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(TRW REPORT No. ER-6041)

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

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

32986

MERCURY RANKINE PROGRAM (SNAP 2)
DEVELOPMENT OF LIQUID-MERCURY-LUBRICATED BEARINGS
VOLUME I
ANALYTICAL DESIGN APPROACH AND STATUS
OF BEARING SYSTEMS (WITH APPENDIXES)

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AEC RESEARCH AND DEVELOPMENT REPORT

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By
OTTO DECKER

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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 on the study. All other Appendixes are also included in Volume I.

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ABSTRACT

Stringent requirements and the limited state of the art complicated the analytical design and development of fluid film bearings lubricated by liquid mercury for systems such as SNAP. The demands for long life, maximum reliability, minimum system weight, and operation in space are reflected in such things as restrictions on permissible supply pressure, flow rate, temperatures, usable materials, and construction. The high rotational speed of shaft, coupled with the low kinematic viscosity of liquid mercury, assures bearing operation beyond the laminar flow regime. Operation was assumed in the little-investigated turbulent and superlaminar flow regimes.

Despite these difficulties, the feasibility of liquid-mercury-lubricated fluid film bearings has been demonstrated by a series of endurance tests on different bearing types. The longest test to date is in excess of 6000 hours on a single set of bearings.

The analytical and experimental activities of the bearing development program, covering the period from January 1961 to June 1963, are covered in this report.

The design approach at TRW Inc. has progressed from the use of empirically corrected laminar flow equations to the use of more nearly fundamental considerations of high Reynolds number (turbulence) effects. Semi-empirical analytical techniques and non-dimensional design curves were generated as a by-product of a predominantly hardware-oriented development program. These techniques and procedures permit the design and performance of hydrodynamic and hybrid bearings operating at Reynolds numbers (based on radial clearance) up to 10,000.

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I. SUMMARY AND RECOMMENDATIONS

Stringent requirements and the limited state of the art complicated the analytical design and development of fluid film bearings lubricated by liquid mercury for systems such as SNAP. The demands for long life, maximum reliability, minimum system weight, and operation in space are reflected in such things as restrictions on permissible supply pressure, flow rate, temperatures, usable materials, and construction. The high rotational speed of the shaft, coupled with the low kinematic viscosity of liquid mercury, assures bearing operation beyond the laminar flow regime. Operation was assumed in the little-investigated turbulent and superlaminar flow regime.

NOTE

The term "turbulent flow" is used throughout this report even though there is considerable evidence that fully developed turbulence does not exist throughout the clearance space of the bearing. Current investigations indicate that the type of flow is very complex for the bearings discussed in this report, lubricated with liquid mercury and operating over the speed range of 0 to 60,000 rpm. At any one time, the clearance space of the bearing may contain a combination of laminar, vortex, and turbulent flows.

The term "superlaminar flow," used by Smith and Fuller (Ref. 6) was an attempt to recognize the various flow regimes that may exist in a bearing operating at Reynolds numbers above the transition period.

Of the bearing configurations investigated during the period covered by this report, the plain cylindrical, the tilting-pad, the three-sector and the three-pad journal, and the spiral-groove thrust bearings are discussed in detail. Of these bearings, the three-pad the the three-sector bearings with orifice compensation are capable of meeting the SNAP 2 journal bearing requirements. The spiral-groove thrust bearing satisfactorily meets the requirements as specified during this period. The self-aligning (tilting-pad) bearing is capable of meeting the lubrication requirements, but requires further development in the area of end seals to assure flooded operation, and demonstration of pivot point integrity over long periods. The plain cylindrical bearing was considered unsatisfactory by virtue of its unstable characteristics.

Detailed performance characteristics of each bearing type are presented. Generally the discussion centers around the following characteristics: load-carrying capacity, attitude angle-eccentricity locus, flow-pressure, flow-pressure-load, power loss, fluid film stiffness, radial film stiffness, damping, resonant frequency of the fluid film and shaft mass system, stability to half-frequency whirl, and life and the influence of cavitation-erosion.

Measured performance is compared to predicted performance wherever possible, and analytical techniques and design curves are modified or generated to explain actual characteristics. The principal conclusions that may be drawn from the results of the liquid-mercury-lubricated bearing development program conducted at TRW are as follows:

- 1) Three-pad and three-sector journal bearings (orifice compensated) and the spiral groove thrust bearing are capable of meeting SNAP requirements.
- 2) Usable design procedures have been established by which the performance of high speed (turbulent) liquid mercury lubricated bearings can be predicted with acceptable accuracy.
- 3) Reasonable agreement has been obtained between experimental results and theoretical predictions for major bearing characteristics such as load capacity, power loss, attitude angle, and instability thresholds (inversion or synchronous and fractional frequency whirl). Where agreement was less than satisfactory, semi-empirical factors which take Reynolds number into consideration were established.
- 4) Mercury-lubricated bearings (similar to the majority of loaded fluid film bearings) suffer separation, starvation, or cavitation in the diverging section of the clearance, downstream of the minimum film thickness. This phenomenon caused minor surface damage to the three-pad and plain cylindrical bearings, particularly during tests when large clearance bearings were subjected to large loads (combined uni-directional and rotating).

Although techniques are now available, as indicated above, to predict performance of high speed, superlaminar or turbulent flow bearings, anyone who has worked in or been associated with the field of liquid metal (inorganic) lubrication is aware of the limited basic and applied research that has been performed. Several approaches are open to improve fundamental understanding of this type of lubrication. The purely theoretical approach is not recommended at this time, principally because too many simplifying assumptions must be made to solve the fluid-flow equations. For instance, the type of flow and its dependence on factors other than Reynolds number remains unknown. Yet this knowledge must be available before descriptive equations can be written. Similarly, the purely experimental program is not recommended because the information and data obtained are too limited and expensive, and cannot be used

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for different sizes or types of bearings. The approach that holds the most promise to improve basic understanding and is general enough to apply to a variety of bearing sizes and configurations, while at the same time permitting development of bearings to stay in step with rotating machinery advances, is a combined theoretical and experimental program similar to the one being conducted at TRW in support of SNAP 2. The recommended program consists of three phases.

Phase I. The turbulent flow factors necessary to permit prediction of load capacity, flow, power loss, and attitude angle must be obtained. Whether these factors are mixing length coefficients, k , or overall turbulent factors, K , they must reflect the influence of a varying Reynolds number in the bearing clearance space and the L/D ratio to include end leakage. Phase I would be a continuation of the present TRW program and would provide the necessary design procedures for current or near-future applications.

The fundamental experimental and analytical program must include the following cases:

- 1) Bearings at higher Reynolds Numbers ($Re > 6000$) to assure encountering turbulent flow regimes;
- 2) Bearings operating under large loads ($\epsilon > 0.6$) where both laminar and turbulent flow may exist;
- 3) Bearings having large L/D ratios where present assumptions of axial and circumferential pressure gradients produce inaccurate results;
- 4) Bearings operating predominantly under the influence of dynamic forces such as rotating loads, in or out of phase with shaft frequency; and
- 5) Hybrid bearings that are part hydrostatic and part hydrodynamic. In the case of these types of bearings the non-dimensional design curves should be complemented with semi-empirical equations containing overall turbulent factors. Before this is possible a practical technique for predicting laminar performance (which includes speed effects) must be established.

Phase II. Before establishing a realistic and usable turbulent flow theory, the following basic relationships must be obtained from carefully controlled experiments:

- 1) Onset of unstable flow, or the critical Reynolds number at which the flow regime departs from pure laminar flow in the clearance space;
- 2) Influence of eccentricity ratio and other factors such as bearing type, configuration, supply sources and pressures on establishing the onset of unstable flow;

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- 3) Actual type of flow that exists in bearings as functions of local Reynolds numbers, i. e., laminar, Taylor vortices, turbulence or a combination of these; and
- 4) Turbulent flow end leakage factors (relationship of the velocity components in circumferential and axial directions).

Phases I and II will establish the fundamental relationships needed to permit solution of the currently unsolvable fluid flow equations and will provide improved analytical design procedures.

Phase III. Verification tests of established theory and design procedure will then provide the necessary understanding of turbulent-flow, liquid-metal lubrication most expeditiously and economically.

In conjunction with the combined analytical and experimental program discussed in the above phases, instrumentation must be developed that can measure and detect accurately such variables as flow, film pressures, shaft position (eccentricity ratio, film thickness and attitude angle), power loss, and temperature. Existing instrumentation suffers from temperature effects, sensitivity to contamination, and being removed from close proximity to the bearing. Since the success of the entire program to develop bearings and lubrication technology depends largely on the accuracy of the instrumentation and the measurements, considerable effort is recommended in establishing the best available methods.

In terms of suitable bearings, both the three-pad and three-sector orifice compensated bearings are recommended for evaluation in the SNAP 2 turboalternators, Combined Rotating Units (CRU's). Endurance capability, the type of bearing and the operating conditions which either eliminate or minimize cavitation-erosion damage to an acceptable level should be established on component and package tests.

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II. INTRODUCTION

As in any turbomachinery application, the successful development of the SNAP 2 Combined Rotating Unit (CRU) is dependent on the availability of bearings to support the shaft assembly. Major considerations in the selection of fluid-film bearings for the CRU were the long-life requirement and use of cycle working fluid as the lubricant. These two considerations eliminated the use of and need for seals and the difficult development problems associated with minimizing the leakage rate, power loss, and the performance degradation characteristics of such devices. Though the use of fluid-film bearings lubricated with liquid mercury posed many questions, there was no doubt that their development and application would be successful more quickly than development of seals compatible with mercury.

To establish the boundaries and restrictions pertinent to the development and application of fluid-film bearings lubricated with liquid mercury, the following conditions imposed by the SNAP 2 system and CRU concept were considered:

- 1) Long life without maintenance and the use of mercury require that bearing materials and geometry be corrosion resistant, erosion resistant, and insensitive to contaminants resulting from corrosion products.
- 2) Low system weight to power ratio as manifested by high system efficiency dictates that permissible bearing friction power loss and available lubricant flow and supply pressure be restricted.
- 3) Startup of the system when orbit is achieved, coupled with system weight limitations, requires a minimum lubricant inventory and imposes transient thermal effects on lubricant and bearing.
- 4) The CRU concept of a single shaft assembly to minimize auxiliary rotating machinery provides maximum efficiency and reliability. However, the concept imposes a severe thermal environment on the bearing and requires compatibility of thermal expansion characteristics of the bearing, housing, journal, and shaft materials.
- 5) Operation on the ground as well as in orbit imposes a range of loading conditions not found in conventional bearing applications. In particular, development testing requires some design compromise to achieve rapid and economical replacement of bearings.

These technical considerations, as well as considerations of budget and total program schedule, have influenced the details of the SNAP 2 bearing investigation.

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The requirements of the SNAP 2 system and the CRU concepts are manifested in specific bearing characteristics that may be categorized as follows:

- 1) Load-carrying capacity is limited by minimum film thickness or closest approach of journal to bearing. Because the magnitude, direction, and character, i. e., rotating or unidirectional, of the load cannot be predicted precisely nor controlled within the assembly, this requirement cannot be specified exactly. Moreover, the minimum permissible film thickness, except for zero values, cannot be determined easily, particularly within the CRU.
- 2) Fluid-film stiffness, defined as the resistance of the lubricant film to lateral motion of the journal in the bearing, is related to the several natural frequencies of the shaft-bearing system. Ideally, the stiffness or spring rate of the fluid film should be high so that the lowest natural frequency is above the operating speed of the shaft. The stiffness may also be very small so that operation is always above the critical speed. However, if the fluid film has a high damping coefficient which limits the resonant amplitude, the stiffness becomes less important.
- 3) Fluid-film stability is the resistance to the self-excited vibration known as fractional-frequency whirl. This phenomenon is manifested in whirling of a lightly loaded journal about the bushing center at approximately half the journal speed. The neutral gravity and the absence of a predictable load intentionally imposed makes this stability characteristic important. Theoretical load capacity is zero for this instability.
- 4) Film continuity is related to heat transfer from the shaft, to cavitation-erosion, and to rotating load capacity. In the CRU, the lubricant is used to remove heat conducted along the shaft from the turbine and NaK pump, as well as that generated by viscous shear of the film. Since the amount of available lubricant flow is restricted by other system considerations, a minimum flow characteristic is desired.
- 5) Insensitivity to mechanical distortion is required, since temperature and corrosion considerations preclude the use of conventional bearing materials having good conformability. The mechanical distortions caused by thermoelastic effects are misalignment, out-of-roundness, and changes in clearance. While the distortions can be minimized by the design of the CRU housing and by control of the bearing environment, fabrication simplicity, among other considerations, requires selection of a bearing which has maximum tolerance to distortion.

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HOLE FED JOURNAL BEARING

AXIAL GROOVE JOURNAL BEARING

TANGENTIAL INLET JOURNAL BEARING

ANNULAR GROOVE BEARING

LOAD-PAD TYPE JOURNAL BEARING

THREE-SECTOR JOURNAL BEARING

THREE-PAD JOURNAL BEARING

TILTING-PAD BEARING

Figure 1. Typical Journal Bearing Geometries

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Table I lists the detailed bearing specifications arising from the requirements established by the SNAP 2 system and current ground and flight CRU designs.

TABLE 1
SNAP 2 BEARING SPECIFICATIONS

Lubricant	Mercury at $425 \pm 25^\circ\text{F}$	(nominal design steady state)
Speed	36,000 rpm $\pm 10\%$	(originally 40,000 rpm)
Maximum available lubricant supply pressure	450 ± 25 psi	
	<u>Alternator</u>	<u>Turbine</u>
Unidirectional radial load (max)	15 lb	15 lb
Rotating radial load (max)	15 lb	15 lb
Thrust loads (net)	24 lb toward Hg pump - Condition 1 23 lb toward turbine - Condition 2	

Figures 1 and 2 show some of the standard journal and thrust bearing types considered for low viscosity lubricant applications. Reference 1 summarizes the development and performance characteristics of the hole fed, axial groove, tangential feed, annular groove, and load-pad journal bearings and of the hydrodynamic pocket (Rayleigh step-land) thrust bearing. The above journal bearing types were modifications of the plain journal bearing and were designed to improve life, stability, flow, and load characteristics. At the conclusion of the work described in Reference 1, TRW had decided that fluid-film journal bearings and the spiral groove thrust bearing were feasible for the SNAP 2 application.

As part of TRW's SNAP 2 development program since 1961, hydrodynamic, hydrostatic, hybrid (both hydrodynamic and hydrostatic) types of journal bearings, and the hybrid spiral groove thrust bearing were evaluated. A typical hydrodynamic journal bearing type that received additional evaluation during this period was the axial groove plain journal bearing. A three-pad tilting-pad bearing, another hydrodynamic (self-acting) bearing, was also developed. Flooded operation of this type of bearing was assured by restricting the outward flow of lubricant by means of screw-type seals acting as viscosity pumps. Typical hybrid types considered were the three-sector and the three-pad bearings. Several variations of the three-sector and the three-pad bearing were evaluated. In each case the three-sector bearing consisted of three separate sectors divided by an axial drain groove. Each sector had either three supply orifices or a pad supplied by a single orifice. The supply orifices were initially uncompensated; later the supply orifices were compensated. The three-pad bearing was identical to the three-sector orifice-compensated bearing except that the axial drain grooves were removed.

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Figure 2. Thrust Bearings

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The results of the analytical and experimental investigations conducted to achieve required characteristics in the above bearings and the contributions of the SNAP 2 program to the state-of-the-art in turbulent-flow, fluid-film bearings is the subject of this report. TRW experimental testing of liquid-mercury-lubricated bearings from January 1959 through 1964 is shown in Figure 3. The different tests, their designations, and their time sequence are shown in Figure 4. This report deals primarily with the development program covering the period from January 1, 1961 to June 1963. The major exception is the DBTR 6 test series which was conducted shortly after the period ended. However, it is included since it was a continuation of the DBTR 5 test series. Figure 5 shows the Sequence of Bearing Development covering 1959 through 1963. The initial phases of the bearing development program were concentrated on analytical studies and short parametric verification tests. The need for extended periods of running (endurance tests) did not arise until suitable bearings for the SNAP 2 application had demonstrated their ability to meet load, flow-pressure, power loss, and critical speed requirements. Table 2 is a summary of SNAP 2 developmental bearing test activity.

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Figure 3. Bearing Development Activity - Test Hours

Figure 4. Bearing Development Activity - Number of Tests

BEARING TYPE	1959	1960	1961	1962	1963
PLAIN JOURNAL
THREE-SECTOR (Uncompensated)
THREE-PAD (Compensated)
THREE-SECTOR (Compensated)
TILTING-PAD
SPIRAL-GROOVE THRUST BEARING

Figure 5. Sequence of Bearing Development

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TABLE 2

SNAP 2 BEARING TEST SUMMARY

<u>Bearing Type</u>	<u>Test Number</u>
Annular	BCR 1; FRR 4, 4-1, 5, 5-1
Plain with Hole	BCR 2
Axial Pad with Tangential Feed	FRR 2, 3, 4, 4-1
Load Pad	FRR 6, 7, 8, 9
*Plain with Axial Pads	BCR 3; MDBTR 1, 2; FRR 1, 1-1, 2, 3, 10, 11; <u>BETR 4, 4A, 4B</u>
*Tilting-Pad	<u>MDBTF 1-0, 1-1, II-1, II-1A, II-1B, II-2, II-2A, II-2B, II-2C, III-1, III-2, III-2A, III-3</u>
*Three-Sector	<u>FRR 11, 12, 12A, 12B, 12B-1, 13, 13A, 14, 14-1, 15, 16, 17, 17-1, 18, 19, 19-1, 20, 21, 21-1, 22-1, 23, 24, 25, 26, 27, 28, 28-1, 29, 30; BETR 1, 2, 3; TFR 1, 2; DBTR 1, 1A; AI Pump; DBTR 5, 5A, 5C, 6, 6A, 6C; CRU II-1, II-2, II-2A, II-2B, II-3</u>
*Three-Pad	<u>DBTR 2, 3, 3A, 4, 4A, 4H, 4M; BETR 5, 5E</u>
Thrust Step Land	PTR 1, 2
*Spiral-Groove Thrust	<u>DBTR 4-1, 4-2, 4-3</u>

BCR = Bearing Concept Rig	MDBTR = Modified Dual Bearing Test Rig
BETR = Bearing Endurance Test Rig	MBTR = Modified Bearing Test Rig
DBTR = Dual Bearing Test Rig	PTR = Pocket Thrust Rig
FRR = Free Running Rig	TFR = Transparent Flow Rig
CRU = Combined Rotating Unit	

* Only these bearing types are reviewed in this report. The data from the underlined test numbers are used to present performance.

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Figure 6. Properties of Liquid Mercury

III. ANALYTICAL DESIGN APPROACH

A. FACTORS COMPLICATING THE ANALYTICAL DESIGN

Lubricant properties, turbulent flow, geometry in the case of the three-sector and three-pad bearings, and the difficulty of separating hydrodynamic and hydrostatic effects are complexities not found in conventional applications of liquid-film journal bearings. To simplify the mathematical derivation of the hydrodynamic bearing equations, some of the properties of liquid mercury as a lubricant are not included in the classical theory. Consider the normal assumptions made in the development of the Reynolds equation as related to the lubricating properties of liquid mercury:

<u>Normal Assumptions</u>	<u>Liquid Mercury Lubrication</u>
Flow is laminar:	The low kinematic viscosity of liquid mercury coupled with the high shaft speeds results in Reynolds numbers which indicate turbulent or nonlaminar flow.
Fluid inertia terms are negligible:	The density of mercury is an order of magnitude higher than that of conventional lubricants; so the inertia terms are not negligible.
No slip occurs at the liquid-solid interface:	A reduction in hydrodynamic bearing friction was reported to occur with oil-repellant (non-wetting) bearing surfaces (Ref. 2), and this is believed to be due to slip at the lubricant-bearing interface. However, no significant difference in the efflux times of glycerol-water solution through capillaries of differing wettability was observed (Ref. 2). This observation, coupled with others which indicate that the Hagen-Poiseuille or Blasius laws can be applied to calculate friction pressure drop of nonwetting liquid mercury flowing in tubes, implies that slip at the bearing-lubricant interface requires no significant modifications in classical bearing analysis. However, in determination of film continuity or cavitation, non-wetting, or more properly surface tension, may be a significant factor. Under similar conditions, the surface tension of liquid mercury is 487 dynes/cm compared to a surface tension of 72.8 dynes/cm for water.

Analytically, liquid mercury has the desirable advantage over conventional lubricants of nearly constant absolute viscosity over a wide range of temperatures. The properties of liquid mercury which are pertinent to hydrodynamic lubrication are plotted in Figure 6 as a function of temperature.

Figure 7. Reynolds Number vs Speed

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The low kinematic viscosity of liquid mercury coupled with high shaft speeds assures that the flow in the bearing will not be completely laminar. For example, the Reynolds number for the SNAP 2 bearings based on a diametral clearance of 0.001 inch is 4500. Using the classical instability criterion of G. I. Taylor (Ref. 3), the critical Reynolds number at which the flow changes from laminar to vortical is 1030 for this case. Based on Taylor's concentric cylinder experiments, the critical Reynolds number (Re_{cr}), that Reynolds number at which laminar flow breaks down, is given by

$$Re_{cr} = \frac{\rho U_{cr} C_R}{\mu} = 41.1 \sqrt{\left(\frac{R}{C_R}\right)} \quad \dots 1$$

In terms of transition or critical speed, the above equation can be modified to

$$N_{tr} = \frac{77,400 \nu}{m^{3/2} D^2} = \frac{1.57 \times 10^3 \mu g}{\rho m^{3/2} D^2} \quad \dots 2$$

Figures 7, 8, and 9 show these relationships as functions of speed and different clearance moduli. The analytical design approach considering turbulence is discussed in Paragraph II-C.

The geometric configuration of the three-sector bearing or other hybrid bearings complicates the analysis to the extent that it consists of three interacting partial bearings. The available analytical solution for this bearing configuration (Ref. 4) without hydrostatic augment is based on laminar flow theory.

The plain cylindrical bearing and similar hydrodynamic types should offer no problem since this is the configuration assumed in classical solutions. However, a pressurized feed source introduces another degree of complication that is not included in the classical approach.

B. REVIEW OF AVAILABLE THEORIES

Limited analytical and fundamental understanding exists in the field of turbulent-flow and liquid-mercury lubrication, if the number of technical papers on the subject serves as a guide. To be sure, a general knowledge of the over-all effects of turbulence on bearing behavior and performance is available. For example, when flow is turbulent, load capacity increases, flow goes down, and power loss goes up. The general effect of these variations is to decrease stability and increase the temperature rise and attitude angle. What is not known within acceptable accuracy is how much or how little. In fact, the few investigators in the field point out that the degree of turbulence required to influence performance is not even agreed upon.

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Figure 8. Ratio of Operating Speed to Critical Speed vs Operating Speed
For Various Clearance Moduli

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Figure 9. Transition Speed as a Function of Clearance Modulus (m)

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Investigations in turbulent-flow lubrication are comparatively new, although the classic criterion for departure from laminar flow in bearings was established by G.I. Taylor in 1923 (Ref. 3). The existing state-of-the-art can be summarized as follows:

- 1) Wilcock and Booser (Ref. 5) present empirical corrections to laminar flow equations for power loss and flow.
- 2) Smith and Fuller's solution (Ref. 6) assumes that the circumferential pressure gradient is proportional to the square of the average film velocity in an infinitely long bearing. Based on their experimental work, turbulent load performance can be explained by correcting laminar end leakage factors and power loss by means of an empirical equation.
- 3) Chou and Saibel (Ref. 7) obtain a solution for the turbulent slider bearing by assuming no side leakage and relating the fluctuating and mean velocity components by means of Prandtl's mixing length theory. The maximum pressure ($P_{\max}$), load capacity, and power loss are established for turbulent flow as a function of an experimental, empirical constant, k , i.e., a Prandtl's mixing length constant.
- 4) Tao (Ref. 8) modifies Dubois and Ocvirk's laminar short-bearing approximation (Ref. 9) by assuming a negligible circumferential pressure profile and turbulent axial flow based on the "Power Law", i.e., the average velocity in the annulus related to the pressure gradient is derived from Blasius' law of friction with the velocity profile of one-seventh power. The result is a turbulent capacity number which is a function of the Sommerfeld number (S), length-to-diameter ratio (L/D), and Reynolds number (Re).
- 5) Constantinescu (Ref. 10) also uses Prandtl's mixing length theory and establishes effective viscosities dependent on empirical constants and Reynolds number which are then applied to the short journal bearing approximation (Ref. 9). The value of the constant he selects is $k = 0.4$.
- 6) Sternlicht and Arwas (Ref. 11) establish the turbulent-flow bearing equations and solve them by using Prandtl's mixing length coefficient (k). A value of $k = 0.4$ is used, based on the work of Laufer (Ref. 12), to establish performance for infinitely long bearings, i.e., no side leakage. An empirical equation was obtained for side leakage based on calculated and actual turbulent performance. Smith and Fuller's experimental data (Ref. 6) were used.

