- 10) Even though a clearance increase of ≈ 0.01 inch was experienced on the end with the 3/32 inch diameter balls, the bearing did not fail and supported applied loads in excess of 10 lb. It did not exhibit half-frequency whirl instability.
- 11) A turbulent load factor for the longer bearing (based on an improved laminar analysis) was obtained as follows: (See equation 26)

$$K = 0.0397 \text{ Re}_{h2}^{0.589} \quad . . . 26$$

This matched the previously established factor very well to Reynolds numbers of 10,000.

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III. MAJOR CONCLUSIONS

The analytical and experimental bearing development program indicated that the tilting-pad journal bearing eliminated several disadvantages and undesirable characteristics associated with such fixed geometry journal bearings as the plain, three-sector, and three-pad.

- 1) The tilting-pad bearing remained free from half-frequency whirl at all tested speeds, loads, and clearances.
- 2) Sensitivity to clearance change was negligible for load capacity and stability.
- 3) Flow-pressure requirements were well within system capability, and supply pump pressure could drop off drastically without influencing bearing performance.
- 4) Selection of preload permitted critical speed control.
- 5) Misalignment due to thermal gradients could be easily absorbed by the tilting-pads (within the limits of the seals and fixed thrust bearing).
- 6) Larger clearances permitted greater tolerance to particles.
- 7) Adequate design procedures were developed to predict performance for a wide range of sizes, clearances, and speeds.

The component development program established the ability of the tilting-pad bearing to meet the SNAP 2 requirements of load, flow-pressure, stability, etc.; and a single CRU IVM-4 test demonstrated that it could be integrated into a CRU and operate successfully at design conditions. Nevertheless, several developmental problems required resolution before the tilting-pad bearing could be integrated into the CRU's with confidence of meeting the life requirements. The major problem areas were:

- 1) Erosion damage in the seal area
- 2) Wear at the pivot.

The elimination of these problems by selection of the correct configuration and material had to be verified. The tilting-pad bearing, however desirable, therefore had to be classified as life-limited until proven otherwise.

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