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- 7) AiResearch (Ref. 13) used Prandtl's mixing length coefficient but selected a value of k which is a function of eccentricity ratio. By selecting a value of $k = 0.2$ at $\epsilon = 0$ and decreasing k as $\epsilon \rightarrow 1.0$, AiResearch obtained good correlation with the experimental data of Reference 6.

While the basis and reasoning in these approaches are more nearly fundamental than the preceding statements would imply, the results are as stated. None are true representations of the physical phenomenon that occurs in the film in the transition region between laminar and turbulent flow. All require empirically determined factors. A significant phenomenon is that in a single bearing under load the flow may range from laminar, through transition, to turbulent flow regimes as one progresses around the clearance in the bearing.

Perhaps the most significant point is whether the turbulent flow in the clearance can be treated as turbulent flow in a pipe or if Prandtl's mixing length theory can be applied to turbulent bearing flow. This must be verified experimentally.

The above analyses are for turbulent flow and unidirectional (static) loading. For turbulent flow with dynamic (pulsating) loading, including instability, little or no analysis is available. In high-speed space turbomachinery operating on liquid-mercury working fluids, the fluid-film bearings are subjected to turbulent flow with dynamic loading.

C. TRW'S DESIGN APPROACH

Because the available analyses and the understanding of the turbulent flow phenomenon in high-speed, fluid-film bearings were even less well known at the start of the SNAP 2 bearing investigation in 1957 than they are today, the work reported in TRW ER-4102 (Ref. 1) was based on gross corrections to laminar flow design equations for turbulence. The results reported in Reference 1 show that turbulent flow bearings can be designed to operate with reasonable success, despite the lack of physical understanding. That this situation is not uncommon is evident from the experience of other investigators. The basic parameters of importance in design of the bearing are listed below:

- 1) Load capacity as defined by a generalized bearing number such as the reciprocal Sommerfeld number
- 2) Attitude angle-eccentricity ratio locus as associated with load capacity, film stiffness, and stability
- 3) Lubricant flow rate as affected by bearing geometry, speed, supply pressure, temperature, and load
- 4) Power loss as affected by bearing geometry, speed, temperature, and load

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Because these parameters are not easily expressed for rotating loads, the analysis was generally confined to unidirectional loads.

Lacking verified analytical techniques for predicting performance of turbulent-flow bearings, TRW utilized the theories that were available to design the early SNAP 2 bearings. These theories were then modified and improved by using semi-empirical turbulent factors obtained by comparing experimental and predicted performance. The hydrodynamic bearings lent themselves to this approach because laminar performance could be predicted with good accuracy. Consequently, the plain cylindrical bearing and the purely hydrodynamic tilting-pad bearing were carefully analyzed using available techniques which were improved and generalized by experimental results. Unfortunately, the hybrid bearings, i.e., partially hydrodynamic and partially hydrostatic, did not lend themselves to a similar approach since laminar flow predictions were not based on rigorous analytical techniques. Consequently, the three-sector and three-pad bearings were treated as hydrodynamic partial arc bearings approaching the Sommerfeld bearing to account for the high supply pressure.

1. Plain, Cylindrical Bearing

a. Initial Approach - Modified Laminar Case

The initial design approach used at TRW was based on the short journal bearing approximation corrected for the effects of turbulence on flow and power loss.

Load-Carrying Capacity

As a first approximation the equations derived by DuBois and Ocvirk were used since they predicted larger load capacities than were measured experimentally. Since the length-to-diameter ratio of the plain bearing was less than unity, it qualified as a short bearing.

$$\frac{1}{C_n} = \frac{1}{S} \left(\frac{D}{L} \right)^2 = \frac{\pi \epsilon}{(1 - \epsilon^2)^2} \sqrt{\pi^2 (1 - \epsilon^2) + 16 \epsilon^2} \quad \dots 3$$

where $\frac{1}{S} = \left(\frac{W}{LD} \right) \cdot \left(\frac{1}{\mu N'} \right) \cdot \left(\frac{C_D}{D} \right) \quad \dots 4$

Attitude Angle

For consistency, the equation derived for attitude angle in terms of eccentricity ratio for short journal bearings was used. Again, the experimental results of Dubois and Ocvirk deviated from the results predicted by their equation in the same direction as turbulence would cause a difference from the laminar case. The observed attitude angle for a given eccentricity was less than would be indicated by their equation:

$$\phi = \arctan \frac{\pi}{4} \frac{\sqrt{1 - \epsilon^2}}{\epsilon} \quad \dots 5$$

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Flow Rate

The empirical equation of Wilcock and Booser (Ref. 5) for turbulent bearings was used:

$$Q_{\text{tot}} = Q_{\text{tr}} + \frac{60 N' - N_{\text{tr}}}{N_{\text{tr}}} (Q_{\text{tr}} - Q_0) \quad \dots 6$$

Where:

N_{tr} = shaft speed at which flow ceases to be laminar according to G. I. Taylor's criterion

$$N_{\text{tr}} = 1570 \frac{\nu}{\sqrt{DC_D^3}} \text{ rpm} \quad \dots 7$$

$$Q_{\text{tr}} = q \cdot \frac{\pi}{120} DC_D LN_{\text{tr}} \epsilon \text{ in.}^3/\text{sec} \quad \dots 8$$

In Equation 8, the eccentricity ratio corresponded to that at the transition speed N_{tr} :

$$Q_0 = \frac{\pi}{48} \frac{DC_D^3 P_s}{\mu L} \text{ in.}^3/\text{sec} \quad \dots 9$$

which is the Hagen-Poiseuille equation for flow through the ends of a concentric annular passage with the source at the mid-length.

Since no definitions for Q_{tr} and Q_0 were specified by Wilcock and Booser (Ref. 5), those defined by Equations 8 and 9 were used.

Power Loss

An empirical correction factor was used to correct the laminar computation for turbulence. This multiplying "j" factor was established to be a function of the rate $60 N'/N_t$ and was applied to Petroff's equation for concentric bearings. Correction for eccentricity was made using the results of Dubois and Ocvirk.

$$H = j \cdot \frac{7\mu LD^3 N'^2}{C_D} \cdot \frac{1}{\sqrt{1 - \epsilon^2}} \text{ watts} \quad \dots 10$$

Although this technique is simple, its validity is obviously questionable since the use of the empirical factors was extrapolated well beyond the test conditions for which they were derived. Moreover, these factors give no insight as to the phenomenon of nonlaminar or turbulent flow.

b. Modified Approach After Smith and Fuller

To overcome the shortcomings of this preliminary approach and to account for the variations from actual performance, the analytical technique was revised to include the conclusions derived from the work of Smith and Fuller (Ref. 6). Although these conclusions do not give insight into the microscopic mechanism of turbulent flow in bearings, the results provide a reasonable correlation between the laminar prediction and the observation of turbulent performance. Professor D. D. Fuller helped establish new design equations based on the SNAP 2 experimental observations.

Load-Carrying Capacity

In the Smith and Fuller experiment it was observed that as the shaft speed increased beyond the critical value, i.e., beyond the laminar region, the load-carrying capacity of the bearing ($L/D = 1$) approached that of an infinitely long, laminar bearing having 180 degrees of active arc. This observation was used to correct the end leakage factors for turbulence (as shown in the family of curves on page 202 of Ref. 14). Thus,

$$(\eta_{L/D})_t = (\eta_{L/D})_{\infty} \cdot \frac{1}{(\eta_{L/D=1})_{\infty}} \quad \dots 11$$

Since lubricant is supplied to the SNAP 2 bearings at high inlet pressures so that the film condition theoretically approaches that of the Sommerfeld bearing, the load-carrying capacities obtained (as shown in the family of curves on page 204 of Ref. 14) were doubled. The capacities given in the reference are for partial bearings, since it is assumed that the film pressure on the unloaded side of the bearing cannot be below ambient. With high supply pressure it is thought that the Gumbel half bearing ($2\beta = 180$ degrees) assumed by Smith and Fuller will approach the full Sommerfeld bearing ($2\beta = 360$ degrees). Since for laminar flow

$$\frac{1}{S} = \pi A_{\beta=90 \text{ degrees}} \quad \dots 12$$

Where:

$$A = \frac{132}{\eta_{L/D}} [1000 \text{ m}]^2 \frac{P}{zN} \quad \dots 13$$

Therefore for turbulence and high supply pressure

$$\frac{1}{S} = 2 \cdot (\eta_{L/D})_t \cdot \pi A_{\beta=90 \text{ degrees}} \quad \dots 14$$

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The alternate method of analyzing load capacity based on Smith and Fuller's work is to assume that all the high inlet pressure contributes is an internal load on the shaft and to include merely the end leakage correction. This approach results in a turbulent Sommerfeld number exactly half of that resulting from the above equation:

$$\frac{1}{S} = \left(\eta_{L/D} \right)_t \cdot \pi A_{\beta = 90^\circ} \quad \dots 15$$

The influence of the high supply pressure, as indicated later, will then merely be part of the total applied load, i.e., sum of internal plus external loads.

Attitude Angle

Available SNAP 2 experimental data indicated that the best fit for the attitude-eccentricity locus was one derived for a 360 degree gas bearing (Ref. 15). Aside from the fit, the characteristic of the gas bearing locus (showing attitude angles approaching 90 degrees for small ϵ) matched that predicted for turbulent flow liquid bearings. The particular case selected was for an $L/D = 1.0$ and $\Lambda = 6 \mu UL/P_S = 0.06$. It should be noted the Raimondi's compressibility number λ equals $\Lambda/6$, although Λ is in more nearly common usage as the compressibility number. In Reference 15, the low value of the compressibility number for the case selected is shown to make the gas bearing solution applicable to liquid bearings with no discrepancy. Since the solution is a numerical one, no equation is presented here.

Flow Rate

Because the results of Smith and Fuller did not include data on flow and did not present a theoretical expression, flow rate was not predicted analytically but was empirically established on the basis of prior experiments.

Power Loss

Although the modifications which had been made to arrive at the assumed bearing characteristics were a departure from the experimental results of Smith and Fuller, their observed friction factor was used to estimate power loss. Thus, in the turbulent region

$$H_t = 0.43 \frac{\rho D^4 L N^3}{Re^{0.43}} \text{ watts or } H_t = 0.039 Re^{0.57} H_l \quad \dots 16$$

More detailed experimental results indicate this technique is still empirical and does not have a good foundation in the physics of turbulent flow. A specific deficiency is the omission of Reynolds number effects on load-carrying capacity and attitude angle.

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c. Simplified Turbulent Theory After Tao

To include the influence of Reynolds number, which the Smith and Fuller approach does not, Tao's work (Ref. 8), a modification of the laminar short journal bearing approximation, was used. The change from laminar to turbulent flow is accomplished by relating "the average velocity (in the axial direction) in the annulus ... to the pressure ... gradient ... derived from Blasius' law of friction with the velocity profile of one-seventh power." The subsequent mathematics is considerably more complex than for the laminar case.

Load-carrying Capacity

$$\frac{1}{S} = \left(\frac{L}{D}\right)^{2.75} \cdot Re^{0.75} \cdot \frac{0.326 \pi \epsilon^{1.75}}{16 \times 3.75} \sqrt{A^2 + B^2} \quad \dots 17$$

Where:

$$A = \sqrt{\pi} \frac{3 \epsilon}{4 - \epsilon} \cdot \frac{\Gamma(1.375)}{\Gamma(1.875)} \cdot F(2.5; 2; 2.875; \epsilon^2) \quad \dots 18$$

$$B = \sqrt{\pi} \frac{\Gamma(1.875)}{\Gamma(2.375)} \cdot F(1.5; 2; 2.375; \epsilon^2) \quad \dots 19$$

$$Re = \frac{\pi DN' C_D}{2 \nu} = \frac{U C_D}{2 \nu} \quad \dots 20$$

where $\Gamma(x)$ is gamma function and $F(a; b; c; x)$ is hypergeometric function.

Tao presents his solution of eccentricity ratio as a function of the dimensionless quantity

$$S \left(\frac{L}{D}\right)^{3-n} (Re)^{1-n} \quad \dots 21$$

For turbulent flow, $n = 0.25$. For laminar flow, $n = 1$ and the quantity above reduces to the $S(L/D)^2$ which is the capacity number. Hence for turbulence Tao's dimensionless quantity is

$$S \left(\frac{L}{D}\right)^{2.75} Re^{0.75} \quad \dots 22$$

where $S = \frac{\mu N}{P} \left(\frac{D}{C_D} \right)^2$. . . 23

Attitude Angle

$\phi = \text{arc tan } B/A$. . . 24

Flow Rate

Although no equation for flow rate was presented in Tao's paper, there is reason to believe that analogous operations as performed in the Dubois and Ocvirk paper will yield a flow factor. However, the relationship between this factor and the modified Sommerfeld number would have to be established.

Power Loss

Since Tao's assumption concerning the velocity profile in the film in the circumferential direction is that assumed in the laminar short bearing approximation case, Dubois and Ocvirk's equation for power loss results. That this equation yields power loss values which are too low for turbulent bearings has been shown by other investigators.

d. Turbulent Theory After Constantinescu

One of the more active investigators in the field of turbulent flow bearings is V. N. Constantinescu (Ref. 10). Unfortunately, few of his papers have appeared in English translation, and those that have are not clear in their definition of symbols. Despite this drawback, the basis of his analysis appears more nearly fundamental than that of the three investigations already mentioned. The inclusion of "apparent" Reynolds stresses in the Navier-Stokes equation, which are estimated using Prandtl's mixing length theory, is basic. Constantinescu's result is a modified Reynolds equation:

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu k_z} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu k_z} \right) \frac{\partial p}{\partial z} = \frac{U}{z} \frac{\partial h}{\partial x} \quad . . . 25$$

The coefficients k_x and k_z are functions of an empirical constant and Reynolds number of the form

$$k_i = 12 + a_i k^2 \left(\frac{Uh}{\nu} \right)^{\alpha_i} \quad . . . 26$$

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From this basis, Constantinescu presents solutions in which $\partial p / \partial x \approx 0$; i.e., for a short bearing. When compared to the results of Smith and Fuller, his prediction is in remarkably good agreement for both load-eccentricity ratio and attitude angle-eccentricity ratio. To estimate power loss, Constantinescu uses Equation 16. Unfortunately, the method proposed is difficult to apply without access to his previous work from which some of the necessary curves were derived.

2. Tilting-Pad Bearings

Tilting-pad bearings were selected for evaluation because they are inherently stable, are capable of tolerating misalignment and distortion, and can operate without high supply pressures.

a. Initial Approach - Laminar Flow

For the first round of calculations the procedure was to adhere to laminar flow conditions with the objective of modifying the results later to make them more representative of the turbulent flow conditions that do prevail in the bearing clearance space. This approach was considered satisfactory since the major performance predictions based on laminar flow would be conservative in each case. Four specific bearing and system characteristics were investigated:

- 1) Load-carrying capacity
- 2) Pitching and rocking frequency of individual shoes
- 3) Synchronous whirl frequency of undeflected shaft in the journal bearing
- 4) Friction losses

The occurrence or onset of fractional-frequency whirl was not investigated since the individual shoes tilt until the resultant force is in line with the applied force, i.e., normal to the shaft. Consequently, there are no tangential forces which could cause instability. Turbulence in the clearance space is assumed not to alter the resultant locus of the shaft center under load from the locus in the laminar case.

One further important assumption made in the initial approach was that with the shaft at an eccentricity ratio of zero, i.e., in the center of the bearing, the center of the shaft coincides with the center of curvature of each shoe. This means that the "pre-load" was zero.

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Load-Carrying Capacity

From page 176 of Reference 14:

$$W = \frac{6 \mu U L^2 B K_p \eta}{h_o^2} \quad \text{lb per shoe} \quad \dots 27$$

With load applied between two pivot points, the net load capacity of the three-shoe tilting-pad bearing is

$$W_{\text{net}} = W_1 - W_2 \cos \theta_2 - W_3 \cos \theta_3 \quad \dots 28$$

since the unloaded shoes impose a load on the shaft.

Rocking Frequency of Shoe (Pitching Frequency)

From Reference 17, the natural frequency is

$$f = \frac{60}{2 \pi} \sqrt{\frac{S_A}{I}} \quad \dots 29$$

where S_A is the rocking stiffness.

Synchronous Whirl Frequency (for Undeflected Shaft)

$$f = \frac{60}{2 \pi} \sqrt{\frac{S_L}{M}} \quad \text{rpm} \quad \dots 30$$

where S_L is bearing spring rate dW/de .

Friction Loss

From page 180 of Reference 14, frictional force is given by

$$F = \mu B L \frac{U}{h_o} k_{fr}, \quad \text{lb per shoe} \quad \dots 31$$

b. Analytical Technique Including Pre-Load

To avoid flutter of the unloaded shoes as a result of the presence of a diverging-converging film instead of a converging-diverging film, self-aligning bearings are generally assembled with pre-load. Consequently, the center of the shaft does not coincide with the center of curvature of the individual shoes. In fact, with the shaft located in the center, the pre-loaded tilting pad is identical to a multiple "lobe" bearing except that each "lobe" or pre-loaded shoe is free to tilt.

The built-in pre-load results in different performance characteristics. The difference in load-carrying capacity is most noticeable. Consequently, a method for analyzing pre-loaded, self-aligning bearings was developed at TRW. The analytical procedure developed was based on the theory of a crowned slider bearing. Laminar flow was still assumed. The crowned slider theory was presented in a paper by Abramovitz (Ref. 18). To apply his theory to the pre-loaded tilting-pad bearing it was necessary to assume that each pad acts as a crowned slider bearing. The Abramovitz assumption of negligible side leakage was modified to include the effect of flow in that direction. The analytical procedure used to establish load capacity, including the influence of pre-load, is given in Appendix C.

The final results of total (or net) load-carrying capacity, total power loss, and bearing stiffness were expressed in dimensionless form as functions of eccentricity ratio, ϵ , and pre-load number, m , where

$$\epsilon = \frac{e}{C_R} \quad \dots 32$$

$$m = \frac{C_{MR}}{R} \quad \dots 33$$

Load-Carrying Capacity

The load-carrying capacity of the pre-loaded tilting-pad journal bearing is the vector summation of the forces exerted by all the pads. It is assumed that these forces are always normal to the pivot points, resulting in a shaft locus that is always in line with the direction of the applied load. Then the bearing load capacity is given by

$$W = \sum_{i=1}^n W_i \cos \theta_i \quad \dots 34$$

where θ_i is the angle measured clockwise from the applied load W to the pad pivot point:

$$W_i = B \int_0^1 P L dx_{1_i} \quad \dots 35$$

Solving for P, integrating, and non-dimensionalizing

$$\bar{W}_{\text{tot}} = \frac{W_{\text{tot}}}{\mu N R L f_s} \left(\frac{C_{MR}}{R} \right)^2 = \frac{\pi^3 \theta_i C_p [a(1 - \bar{x}_1) + \bar{x}_1]^2}{9.72 \times 10^5 \left[1 - \left\{ m + \epsilon (1 - m) \cos \theta_p \right\} \cos \frac{\theta_i}{2} \right]^2}$$

Friction Power Loss

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The total bearing friction power is the algebraic summation of the power loss of all pads:

$$H = \sum_{i=1}^n H_i \quad \dots 37$$

Solving for H_i , integrating, and non-dimensionalizing:

$$\bar{H}_{\text{tot}} = \frac{H_{\text{tot}}}{\mu R^2 N^2 L} \left(\frac{C_{MR}}{R} \right) = \frac{2.16 \times 10^{-3} [a(1 - \bar{x}_1) + \bar{x}_1]}{\left[1 - \left\{ m + \epsilon (1 - m) \cos \theta_p \right\} \cos \frac{\theta_i}{2} \right]} \quad \dots 39$$

Fluid Film Stiffness

The fluid film stiffness, which in the case of the tilting-pad bearing is identical to the radial film stiffness since the attitude angle is always zero, is given in the usual manner:

$$K = \frac{dW}{de} = \frac{1}{C_R} \cdot \frac{d \left(\sum_{i=1}^n W_i \cos \theta_i \right)}{d \epsilon} \quad \dots 40$$

Solving for W_i , integrating, and non-dimensionalizing:

$$\bar{K} = \frac{K}{\mu N L f_s} \left(\frac{C_{MR}}{R} \right)^2 \left(\frac{C_R}{R} \right) \quad \dots 41$$

c. Effect of Turbulence

Prior to verifying the experimental performance of a variety of three-shoe tilting-pad bearings, the influence of turbulence on performance was partially ignored except that it was considered to result in a comfortable margin of safety. The work of Chou and Saibel (Ref 7) was used to predict tilting-pad journal bearing performance including turbulence but neglecting pre-load. Details of this computer program analysis are presented in Appendix D. Appendix D also shows the method employed to extend the usefulness of Chou and Saibel's approach by deriving a semi-empirical equation for the turbulent factor k as a function of fluid density and Reynolds number.

The entire analytical technique for predicting turbulent-flow, tilting-pad bearing performance has been revised as a result of the measured performance. This is discussed in Volume III.

3. Three-Sector (Hybrid) Bearing

For a number of reasons, based on approximate solutions and on qualitative judgments, much of the SNAP 2 bearing investigation has centered around the three-sector bearing. From an analytical point of view, the end-fed, plain cylindrical bearing would have been preferred. However, the undesirable characteristics of the plain bearing, e. g., demonstrated susceptibility to half-frequency whirl and high supply pressure requirements for flow, made consideration of the three-sector bearing necessary. The initial design approach was based on Fuller's method for the plain bearing described in Paragraph C-1-b.

Load-Carrying Capacity

The end leakage factors for a laminar flow bearing are obtained from the family of curves on page 202 of Reference 14. The load-carrying capacity for a 120 degree ($\beta = 60$ degrees) arc partial bearing is obtained from the family of curves on page 204 of Reference 14. Assuming that the other two partial bearings contribute in a minor way to the load-carrying capacity, the single 120-degree partial bearing is used. By applying the correction defined by Equation 15, we have:

$$\frac{1}{S} = (\eta L/D)_t \cdot \pi A \beta = 60^\circ \quad \dots 42$$

A particular problem posed by the geometry of the three-sector bearing is that the load direction influences its performance. As a limiting case, the performance of the bearing with the load direction into the groove is of interest. Pinkus (Ref. 4) had reported loading into only one of the sectors.

Attitude Angle

The attitude-eccentricity ratio which appeared to best fit the locus implied from initial SNAP 2 test data on stability was that derived by Pinkus (Ref. 4) for a three-sector hydrodynamic bearing. The assumption was made that Equation 42 adequately defined the load-carrying capacity. Attitude angle correction for turbulence is made by assuming that the three-sector bearing is affected in the same manner as the plain bearing:

$$\phi_{3St} = \phi_{3S\ell} \text{ (Pinkus)} \cdot \frac{\phi_{Pt} \text{ (Smith \& Fuller) Ref. 6}}{\phi_{P\ell} \text{ (Pinkus and Sternlicht) Ref. 16}} \quad \dots 43$$

Flow Rate

As a first-order approximation, it was assumed that there was no end leakage and that the shaft and bushing were concentric. Further, it was assumed that lubricant was not carried from one pad to the next, so that total bearing flow is

$$Q_{\text{tot}} = \frac{3\pi}{4} DL C_D N' \text{ in.}^3/\text{sec} \quad \dots 44$$

Power Loss

The semi-empirical expression for the turbulent plain bearing shown in Equation 16 was used to predict frictional power loss.

Although the procedures for predicting performance are somewhat arbitrary, these procedures attempted to allow for the interaction between hydrodynamic and hydrostatic action, effects of bearing geometry, influence of high supply pressure, and the varying flow regimes within the clearance space of the bearing.

4. Three-Pad (Hybrid) Bearing

To improve the stability characteristics of the fixed geometry (plain and three-sector) bearings mentioned above, design modifications were made to utilize the available high pressure lubricant supply, thereby increasing the stabilizing hydrostatic component. This was accomplished by eliminating the three axial grooves from the three-sector bearing which permitted the internal pressure to bleed down to the drain pressure at three locations in the bearing. Stable operation was further enhanced by using orifice-compensated supply inlets.

The resultant design, designated three-pad, was between a plain cylindrical bearing and a three-sector bearing. It contained three compensated supply orifices with a recessed pad around each, separated by 120 degrees of uninterrupted bearing surface. Diameter, length, and the recessed supply pads were identical to those in the three-sector bearing.

The analysis for this partially hydrostatic, partially hydrodynamic (hybrid) bearing was based on the same assumptions made for the plain and three-sector bearings.

Load-Carrying Capacity

The laminar end leakage factors for this bearing were corrected for turbulence as they were for the three-sector bearing. However, since this bearing had no axial grooves, the effective area of the bearing was considered to be 180 degrees ($\beta = 90$ degrees), i.e., the same as a 360-degree plain bearing. Applying this correction resulted in the following equation:

$$\frac{1}{S} = (\eta L/D)_t \cdot \pi A \beta = 90^\circ \quad \dots 45$$

To compensate for the hydrostatic influence and its contribution to load-carrying capacity, the same approach as used for the high ambient pressure in the plain cylindrical bearing was applied. Therefore for high values of supply pressure, where the three-pad bearing approaches the full Sommerfeld bearing, Equation 45 is modified as follows:

$$\frac{1}{S} = 2 (\eta L/D)_t \cdot \pi A_{\beta = 90^\circ} \quad \dots 46$$

Attitude Angle

The attitude-eccentricity locus selected for this turbulent hybrid bearing was based on the assumption that the bearing geometry simulated that of a pressurized plain journal bearing. However, the orifice compensation and its tendency to restore the shaft in the direction of its displacement resulted in a narrow locus which tended to overcome the influence of turbulence. Consequently, in the absence of a measured locus the values selected corresponded to a partial bearing with a 120-degree arc.

Flow Rate

Since the flow rate was a function of the number of inlet holes, among other things, the value was estimated based upon the assumptions made for the plain cylindrical bearing but increased by a factor of three.

Power Loss

The semi-empirical expression for the turbulent plain journal bearing (Equation 16) was used to predict power loss.

5. Dynamic Characteristics in Journal Bearings

In addition to the difficulties associated with mathematical description and analytical solution of the behavior of the bearing under dynamic loading, the inconsistent definitions of the instability phenomena by various investigators complicate the study of the dynamic behavior of fluid-film bearings. In the discussion that follows, attention is paid to steady-state rotating loads at shaft frequency, natural frequency of the bearing-shaft combination as a simple spring-mass system, and self-excited hydrodynamic instability or half-frequency whirl.

The basic form of the Reynolds equation which applies in these cases for a fixed bearing and rotating shaft (Ref. 16) is

$$\frac{\partial}{\partial \theta} \left(\frac{h^3}{\mu} \frac{\partial P}{\partial \theta} \right) + \frac{D^2}{4} \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial P}{\partial z} \right) = \frac{3}{4} C_D \left(\frac{D}{C_D} \right)^2 \left[-\epsilon (\omega - 2\dot{\alpha}) \sin \theta + 2\dot{\epsilon} \cos \theta \right] \quad \dots 47$$

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$\dot{\alpha}$ = angular velocity of the line of centers

$\frac{C_D}{2} \dot{\epsilon}$ = linear velocity of the shaft center with respect to the bearing center along the line joining them

θ = angle to the line of centers from a fixed reference axis

In addition to the inherent difficulty of obtaining an analytical solution to the Reynolds equation, the coupling between bearing and shaft and the cross-coupling between bearings make the solutions of even simplified cases difficult to obtain; for example:

- 1) Rotating loads at shaft frequency (as produced by shaft unbalance) will cause the shaft center to orbit around some fixed point. As a first approximation where the unidirectional load is small, the radius of the orbit can be assumed equal to the eccentricity which would result if the magnitude of the rotating load were applied as a unidirectional load. For large, unidirectional loads a greater error is introduced by this assumption, although the effect of increasing load-carrying capacity due to squeeze film will tend to reduce the error. Solutions in which more detailed considerations are given to the relationship between the kinematics of the shaft center and hydrodynamic pressures are available for simplified geometries (Ref. 16). However, their use requires that $\dot{\alpha}$ and $\dot{\epsilon}$ be specified. Since these parameters are functions of the shaft-bearing system, the required matching of the motion to the forces is a trial and error procedure. For the purpose of initial design, the first approximation mentioned above gives adequate results.
- 2) The natural frequency of the system consisting of the "elastic" lubricant film and the shaft mass acting at the bearing is defined by analogy to the simple spring-mass system. This "rigid body critical," "inversion speed," or "synchronous whirl frequency," as it is variously called, can be expressed quasi-statically as:

$$\omega_{cr}^2 = \frac{K}{M} \text{ rad}^2/\text{sec}^2 \quad \dots 48$$

where K is the "spring rate" of the fluid film.

The tangent of the load (W) versus eccentricity ratio (ϵ) curve, or the reciprocal Sommerfeld number ($\frac{1}{S}$) versus eccentricity ratio (ϵ) curve represents the instantaneous spring rate of the fluid film. Consequently the spring rate (K) is proportional to the slope as follows:

$$\text{slope} = \frac{d \left(\frac{1}{S} \right)}{d \epsilon} \quad \dots 49$$

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$$\text{But } \frac{1}{S} = \frac{P}{\mu N'} \left(\frac{C_D}{D} \right)^2 = \frac{60 W}{\mu N L D} \left(\frac{C_D}{D} \right)^2 \quad \dots 50$$

$$\text{and } \epsilon = \frac{2 e}{C_D} \quad \dots 51$$

$$\therefore \text{ slope } = \left(\frac{C_D}{D} \right)^2 \frac{60 W}{L D \mu N} \frac{C_D}{2 e}$$

$$\text{or } \text{ slope } = \left(\frac{C_D}{D} \right)^3 \frac{30}{L \mu N} \left[\frac{W}{e} \right] \quad \dots 52$$

but $\frac{W}{e}$ is the stiffness K

$$\therefore K = \text{ slope } \left(\frac{D}{C_D} \right)^3 \left(\frac{\mu L N}{30} \right) \quad \dots 53$$

- 3) Half-frequency or fractional-frequency whirl is caused by hydrodynamically generated forces in the film which cause the axis of a lightly loaded shaft to orbit around the bearing axis at a frequency equal to half the shaft speed or less. The stability criterion established by Boeker and Sternlicht (Ref. 19) is

$$M \omega_{cr}^2 \left[\frac{1}{K_S} + \frac{1}{K_R} \right] = 4 \quad \dots 54$$

Where K_S = "spring rate" of the shaft, K_R radial film stiffness or

$$M_S \omega_{cr}^2 \text{ lb/inch}$$

If it can be assumed that the shaft moves as a rigid body, i.e., $K_S = \infty$, then

$$\omega_{cr} = 2 \sqrt{\frac{K_R}{M}} \text{ rad/sec} \quad \dots 55$$

Where K_R = "stiffness" (along line of bearing and shaft center) of the fluid film or radial film stiffness.

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To establish the radial film stiffness K_R it is necessary to determine the reactive load-carrying capacity of the film in the radial direction. Designating this radial force as F_R , it represents the bearing force acting along the line joining the bearing and shaft centers. Consequently, F_R is related to the applied load W and the attitude angle ϕ (where ϕ is the angle between the load line and the line joining the bearing and shaft centers). From the following diagram it follows that $F_R = W \cos \phi$.

Observe that the second reactive force F_θ acts perpendicular to the line joining the bearing (O_B) and shaft (O_S) centers. This tangential component, $F_\theta = W \sin \phi$, is responsible for the half-frequency whirl instability in hydrodynamic journal bearings, unless the bearing forces such as F_R are sufficiently large to overcome it.

Representing radial film stiffness K_R in a similar manner to the fluid-film stiffness K gives:

$$\text{Slope (of } \frac{1}{S} \cos \phi \text{ vs } \epsilon) = \frac{d\left(\frac{1}{S}\right) \cos \phi}{d\epsilon} \quad \dots 56$$

$$\therefore \text{ slope} = \left(\frac{C_D}{D}\right)^3 \frac{30}{\mu NL} \cdot \left[\frac{W \cos \phi}{e}\right] \quad \dots 57$$

$$\text{but } \frac{W \cos \phi}{e} = \text{radial film stiffness } K_R$$

$$\therefore K_R = [\text{slope}] \cdot \frac{\mu NL}{30} \left(\frac{D}{C_D}\right)^3 \quad \dots 58$$

The value of K_R which corresponds to the whirl threshold can be written as:

$$K_R = \text{slope} \left(\frac{\mu N_{cr} L}{30}\right) \left(\frac{D}{C_D}\right)^3 \quad \dots 59$$

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Since the width of the bearing was small compared to the mean radius, only a small error was introduced by treating the spiral grooves as parallel straight lines instead of concentric spirals. The small error so introduced could be summarized by using polar coordinates instead of Cartesian coordinates to take into account the curvature. The analysis was based on laminar flow even though turbulent flow was predicted. Again, the resulting estimate of load capacity was conservative while the estimates of flow and power loss were optimistic.

Thrust Load Capacity

Solving the general Reynolds equation for the inclined groove bearing resulted in the following pressure profile:

$$\frac{dP}{dy} = E - RJ \quad \dots 63$$

which for incompressible flow reduces to

$$P - P_o = (E - RJ) y \quad \dots 64$$

Where E = driving force

$$= \frac{6 \mu U_o a_1 a_2 (h_1^3 - h_2^3) (h_1 - h_2) \sin 2\theta}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2\theta} \quad \dots 65$$

and R is the resistance to flow, so that

$$R = \frac{12\mu}{\rho} \frac{2(a_1 + a_2)(h_1^3 a_2 + h_2^3 a_1)}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2\theta} \quad \dots 66$$

Where J = mean mass flow in y direction.

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Applying boundary conditions as shown in Figure 10:

$$P_1 - P_o = (E - RJ) b \quad \dots 67$$

Integration of the pressure profile will establish the thrust capacity

$$\text{i.e., } T = \int PdA \quad \dots 68$$

Or for a double-acting thrust bearing

$$T = \int PdA_{\text{loaded}} - \int PdA_{\text{unloaded}} \quad \dots 69$$

Substituting for P and A and integrating results in

$$T = \frac{1}{2} \left[\frac{R' bcE (b + c) + bc (R' - R) (P_o - P_2)}{Rb + R'c} \right] \quad \dots 70$$

$$\text{where } R' = \frac{12 \mu}{h_2^3 \rho} \quad \dots 71$$

Mass Flow

Mass flow is given by

$$\int_0^h \rho u dz \text{ in } x \text{ direction} \quad \dots 72$$

$$\int_0^h \rho v dz \text{ in } y \text{ direction} \quad \dots 73$$

Substituting for u and v

$$J_{(x)} = \rho \left[-\frac{h^3}{12 \mu} \frac{dP}{dx} + \frac{h}{2} u_o \right] \quad \dots 74$$

$$J_{(y)} = -\frac{h^3}{12 \mu} \rho \frac{dP}{dz} \quad \dots 75$$

Solving for J gives

$$J = \frac{E b - (P_2 - P_o)}{Rb + R' c} \left(\frac{\text{lb}}{\text{sec-in.}} \right) \quad \dots 76$$

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Friction Loss

The friction per unit area of sliding plate is given by

$$H = \mu \left. \frac{du}{dz} \right|_{z=h} = \frac{h}{2} \frac{dP}{dx} + \frac{\mu U_o}{h} \quad \dots 77$$

Average friction $\bar{H}$ is given by:

$$\bar{H} = \frac{1}{a_1 + a_2} \int_0^{a_1 + a_2} H dx \quad \dots 78$$

Solving

$$H = \frac{\mu U_o}{(a_1 + a_2)} \left[\frac{a_1}{h_1} + \frac{a_2}{h_2} + \frac{3a_1 a_2 (h_1 - h_2)^2 \cos^2 \theta}{h_1^3 a_2 + h_2^3 a_1} \right] + \left(\frac{a_1 + a_2}{U_o \rho} \right) \frac{E}{R} (E-1.5) \quad \dots 79$$

Assuming a smooth disk and laminar flow, power loss may be established by applying the fundamental equation of viscous shear force.

$$dF = \tau dA \quad \dots 80$$

Where $\tau = \mu \frac{du}{dh}$ for a Newtonian fluid \dots 81

For laminar flow, assuming a linear velocity profile, the viscous power loss equation for a smooth disk having inside and outside radii of R_o and R_i respectively reduces to

$$H_{\ell} = \frac{2 \pi^3 \mu (N')^2}{h} (R_o^4 - R_i^4) \frac{\text{in.} \cdot \text{lb}}{\text{sec}} \text{ per side} \quad \dots 82$$

Based on the work of Constantinescu (Ref. 10) the turbulent power loss for a thrust bearing is given by:

$$H_t = H_{\ell} \times 0.009 (\text{Re})^{0.775} \quad \dots 83$$

The above laminar analysis is somewhat conservative since it assumes a smooth disk. The spiral groove thrust bearing with its recesses will therefore have a higher loss than predicted above.

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D. STATUS OF LIQUID MERCURY LUBRICATED BEARING SYSTEMS

The high rotational speed of the shaft coupled with the low kinematic viscosity of liquid mercury assures operation of the SNAP 2 bearings beyond the laminar flow regime. During the period covered by this report the assumption was that flow at the resultant Reynolds numbers was turbulent. But based on the experimental work of Smith and Fuller and work at TRW, it became apparent that fully developed turbulence did not exist throughout the bearing. At large eccentricity ratios the bearing approached laminar performance, and the influence of high pressure supply pads and drain grooves minimized the influence of Reynolds number.

The existence of laminar, laminar plus vortex, turbulent plus vortex, and turbulent flow is now considered to be possible in the bearing. The actual flow may be a combination depending on the type of bearing (hydrodynamic or hybrid), the operating characteristics (Reynolds number, eccentricity ratio, etc.), and the type of lubricant. How these variables affect the flow regime cannot currently be assessed. However, on the basis of information available on rotation of cylinders, the assumption of similarity between high speed lubricant flows and turbulent channel flows seems unreasonable at the present time.

Fortunately the experimental data obtained during this report period are not dependent on the type of flow that exists. Furthermore, the design curves and the semi-empirical equations derived from the experimental data to help predict bearing performance are not based on the presence of turbulent or any other type of flow. Consequently, the data, the design curves and the semi-empirical equations presented in this section and throughout the report are valid for the type of bearings to which they apply and over the range for which they were obtained. General equations describing bearing performance which are limited to a specific operating range or bearing geometry must await information on the type of flow and the influence on the flow regime of such major variables as grooves, supply pads, supply pressures, operating eccentricity ratio arc length, and type of loads.

The lack of rigorous analytical design techniques and complete understanding of the flow regime in the clearance space was not a deterrent to developing several successful bearings for mercury Rankine machinery. Despite these unknowns and the difficulties arising from operation with mercury as the lubricant, the feasibility of mercury-lubricated fluid film bearings was demonstrated by TRW in runs of 2500 hours with hydrospheres and > 1000 hours with journal bearings. (In the meantime, single sets of journal bearings have been operated in excess of 6000 hours.)

1. Analytical Status

The design approach at TRW has progressed from the use of empirically corrected laminar flow equations to the use of more nearly fundamental considerations of

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superlaminar (turbulent) effects. TRW's approach to turbulent flow bearing design is described in Paragraph III-C of this report. When coupled with an accurate experimental program, these techniques provide TRW with useful and satisfactory design guides.

Analysis of experimental data compiled during this report period, and presented in Volumes II through VI, provided information which was used to evaluate conclusions by investigators in the field. In addition, TRW's approach was modified and improved as well as extended to include both hydrodynamic and hybrid bearings of different types. The following paragraphs will summarize these conclusions and the useful techniques available at the present to predict performance of liquid-mercury-lubricated bearings.

- 1) Reasonable correlation was achieved with Wilcock and Booser's empirical power loss equation for plain journal bearings over a narrow clearance range.

$$H_T = j H_L \quad \dots 84$$

For hybrid bearings the trend of power loss with clearance predicted by the above equation was opposite to the measured variation. Whereas Wilcock and Booser's equation predicted an increase in power loss with increasing clearance, the actual trend was in the opposite direction. Similarly the trend in flow predicted by these investigators was opposite to that measured.

- 2) Smith and Fuller's approach to modified end leakage resulted in a load-carrying capacity which was in good agreement with actual performance for the plain journal bearing, but only over a narrow Reynolds number range, i.e., $Re \approx 6000$. Based on their experimental work the end leakage term corrected for turbulence, (see Equation 11)

$$(\eta_{L/D})_T = \left(\eta_{L/D} \right)_L \cdot \frac{1}{(\eta_{L/D} = 1)_L}$$

suffered because the resultant predicted performance is independent of Reynolds number. Experimental data indicate that the turbulent load capacity is a function of

$$W = f \left[L ; D ; \frac{1}{C_D} ; \mu ; \eta ; (Re)^a \right] \quad \dots 85$$

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The end leakage term, η , must be obtained as a function of Reynolds number to improve the above approach. Good agreement was obtained using their experimentally determined Equation 16 for turbulent power loss

$$H_T = 0.039 (\text{Re})^{0.57} \times \text{laminar power loss (Petroff)}$$

The above equation was verified for both hydrodynamic and hybrid bearings and followed the measured trend as clearances were varied.

- 3) The usefulness of Chou and Saibel's approach in predicting load capacity of slider bearings was increased by applying it to a curved slider and establishing the relationship of Prandtl's mixing length constant as a function of the lubricant density and Reynolds number as follows:

$$k = f \left[\sqrt{\rho}; (\text{Re})^b \right] \quad \dots 86$$

From a series of experimental tests on self-aligning hydrodynamic bearings covering a variety of clearances and speeds, the actual equation of "k" as a function of density and Reynolds number was

$$k = 3.25 \times 10^{-6} \sqrt{\rho} \times (\text{Re})^{0.46} \quad \dots 87$$

However, this same series of tests indicated that the above empirical equation was limited to a range of Reynolds numbers from 5000 to approximately 15,000. No data were available at higher Reynolds numbers. At lower values the relationship broke down.

- 4) After establishing an apparent influence of Reynolds number on load capacity, the experimental data were compared to the predicted performance based on Tao's modified Sommerfeld number relationship (see Equation 21):

$$S (L/D)^{3-n} (\text{Re})^{1-n}$$

Correlation was not good. At low eccentricity ratios the predicted load capacity was conservative, indicating that the assumption for a negligible circumferential pressure gradient was not realistic. At large eccentricity ratios, the predicted load performance was optimistic, indicating that it might be the Reynolds number based on minimum film thickness that controls, for at large shaft displacements the performance approaches that of a laminar bearing. Smith and Fuller's experimental work indicated that for $\epsilon \rightarrow 1.0$, the performance approaches that predicted by laminar analysis.

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- 5) The approach of Constantinescu and Sternlicht and Arwas gave reasonable load capacity prediction using a Prandtl's mixing length constant of 0.4. However, agreement of hydrodynamic bearing capacity was good only at small values of ϵ . This was to be expected since the value of Prandtl's coefficient and the end leakage term were based on matching Smith and Fuller's experimental data. However, at larger shaft displacements (at $\epsilon > 0.5$) the value of the coefficient seems to behave as predicted by AiResearch:

$$k = f(\epsilon)$$

or $k = f(\text{Re based on minimum film thickness}) \quad \dots 88$

- 6) This general approach was expanded by TRW to establish an empirical turbulent flow equation which presents an overall turbulent load factor, K , as follows:

$$K = C(\text{Re})^a \quad \dots 89$$

Where

$$K = \frac{W_T}{W_L} \quad \dots 90$$

Using the experimental plain journal bearing data, the value of K was found to be

$$K = \frac{W_T}{W_L} = 0.0301 (\text{Re}_{ho})^{0.553} \quad \dots 91$$

Observe that the value of K is a function of Reynolds number at the point of minimum film thickness, h_o .

The hydrodynamic tilting-pad bearing data were similarly reduced to establish the overall turbulent load factor, K . For this type of bearing

$$K = 0.0397 (\text{Re}_{h2})^{0.589} \text{ for } B/L = 0.75 \quad \dots 92$$

For $B/L \simeq 0.6$, the curve of K vs Re_{h2} was almost identical indicating a weak dependence of B/L ratio. (The value of K was obtained as a function of trailing edge film thickness of the loaded pad. Different constants exist for leading edge or minimum film thicknesses.)

The above technique can be extended to establish factors for power loss, flow, attitude angle, or any other independent bearing parameter. To broaden the utility of the technique values of turbulent factors, K can be established for different L/D ratios and for different types of hydrodynamic bearing (varying arcs, for instance). The net result would be tables of turbulent factors which are influenced by eccentricity ratio and L/D ratios. The objective is to obtain the equation of K as a function of Reynolds number and L/D ratio, which would then include the effects of the flow regime in the clearance space and the end leakage under those conditions.

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$$K = f \left[(Re)^n; (L/D)^m \right]$$

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The final equation will be similar to Tao's modified Sommerfeld number. The usefulness of this intermediate step in being able to analyze and predict performance of super-laminar flow bearings must be verified by many more carefully conducted and controlled tests on a variety of basic bearing types of different sizes.

The above approach represents an intermediate step in the evolution of more fundamental, but still useful, theories and techniques to predict performance of bearings operating at high Reynolds numbers, beyond the laminar regime. It is not suggested as a replacement to fundamental theory, but merely as a useful tool to assist in design and prediction during the interim between the need to develop bearings and the time when fundamental and general analytical techniques become available.

Hybrid bearings may be treated the same as hydrodynamic bearings. However, laminar performance predictions which include the influence of speed (hydrodynamic effects) on the hydrostatic performance are still in preliminary stages with many simplifying assumptions to permit solution of the complicated fluid equations. Consequently, experimental verification of the laminar performance of hybrid bearings is still unavailable.

To overcome this limitation, the experimental data of the different types of hybrid bearings were non-dimensionalized to establish design performance curves. These design curves were modified for operating Reynolds number to account for the non-laminar flow regime influence.

Non-dimensional plots of modified bearing number versus modified Sommerfeld number were generated to account for load capacity. These non-dimensional parameters are defined as:

$$\Lambda^* = \frac{6 \mu \omega}{\Delta P} \left(\frac{D}{C_D} \right)^2 C' (Re)^{a'} \quad . . . 94$$

and

$$\frac{1}{S_n} = \frac{W'}{\mu N'} \left(\frac{C_D}{D} \right)^2 \quad . . . 95$$

Where

ΔP = supply pressure drop across bearing

$$C' = 10^{-2}$$

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$$a' = -0.6$$

$$W' = W/L \times D$$

Similarly, non-dimensional plots of flow characteristics were developed to describe flow variation as functions of supply pressure, speed and clearance.

Flow factor q_n was defined as

$$q_n = \frac{2 Q}{\rho \pi D L C_D N} \quad . . . 96$$

and the above was plotted against the modified bearing number, $\Lambda * C'' (Re)^{a''}$, where $C'' = 100$ and $a'' = 0.6$.

The usefulness of these various design plots is that they cover relatively large ranges of speed (Reynolds number) (10,000 to 40,000 rpm), eccentricity ratio (0 to 0.8), clearance (0.0015 to 0.002 inch), load (0 to 75 lb), flow (0 to 15 lb/min), and pressure (0 to 600 psi).

Data for both types of hybrid bearings developed during this report period, three-pad and three-sector, have lent themselves to reduction to non-dimensional design plots. These plots in turn can be used to predict performance of similar bearing types.

2. Development Status

The extensive experimental and analytical program conducted at TRW has led to the development of bearings for space power systems such as SNAP 1/SPUD and Sunflower as well as SNAP 2; to evaluation and modification of available analytical techniques as discussed above; and to experience in conducting liquid-mercury-lubricated bearing programs. Figures 11, 12, and 13 show schematics of different turboalternators using liquid mercury lubricated bearings developed at TRW.

Several different bearing types have been developed. They are considered satisfactory in meeting the lubrication requirements of the different systems for which they are intended. They include hydrospheres for SNAP 1/SPUD, Figure 11, inherently compensated three-sectors for Sunflower, Figure 12, and hydrodynamic tilting-pad and hybrid orifice compensated three-sectors and three-pads for SNAP 2.

Despite the problems associated with meeting the stringent lubrication requirements of auxiliary space power systems, the feasibility of liquid-mercury-lubricated, fluid film bearings has been demonstrated by a series of long endurance tests. Examples are:

Figure 11. SNAP 1/SPUD Turboalternator

Figure 12. Sunflower Turboalternator

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Figure 13. SNAP 2 Modified CRU IV

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2500 hours with hydrospheres in SNAP 1/SPUD

> 6000 hours with three-sectors and spiral groove thrust bearings in Sunflower

> 6000 hours with three-pad and spiral groove thrust bearings in SNAP 2

These results and others confirm that there are no fundamental problems or limitations to the development of liquid-mercury-lubricated bearings. With the correct selection of bearing type and material, there is no reason known today which will prevent one year life using bearing lubricants that result in turbulent flow in the clearance space.

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IV. CORRELATION OF EXPERIMENTAL RESULTS WITH PREDICTED PERFORMANCE

Each of the bearing types investigated has advantages and disadvantages. In most instances the inherent advantages and disadvantages are the same as those for laminar flow, organic lubricants, and more nearly standard applications. However, the space power application and the resultant requirements create several other advantages and disadvantages which may have an overriding influence on whether a particular bearing type is satisfactory. These factors, which play an important role in influencing life capabilities, are discussed below:

- 1) Turbulent Flow: Flow in the bearing clearance space may be turbulent at design speed. The result is a higher load capacity and power loss and a lower flow for a given pressure differential in comparison to a laminar flow bearing. The resulting decrease in stability and increase in temperature gradient tend to lower bearing life unless counteracted by the selection of the bearing.
- 2) Instability (Half Frequency Whirl): Operation at zero gravity combined with high rotating speeds requires that the selected bearing have "built-in" provisions preventing destructive half-frequency whirl.
- 3) Critical Speed: The selected turbomachinery concept, the types of bearings, and the lubricant normally used result in passing through at least the first and frequently the second mode critical speed. Consequently, actual bearing fluid-film stiffness and damping characteristics assume relatively important roles.
- 4) Types of Loads: SNAP 2 bearings must support both unidirectional and rotating loads. Applied loads are listed in Table 1.

Since the turboalternator will impose predominantly rotating or fluctuating loads in space (often of unknown magnitude and direction), the bearing must be insensitive to these conditions. Preferential load orientation, i. e., plain journal bearing with axial supply, or pivot points (self-aligned bearings) which may wear or suffer fretting-corrosion damage constitute definite disadvantages.

- 5) Nature of the Lubricant: The high density of mercury combined with its low vapor pressure may result in cavitation in the bearing and subsequent cavitation-erosion damage. This would limit the long-term integrity of the system. Bearings which permit continuous formation and then collapse of vapor bubbles inside the clearance space will not have the necessary endurance capability even though in all other respects they are satisfactory. The high density of the lubricant, which makes consideration of fluid inertia necessary, creates problems for certain bearings where

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none would exist under normal circumstances. However, performance of a bearing type that utilizes the effect of the fluid inertia is considerably improved.

- 6) Cycle Conditions: Lubricant inlet pressures, temperatures, and flow; temperature and pressure gradients in the bearing clearance space; and drain pressures and temperatures must be carefully controlled. The combination of high temperature, low pressures, and low flow will result in vaporization, the possibility of cavitation, and limited endurance capability. The use of mercury as the lubricant creates a problem in the clearance because two-phase flow, i. e., liquid and vapor, may exist. End leakage, high and low pressure regions, and high temperatures cause vapor bubbles to form in the unloaded portion of the bearing. If allowed to collapse in the bearing clearance space these bubbles will cause cavitation-erosion damage and limit life. This type of damage is greatest in bearings that generate hydrodynamic pressures and have high supply pressure sources in one or several places, i. e., mostly in plain cylindrical and hybrid types. Generally these bearings exhibit cavitation-erosion damage upstream of the supply pads. This is understandable since it is here that bubbles formed in the unloaded bearing region due to end leakage, low pressure, and high temperature tend to collapse upon encountering the high inlet pressures. If the vapor bubbles pass from the bearing without collapsing, via slots or grooves referenced to drain pressures, i. e., as in axial grooved types, damage caused by this phenomenon is eliminated or reduced to permit reliable long-term operation.

An additional problem created by two-phase flow in the clearance space is reduction in load capacity if the vapor is carried into the loaded portion of the bearing or if loads are rotating.

- 7) Start-Stop Cycles: Hydrodynamic bearings tend to start with metal-to-metal contact, while hydrostatic type bearings, depending on available supply pressure, start with separated journal and bearing surfaces. For a non-wetting lubricant, the hydrodynamic bearing will not even have the benefit of a layer of residual lubricant. These characteristics hold true during the shutdown cycle. For ground operation, or if restart capability is required during space operation, any wear during repeated start-stop cycles will limit the useful life capabilities of the bearings. Careful selection of materials will minimize the effect of metal-to-metal contact in the absence of a hydrostatic start. One hundred and twenty hydrodynamic starts were made using Bluechip (AMS 18-4-1 Tool Steel) without any measurable wear.
- 8) Bearing Materials: Selection of bearing and journal materials has an increased influence on life capabilities when start-stop cycles occur with metal-to-metal contacts, i. e., as in hydrodynamic bearings or if adequate

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lubricant flow is dependent on rotor speed, if conditions for two-phase flow exist in the bearings, or if rust and crud tend to deposit on the bearing and journal surfaces during operative and nonoperative phases. Resistance to wear, cavitation-erosion damage, and rust and corrosion product (crud) deposition can be increased by careful selection of bearing and journal materials and by bearing geometry.

There are many desirable bearing characteristics; some important, some not as important. For different applications, the relative importance of these characteristics changes drastically. For instance, the bearings for a one-shot missile must be extremely reliable yet have a life requirement that is measured in terms of minutes. In a space power application such as SNAP 2, the reliability requirement again is high, but the specified life is measured in terms of a year or thousands of hours. Without attempting to signify order of importance, the major bearing characteristics sought for in the SNAP 2 application are listed below:

- 1) Stability to half-frequency and synchronous whirl
- 2) Load capacity adequate for both ground and space operation
- 3) Adequate life, including such factors as hydrostatic start-stop cycles and resistance to cavitation-erosion
- 4) Low power loss
- 5) Tolerance to clearance changes from cold to hot
- 6) Fluid film stiffness control via supply pressure to regulate rotor-bearing criticals and damping
- 7) Flow-pressure requirements within the system capability

Many other desirable bearing characteristics exist, such as tolerance to particles, damping, ease of manufacture and integration into the rotating package, and small mechanical (elastic) and thermal distortion.

For each bearing selected as promising, based on analysis, past experience, and an evaluation of its characteristics, a series of parametric tests was conducted. The bearing dimensions that were varied include length-to-diameter ratio, diametral clearance, and misalignment (taper or out-of-roundness). These dimensions were selected in an effort to optimize the selected bearing for the following considerations: load capacity, stability, tolerance to misalignment and clearance changes, critical speed(s), and many other important bearing characteristics. Component tests were then conducted to determine performance of the bearings with minimum interaction from external effects. Speed, load, temperature, flow, and pressure were the independent variables. These tests were conducted on just the bearings and established

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the influence of cold and hot operation, the effect of rotating loads, and vibration on performance. Package simulation tests were then conducted; if successful, these tests were extended to endurance tests. Finally, the performance of the selected bearings was verified in actual cold and hot package tests.

The following bearings were selected for evaluation and development during the 1961 through 1963 period:

- 1) Plain with axial lube supply pad,
- 2) Tilting pad with three separately pivoted shoes,
- 3) Three sector without and with orifice compensation,
- 4) Three pad with orifice compensation, and
- 5) Spiral groove thrust bearing.

The data in Volumes II through VI compares the significant performance characteristics of each bearing with analytical predictions, wherever applicable.

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V. BEARING MATERIALS

Material requirements when using liquid mercury lubricants are different from those for organic lubricants. The physical and chemical properties of mercury and other inorganic fluids used as lubricants preclude the use of conventional bearing materials. In particular, the softer metals which have the desirable properties of conformability (ability to adjust to geometric errors including misalignment) and imbedability (ability to tolerate foreign particles without severe scoring) are excluded. For application in high speed turbomachinery lubricated with mercury, the properties of bearing materials described below have been accepted as important requirements.

A. RESISTANCE TO CORROSION

This requirement includes not only structural integrity but minimum production of contaminants. The formation of contaminants becomes a serious problem for long term operation because contaminants generally circulate through the system and "plate out" at the most undesirable places. For instance, small orifices in the bearings, jet pumps or any other restricted passage which are at a lower fluid temperature may act as "cold traps" for these contaminants. The standard screening test of bearing materials for this property is conducted in bent reflux tubes of the type shown in Figure 14. Materials are classified on the basis of weight loss and observed dissolution. Class I materials have the highest resistance; Class II materials have adequate resistance for the required life; Class III and IV materials have poor resistance to the point of non-usability; and Class V materials are completely non-resistant to corrosion. The binder matrix in non-metallics such as carbides most strongly influences corrosion resistance. Corrosion tests are conducted with 900°F liquid mercury for 12, 30, and 60 days. The over-all TRW materials evaluation program and results are detailed in Reference 1.

B. RESISTANCE TO SCORING

This requirement is especially important when non-wetting lubricants such as mercury are used and bearing types are selected that have metal-to-metal contact during start and stop cycles. Bearing materials were investigated for anti-scoring properties by the New Departure Division (Reference 26) on a high speed, high temperature friction and wear tester shown schematically in Figure 15. Some 20 material combinations were tested at 70°F and 500°F with and without liquid mercury at the point of contact. Test conditions were as follows:

Speed	5250 $\pm$ 150 fpm (20,000 rpm for roller having a one-inch diameter)
Static Load	1 pound
Roller Specimen	1-inch diameter
Flat Specimen	1/2 inch x 1/2 inch x 1/4 inch

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Figure 14. Bent Reflux Tubes

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Figure 15. Apparatus for Establishing Material Score Characteristics

Test Duration	30 minutes
Lubrication	Triple distilled mercury
Atmosphere	Nitrogen

The major conclusions of this test series, summarized in Tables 3 and 4, indicated that one material combination was outstanding when compared with the others tested. This was BG 41 stainless steel rotating against CDJ graphite. Four other material combinations, consisting mainly of tool steels, performed similarly showing only slight differences in their cumulative ratings. The remaining combinations were poor in comparison to those cited above. Reference 26 provides complete details of the test procedure and measuring and evaluation techniques.

Measurement of friction force and Talyrond surface micrographs of the specimens were recorded. Classifications of the materials were based on wear spot diameter, appearance, and pick-up height. Figure 16 shows typical specimen appearance.

C. RESISTANCE TO RUSTING

While this requirement is not important in the final hermetically-sealed system, it is desirable in the development of the bearings and the turbomachine. The nature of development rig use precludes economical sealing. When mercury and the bearing material combine in the presence of oxygen to form rust, clearance spaces in the bearing, orifices, and even pumping grooves become ineffective.

Only BG 41 (modified 440 stainless steel), aluminum oxide (Al_2O_3) and CDJ graphite have good resistance to rust. All others showed poor resistance.

D. COMPATIBLE COEFFICIENT OF EXPANSION

Since bearing clearance has a significant effect on performance, this requirement is an essential one. Moreover, the thermal environment of the application requires that the bearing materials have expansion rates compatible with those of the housing and shaft to minimize thermal stresses which tend to cause distortions. Table 5 presents the differential coefficient of expansion between the journal and the bushing and the bushing and the housing (assuming the housing material to be Haynes 25). Generally the tool and stainless steels have compatible expansion rates with the Haynes housing. However, the graphite and ceramics present serious mechanical problems in compensating for the large differential expansion rates. Since the bearings can sense temperatures from $-65^{\circ}F$ to $+600^{\circ}F$, it follows that graphite (CDJ), aluminum oxide (Al_2O_3), and Kenometal (K501) impose difficult assembly problems to assure adequate contact during start-up and stop, thermal transients and operation at design point. Generally, these problems can only be reliably overcome by large interference fits, causing high internal surface stresses. It is these high internal stresses which should be avoided to minimize distortions of the bearing bore leading to loss in load capacity.

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TABLE 3
RATINGS OF MATERIAL SPECIMEN COMBINATIONS

<u>Overall Rating</u>	<u>Material Specimen Combination</u>	<u>Initial Oven Ambient Temperature</u>	<u>Roller Appearance</u>	<u>Flat Sample Appearance</u>	<u>Flat Sample Wear Spot Diameter</u>	<u>Cumulative Rating</u>	<u>Surface Mating Characteristics</u>
1	BG-41 Stainless Steel Roller CDJ Graphite Flat	500° F	3	1	1	5	Smooth
2	T-15 Tool Steel Roller T-15 Tool Steel Flat	500° F	8	2	3	13	Smooth
2	M-4 Tool Steel Roller M-4 Tool Steel Flat	500° F	7	3	3	13	Fair
4	18-4-1 Tool Steel Roller BG-41 Stainless Steel Flat	500° F	9	2	3	14	Fair
4	M-50 Tool Steel Roller M-50 Tool Steel Flat	500° F	9	2	3	14	Fair
6	AM18-4-1 Tool Steel Roller AM18-4-1 Tool Steel Flat	500° F	9	4	3	16	Fair
6	18-4-1 Tool Steel Roller Aluminum Oxide Flat	500° F	11	3	2	16	Poor
8	18-4-1 Tool Steel Roller 18-4-1 Tool Steel Flat (Rc 53.0)	500° F	10	4	3	17	Fair
8	BG-41 Stainless Steel Roller K501 Tungsten Carbide Flat	500° F	12	3	2	17	Rough
10	BG-41 Stainless Steel Roller BG-41 Stainless Steel Flat	500° F	12	3	3	18	Rough

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TABLE 4
LIQUID-MERCURY-LUBRICATED FRICTIONAL WEAR AND SCORING RESISTANCE TESTS

Test Conditions
Wear Surface Contact Speed 5250 ± 150 fpm, Static Load = one pound, Triple Distilled Liquid Mercury Lubrication,
Nitrogen Atmosphere, Roller Specimen - One inch diameter, Flat Specimen - 1/2 inch x 1/2 inch x 1/4 inch, Test Time 30 min

Test Results

Compar- ative Rating	ROLLER			FLAT			Surface Wear (gms) x 10 ⁻⁴	Initial Oven Ambient Temperature	Average Friction Coefficient	Surface Mating Character- istics	
	Material	RC Hardness @ 80°F		Material	RC Hardness @ 80°F						
		Before Test	After Test		Before Test	After Test					
75	1	BG-41 Stainless Steel	62.8	62.5	CDJ Graphite	Not Taken		0.3247	500°F	0.12	Smooth
	2	T-15 Tool Steel	64.8	64.8	T-15 Tool Steel	63.6	63.7	18.1117	500°F	0.17	Smooth
		M-4 Tool Steel	63.5	63.2	M-4 Tool Steel	64.7	65.0	18.1117		0.25	Fair
	4	18-4-1 Tool Steel	62.9	62.8	BG-41 Stainless Steel	63.2	63.5	2.5147	500°F	0.17	Smooth
		M-50 Tool Steel	63.6	64.1	M-50 Tool Steel	65.0	64.7	32.1771		0.22	Fair
	6	18-4-1 Tool Steel	61.7	61.5	18-4-1 Tool Steel	61.2	61.5	128.3645	500°F	0.31	Fair
		18-4-1 Tool Steel	62.8	62.9	Aluminum Oxide	Not Taken		2.4962		0.25	Fair
	8	18-4-1 Tool Steel	63.0	62.8	18-4-1 Tool Steel	53.0	52.8	156.9004	500°F	Not Available	Fair
		BG-41 Stainless Steel	63.2	62.9	K501 Tungsten Carbide	Not Taken		8.8162		0.22	Rough
	10	BG-41 Stainless Steel	63.1	63.0	BG-41 Stainless Steel	63.0	63.3	32.1771	500°F	0.22	Rough

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TEST CONDITIONS: LIQUID MERCURY LUBRICATION - ONE POUND LOAD

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ROLLER AND FLAT SPECIMEN MATERIALS RUN AT 500°F INITIAL OVEN AMBIENT TEMPERATURE
RANKED IN ORDER OF PERFORMANCE RATING

Figure 16. Material Test Specimens

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TABLE 5
BEARING MATERIALS RATING CHART

Material Combinations (Journal on Bushing)	Corrosion Class	Anti-Score Rating	Resistance to Rust	Differential Expansion (in/in)	
				Between Journal & Bushing	Between Bushing & Housing Material Haynes 25
BG41 on CDJ Graphite	2-1	1	good-good	4.06×10^{-6}	5.20×10^{-6}
T15 on T15	2-2	2	poor	0	
M-4 on M-4	2-2	2	poor	0	0.13×10^{-6}
AM18-4-1 on BG41	2-2	4	poor-good	0.63×10^{-6}	1.14×10^{-6}
M50 on M50	2-2	4	poor	0	0.74×10^{-6}
AM 18-4-1 on AM 18-4-1	2-2	6	poor	0	0.5×10^{-6}
AM 18-4-1 on Al_2O_3	2-1	6	poor-good	2.1×10^{-6}	3.56×10^{-6}
AM 18-4-1 on AM 18-4-1 (Rc53)	2-2	8	poor	0	0.5×10^{-6}
BG-41 on K501	2-3	8	good-fair	3.4×10^{-6}	4.50×10^{-6}
BG-41 on BG-41	2-2	10	good	0	1.14×10^{-6}

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The candidate bearing and journal combinations were rated in terms of corrosion, wear, and resistance to rust. Table 5 presents a summary of the results combined with the thermal expansion data.

With the exception of Kenemetal (K501), all materials demonstrated adequate resistance to corrosion for the required life. A large variation in the wear rate (scoring) exists between the various material combinations. However, in the fluid film bearing the bearing and journal surfaces are separated by a fluid (mercury) film as opposed to the metal-to-metal contact in a rolling element bearing. Consequently, since fluid film bearings were selected, wear rate assumes secondary importance for this application. Table 5 shows that of the materials which demonstrated good resistance to rust, only BG 41 had a reasonable differential expansion.

E. RESISTANCE TO CAVITATION-EROSION DAMAGE

Cavitation-erosion damage in the bearing clearance space (caused by the formation and implosion of vapor bubbles) may occur as a result of the lubricant properties, the operating specifications, and the bearing type selected. Consequently, the material selected for the journal and the bearing must provide the maximum resistance to cavitation damage. The high density of mercury combined with the high fluid turbulence and inertia have an eroding effect once the surface is damaged by cavitation occurring in the clearance space. Hence, the name cavitation-erosion is used to describe the form of damage encountered by some mercury-lubricated bearing types.

On the basis of pump experience, hard, yet ductile, materials have been found to be most resistant to cavitation. Unfortunately, no techniques other than direct experimental comparison and past experience are available to establish the cavitation performance of different materials.

Other desirable materials properties include high thermal conductivity and mechanical and fatigue strength at elevated temperatures. All these necessary and desirable properties and requirements restrict selection of bearing materials and result in a compromise, as does the bearing type itself. A compromise is necessary since no single material combination has all the desirable properties discussed above and required for the SNAP 2 application.

Two bearing materials have generally been used during component development and for CRU's. At the beginning of the SNAP program, Bluechip (18-4-1) tool steel was used exclusively. This material, used as the bearing bushing and journal sleeve, behaved satisfactorily except that it tended to rust. On several occasions during both component and CRU testing, the rust deposits prevented initiation of rotation. This drawback combined with a material change in the bearing housing led to the early selection of BG 41 (modified 440C stainless steel) to replace the Bluechip tool steel. This material, hardened to Rc 55 and 60, has been used for the majority of bearings during this report period for both component tests and bearings assembled in CRU's. Its selection was made for the following major reasons:

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- 1) It had a good rating against corrosion.
- 2) Its resistance to rust was good.
- 3) The thermal coefficient of expansion matched the shaft and housing materials so that the preliminary interference fits permitted easy assembly and the contract stresses were at an acceptable level to minimize distortion at the extremes of the temperature spectrum.
- 4) Selection of a hybrid bearing minimized the importance of the anti-score (wear) rating since start-stop phases would be accomplished with a floating shaft, i. e. , no metal-to-metal contact.
- 5) The only other corrosion- , wear- , and rust-acceptable material combinations that had been evaluated to that time were CDJ graphite and aluminum oxide (Al_2O_3). Both materials were eliminated because they were brittle and had unfavorable differential expansion characteristics with the shaft and housing materials. It was felt that these materials could not meet the severe shock and vibration specification encountered during launch without chipping. In addition, the rather complicated bearing configuration, which included recess pads, orifices and axial slots maintained to close tolerances, were not easily attainable in graphite and aluminum oxide which are both difficult to machine. The need to control or predict accurately the operating clearance was not possible with the above material combination because of their initial interference fits. Furthermore, to retain sufficient press fit when the operating temperatures approached their maximum resulted in large interference fits at room temperature. These in turn increased assembly complexity and approached the allowable stress limit of graphite and aluminum oxide.

All bearings tested during this report period used BG 41 and BG 42 (modified 440C) except the tilting-pad journal bearings reviewed in Volume III and some early thrust bearings, which were made out of a variety of materials including 17-4 stainless (Rc 30), AISI 52100 (Rc 60 to 67), and Bluechip 18-4-1 (Rc 55 to 60). As a material BG 41 (later BG 42) has performed satisfactorily. No corrosion or rusting has been evident. When metal-to-metal contact has occurred due to loss in load capacity, BG 41 has exhibited varying characteristics. In some cases only wear has taken place. In others welding and seizure, typical of stainless rubbing on stainless, have resulted. Its resistance to cavitation-erosion damage above that of other material combinations, cannot be established at this time. BG 41 and BG 42 do suffer this form of damage when the operating conditions for cavitation exist. However, the material has demonstrated the capability of achieving the life requirement when load capacity is adequate and when the conditions of cavitation in the bearing clearance space and drain area are minimized.

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Figure 17. Single Hydrosphere Bearing Test Rig

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VI. EXPERIMENTAL EQUIPMENT AND INSTRUMENTATION

A. TEST FIXTURES* AND RIGS**

Experimental programs needed to achieve a fundamental understanding of turbulent flow bearing behavior and to develop bearings for a difficult application are related, but generally require different test fixtures and different test conditions. For instance, to obtain basic bearing performance data which will lead to improved understanding of a particular phenomenon that will be applied to establish analytical equations and design procedures, all such external influences on the test bearing as rotor dynamics must be eliminated or isolated. Normally, a single bearing fixture is used with carefully controlled test conditions to obtain fundamental and accurate performance data. However, no bearing is considered satisfactory until it has demonstrated successful operation in the actual machine or in a fixture that simulates actual operations conditions.

The experimental programs described in Paragraph IV and Volumes II through VI were conducted on fixtures that range from a simple one in which the flow patterns of the lubricant were observed with the aid of high speed photography, to a turbo-machine instrumented to measure shaft deflection and displacement.

Many of the initial journal bearing data (Reference 1) were obtained from the single bearing test fixture shown in Figure 17. The test bearing was mounted in a free-floating piston, supported on externally-pressurized gas bearings. A torque-meter attached to the piston yielded friction power loss data. The journal was mounted on a vertical grinder spindle which was driven by an axial flow turbine. The spindle was suspended on a self-aligning bearing located far enough away that any misalignment transferred to the test bearing would be small compared to its clearance. A load cell was used to apply radial loads to the spindle, and eccentricity was measured with inductance-type proximity transducers mounted on the piston. For thrust bearing tests, axial lift was measured with a telescopic microscope. Although this fixture yielded much useful data, it was discarded because it was difficult to set up the fixture and maintain a hermetically sealed system. Establishment of initial alignment was a particularly tedious task.

The experimental work conducted during the period covered by this report utilized three basic test fixtures. The primary function of the first type was to establish bearing parametric performance. The majority of the test fixtures were modifications of a relatively simple dual bearing fixture. The second type of fixture was to be used for endurance testing, although parametric performance could be checked before and after the endurance test. The third fixture was a modified turbomachine which simulated the CRU design.

*A test fixture is the item or assembly being tested.

**A test rig includes everything enclosed in the test booth and the connected consoles.

TABLE 6
TEST FIXTURE CAPABILITIES

Test Fixture	ϵx	ϵy	δ_{xx}	Rig Capabilities				Type of Test
	Shaft Measurements			Max. Lubricant Pressure, psi	Load Range, lb	Maximum Lubricant Temperature, °F	Maximum Speed, rpm	
Single Bearing Test Fixture (BCF)	X	X		300	0-10	100	40,000	Parametric
Dual Bearing Test Fixture (DBTF)	X	X	XX	550	0-80 (Axial) 0-50 (Radial)	450	60,000	Parametric and Endurance
Simulated Turbomachine (CRU II)	X	X	XX	450	0-10	400	60,000	CRU Simulation
Instrumented Turbomachine (CRU III)			XX	450	----	450	45,000	CRU Simulation
Flow Observation (TFF)	XXX	XXX		300	----	100	40,000	Parametric
Bearing Endurance Test Fixture (BETF)	X	X	XX	550	0-80 (Axial) 0-50 (Radial)	450	60,000	Parametric and Endurance
Modified Dual Bearing Test Fixture (MBTF)	X	X	XX	550	0-80 (Axial) 0-50 (Radial)	450	60,000	Parametric and Endurance
Bearing Assembly Checkout Fixture (BACOF)	X	X	XX	550	0-80 (Axial) 0-50 (Radial)	450	30,000 (1" Bearing) 60,000 (Regular Brg)	Parametric and Endurance Parametric

- X - may be made
XX - shaft deflection in several planes
XXX - pre-set
 ϵx - horizontal displacement
 ϵy - vertical displacement
 δ_{xx} - horizontal and vertical displacement

Table 6 presents the test fixtures utilized during various phases of the program and summarizes their capabilities. By incorporating a shaft-mounted pump, several of the fixtures were capable of both parametric and endurance testing.

1. Bearing Test Fixtures

The Dual Bearing Test Fixture (DBTF) represents a typical fixture used during the program to establish bearing performance.

The dual bearing test fixture, shown schematically in Figure 18 and in an external view in Figures 19 and 20, provided a great deal more versatility and closer simulation to the final application than did the BCF. In this fixture, the shaft assembly can incorporate a centrifugal pump in addition to the journal and thrust bearings. The center cylinder, containing the Terry-type turbine, can be loaded upward or downward by externally pressurized air pads. The shaft may be unloaded (simulating zero g operation) by balancing its weight with air pressure. That these air pads do not affect the test bearing characteristics may be inferred from the large clearances in the pads. This was demonstrated in tests in which a given load was obtained with different pressure levels in the upper and lower pads. Capacitance-type proximity transducers were used to measure bearing eccentricity, attitude angle, and axial movement of the shaft. An elaborate nitrogen bleed system was incorporated in the housing to minimize air contamination of the mercury lubricant. Lubricant temperatures to 450°F can be obtained. Speeds to 60,000 rpm are possible, and the fixture may be used to demonstrate the endurance capability of bearings and pumps. Several modifications were made to this fixture to increase its versatility and permit evaluation of improved centrifugal pumps.

The Bearing Endurance Test Fixture (BETF) and Modified Dual Bearing Test Fixtures (MDBTF) were of similar design in the DBTF but were modified to accept different bearings and operate in different test rigs.

The Bearing Assembly Check Out Fixture (BACOF) was designed to establish performance of complete CRU bearing assemblies. Figure 21 shows the layout. The fixture could be utilized to establish performance prior to a CRU test and then check the performance at the conclusion of the test. Except for the end caps (bearing caps), the fixture design did not vary much from the other parametric fixtures. A unique feature of the BACOF was the bearing caps which were designed to permit installation of split or one piece CRU bearing assemblies. Pedestal bearing supports similar to CRU's were used, and lubricant was introduced to the bearing in the same manner. This fixture was also capable of endurance testing and testing of larger than currently used CRU bearings.

Because observation of film continuity was desired, the transparent bearing fixture, shown in Figures 22 and 23 was built. Although the drive shaft and bearings are fixed relative to each other, eccentricity and misalignment can be pre-set. The drive is the same as that in the single bearing test fixture. Flow patterns are observed visually and recorded with the aid of a high speed Fastax camera.

Figure 18. High Temperature Bearing Pump Test Rig, Dual Bearing Test Fixture (DBTF)

Figure 19. Dual Bearing Test Fixture, External View

Figure 20. Installed Dual Bearing Test Fixture

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Figure 21. Bearing Assembly Check Out Fixture (BACOF)

Figure 22. Transparent Flow Fixture - Overall View

Figure 23. Close-Up of Transparent Flow Fixture

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Figure 24. Combined Rotating Unit (CRU) Simulation Rig (Modified CRU II)

The CRU II test fixture was designed to permit improved simulation of the CRU. This was accomplished by replacing the alternator rotor and stator with dummy cylinders in an early version of the SNAP 2 turbomachine, the CRU II. Figure 24 shows the converted test rig. The second-stage axial flow turbine of the turbomachine was retained to replace the Terry-type turbine of the previous test fixture; the first-stage turbine wheel and nozzle were removed to facilitate assembly and disassembly; and a solid cylinder was substituted for the sodium pump rotor of the original machine. Three planes of capacitance-type proximity transducers were used to measure eccentricity and attitude angle at each bearing, conical or translatory whirl, and shaft deflection. The housing, turbine air and lubricant can be heated to simulate temperature gradients found in the actual turbomachine. Although simulation of the current SNAP 2 turbomachine is not exact, the fixture is capable of providing design data, including shaft and housing effects on bearing performance. Substitution of dummy cylinders for the alternator rotor and stator minimizes the loading capability but provides a motoring device for long endurance runs. A description of the instrumentation is discussed later, but the method of determining shaft deflection is presented below.

The shaft deflection was measured at three locations, as shown in the following sketch, by capacitance probes. The probe output was amplified and displayed on a cathode-ray-oscilloscope. Polaroid pictures were taken of the scope traces.

This shows that probes A and C are not in the same plane with probe T. This is due to fabrication problems which made it impossible to locate the C and A probes along the vertical center line. The scribe lines on the rotor were located so they would appear under their respective probes all at the same time.

The procedure used to calibrate the capacitance probes was as follows:

- 1) The shaft was lifted to the top of the bearing by means of the lower load pad.
- 2) With the shaft topped, the probes were "nulled out", and this point was designated zero on the oscilloscope.
- 3) The shaft was then bottomed by means of the upper load pad.
- 4) The scope read-out was then calibrated by means of the amplitude calibration control to give:

Probe A 20 scope units = ($C_D = 0.0010$ inch)

Probe T 20 scope units = ($C_D = 0.0010$ inch)

Probe C 10 scope units = ($C_D = 0.0010$ inch)

The above scope settings resulted in an observational error of 5 percent for both the 20 unit and the 10 unit calibration. No attempt was made to correct the calibration due to A and C probes not being in line with the vertical.

The dynamic shaft deflection was taken to be one-half the peak to valley distance of the sine wave. This is shown schematically in the following sketch. To correct for runout the dynamic deflection at 5,000 rpm was subtracted from the dynamic deflection at higher speeds. Since the runout phase angle was not appreciably different from the shaft deflection phase angle and the magnitude was so small, the phase angle was not taken into account in subtracting the runout.

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To determine the phase angle between probes, the "pips" due to the scribe line on the rotors were used. This was used to determine if an output from one probe in one plane was in phase or out of phase with another probe in another plane and to determine the presence of half-frequency whirl. A Bruel and Kajar wave analyzer was used in addition to the "pips" to detect half-frequency whirl.

A later version of the SNAP 2 turbomachine incorporating the NaK pump was instrumented as described above. The unit, shown in Figure 25, closely simulates the current design and, with the exception of the loading capability, can be used in the same manner as the simulated turbomachine described previously.

2. Bearing Test Rigs

Two test facilities identified as test rigs were utilized during this report period to develop mercury lubricated bearings.

Free Running Bearing Test Rig

The free running bearing test rig was designed to test a mercury pump and mercury-lubricated journal and thrust bearings. This facility may also be used to demonstrate the endurance capability of bearings and pumps. Speeds in excess of 60,000 rpm can be attained.

The test rig consists of a Terry-type turbine drive package, a separator tank, an accumulator, an air preheater, and a Chem-pump or a pressurized mercury reservoir used in conjunction with a Lapp pump to supply mercury to the fixture through a filtering system. Mercury heated to 450°F can be supplied to the bearings. To minimize air contamination of the mercury lubricant, the facility has a provision for supplying nitrogen to the package. A separate pump loop supplies the jet pump with mercury at temperatures up to 500°F and is used to record impeller performance. External axial loads of approximately 80 pounds and radial loads of approximately 50 pounds can be applied either upward or downward. A set of dynamic micrometers is used in conjunction with an oscilloscope to measure radial displacement to an accuracy of 0.0001 inch. The facility is also equipped to record supply pressures, flows, temperatures, drain pressure, and speed.

Bearing Endurance Test Rig

The Bearing Endurance Test Rig (BETR) is similar to the free running bearing test rig described above. The main difference is that the endurance test rig uses a series of Lapp pumps to supply mercury to the fixture. No separate pump loop is provided, but the rig can test a bearing-pump package. The bearing endurance test rig is equipped to record supply pressures, temperatures, and flows automatically. A schematic of the rig is shown in Figure 26.

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Figure 25. Instrumented Combined Rotating Unit, CRU III

Figure 26. Bearing Endurance Test Rig, Mercury and Vacuum System

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Frequency Response -- 0-10 KC
Oscillator Tuner -- 600-900 KC
Output E -- ± 10 V Max.

Figure 27. Schematic of Shaft Displacement Measuring Equipment

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B. INSTRUMENTATION AND METHODS OF MEASUREMENT

As important as test fixtures and rigs are in obtaining fundamental and accurate performance data, the success or failure of an experimental program depends primarily on the type of instrumentation used, and the reliability and accuracy with which the measurements are made. The instrumentation is required to measure shaft position and motion in the bearing, power loss, damping characteristics and flow-pressure relationships.

The system used to measure the dynamic position and motion of the journal in the bearing is the most important and normally the most difficult to obtain accurately and reliably. A Photocon proximity probe was used to measure minute static and dynamic changes of the shaft. The probe is one plate of the condenser, and the journal (shaft) is the other plate. The probe is fixed, and movement of the shaft varies the space between the probe and the shaft, resulting in a change of capacitance and impedance to an impressed voltage. The voltage variation is amplified and displayed on a Dumont Type 411 R or a Tektronics Type 502 dual beam oscilloscope. Two probes at right angles were used in one plane to give shaft position in two dimensions. A schematic diagram of the system is shown in Figure 27. This system provided a direct reading of shaft position in the bearing clearance.

Under static operation the vertical and horizontal probe signals, i. e., shaft position in the clearance space, yields a stable dot. During dynamic operation, however, the output of the two probes may be a dot on the CRO (cathode ray oscilloscope), but more generally, the shaft traced a path which is a result of shaft run-out due to imperfect machining, rotating load due to residual unbalance, shaft bending, and possibly shaft whirling due to instability. To enable determination of the frequency of whirl in the bearing clearance space, a small notch, 0.030-inch wide by 0.030-inch deep, was made on the shaft. This notch caused a small blip on the CRO trace. When the shaft whirled at the frequency of the shaft rotating speed, one blip from each probe or two blips per whirl cycle were observed. With half-shaft speed whirl, four blips were observed on the CRO. The amplitude of whirl was also observed from the trace. The blip was also used to establish phase angle at different speeds. A comparison of the phase angle of probes located at different axial positions established the rotational mode i. e., translatory or conical, of the shaft and helped to pin-point critical speeds.

Calibration of the proximity instrumentation was accomplished as follows:

- 1) The gap between the transducer probe end and the shaft was set so that the transducer output over the displacement defined by the bearing clearance was linear (usually about 0.008 inch).
- 2) The transducer output and the CRO gain were adjusted so that the total shaft displacement (bearing clearance) represented a CRO beam deflection of about 75 percent of the screen useful diameter.

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Figure 28. Calibration Curve - Shaft Displacement
vs Transducer Output

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- 3) The transducer was calibrated (shaft movement vs CRO beam deflection) by displacing the shaft a known amount.
- 4) The zero speed shaft position was adjusted to the desired location on the CRO screen.

The frequency response of the proximity meter is flat up to 15,000 cps and usable up to 50,000 cps. This allows measurement of dynamic phenomena occurring at the rate of 3,000,000 cpm. Movements as small as one micro-inch can be measured. Linearity is dependent on the transducer and cable length. A typical calibration curve for a 3/8-inch diameter proximity transducer is shown in Figure 28. With a gap of 0.012-inch between shaft and probe, the transducer output is quite linear for a displacement of 0.002 inch. The maximum deviation is seen to be about five percent as related to bearing clearance. Special effort was made to keep mercury liquid or vapor out of the gap between the proximity probe and shaft, since this would change the dielectric constant with a resultant change in calibration. This was accomplished by bleeding nitrogen gas into the gap area, thus providing a constant nitrogen blanket at the probes. Shaft displacement as a function of oscilloscope movement is shown in Figure 29; the relationship is again linear for the probe clearance selected. If mercury vapor existed in the gap displacement measurements could still be taken by considering the effect on the dielectric constant and the calibration. Figures 30 and 31 show the effect of mercury vapor on the calibration. In the bearing test fixtures, mercury vapor is kept away from the displacement probes as much as possible by purging the ends of the probes with constantly circulating nitrogen.

To overcome the temperature sensitivity of proximity probes, water jackets were provided which tend to keep the probes at an even temperature. This also permitted the use of these probes at elevated temperatures where it is normally difficult to obtain shaft behavior data.

Some of the other standard measurements included power loss, which was measured by speed decay method; flow-pressure relationships, measured by calibrated, flat plate orifices and pressure gauges; and applied load, determined by calibrated load pads. Damping characteristics of the bearing fluid film were obtained for the static and dynamic cases. Static damping coefficients were obtained for hybrid bearings by "shocking" the shaft, measuring its decay trace, and applying

$$X = X_0 e^{-\left(\frac{C}{2m}\right)t}$$

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Figure 29. Calibration Curve - Shaft Displacement
vs Oscilloscope Movement

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Figure 30. Displacement Transducer Output as a Function of Temperature in Air and Mercury Vapor Surroundings

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Figure 31. Calibration Correction for Presence of Mercury Vapor

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Comparison with the critically damped case was made by applying

$$\gamma = C/C_{cr} = \frac{\ell_n A_1/A_n}{\sqrt{\pi^2 n^2 + \ell_n (A_1/A_n)^2}}$$

Dynamic damping characteristics were obtained qualitatively by establishing the frequency range required to pass through a critical speed, i. e., obtain a 180 degree phase change, and then by comparing this frequency range to standard curves for various fractions of critically damped systems. The use of rotating loads and the measured shaft locus were also used to establish damping coefficients.

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APPENDIX A

NOMENCLATURE

A	Area or load factor
B	Circumferential length, in.
c_p	Specific heat at constant pressure, $\frac{\text{Btu}}{\text{lb-}^\circ\text{F}}$
C	Damping Coefficient lb-sec/in.
C_D	Diametral clearance, in.
C_{MD}	Machined-in diametral clearance, in.
C_{MR}	Machined-in radial clearance, in.
C_n	Capacity number, dimensionless
C_R	Radial clearance, in.
D	Diameter of bearing, in.
d	orifice diameter, in.
E	Driving force
e	Eccentricity, dimensionless
F	Hypergeometric function or friction force
F_R	Radial force, lb
f	Frequency, rpm
f_s	End leakage, dimensionless
g	Gravity, in./sec ²
H	Power loss, watts
h	Film thickness, in.
h_o	Minimum film thickness, in.
J	Mean mass flow in y direction, lb/sec-in.
j	Friction factor, function of the rate $60 N'/N_{tr}$, dimensionless
I	Moment of inertia of the shoe
K	Fluid film stiffness, lb/in.
K_R	Radial film stiffness, lb/in.
K_s	Spring rate of shaft, lb/in.
K_p	Load factor, dimensionless
k	Pradtl's mixing length constant, dimensionless

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k_{fr}	Friction factor, dimensionless
L	Axial length, in.
M	Mass, lb
m	Clearance modulus, in./in.
N	Speed, rpm
N'	Speed, rps
P	Pressure, lb/in. ²
p	Unit pressure, W/LD, lb/in. ²
q	Flow factor, dimensionless
Q	Flow, in. ³ /sec
Q_W	Weight flow, lb/min
R	Radius, in.
Re	Reynolds number, dimensionless
S	Sommerfield number, dimensionless
S_A	Rocking stiffness, lb/in.
S_L	Bearing spring rate, lb/in.
T	Thrust capacity, temperature
U	Surface velocity, in./sec
u	Velocity gradient in x direction, in./sec
v	Velocity gradient in y direction, in./sec
W	Applied load, lb
Z	Absolute viscosity, centipoise

Subscripts

cr	Critical
d	drain
i	Initial
l	Laminar
min	minimum
n	number
o	outlet or zero speed
R	Radial

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s Supply
t Turbulent
tr Transition
tot Total
x, y, z Direction of the axis

Greek

α Angular velocity, rad/sec
 β Bearing angle, degrees
 ϵ Eccentricity ratio, dimensionless
 η End leakage, dimensionless
 θ_i Angle measured clockwise from the applied load to the pad pivot point, degrees
 Λ Bearing number, dimensionless
 λ Raimondi's compressibility number, dimensionless
 μ Absolute viscosity, lb-sec/in.² or rehns
 ν Kinematic viscosity, in./sec or centistokes
 ρ Density, lb/in.³
 ϕ Attitude angle, degrees
 ϕ_{3SL} Three-sector bearing laminar
 ϕ_{3St} Three-sector bearing turbulent
 ϕ_{Pl} Plain bearing laminar
 ϕ_{Pt} Plain bearing turbulent
 ω Speed, radians/sec

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APPENDIX B

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APPENDIX C

ANALYTICAL PROCEDURE TO ESTABLISH THE LOAD CARRYING
CAPACITY OF TILTING-PAD BEARINGS INCLUDING PRE-LOAD

An analytical procedure to establish the load capacity of the three-pad tilting-pad journal bearing including pre-load was developed. It is assumed that the lubricant film is laminar, viscosity remains constant, and a slider bearing can be substituted for the curved pad. To account for the pre-load in the bearing the crowned slider bearing (Ref. 18) is used for substitution. It is assumed that pre-load with a curved slider is similar to crowning in a flat slider. S. Abramovitz derived the load capacity of the single slider to be

$$W = \frac{\mu U B^2 L}{h_2^2} C_P \quad . . . 97$$

Where:

μ = the lubricant viscosity, lb-sec/in.²

U = the surface speed, in./sec

B = the pad circumferential length, in.

L = the pad width (or bearing axial length), in.

h_2 = the trailing edge film thickness, in.

C_P = a result of numerical integration and a function of "a" (h_1/h_2) and "b" (H_c/h_2)

A schematic of the crowned slider bearing is shown in Figure 32 with the corresponding nomenclature. Values of C_P are given in Reference 18 for various values of "a" and "b". The tilt of the pad, "a", is a function of pivot point location and "b". Reference 18 also gives values of pivot point location as a function of "a" and "b".

The incorporation of the slider bearing into the journal bearing is shown in Figure 33. The center point of each pad is shown as O_1 , O_2 , and O_3 a distance, d , away from the center of the shaft in the concentric position. The effective height of the crown of the slider, H_c , is seen to be a function of shaft eccentricity, pivot point clearance, C_m , and the distance, d . The attitude-eccentricity locus is assumed to be a straight line in the direction of the applied load. Experimental evidence proves this assumption to be relatively good.

Figure 32. Crowned Slider Bearing Schematic

Figure 33. Tilting-Pad Journal Bearing Schematic

The effective crown height for the pads is obtained from

$$H_c = h_c - h_p \quad \dots 98$$

There are two limiting conditions of loading: (1) loading directly into the pivot points and (2) loading away from the pivot point.

Loading Down

For loading down (into the pivot point) the effective crown height for pad No. 1 is obtained as follows:

$$H_{c1} = h_{c1} - h_{p1} \quad \dots 99$$

Where:

h_{c1} = the film thickness at the end of the pad with the pad in the untilted position

h_{p1} = the film thickness at the pivot point

Since the film thickness in any bearing is given by:

$$h = C (1 - \epsilon \cos \theta) \quad \dots 100$$

h_{c1} can be expressed as:

$$h_{c1} = (C_m + d) \left(1 - \frac{e_1}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 10p$$

Where:

C_m = the pivot point clearance

e_1 = the shaft eccentricity with respect to O_1

θ_i = the arc length of the pad

Noting that

$$e_1 = e + d \quad \dots 102$$

and letting:

$$C = C_m + d \quad \dots 103$$

$$\epsilon = e/C_m \quad \dots 104$$

$$m = d/C_m \quad \dots 105$$

then,

$$h_{c1} = C \left(1 - \frac{\epsilon + m}{1 + m} \cos \frac{\theta_i}{2} \right) \quad \dots 106$$

The pivot point film thickness is given by:

$$h_{P1} = C \left(1 - \frac{\epsilon + m}{1 + m} \right) \quad \dots 107$$

For this type of loading, i. e., directly toward the pivot point of pad No. 1, effects of pad No. 2 and 3 are equal; therefore, only pad No. 2 is considered. The crown height of this shoe is again expressed as:

$$H_{c2} = h_{c2} - h_{P2} \quad \dots 108$$

However, h_{c2} must represent the film thickness at both the leading and trailing edges of the pad with no effective tilt. This is obtained by the following expression:

$$h_{c2} = (C_m + d) \left(1 - \frac{e'_2}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 109$$

From Figure 33:

$$e'_2 = C_m + d - h_{P2} \quad \dots 110$$

so that

$$h_{c2} = (C_m + d) \left(1 - \frac{C_m + d - h_{P2}}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 111$$

or

$$h_{c2} = C \left(1 - \frac{C - h_{P2}}{C} \cos \frac{\theta_i}{2} \right) \quad \dots 112$$

where the pivot point film thickness h_{P2} is

$$h_{P2} = C_m (1 - \epsilon \cos \theta_{P2}) \quad \dots 113$$

The trailing edge film thicknesses of the pads 1 and 2 can be expressed as

$$h_{21} = \frac{h_{P1} + H_{c1}}{1 + \bar{X}/B (a - 1)} \quad \text{for pad 1} \quad \dots 114$$

And

$$h_{22} = \frac{h_{P2} + H_{c2}}{1 + \bar{X}/B (a - 1)} \quad \text{for pad 2} \quad \dots 115$$

But a and $\bar{X}/B$ are dependent on each other so that the above equations are solved by a method of trial and error.

- 1) Assume a value of a .
- 2) Calculate h_2 for a known pivot point location.
- 3) Calculate b .
- 4) Check to see if the variables a , b , and $\bar{X}/B$ satisfy the results of Reference 18.
- 5) Assume new a in the direction as indicated by the error and continue until Reference 18 is satisfied.

With the values of a , b , and $\bar{X}/B$ obtained, Reference 18 also gives C_p . Now the pad loads W_1 , W_2 , and W_3 can be calculated. The net load capacity of the bearing is the vectorial summation of the contribution of all three pads:

$$W_{\text{net (down)}} = W_1 + W_2 \cos \theta_{P2} + W_3 \cos \theta_{P3} \quad \dots 116$$

For the three-pad bearing, $\theta_{P2} = 120$ degrees; $\theta_{P3} = 240$ degrees; and $W_2 = W_3$.
Then

$$W_{\text{net (down)}} = W_1 - \frac{1}{2} W_2 - \frac{1}{2} W_2$$

$$W_{\text{net (down)}} = W_1 - W_2 \quad \dots 117$$

So far no side leakage has been considered. The best side leakage factor obtained to date is that of Kingsbury. W_{net} , then, must be multiplied by this factor (side leakage factor = f_s):

$$W_{\text{net (down)}} = (W_1 - W_2) f_s \quad \dots 118$$

Loading Up

For loading up (away from pivot point) the same procedure is followed with only slight modification. For pad No. 1 (from equation 99)

$$H_{c1} = h_{c1} - h_{P1} \quad \dots 99$$

So that

$$h_{c1} = C \left(1 - \frac{e_1}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 101$$

But $e_1 = d - e \quad \dots 119$

So that $h_{c1} = C \left(1 - \frac{d - e}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 120$

or $h_{c1} = C \left(1 - \frac{m - \epsilon}{1 + m} \cos \frac{\theta_i}{2} \right) \quad \dots 121$

The pivot point film thickness is given by equation 107

$$h_{P1} = C \left(1 - \frac{m + \epsilon}{1 + m} \right) \quad \dots 107$$

For pad No. 2 (from equation 108)

$$H_{c2} = h_{c2} - h_{P2} \quad \dots 108$$

So that

$$h_{c2} = C \left(1 - \frac{e'_2}{C_m + d} \cos \frac{\theta_i}{2} \right) \quad \dots 109$$

Again (from equation 110)

$$e'_2 = C_m + d - h_{P2} \quad \dots 110$$

then, as in equation 112

$$h_{c2} = C \left(1 - \frac{C - h_{P2}}{C} \cos \frac{\theta_i}{2} \right) \quad \dots 112$$

$$\text{and } h_{P2} = C_m (1 + \epsilon \cos \theta_{P2}) \quad \dots 122$$

From here on a, b, h_2 , and W are calculated as before. The net load capacity upward is

$$W_{\text{net-up}} = (W_2 - W_1) f_s \quad \dots 123$$

A non-dimensional load can be defined as:

$$\bar{W}_n = \frac{W_n}{\mu \text{ NLR } f_s} \left(\frac{C_{MR}}{r} \right)^2 \quad \dots 124$$

$$\text{or } \bar{W}'_n = \frac{W_n}{\mu \text{ NLR } f_s} \left(\frac{C_R}{R} \right)^2 \quad \dots 125$$

$$\text{Since } C_R = C_{MR} (1 - m) \quad \dots 126$$

$$\therefore \bar{W}'_n = (1 - m)^2 W_n \quad \dots 127$$

Power Loss

The shear force for the crowned slides is given by:

$$F = \frac{\mu \text{ UBL}}{h_2} C_f \quad \dots 128$$

Therefore, the power loss for a single pad P_n is given as:

$$P_n = F_n \times U; \text{ or } P_n = \left[\frac{\mu U^2 BL}{h_2} \right] C_f \frac{\text{in.-lb}}{\text{sec}} \quad \dots 129$$

Note: $1 \frac{\text{in.-lb}}{\text{sec}} = 0.1129 \text{ watts}$

Consequently, the total power loss for 'n' pad journal bearing will be:

$$P_T = P_1 + P_2 + \dots + P_n \quad \dots 130$$

A non-dimensional power loss $\bar{P}_n$ can be defined as:

$$\bar{P}_n = \frac{P_n}{\mu R^2 N^2 L} \left(\frac{C_{MR}}{r} \right) \text{ for a single pad} \quad \dots 131$$

Dimensionless Bearing Stiffness

A dimensionless bearing stiffness can be defined as

$$\bar{K} = \frac{d\bar{W}}{d\epsilon} \quad \dots 132$$

$$\text{Since } \bar{W} = \frac{W}{\mu R N L f_s} \left(\frac{C_{MR}}{r} \right)^2 \quad \dots 133$$

$$\text{and } \epsilon = \frac{e}{C_R} \quad \dots 134$$

$$\therefore \bar{K} = \frac{K}{\mu N L f_s} \left(\frac{C_{MR}}{r} \right)^2 \left(\frac{C_R}{R} \right) \quad \dots 135$$

Influence of Radial Clearance on Performance

Use of the design procedure developed for the pre-loaded tilting-pad bearing is illustrated in the following analysis conducted to establish the influence of radial clearance on such parameters as load capacity, power loss, film stiffness, and minimum film thickness.

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The analysis is based on the following parameters with mercury as the lubricant.

Speed (N) = 36,000 rpm
Radius (R) = 0.3125 inch
Axial Length (L) = 0.427 inch
Pre-Load (m) = 0.5
Lube Temperature = 475°F (considered as mean in bearing)
Three equal shoes = 98° arc
Pivot Location $\bar{X}/B$ = 0.575 inch from leading edge

Radial pivot point clearance is varied from 0.0004 to 0.002 inch and applied load up to 7 lb assuming laminar flow. Based on previous experience, this is equivalent to some 20 lb load under turbulent flow, and corresponds to the range of loads anticipated in the actual turbomachine.

Dimensionless load, power loss, and stiffness curves shown in Figures 34, 35, and 36 are used to obtain Figures 37, 38, and 39 for the pre-load of 0.5

For the given bearing with the various clearances and under the specified load, the predicted performance is summarized in Figures 40 through 45.

From this laminar analysis the following conclusions can be drawn:

- 1) Power Loss increases with increasing load, is larger for a load midway between the pivots, is minimum for C_R greater than 0.0010 inch.
- 2) Bearing Stiffness increases with increasing load, is smaller for a load midway between the pivots, except for large values of C_R , is minimum for C_R greater than 0.0008 inch.
- 3) Minimum Film Thickness decreases with increasing load, is larger for a load midway between the pivots, has a maximum value between values of 0.0006 and 0.0012 inch for C_R .

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Figure 34. Dimensionless Load Capacity ($\bar{W}_T^{-1}$) Versus Eccentricity Ratio (ϵ)
For a Three-Pad Tilting-Pad Journal Bearing

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Figure 35. Dimensionless Power Loss ($\bar{P}_T$) Versus Eccentricity Ratio (ϵ)
For a Three-Pad Tilting-Pad Journal Bearing

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Figure 36. Dimensionless Bearing Stiffness ($\bar{K}$) Versus Eccentricity Ratio (ϵ)
For a Three-Pad Tilting-Pad Journal Bearing

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Figure 37. Dimensionless Total Load Capacity ($\bar{W}_T$) Versus Eccentricity Ratio (ϵ)
 For A Three-Pad Tilting-Pad Journal Bearing
 For Pre-Load (m) = 0.5

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Figure 38. Dimensionless Total Power Loss ($\bar{P}_T$) Versus Eccentricity Ratio ()
 For A Three-Pad Tilting-Pad Journal Bearing
 For A Pre-Load (m) = 0.5

Figure 39. Dimensionless Bearing Stiffness ($\bar{K}$) Versus Eccentricity Ratio (ϵ)
 For A Three-Pad Tilting-Pad Journal Bearing
 For A Pre-Load (m) = 0.5

Figure 40. Minimum Film Thickness ($h_{\min}$) Versus Radial Clearance (C_R)
For A Three-Pad Tilting-Pad Journal Bearing

Figure 41. Minimum Film Thickness ($h_{\min}$) Versus Radial Clearance (C_R)
For A Three-Pad Tilting-Pad Journal Bearing

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Figure 42. Power Loss (H_T) Versus Radial Clearance (C_R)
For A Three-Pad Tilting-Pad Journal Bearing

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Figure 43. Power Loss (H_T) Versus Radial Clearance (C_R)
 For A Three-Pad Tilting-Pad Journal Bearing

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Figure 44. Bearing Stiffness (K) Versus Radial Clearance (C_R)
For A Three-Pad Tilting-Pad Journal Bearing

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Figure 45. Bearing Stiffness (K) Versus Radial Clearance (C_R)
 For A Three-Pad Tilting Pad Journal Bearing

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NOMENCLATURE FOR APPENDIX C

a	= h_1/h_2
b	= H_c/h_2
B	= pad circumferential length, in.
C	= $C_m + d = C_{MD}/2$, in.
C_D	= bearing diametral clearance, in.
C_m	= $C_D/2$, in.
C_{MD}	= machined-in diametral clearance, in.
C_P	= load factor
C_f	= power loss factor
d	= distance between 0 and center of shaft in the centered position, in.
D	= diameter, in.
e	= shaft eccentricity, in.
f_s	= side flow factor
h_1	= pad leading edge film thickness, in.
h_2	= pad trailing edge film thickness, in.
h_c	= pad leading edge film thickness on the untilted position, in.
H_c	= slider crown height, in.
h_P	= pivot point film thickness, in.
K	= bearing stiffness, lb/in.
L	= pad axial length, in.

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O = center of curvature of pad
 R = shaft radius, in.
 r = radius of pad, $(R + d)$, in.
 U = surface speed, in./sec
 W = load, lb
 $\bar{X}$ = distance of pivot point from trailing edge, in.
 ϵ = e/C_m , eccentricity ratio
 m = d/C_m , preload number
 μ = viscosity, lb-sec/in.²

Subscripts

x = experimental
 ℓ = theoretical laminar
 $1, 2, 3$ = refer to pads no. 1, 2, and 3 if added to already existing subscripts

APPENDIX D

ANALYSIS OF TILTING-PAD BEARINGS INCLUDING TURBULENCE

High density, relatively low viscosity, and high operating speed make the mercury lubricant film quite turbulent in a 5/8-inch diameter tilting-pad journal bearing. Consequently, analysis based on laminar film is not applicable. Chou and Saibel (Ref. 7) present an analytical solution to turbulent slider bearing lubrication based on Prandtl's mixing-length mechanism which includes a constant "k". The value of k must be deduced from experiment.

The main assumptions in Chou and Saibel's analysis are:

- 1) There is no side leakage.
- 2) With a large Reynolds number, as in the case of turbulent flow, the turbulent friction is much larger than laminar friction so that

$$\mu \frac{\partial^2 \bar{u}}{\partial y^2} \approx 0 \quad \dots 136$$

- 3) Prandtl's mixing length theory for the boundary conditions of the slider bearing gives

$$\rho \ell^2 = k^2 y (h - y) \quad \dots 137$$

where ρ is the density and ℓ is Prandtl's mixing length.

Substituting these equations into Reynolds equations of motion for turbulent flow of an incompressible fluid and solving for the pressure equation, load-carrying capacity, and power loss of a fixed slider bearing, Chou and Saibel derived the following equations:

Load-Carrying Capacity

$$W = \frac{k^2 U^2 B^2 L (a+1)}{h_1^2 \pi^2 a^2 (a-1)^3} \left(m + n \ln a + s \frac{a-1}{a+1} \right) \quad \dots 138$$

Where: W = load, lb

$$m = \frac{(a-1)^2}{(a+1)} r + st \quad \dots 139$$

$$n = \frac{s(2at - a + 1) + (a-1)^2 r}{(a^2 - 1)} \quad \dots 140$$

$$r = -24 a^2 \quad \dots 141$$

$$s = \frac{1}{h_1^2} a (a-1) c (c + 10 a h_1) \quad \dots 142$$

$$t = \frac{a-1}{a} \left[1 - \frac{c}{2(1 + 10 a h_1)} \right] \quad \dots 143$$

$$c = -10 h_1 \frac{a}{a+1} \left\{ 1 - \left[1 - \frac{48}{100} \left(\frac{a+1}{a-1} \right) \ln a \right]^{1/2} \right\} \quad \dots 144$$

And:

U = relative velocity, in./sec

k = experimental constant

B = slider circumferential length, in.

L = slider axial width, in.

$a = h_2/h_1$

h_2 = inlet film thickness, in.

h_1 = outlet film thickness, in.

Power Loss

$$H = \frac{k^2 U^3 B L}{\pi^2 h_1 (a - 1)} \left[3 c \ln a + \frac{(36 h_1^2 a + c^2)}{4 h_1 a} (a - 1) \right] \dots 145$$

where H = power loss, in.-lb/sec.

In applying this slider bearing analysis to a three-pad, tilting-pad, journal bearing the following assumptions are made:

- 1) The pivots are equally spaced about the circumference at 120 degree intervals.
- 2) The pivot is placed at a point $4/9$ of the length of the shoe from the trailing edge.

The outlet film thickness can be expressed by the following equation.

$$h_1 = \frac{h_P}{1 + \bar{X}/B (a - 1)} \dots 146$$

The clearance at the pivot, h_P , is also defined in terms of the angular position of the pivot and the eccentricity of the journal in the bearing:

$$h_P = \frac{C_D}{2} (1 - \epsilon \cos \theta_P) \dots 147$$

Where: C_D = diametral clearance, in.

ϵ = eccentricity ratio

θ_P = angular position of pivot

Net-Load Carrying Capacity

The net load-carrying capacity is the result of forces exerted by all three pads:

$$W_{\text{net}} = (W_1 - W_2 \cos \theta_2 - W_3 \cos \theta_3) \eta \quad \dots 148$$

Where η is the side leakage factor corresponding to the B/L ratio.

But in this case

$$\cos \theta_2 = \cos \theta_3 \text{ and } W_2 = W_3 \quad \dots 149$$

so that ($\theta_2 = 120$ degrees) .

$$W_{\text{net}} = (W_1 - W_2) \quad \dots 150$$

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To obtain the net load capacity W is calculated for pads No. 1 and 2 from the slider bearing equation 138 through 144 and equations 146 and 147. In the case of the journal bearing, however,

$$U = \pi D N' \quad \dots 151$$

Where: D = diameter, in.

N' = speed, revolutions per second

and the pad length

$$B = \pi D \times \frac{98}{360} \quad \dots 152$$

Power Loss

$$H_{\text{total}} = (H_1 + 2 H_2) \times \frac{1}{8.85} \text{ watts} \quad \dots 153$$

The net load capacity and the power loss equations were programmed on the Burroughs 205 digital computer for the three-pad, tilting-pad journal bearing.

The basic program inputs were:

D = diameter, in.

N' = speed, rps

θ_1 = angular position of pad No. 1

θ_2 = angular position of pad No. 2

a = h_2/h_1

$\bar{X}/B$ = pivot position

The variable inputs were:

C_D = diametral clearance, in.

ϵ = eccentricity ratio

L = pad axial width, in.

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η = side leakage factor (corresponding to L/B)

k = empirical constant

Computer runs were made for a 5/8-inch diameter bearing using experimental data available from MDBTF-I and II test series. The following parameters were fed into the computer:

a = 2

D = 0.625 in.

N = 333, 500, 667 cps

θ_1 = 0°

θ_2 = 120°

C_D = 0.0016, 0.002, 0.003 in.

ϵ = 0, 0.2, 0.4, 0.6, 0.8

k = 0.0002, 0.0005, 0.0008

L = 0.32 in.

η = 0.241

$\bar{X}/B$ = 4/9

From the computer results figures are plotted which show load capacity as a function of " K " and " ϵ ". Figures 46 and 47 are examples at 40,000 rpm and $C_D = 0.002$ and 0.003 inches. These figures are then compared with experimental data as shown for MDBTF I results to obtain the experimental value of k .

This procedure is repeated for other speed and other clearances for which experimental data are available and then compared. The net result is a series of k 's for different speeds, and eccentricity ratios, which are then treated as functions of Reynolds number and density as shown below:

From a dimensional analysis, k is a function of the following variables:

$k = k(U, \rho, \nu, c)$. . . 154

i. e., velocity, density, viscosity, and clearance.

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Figure 46. Load Capacity As A Function Of Turbulence Factor
For A Three-Pad, Tilting-Pad Journal Bearing

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Figure 47. Load Capacity As A Function Of Turbulence Factor
For A Three-Pad, Tilting-Pad Journal Bearing

Figure 48. Prandtl's Mixing Length Constant vs Reynolds Number
 For Tilting-Pad Journal Bearing
 (MDBTR II-1 Test Series)

As can be seen from Prandtl's mixing length theory, k has the dimensions of $\sqrt{\rho}$. Then if the above equation is expressed in the fundamental unit system of mass (M), length (L), and time (T), the following equation results:

$$k = C U^\alpha \rho^\beta \nu^\gamma L^\delta \quad \dots 155$$

or in terms of M, T, and L

$$\frac{M^{1/2}}{L^{3/2}} = \left(\frac{L}{T}\right)^\alpha \left(\frac{M}{L^3}\right)^\beta \left(\frac{L^2}{T}\right)^\gamma L^\delta \quad \dots 156$$

$$= L^\alpha T^{-\alpha} M^\beta L^{-3\beta} L^{2\gamma} T^{-\gamma} L^\delta \quad \dots 157$$

$$M^{1/2} L^{-3/2} T^0 = L^{\alpha - 3\beta + 2\gamma + \delta} T^{-\alpha - \gamma} M^\beta \quad \dots 158$$

$$\alpha - 3\beta + 2\gamma + \delta = -3/2 \quad \dots 159$$

$$-\alpha - \gamma = 0 \quad \dots 160$$

$$\beta = 1/2 \quad \dots 161$$

Solution of the simultaneous equations above gives:

$$\alpha = \delta \quad \dots 162$$

$$\beta = 1/2 \quad \dots 163$$

$$\gamma = \delta \quad \dots 164$$

Then upon substitution:

$$k = C U^\delta \rho^{1/2} \nu^{-\delta} C^\delta \quad \dots 165$$

$$= C \left(\frac{Uc}{\nu}\right)^\delta \sqrt{\rho}$$

or $k = C \sqrt{\rho} R_e^\delta$ \dots 166

To obtain the actual relationship, $k/\sqrt{\rho}$ is plotted versus Reynold's No. as shown in Figure 48.

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Figure 49. Oxidation and Wear Bands on 52100 Pivot Balls. (Approx 20X)

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APPENDIX E

ANALYSIS OF PIVOT BALLS AFTER MDBTF II-2 TEST

Six 52100 steel balls used in the MDBTF II-2 tilting pad bearing test were submitted for evaluation of wear after a run time of 28 hours and 31 minutes at speeds ranging from 20,000 to 46,500 rpm. Macroscopic examination of the steel balls was made at about 30 magnification. Two observations were made:

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

These are illustrated in Figure 49. Positive indication that the wear band resulted after the corrosion attack is significant since it indicates that a corrosive attack probably occurred prior to operation. If this was the cause, it probably was oxidation and not mercury corrosion.

Metallographic examination substantiates this postulation of oxidation corrosion. Pitting and an oxidation layer were present as shown in the photomicrograph of Figure 50. Figure 51 illustrates the pitting and the typical microstructure of this alloy.

A hardness survey using a Vickers Hardness Tester (1000 gm load) was made. All of the readings were uniform from the center to the edge of the balls. A Vickers hardness (DPH) of 772 (Rockwell 63.5) was obtained.

No evidence of material removal by wear was noted. It is concluded that the oxidation which occurred prior to operation obscured any wear damage other than that shown in Figure 49.

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RDM 5730

500X

Figure 50. Oxide on Surface of 52100 Pivot Ball (Unetched)

RDM 5753

500X

Figure 51. Microstructure of 52100 Pivot Ball (Picral Etch)

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APPENDIX F

TILTING-PAD BEARINGS - MDBTF II-2A TEST ANALYSIS

Bearing hardware from MDBTF II-2, II-2A and vibration test were received by Advanced Materials Development for analysis. The hardware consisted of twelve 3/32-inch diameter balls (440C stainless steel), 6 tilting-pad bearing shoes (18-4-1), 2 bearing sleeves (BG-41), 2 bearing journals (18-4-1 tool steel), and 2 retaining plates (17-4PH). Four tests had been performed on the assembled bearings. These were designated shake test "A", endurance test "A", shake test "B", and endurance test "B". Each set of three balls had been used in only one of the four tests. The other hardware was designated "A" or "B". "A" hardware was used in both shake and endurance test "A"; "B" hardware had been used for shake and endurance test "B". The requested analysis included evaluation of erosion, fretting, wear, or other damage to the various pieces and hardness measurements on all parts.

Visual, macroscopic, and microscopic examinations were made to assess damage of the bearing components. Hardness of the 440C balls was taken on a Knoop hardness tester using a 100-gram load. Hardness of the remaining pieces was taken on a standard Rockwell Hardness Tester, "C" Scale. Hardness is recorded in Table 7.

Damage to the balls consisted of wear paths from the endurance tests. The balls from endurance test A also showed some discoloration, probably due to the corrosion. The wear depth on one ball from endurance test B was measured as 0.0004 inch (see Figure 52). The balls from the shake test "B" showed little evidence of deterioration other than the beginning of wear tracks. The most severe damage was suffered by balls from shake test "A". A corrosion layer approximately 0.0015-inch thick was present on the surface. Figure 53 is a typical cross section of the corroded surface. The attack is probably oxidation which occurred either prior to operation in the mercury environment or during operation. It has been found that the oxidation reaction is accelerated in a mercury environment; that is, attack by oxygen in the presence of mercury is more rapid and severe than attack in most other oxygen-containing environments (Ref. 24).

The balls were also measured and checked for flat spots. None were detected. Measurements are given in Table 8.

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TABLE 7

HARDNESS OF BEARING PARTS

<u>Part</u>	<u>Material</u>	<u>Hardness</u>		
		<u>Knoop (100 g)</u>	<u>Rockwell "C"</u>	
Ball 1	440C	770	62	Conversion
Ball 2	440C	835	64.5	Conversion
Ball 3	440C	865	65.5	Conversion
Ball 4	440C	850	65	Conversion
Bearing Shoe A	18-4-1		63.4	
Bearing Shoe B	18-4-1		62.4	
Sleeve A	BG-41		61.3	
Sleeve B	BG-41		60.9	
Journal A	18-4-1		61.9	
Journal B	18-4-1		63.9	
Retaining Plate A	17-4PH		30.2	
Retaining Plate B	17-4PH		31.4	

TABLE 8

MEASUREMENT OF BALLS

<u>Ball</u>	<u>Average Diameter</u>	<u>Variation</u>
Shake "A"	0.09381 in.	±0.00011 in.
Endurance "A"	0.09363 in.	{ +0.00005 in.
		{ -0.00002 in.
Shake "B"	0.09372 in.	±0.00002 in.
Endurance "B"	0.09370 in.	-0.00001 in.

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RDM 6048 250X
Unetched

Figure 52. Edge of Ball from Endurance Test "B" Showing Wear Track

0.0014 inch

RDM 6049 250X
Unetched

Figure 53. Edge of Ball from Shake Test "A" Showing Pits and Corrosion Layer

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06761-1

6.25X

Figure 54. Plate "A" - Eroded Area

06761-2

6.25X

Figure 55. Plate "B" - Eroded Area

NAA-SR-6320, VOLUME I
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The bearing sleeves were examined for damage in the pivot grooves, the feed hole and the threads. None was found other than normal wear in the pivot grooves. This wear is estimated to be less than 0.0005-inch deep. (Note: This is the lapped surface and not wear caused during operation.)

Damage to the retaining plates consisted of erosion on one face. The estimated depth is about 0.0005 inch. The affected areas are illustrated in Figure 54 and 55.

One pad from each bearing was sectioned through the pivot. No evidence of cavitation damage was found. The pads from both "A" and "B" had a wear track in the pivot, caused by the original lapping operation. A cross section of the track is illustrated in Figure 56. Depth of this lapped track is 0.0010 inch for pad "B" and about the same or a little less for pad "A".

In the course of examination for wear and cavitation damage, cracks in the tip of the pivot in pad B-1 were discovered. (These are not in the working area of the pivots but at the apex of the cone with which the ball does not come in contact.) The section was then etched, disclosing the structure illustrated in Figure 57. This structure is typical but more severe than damage reported previously as being due to electro-spark machining (Ref. 25). It seems likely that cracking was produced directly by the Eloxing operation and is due to the high stresses induced by the very rapid heating and cooling of the metal. The zone of metal affected is greater than that previously found in similar cases, and it contains the "white surface layer, unaffected by the etchant" described by Opitz (Ref. 25).

Since it is also possible that the cracking may have been caused by mercury embrittlement, i. e., the penetration of mercury concurrent with crack propagation, we examined available chemical tests for small quantities of mercury. There appeared to be no tests which are not affected by the elements present in the base material, with appropriate sensitivity to detect the minute amounts of mercury remaining in the crack subsequent to metallographic preparation. Consequently, we were unable to confirm or reject the mercury embrittlement mechanism.

The high thermally induced residual stresses resulting from Eloxing increase the possibility of premature failure by any one of several mechanisms: propagation of a crack formed during Eloxing, mercury embrittlement, fatigue. The bearing journal from bearing "A" was damaged by a cavitation-erosion mechanism. From the position of the eroded areas on the journal, damage coincides with the sides of the pads in the assembled bearing. A typical eroded area is pictured in Figure 58. Maximum depth of erosion is estimated as 0.001 inch.

Journal "B" was damaged much more extensively than journal "A" both in width and depth of the eroded area. Figure 59 illustrates typical damage. Maximum depth of erosion is about 0.0018 inch.

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RDM 6047

100X

Unetched

Figure 56. Pad "B" Pivot - Wear Track Cross Section

RDM 6060

250X

Marble's Etch.

Figure 57. Pad "B" Pivot - Cracking in Tip of Pivot Hole

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06761-4

6.25

Figure 58. "A" Journal - Cavitation-Erosion Damage

06761-6

6.25X

Figure 59. "B" Journal - Cavitation-Erosion Damage

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Of the damage reported, the most serious is the cavitation damage at the ends of the bearing journals. This can probably be alleviated by a design change which will reduce the amount of cavitation in the lubricant during operation. The next most severe damage is that produced in the pivots in the bearing pads. A substitute for Eloxing as a machining method is recommended to avoid premature failure. The only significant damage to the remaining parts is corrosion of the balls during shake test "A". Exclusion of oxygen from the system during the test will probably eliminate this type of corrosion.

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APPENDIX G

LAMINAR FLOW ANALYSIS OF AN UNPRESSURIZED THREE-SECTOR BEARING

Analytical solutions for non-pressure fed, laminar flow, three-sector bearings are presented in Theory of Hydrodynamic Lubrication by O. Pinkus and B. Sternlicht (Ref. 16, page 12). Numerical values of attitude angle, Sommerfeld number, and end leakage factors are presented with eccentricity ratio as the independent parameter for three-sector bearings of L/D ratios of 1.5; 1.0; 0.5; and 0.25. Although not explicitly stated in the text, it is reasonable to assume that the direction of the load is normal to one of the pads.

Because of its application in the SNAP 2 CRU, this analysis of the three-sector bearing is of interest despite the differences between the conditions assumed in the analysis and those existing in the CRU. Unfortunately, neither the book nor the paper (Ref. 4) in which the analysis appeared originally examine the case for loading into the groove. Since the load direction in a CRU is not a constant, this condition must also be analyzed. Reference 16 gives tabular values of the characteristics of partial bearings from which sector bearings are composed. However, the method of combining the partial bearings to obtain the characteristics of a sector bearing is given only in broad terms.

This appendix presents a method for obtaining the laminar flow characteristics of sector bearings using the tabular values presented in Theory of Hydrodynamic Lubrication. On the basis of the method developed by Oscar Pinkus, the load-carrying capacity of an unpressurized three-sector bearing was determined for the case of the load directed into the groove. Half-frequency stability was also investigated. The example is an analysis of a three-sector bearing whose L/D ratio = 0.5. The geometry of the three-sector bearing is shown in Figure 60.

On pages 93 through 111 of Reference 16 are tabulated the values of α/β , ϕ , S, $\bar{q}_{in}$, $\bar{q}_z$ and $(D/C_D) \times f$ for various ϵ of eccentrically loaded partial bearings of different L/D. These tables were used to derive the characteristics for the values of ϕ , S and q_z for various ϵ of two-, three- and four-groove bearings of different L/D. The results are shown on page 113 of Reference 16.

For a load directed downward into a groove of a three-sector bearing as in Figure 61, integration of the pressure distribution for each pad will give the forces F_1 and F_2 . The contribution of the upper pad to the load is negligible, since the film is divergent, except for large values of attitude angle. Figure 4-11a on page 111 of Reference 16 indicates that a similar assumption is made for the case of the load into the pad. Reduction of the pad arc by the axial grooves is ignored, and β for each of the three sectors is assumed to be 120 degrees.

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Figure 60. Three-Sector Bearing Geometry

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Figure 61. Three-Sector Bearing With Load Into Groove

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$\epsilon = 0.4$
 $L/D = 0.5$
 $\beta = 120^\circ$

Figure 62. Determination of Attitude Angle for Three-Sector Bearing with Load into Groove

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For equilibrium:

$$W = F_1 \sin (\alpha_1 - 30^\circ) + F_2 \cos \alpha_2 \quad \dots 167$$

$$O = F_1 \cos (\alpha_1 - 30^\circ) - F_2 \sin \alpha_2 \quad \dots 168$$

Since

$$\frac{1}{S} = \frac{(C_D/D)^2}{\mu N L D} \cdot W \quad \dots 169$$

and the diametral clearance, the diameter, the length, and speed are the same for the two pads:

$$\frac{1}{S_1} = \frac{(C_D/D)^2}{\mu N L D} F_1 \quad \dots 170$$

$$\frac{1}{S_2} = \frac{(C_D/D)^2}{\mu N L D} F_2 \quad \dots 171$$

The viscosity, μ , is assumed constant throughout the bearing, so that by substitution in Equation 168:

$$\frac{1}{S_1} \cos (\alpha_1 - 30^\circ) = \frac{1}{S_2} \sin \alpha_2 \quad \dots 172$$

The attitude angle of the bearing, as shown in Figure 60 is:

$$\phi = \alpha_1 + \phi_1 - 120 \text{ degrees} \geq 0 \quad \dots 173$$

$$= \alpha_2 + \phi_2 \leq 120 \text{ degrees} \quad \dots 174$$

For a given L/D , the value of ϕ may be found for selected eccentricity ratios by using the partial bearing tables in Reference 16. Thus, as in Figure 62, Equation 172 is solved by plotting the two sides of the equation as functions of ϕ defined by Equations 173 and 174. Their intersection gives the value of ϕ .

Equation 167 may be expressed as

$$\frac{1}{S} = \frac{1}{S_1} \sin (\alpha_1 - 30 \text{ degrees}) + \frac{1}{S_2} \cos \alpha_2 \quad \dots 175$$

The reciprocal of the Sommerfeld number may be established as shown in Figure 62. The friction factor, $(D/C_D) \times f$, and the end leakage factors are also shown.

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Figure 63. Attitude Angle vs Eccentricity Ratio for Unpressurized Three-Sector Bearing (Assuming Laminar Flow)

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The resulting attitude angle-eccentricity ratio curve is shown in Figure 63 which also shows the plot for the short journal bearing approximation of Dubois and Ocvirk (Ref. 9) for comparison. The load directions are upward and downward. The load capacity of the bearing is shown in Figure 64, as a plot of eccentricity ratio versus the reciprocal of the Sommerfeld number.

Because half-frequency instability has been of major concern in the bearing application of the SNAP 2 CRU, the spring rate along the line of centers is of interest. The threshold of instability has been defined (Reference 16) as

$$M \omega^2 \left(\frac{1}{K_S} + \frac{1}{K_R} \right) < 4 \quad \dots 176$$

For a rigid shaft, $1/K_S = 0$ and the speed at which half-frequency whirl will occur is

$$N_{cr} = \frac{1}{\pi} \sqrt{\frac{K_R}{M}} \text{ rps} \quad \dots 177$$

The radial film stiffness is defined as

$$K_R = \frac{2}{C_D} \times \frac{d F_r}{d \epsilon} \quad \dots 178$$

A graphical differentiation of the curves shown in Figure 65 yielded those in Figure 66. As is evident, the spring rate of the short journal bearing is greatest for a given eccentricity, and the spring rate for the three-sector bearing with the load into the groove is smallest. However, since the load establishes the eccentricity ratio, a cross plot of Figures 64 and 66 reveals that the three-sector bearing is stiffer than the short journal bearing for a given load. Figure 67 shows this conclusion.

In equation 177 the conventional solution determines the value of K_R for a given speed, e. g., 40,000 rpm for SNAP 2. The onset to whirl speed is then calculated. Because the radial film stiffness is non-linear, it seems proper to use N_{cr} as the speed at which K_R was evaluated. However, if the conventional technique is valid, Equation 177 may be expressed by

$$60 N_{cr} = \frac{60}{\pi} \sqrt{\frac{2}{M C_D}} \sqrt{\left(\frac{d F_r}{d \epsilon} \right)}_{\epsilon} \quad \dots 179$$

where $(dF_r/d\epsilon)_{\epsilon}$ is the film stiffness at a particular operating condition. As an example, assume that the following conditions apply:

Figure 64. Generalized Performance Characteristics of an Unpressurized Three-Sector Bearing
(Assuming Laminar Flow)

Figure 65. Force Component Along Line of Centers for an Unpressurized Three-Sector Bearing
(Assuming Laminar Flow)

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Figure 66. Radial Film Stiffness for an Unpressurized Three-Sector Bearing
(Assuming Laminar Flow)

Figure 67. Radial Film Stiffness vs Load for an Unpressurized Three-Sector Bearing
(Assuming Laminar Flow)

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$$L = 5/16 \text{ inch}$$

$$D = 5/8 \text{ inch}$$

$$C_D = 0.0015 \text{ inch}$$

$$60 N = 40,000 \text{ rpm}$$

$$\mu = 2.20 \times 10^{-7} \text{ lb-sec/in.}^2$$

$$M = 2.5/386 \text{ lb-sec}^2/\text{in.}$$

$$\begin{aligned} \frac{\mu N L D}{(C_D/D)^2} &= 2.20 \times 10^{-7} \times \frac{40,000}{60} \times \left(\frac{5}{16}\right) \left(\frac{5}{8}\right) \left(\frac{5}{8}\right)^2 \left(\frac{10^3}{1.5}\right)^2 \\ &= 4.97 \end{aligned}$$

$$\frac{60}{\pi} \sqrt{\frac{2}{M \times C_D}} = \frac{60}{\pi} \sqrt{\frac{2 \times 386 \times 10^3}{2.5 \times 1.5}} = 8660$$

Then if

$$W = 2.5; \quad 5; \quad 10; \quad 20 \text{ lb}$$

$$\frac{1}{S} = 0.502; \quad 1.004; \quad 2.01; \quad 4.02$$

From Figure 67 and equation 179

$$60 N_{\text{crtsp}} = 27.4; 37.7; 58.0; 97.3 \text{ K rpm, (tsp} \equiv \text{three sector - into pad)}$$

$$60 N_{\text{crtsg}} = 23.7; 33.5; 54.0; 94.7 \text{ K rpm, (tsg} \equiv \text{three sector - into groove)}$$

$$60 N_{\text{crsjb}} = 27.4; 36.2; 53.0; 85.6 \text{ K rpm. (sjb} \equiv \text{short journal bearings)}$$

Although this analysis is academic to the extent that

- 1) Laminar flow has been assumed
- 2) Hydrostatic effects of feed pads have been ignored
- 3) The partial bearing data were limited

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the technique and results are of interest. Qualitatively, it would seem that turbulent flow considerations would reduce the onset to whirl speed. However, an analysis of laminar and turbulent short journal bearings showed that turbulence permits a bearing to be stable at lower values of eccentricity ratio. Whether turbulence increases load capacity, i.e., lower eccentricities for a given load, was not readily discernible from the analysis.

The hydrostatic effects are more complicated, since the Reynolds equation from which equation 169 was developed did not consider them. Without considering hydrostatic effects, a reduction of diametral clearance for a given load would decrease the Sommerfeld number reciprocal. It is not clear from Figure 67 whether the reduction in the stiffness factor is offset by the effect of reduced clearance. For example, if $C_D = 0.0010$ inch,

$$\frac{\mu N L D}{(C_D/D)^2} = \left(\frac{15}{10}\right)^2 \times 4.97 = 11.2$$

For the loads previously considered:

$$W = 2.5; 5; 10; 20 \text{ lb}$$

$$\frac{1}{S} = 0.223; 0.446; 0.892; 1.784$$

Then, for the three-sector bearing with the load directed into the pad, the stiffness factor from Figure 67, is

$$\frac{(C_D/D)^2}{\mu N L D} \times \frac{d W_r}{d \epsilon} = 1.01; 1.68; 2.69; 8.50$$

While these values are lower than corresponding values for $C_D = 0.0015$ inch:

$$\frac{60}{\pi} \sqrt{\frac{2}{M C_D}} = \sqrt{\frac{15}{10}} \times 8660 = 10,610$$

and in 1000 rpm

$$60 N_{\text{crtsp}} = 34.7; 44.7; 56.5; 100.5$$

The value of N_{cr} will be increased further by the reduced clearance, since the hydrostatic spring rate will be increased as well.

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NOMENCLATURE FOR APPENDIX G

C_D	= diametral clearance, inch
D	= bearing diameter, inch
e	= shaft eccentricity, relative to bushing, inch
f	= coefficient of friction
F	= integrated hydrodynamic radial force, lb
F_r, F_t	= radial and tangential forces, lb.
h_o	= minimum film thickness, inch
K_S	= shaft flexural stiffness, lb/in.
K_R	= radial film stiffness
L	= bearing axial length, inch
M	= portion of mass of shaft supported by bearing, lb-sec ² /in.
N'	= shaft speed, rps
N_{cr}	= shaft speed at which it becomes unstable in half-frequency whirl, rps
$\bar{q}_{in}$	= dimensionless coefficient for lubricant flow into pad
$\bar{q}_z$	= dimensionless coefficient for end leakage from pad
S	= Sommerfeld number $= \frac{\mu N L D}{W} \times \left(\frac{D}{C_D} \right)^2$
W	= load, lb
α	= angle from leading edge of groove to load component, degrees
β	= arc angle of pad, degrees
ϵ	= eccentricity ratio = $2e/C_D$
μ	= absolute viscosity of lubricant, lb-sec/in. ²

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ϕ = altitude angle, from direction of load to line of center of shaft and bushing, degrees

Subscripts

1, 2 = pad number

r = along line of centers (radial)

t = tangential to line of centers

tsp = three-sector bearing, load into pad

tsg = three-sector bearing, load into groove

sjb = short journal bearing of Dubois and Ocvirk

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APPENDIX H

ANALYSIS OF BEARINGS FROM BETR 2 TEST

A failed turbine bearing (sleeve and bushing) and an alternator bearing (sleeve and bushing) which were removed from service were submitted for metallurgical analysis.

Turbine Bearing

The turbine bearing of BG-41 steel had failed by scoring and seizing. Since failures of this type have been covered thoroughly in the past, only a limited study was made of this example to determine the metallurgical soundness.

Metallographic examination indicated that the material was of good quality. No large inclusions, voids, or excessive dirt were evident, indicating that the steel was probably vacuum melted. The microstructure and hardness showed that the material was properly heat treated. Rockwell C hardness values of 56 to 57 (converted from DPH 620 to 627) were obtained on both the sleeve and the bushing.

Alternator Bearing

Visual examination of the sleeve and bushing of the alternator bearing indicated that some corrosion attack of the bushing had occurred; the sleeve did not seem to be affected. The bushing had reddish-brown corrosion products on the circumferential groove and in the supply pads and had pitting on the inner surface, especially around the supply pads.

Figure 68 indicates that the bushing was metallographically examined in the region around a supply pad. The unetched structure at Position A at the surface of the supply pad is shown in Figure 69. The metal is overlaid with an uneven, 0.0008-inch thick zone. This zone is probably a combination of scale and of metal from which the chromium had been leached.

Figure 68. Sketch of Sectioned Bearing. (Positions A and B Refer to the Microstructures Shown in Figures 69 and 70.)

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RDM 4684

AH-666

500X

Figure 69. Unetched Structure of Bearing at Supply Pad
(Position A in Figure 68)

RDM 4714

AH-666

100X, Cb etch.

Figure 70. Microstructure of Bearing (Position B in Figure 68)

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In contrast, the microstructure of the surface at Position B on the circumferential groove showed that corrosion attack was not severe. There does seem to be some chromium depletion since the chromium carbides (white particles) are not as heavily concentrated at the surface as in the interior. Otherwise, the structure is normal for a properly heat treated BG-41 steel. This was confirmed by Rockwell C hardness values of 55 to 58 for both the sleeve and the bushing. The material was low in inclusion content, indicating that the bearing was made of vacuum-melted stock. Spectrographic analysis identified the stock as BG-41.

The cause for pitting around the supply pads of the bushing cannot be definitely answered. The pits may have arisen from a cavitation type of attack which ripped out carbide particles and enlarged the resulting cavities by turbulent action. The pits ranged in diameter to about 0.003 inch, as compared to carbide particle diameters of 0.001 inch.

Spectrographic analysis of the reddish-brown corrosion product indicated that appreciable amounts of mercury, minor amounts of iron, magnesium and aluminum, and traces of copper existed. Of interest is the presence of the nonferrous elements magnesium, aluminum, and copper. These are not contained in BG-41 steel. It was also interesting that no chromium was present in the deposits. X-ray diffraction analysis on the limited material available was not conclusive but did point to mercury oxide and not iron oxides (rust).

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APPENDIX I

METHOD OF DETERMINING DYNAMIC STIFFNESS AND DAMPING

Theoretical Model: (Assumes no unidirectional loading)

The theoretical model to be used is a lumped mass, stiff rotor system with a known unbalance, i. e., the center of mass does not coincide with the center of geometry, as shown below:

Where:

S = shaft geometrical center

B = bearing center

G = mass center of shaft

$\delta = \overline{SB}$ = deflection of shaft center from bearing center,
i. e., the radius of the orbital path

$e = \overline{GS}$ = distance from mass center to geometrical center
of the shaft

ω_r = shaft rotational speed

ω_p = shaft precessional speed, i. e., orbital speed

Thus the forces acting on the shaft are:

- 1) Shaft Loading: This is the centrifugal force due to the shaft center being displaced from the bearing center. This force is in the direction from B to S, radially outward from B. The magnitude is

$$F_s = M \delta \omega_p^2 \quad \dots 180$$

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where $M = 1/2$ the shaft mass (for a two bearing system)

- 2) Unbalance Loading: This is the centrifugal force due to the shaft center of mass not coinciding with the shaft geometric center. This force is in the direction from S to G and leads F_S by the angle θ . The magnitude is

$$F_u = M e \omega_r^2 \quad \dots 181$$

- 3) Radial Bearing Force: This is the force due to the bearing stiffness. Its direction is from S to B, and it has a magnitude of

$$F_R = K_R \delta \quad \dots 182$$

where K_R = radial bearing film stiffness

- 4) Tangential Bearing Force: This is the force due to the viscous damping. This force will lag F_S by 90° in the direction of rotation and has a magnitude of

$$F_d = C \delta \omega_p \quad \dots 183$$

Thus, where c is the dampening coefficient, the force diagram acting on the shaft center S is as follows:

Summing the forces along $\overline{BS}$ gives

$$M \delta \omega_p^2 + M e \omega_r^2 \cos \theta - K_R \delta = 0 \quad \dots 184$$

Summing the forces normal to $\overline{BS}$ gives

$$M e \omega_r^2 \sin \theta - C \delta \omega_p = 0 \quad \dots 185$$

For synchronous whirl with a rotating load, $\omega = \omega_r = \omega_p$; therefore

Radial Stiffness, $K_R = M \omega^2 \left(1 + \frac{e}{\delta} \cos \theta\right) \dots 186$

Dampening Coefficient, $C = M \omega \left(\frac{e}{\delta} \sin \theta\right) \dots 187$

Reduction of Oscilloscope Pictures

The direction of shaft rotation for the present bearing fixtures, i. e., MDBTF, DBTF, BETF, is such that the notch on the shaft passes the horizontal probe 90 degrees before the vertical probe. Figures 71, 72, and 73 show oscilloscope pictures of the shaft locus for shaft speed from 0 to 40,000 rpm for an unbalance of 0.0106 in.-oz. For these photographs the shaft turns in the CCW direction.

The unbalance load, $M e \omega^2$, will be in the direction of the notch or will be referenced with respect to the notch. Since there are two probes (horizontal and vertical), two notches occur on the shaft locus.

When the vertical notch appears on the photograph, the unbalance load, $M e \omega^2$, will be vertical. The centrifugal shaft load, $M \delta \omega^2$, will be in the direction of the vertical notch. The shaft load, $M \delta \omega^2$, will lag the unbalance load, $M e \omega^2$, by the angle θ_s . The same resulting θ_s can be obtained by using the horizontal notch.

Shown below is the force diagram obtained for a sample shaft locus.

(a) 0 rpm

(b) 5K rpm

(c) 10K rpm

(d) 15K rpm

Figure 71. Influence of Unbalance on Shaft Orbit for an Orifice Compensated Three-Pad Bearing
(Unbalance = 0.0106 in-oz. Unidirectional Load = 0)

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(e) 20K rpm

(f) 25K rpm

(g) 30K rpm

(h) 36K rpm

Figure 71. Influence of Unbalance on Shaft Orbit for an Orifice
Compensated Three-Pad Bearing
(Unbalance = 0.0106 in-oz. Unidirectional Load = 0)

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(i) 40K rpm

Figure 71. Influence of Unbalance on Shaft Orbit for an Orifice
Compensated Three-Pad Bearing
(Unbalance = 0.106 in-oz. Unidirectional Load = 0)

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The shaft deflection, δ , is the radius of the shaft locus.

For Figure 73, at 40,000 rpm

$$\theta_s = 102^\circ$$

$$\delta_s = 2.2 \text{ cm} \times \frac{0.0016}{8 \text{ cm}} = 0.00044 \text{ in.}$$

Since the shaft locus is circular, the equation of the locus in the vertical direction is

$$y = \delta \cos(\omega t - \theta_s) \quad \text{where } t = 0 \text{ at notch.} \quad \dots 188$$

This curve is shown below.

If the shaft were perfectly round and concentric to the bearings at the point where the shaft locus is measured, then the values C and K_R could be found once δ and θ_s were found from the photograph. But since runout is generally present, i. e., $\delta \neq 0$ at $N = 0$), the values for θ and δ will have to be corrected by subtracting the runout. The equation for runout is

$$y_r = \delta_r \cos(\omega t - \theta_r) \quad \dots 189$$

The corrected values of δ and θ can now be found from the following equations:

$$y_c = y_s - y_r = \delta_c \cos(\omega t - \theta_c) \quad \dots 190$$

$$y_c = \delta_s \cos(\omega t - \theta_s) - \delta_r \cos(\omega t - \theta_r) \quad \dots 191$$

$$y_c = \delta_s \cos \omega t \cos \theta_s + \delta_s \sin \omega t \sin \theta_s - \delta_r \cos \omega t \cos \theta_r - \delta_r \sin \omega t \sin \theta_r \quad \dots 192$$

$$y_c = \cos \omega t (\delta_s \cos \theta_s - \delta_r \cos \theta_r) + \sin \omega t (\delta_s \sin \theta_s - \delta_r \sin \theta_r) \quad \dots 193$$

NOTE: Subscript c refers to corrected
 Subscript r refers to measured run out
 Subscript s refers to measured data, uncorrected.

At $\omega t = \theta_c$, $y_c = \delta_c$ and $\frac{d(y_c)}{d(\omega t)} = 0$

$$\frac{d(y_c)}{d(\omega t)} = -\sin \theta_c (\delta_s \cos \theta_s - \delta_r \cos \theta_r) + \cos \theta_c (\delta_s \sin \theta_s - \delta_r \sin \theta_r) = 0 \quad \dots 194$$

Therefore:

$$\tan \theta_c = \frac{\delta_s \sin \theta_s - \delta_r \sin \theta_r}{\delta_s \cos \theta_s - \delta_r \cos \theta_r} = \frac{\delta_c \sin \theta_c}{\delta_c \cos \theta_c} \quad \dots 195$$

$$*\theta_c = \tan^{-1} \left(\frac{\delta_s \sin \theta_s - \delta_r \sin \theta_r}{\delta_s \cos \theta_s - \delta_r \cos \theta_r} \right) \quad \dots 196$$

Hence:

$$*\delta_c = \cos \theta_c (\delta_s \cos \theta_s - \delta_r \cos \theta_r) + \sin \theta_c (\delta_s \sin \theta_s - \delta_r \sin \theta_r) \quad \dots 197$$

$$\delta_c = \left[\cos \theta_c \left(\frac{\cos \theta_c}{\sin \theta_c} \right) + \sin \theta_c \right] (\delta_s \sin \theta_s - \delta_r \sin \theta_r) \quad \dots 198$$

$$\delta_c = \left(\frac{\cos^2 \theta_c + \sin^2 \theta_c}{\sin \theta_c} \right) (\delta_s \sin \theta_s - \delta_r \sin \theta_r) \quad \dots 199$$

Hence:

$$\delta_c = \frac{\delta_s \sin \theta_s - \delta_r \sin \theta_r}{\sin \theta_c} \quad \dots 200$$

or

$$\delta_c = \frac{\delta_s \cos \theta_s - \delta_r \cos \theta_r}{\cos \theta_c} \quad \dots 201$$

The runout for the locus in Figure 71 is

$$\delta_r = 0.00005 \text{ (See 0 speed)}$$

$$\theta_r = 0$$

*These equations assume that the unbalance load is in the direction of the notch.

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Data Reduction of Sample Photograph (Figure 73 for 40,000 rpm)

The pertinent data for Figure 73 are as follows:

$$M = 2.5 \text{ lb/g} = 0.00648 \frac{\text{lb-sec}^2}{\text{in.}}$$

$$\text{Unbalance} = 0.0106 \text{ in.-oz}$$

$$\delta_s = 0.00044$$

$$\theta_s = 102^\circ$$

$$\delta_r = 0.00005 \text{ in.}$$

$$\theta_r = 0^\circ$$

$$\omega = 40,000 \frac{\text{rev}}{\text{min}} \times \frac{2\pi}{60} = 4,200 \frac{\text{rad}}{\text{sec}}$$

- 1) Correct the data for runout:

Using Eq. 196 to determine θ_c ,

$$\theta_c = \tan^{-1} \left[\frac{(0.00044) \sin 102 - 0.00005 \sin 0}{(0.00044) \cos 102 - 0.00005 \cos 0} \right]$$

$$\therefore \theta_c = 106.9^\circ$$

Using Eq. 200 to determine δ_c

$$\delta_c = \frac{0.00044 (\sin 102) - 0}{\sin 106.9} = 0.00045 \text{ inch}$$

- 2) Determine e, distance between geometric and mass centers:

$$e = \frac{\text{unbalance (in-oz)}}{\text{Mass (oz)}} = \frac{0.0106 \text{ in-oz}}{2.5 \text{ lb} \times \frac{16 \text{ oz}}{\text{lb}}} = 0.000265 \text{ in.}$$

$$e = 265 \mu \text{ inches}$$

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- 3) Determine K_R , the bearing radial film stiffness:

Using Eq. 186,

$$K_R = 0.00648 (4,200)^2 \left[1 + \frac{0.000265}{0.00045} \cos 106.9^\circ \right]$$

$$K_R = 95,800 \text{ lb/in.}$$

- 4) Determine C, the damping coefficient:

Using Eq. 187,

$$C = (0.00648)(4,200) \left[\frac{0.000265}{0.00045} \sin 106.9^\circ \right]$$

$$C = 15.5 \frac{\text{lb-sec}}{\text{in.}}$$

In the above calculations, the bending of shaft due to the unbalance load was assumed to be negligible.

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APPENDIX J

ANALYSIS OF DOUBLE-ACTING SPIRAL-GROOVE THRUST BEARING

If a set of parallel grooves is cut across the surface of a smooth plate at an inclination to its edges, and a second smooth plate, separated from the first by a liquid lubricant, is moved over the first plate in a direction parallel to the edges, then the lubricant will be dragged along the grooves so that it has a component of flow in the direction of movement of the sliding plate. If the flow is retarded, a pressure gradient will be built up, thereby tending to force the plates apart. This principle is applied to self-pumping bearings.

Figure 74 shows a simple arrangement in which the grooves are all in the same direction and only extend partially across. The analysis is applied to this configuration. In applying the analysis to annular shaped thrust bearings, when the grooves have a spiral form, polar coordinates should be used. However, if the width of the bearing is small compared with its mean radius, only a small error will be incurred by using Cartesian coordinates and treating the edges of the grooves as parallel straight lines instead of concentric spirals. This condition is assumed to apply to the bearing shown in Figure 75.

The analysis is based on the application of the Reynolds equation. The velocity components and pressure gradients are described by the equations:

$$\mu \Delta^2 u = \frac{\delta P}{\delta x} \quad . . . 202$$

$$\mu \Delta^2 v = \frac{\delta P}{\delta y} \quad . . . 203$$

$$\mu \Delta^2 w = \frac{\delta P}{\delta z} \quad . . . 204$$

Where:

μ = Absolute viscosity of lubricant

u_0 = Velocity of moving plate in x-direction

$u=u(z)$ = Longitudinal component of flow of lubricant

$v=v(z)$ = Transverse component of flow of lubricant

$w=w(z)$ = Normal component of flow of lubricant

$h=h(x, y)$ = Separation of plates

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Figure 72. Simplified Spiral-Groove Bearing

Figure 73. Spiral-Groove Thrust Bearing

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In general, velocities normal to the plates will be much smaller than velocities parallel to the plates; that is $w \ll u$ and v . Then, as a first approximation,

$$\frac{\delta P}{\delta z} = 0 \quad \dots 205$$

Also u and v vary much more rapidly in the z -direction than in the x - and y -direction, so that

$$\frac{\delta^2 u}{\delta x^2} \ll \frac{\delta^2 u}{\delta z^2} \quad \dots 206$$

Then equations (202), (203), and (204) reduce to:

$$\mu \frac{\delta^2 u}{\delta z^2} = \frac{\delta P}{\delta x} \quad \dots 207$$

$$\mu \frac{\delta^2 v}{\delta z^2} = \frac{\delta P}{\delta y} \quad \dots 208$$

$$0 = \frac{\delta P}{\delta z} \quad \dots 209$$

Applying the boundary conditions which indicate that when $z = 0$, $u = 0$, and $v = 0$ and when $z = h$, $u = u_o$ and $v = 0$; the solutions of the above equations are

$$u = -\frac{1}{2\mu} z (h - z) \frac{\delta P}{\delta x} + u_o \frac{z}{h} \quad \dots 210$$

$$v = -\frac{1}{2\mu} z (h - z) \frac{\delta P}{\delta y} \quad \dots 211$$

The mass flows in the x - and y -directions are:

$$I = \int_0^h \rho u \, dz = \left[-\frac{h^3}{12\mu} \frac{\delta P}{\delta x} + \frac{h}{2} u_o \right] \rho \quad \dots 212$$

$$J = \int_0^h \rho v \, dz = -\frac{h^3}{12\mu} \frac{\delta P}{\delta y} \rho \quad \dots 213$$

Inserting equations 212 and 213 into the continuity equation

$$\frac{\delta I}{\delta x} + \frac{\delta J}{\delta y} = 0 \quad \dots 214$$

The following equation results

$$\frac{\delta}{\delta x} (h^3 \rho \frac{\delta P}{\delta x}) + \frac{\delta}{\delta y} (h^3 \rho \frac{\delta P}{\delta y}) = 6 \mu u_o \frac{\delta}{\delta x} (h \rho) \quad \dots 215$$

which is known as Reynolds equation. For an incompressible fluid, ρ is constant and can be cancelled from the equation. This equation was solved for the inclined groove bearing, Figure 74, by R. T. P. Whipple (Ref. 20). The solution can be written in the form

$$\frac{dP}{dy} = E - RJ \quad \dots 216$$

Where:

$$E = 6 u_o \mu \frac{a_1 a_2 (h_1^3 - h_2^3) (h_1 - h_2) \sin 2 \theta}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2 h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2 \theta} \quad \dots 217$$

$$R = \frac{12 \mu}{\rho} \frac{2 (a_1 + a_2) (h_1^3 a_2 + h_2^3 a_1)}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2 h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2 \theta} \quad \dots 218$$

Equation 216 is the basic equation. The term E, which is proportional to u_o and μ and independent of ρ , represents the driving force. It vanishes when θ is zero or 90° . The term $-RJ$ corresponds to the resistance.

Applying this to the bearing of Figure 74 where μ and ρ are constant, the general solution is

$$\int_{P_o}^P dP = \int_o^z (E - RJ) dy \quad \dots 219$$

or $P - P_o = (E - RJ)y \quad \dots 220$

Where:

P = Pressure in grooved part, psia

P_o = Supply pressure, psia

J = Mean mass flow in y-direction, lb/sec

y = Distance in y-direction (Figure 74)

In particular where $y = b, P = P_1$ at the end of the grooves

$$P_1 - P_o = (E - RJ)b \quad \dots 221$$

If the pressure is P_2 at the outer edge of the ungrooved portion

$$P_1 - P_2 = R' Jc \quad \dots 222$$

Where:

$$R' = \frac{12\mu}{h_2^3 \rho} \quad (= R \text{ where } h_1 - h_2 = 0)$$

From equations 221 and 222

$$J = \frac{Eb - (P_2 - P_o)}{Rb + R'c}, \quad \frac{\text{lb}}{\text{sec-in.}} \quad \dots 223$$

(For a double face thrust bearing, the total flow will be the sum of the flows through each side obtained from equation 223.)

And

$$(P_1 - P_o) = \frac{R'bc E + Rb (P_2 - P_o)}{Rb + R'c} \quad \frac{\text{lb}}{\text{in.}^2} \quad \dots 224$$

The lubricant, flow, and pressure profile can be calculated from equations 203 and 224.

The drag per unit area of the sliding plate is

$$D = \mu \frac{\delta u}{\delta z} \bigg|_{z=h} = \frac{h}{2} \frac{\delta P}{\delta x} + \frac{\mu u_o}{h} \quad \dots 225$$

Figure 74. Double-Acting Spiral-Groove Thrust Bearing Schematic Showing Pressure Profile

The average drag $\bar{D}$ is

$$\bar{D} = \frac{1}{a_1 + a_2} \int_0^{(a_1 + a_2)} D \, dx \quad \dots 226$$

$$\bar{D} = \frac{h_1 - h_2}{2(a_1 + a_2)} \left[P(a_1) - P(0) \right] + \mu u_o \left(\frac{a_1}{h_1} + \frac{a_2}{h_2} \right) \quad \dots 227$$

$$\bar{D} = \frac{\mu u_o}{a_1 + a_2} \left[\frac{a_1}{h_1} + \frac{a_2}{h_2} + \frac{3 a_1 a_2 (h_1 - h_2)^2 \cos^2 \theta}{h_1^3 a_2 + h_2^3 a_1} \right] + \frac{a_1 + a_2}{u_o \rho} \frac{E}{R} \quad (E-RJ) \quad \frac{\text{lb}}{\text{in.}^2} \quad \dots 228$$

The bearing power loss is calculated from Eq. 228.

A thrust equation is developed for a double face thrust bearing by considering the bearing shown in Figure 76.

Thrust capacity of this dual faced thrust bearing is obtained from the pressure profiles on the two faces. An equation is written for the maximum load capacity, based on a minimum film thickness of 0.0002 inch, assuming that a total axial clearance exists so that there is a linear pressure drop on the unloaded side of the bearing. A sketch of the pressure profiles is given in Figure 76.

Considering the pressure on the outside faces of the thrust washers as part of the applied thrust load, the net thrust capacity of the double thrust bearing can be obtained by analyzing the two inside faces.

On the loaded side of the bearing the pressure increases linearly from P_0 to P_1 in the grooved portion. In the ungrooved portion the pressure drops from P_1 to P_2 linearly. On the unloaded side, however, where the clearance is large, the pressure at the inlet to the grooves, P_0 , is assumed to decrease linearly along the whole width of the bearing face to P_2 at the outer edge. This assumption is reasonable since it must be true when the clearance on that side becomes very large. The pressure distribution available for load carrying is then the difference between the pressures on the two sides.

The thrust equation can then be written as

$$T = \left(\int p \, dA \right)_{\text{Loaded}} - \left(\int p \, dA \right)_{\text{Unloaded}} \quad \dots 229$$

which, when written for the area segments, is

$$T = \left[\frac{1}{2} (P_o + P_1) b + \frac{1}{2} (P_1 + P_2) c \right] - \left[\frac{1}{2} (P_o + P_2) (b + c) \right] \quad \dots 230$$

Where: A = Surface exposed to pressure P_1 , in.²

T = Thrust per inch of mean circumference, lb/in.

Rearranging equation 230 gives

$$T = \frac{1}{2} \left[(P_1 - P_o) (b + c) + (P_o - P_2) b \right] \quad \dots 231$$

Equation 221 is substituted for $(P_1 - P_o)$

$$T = \frac{1}{2} \left[(b + c) \left\{ \frac{R' b c E + R b (P_2 - P_o)}{R b + R' c} \right\} + (P_o - P_2) b \right] \quad \dots 232$$

This simplifies to

$$T = \frac{1}{2} \left[\frac{R' b c E (b + c) + b c (R' - R) (P_o - P_2)}{R b + R' c} \right] \quad \dots 233$$

This then represents the maximum thrust capacity of the double face thrust bearing ($h_2 = 0.0002$ inch minimum).

In summary, the derivation of pressure distribution, lubricant flow, and drag equations is given for a single thrust face.

- 1) The pressure rise across the grooved portion is linear and is given by equation 224

$$P_1 = P_o + \frac{R' b c E + R b (P_2 - P_o)}{R b + R' c}$$

- 2) Drag is given by equation 228

$$\bar{D} = \frac{\mu u_o}{a_1 + a_2} \left[\frac{a_1}{h_1} + \frac{a_2}{h_2} + \frac{3 a_1 a_2 (h_1 - h_2)^2 \cos^2 \theta}{h_1^3 a_1 + h_2^3 a_1} \right] \frac{a_1 + a_2}{u_o \rho} \frac{E}{R} (E-RJ)$$

- 3) The lubricant flow is given by equation 223

$$J = \frac{Eb - (P_2 - P_o)}{Rb + R'c}$$

- 4) Equation 223 was derived for maximum thrust capacity based on minimum film thickness for a double faced thrust bearing.

$$T_{\max} = 1/2 \left[\frac{R'bcE (b+c) + bc (R'-R) (P_o - P_2)}{Rb + R'c} \right]$$

These relationships have been established for the spiral groove thrust bearing neglecting the effect of the annular orifice.

Hydrostatic Thrust Capacity

To obtain the theoretical hydrostatic load capacity of this bearing the flow characteristics of a 0.0048-inch diametral restriction must be evaluated. The pressure drop across an annular orifice restriction is given by

$$(P_s - P_o) = 0.582 \times 10^{-6} \left[\frac{\frac{\lambda}{2s} + 1.5}{\rho (D_i^2 - D_2^2)^2} \right] Q^2 \quad \dots 234$$

Where:

$(P_s - P_o)$ = Pressure drop across annular orifice

Q = Mercury flow, lb/min

ρ = Density, lb/in.³

λ = Length, in.

D_i = Bore diameter, in.

D_2 = Shaft diameter, in.

s = Radical Clearance, in.

$$\lambda = \frac{96}{R_e} \quad (R_e < 2320); \quad \lambda = \frac{0.427}{R_e^{1/4}} \quad (R_e > 2320)$$

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For the concentric case and fully eccentric cases, respectively:

$$\lambda = \frac{96}{2.5 R_e} \quad (R_e < 2320) \text{ and } \lambda = \frac{0.427}{1.21 R_e^{1/4}} \quad (R_e > 2320)$$

Reynolds number is given by

$$R_e = \frac{2 s v}{\gamma} \quad \dots 235$$

Where:

v = velocity, ft/min

γ = kin. visc, ft^2/hr

The velocity v is obtained from

$$v = \frac{Q/\rho}{A} \quad \dots 236$$

Evaluation of equation 236 for this bearing with mercury at 70°F results in

$$v = 75.8 Q \text{ ft/min} \quad \dots 237$$

Substitution of equation 237 into equation 235 gives

$$R_e = 437 Q \quad \dots 238$$

Substitution of the selected values of λ , s , D_1 , D_2 and ρ in equation 234 gives

$$(P_s - P_o) = 0.142 (18.75 \lambda + 1.5) Q^2 \quad \dots 239$$

Evaluation of equations 238 and 239 is given in Table 9 for various flow rates of mercury. Only the concentric case has been taken since the clearance of the annular orifice is more than 3 times that of the journal bearings. If the journal bearing is operating at an eccentricity ratio of 0.5, the annular orifice eccentricity is less than 0.16, which is negligible for effect on flow. The flow curve of this annular orifice is plotted in Figure 77.

Another restriction to flow of the mercury is the clearance between the thrust faces. The flow through this restriction, which is in series with the annular restriction, must be evaluated for the condition of zero speed. The flow through one face is expressed by equation 223.

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TABLE 9

FLOW OF MERCURY THROUGH ANNULAR ORIFICE

D = 0.030 inch
 $C_D = 0.0048 \text{ inch} = 2s$
 $\lambda = 0.090 \text{ inch}$
 T = 70°F

$\frac{Q \text{ lb/min}}{\mu}$	$\frac{Re}{\mu}$	λ_c	$\frac{18.75 \lambda_c}{\mu}$	$\frac{18.75 \lambda_c + 1.5}{\mu}$	$\frac{P}{\text{psi}}$
2	874	0.110	2.06	3.56	2.0
4	1750	0.0549	1.03	2.56	5.7
6	2620	0.0595	1.11	2.61	13.2
8	3500	0.0555	1.04	2.54	23.0

TABLE 10

FLOW OF 70°F MERCURY THROUGH BEARING CLEARANCE

$\frac{h_2}{\mu}$	$\frac{E}{\mu}$	$\frac{R}{\mu}$	$\frac{R^1}{\mu}$
0.0002	0	58,200	688,000
0.0004	0	23,600	85,800
0.0006	0	10,000	25,400
0.0008	0	4,970	10,700

When $h_2 = 0.0002$

$\frac{(p_0 - p_2)}{\mu}$	$\frac{J, \text{ lb/sec-in.}}{\mu}$	$\frac{Q \text{ lb/min}}{\mu}$
0	0	0
50	0.00129	0.113
100	0.00258	0.226
150	0.00387	0.338
200	0.00517	0.452

$h_2 = 0.0004$

0	0	0
50	0.00755	0.66
100	0.0151	1.32
150	0.0226	1.98
200	0.0302	2.64

$h_2 = 0.0006$

0	0	0
50	0.0218	1.91
100	0.0437	3.82
150	0.0655	5.73
200	0.0874	7.65

$h_2 = 0.0008$

0	0	0
50	0.0476	4.16
100	0.0952	8.33
150	0.143	12.5
200	0.191	16.7

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Figure 75. Hydrostatic Flow Curves (Theoretical) For Spiral-Groove Thrust Bearing

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The mercury flow for this bearing has been evaluated by a computer; it is tabulated in Table 10. The flow of mercury for the various clearances is superimposed on the annular orifice flow curve in Figure 77.

Since the two restrictions are in series and the flow through one must be equal to the flow through the other, the intersection of the flow curves represents the flow through both restrictions with their respective pressure drops. Their values are summarized below:

h_2	Q	$(p_s - P_0), \text{ psi}$	$p_0, \text{ psi}$	
0.0002	≈ 0.7	≈ 0.5	214	} For $P_s = 214 \text{ psi}$
0.0004	3.0	3.8	210	
0.0006	7.6	21.0	193	
0.0008	12.8	60.0	154	

Therefore, two points of operation are available for the dual bearing with a total axial clearance of 0.0010 inch.

h_{2a}	h_{2b}	Q_a	Q_b	Q_{tot}	p_{0a}	p_{0b}
0.0002	0.0008	0.7	12.8	13.5	214	154
0.0004	0.0006	3.0	7.6	10.6	210	193

To determine the thrust capacity of the bearing, the pressure profile along the thrust face radius has to be calculated. The pressure at the end of the grooves, P_1 , is calculated from equation 224.

h_2	p_0	p_1
0.0002	214	178
0.0004	210	128
0.0006	193	100
0.0008	154	74

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The thrust capacity is then calculated from equation 230.

h_{2a}	h_{2b}	T_a	T_b	T_{net}
0.0002	0.0008	37.3	20.5	16.8 lb
0.0004	0.0006	30.9	26.4	4.5 lb

The theoretical hydrostatic load capacity of this bearing is plotted in Figure 76.

Figure 76. Predicted Hydrostatic Load Capacity of Spiral-Groove Thrust Bearing

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NOMENCLATURE FOR APPENDIX J

A = Bearing area, in.²

$a_1 \cos \theta$ = Width of grooves, in.

$a_2 \cos \theta$ = Width of ridges, in.

b = Width of grooved part of plate, in.

c = Width of ungrooved part of plate, in.

D = Drag per unit area of sliding plate, lb/in.²

$\bar{D}$ = Average drag, lb/in.²

D_o = Bearing OD, in.

D_i = Bearing ID, in.

D_m = Bearing mean diameter, in.

D_2 = Diameter of shaft, in.

$$E = \frac{6 \mu u_o a_1 a_2 (h_1^3 - h_2^3) (h_1 - h_2) \sin 2 \theta}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2 h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2 \theta}$$

h_1 = Separation in grooves, in.

h_2 = Separation outside grooves, in.

I = Mass flow in x-direction

J = Mass flow in y-direction, lb/sec-in.

ℓ = Annular orifice length, in.

P = Pressure

P_o = Pressure at inlet to grooves, lb/in.²

P_s = Mercury supply pressure, lb/in.²

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P_1 = Pressure at inner end of grooves, psi

P_2 = Drain pressure, psi

$$R = \frac{12 \mu}{\rho} \frac{2 (a_1 + a_2) (h_1^3 a_2 + h_2^3 a_1)}{(h_1^3 + h_2^3)^2 a_1 a_2 + 2 h_1^3 h_2^3 (a_1^2 + a_2^2) + (h_1^3 - h_2^3)^2 a_1 a_2 \cos 2 \theta}$$

$$R^1 = \frac{12 \mu}{\rho h_2^3}$$

R_e = Reynolds number, $\frac{2 s V}{\nu}$

s = Radical clearance of annular orifice, in.

$T_{\max}$ = Maximum thrust capacity of double faced thrust bearing, lb/in.

v = Lubricant velocity through annular orifice, ft/min

u_o = Velocity of sliding plate, in./sec

u, v, w = Components of velocity

Q = Mercury mass flow, lb/min

x, y, z = Space coordinates

μ = Lubricant viscosity, lb-sec/in.²

ν = Kinematic viscosity, ft²/hr

θ = Groove angle

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