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THE SNAP II POWER CONVERSION SYSTEM
TOPICAL REPORT No. 6 BEARING DESIGN & DEVELOPMENT

STAFF RESEARCH AND DEVELOPMENT

THOMPSON PRODUCTS DIVISIONS

Thompson Ramo Wooldridge Inc.

CLEVELAND, OHIO

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BEARING DESIGN & DEVELOPMENT

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ABSTRACT

SNAP II is the designation for a 3KW nuclear auxiliary power unit to be used in a satellite vehicle. The SNAP II system consists of a reactor heat source, a mercury Rankine power conversion system and controls. The system rotating components, which include the turbine, alternator, and pumps, are supported on a common shaft by mercury lubricated bearings. Bearing design details and test results are presented in this report. This work was performed under a subcontract to Atomics International as part of the Atomic Energy Commission Contract (At 911-1)Gen-8.

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1.0 SUMMARY

The SNAP II system is a 3KW nuclear auxiliary power unit intended to be used in a satellite vehicle. The SNAP II power plant utilizes a mercury vapor Rankine cycle with a reactor as a heat source. Useful electrical power output is obtained from an alternator directly coupled to the turbine shaft.

The work presented in this report was performed under a subcontract to Atomics International with three prime objectives:

1. Select a rotating unit bearing system that would best suit the SNAP II requirements.
2. Design, fabricate, and test such bearings in order to determine operating characteristics.
3. Determine areas of possible bearing improvements.

A preliminary analysis conducted on various types of bearings has indicated that hydrodynamic type journal and thrust bearings lubricated with a portion of the mercury from the condensate return pump would best suit the SNAP II requirements.

Experimental results have confirmed the bearing design approach. Stable bearing operation has been obtained at speeds in excess of the 40,000 rpm design objective with simulated loads of 1 to 10g's in the radial direction and 0-2g's in the axial direction. Total power consumption of the bearing system is approximately 550 watts at the design speed.

2.0 INTRODUCTION

For the requirements of the SNAP II Combined Rotating Unit, preliminary investigations resulted in the selection of liquid mercury lubricated journal and thrust bearings. The object of this report is to review the reasons for the selection, present the design, trace the background and development history of the bearings, discuss the test results, and summarize the bearing operating characteristics.

3.0 DESIGN

3.1 Specification

The SNAP II system operating parameters required that the bearings operate under the following conditions:

Speed	40,000 rpm
Lubricant	Liquid mercury
Lubricant - Supply temperature	430°F
Lubricant - Supply pressure	140 psia
Lubricant - Drain pressure	6 psia
Lubricant - Flow rate	20 ppm (Max.)
Radial load per bearing	0-25 lb
Axial load	0-15 lb
Maximum power loss	600 watts
Life (continuous operation)	1 year

With the exception of the bearing loads, the above conditions are a result of thermodynamic considerations in the SNAP II power conversion system.

The combined rotating unit (CRU) shaft assembly (shown in Figure 1) weight was estimated to be 5 pounds with the center of mass midway between bearing supports. With launch accelerations in a plane normal to the shaft equal to 10g's, the resultant radial load per bearing is 25 pounds. During orbital operation, the radial load approaches zero with only negligible internal and unbalance loads being applied.

Net internal axial loading on the shaft (from turbine, Hg pump, and Nak pump) of 5 pounds maximum plus a 2g launch acceleration in a plane parallel to the shaft cause a maximum of 15 pounds thrust loading.

3.2 Selection of Bearing Type

Selection of the type bearing used for the SNAP II application was governed by the following power system design requirements:

1. The combined rotating unit is required to operate in space continuously for a period of one year. During this time no maintenance is possible.
2. The power conversion system must be hermetically sealed.
3. The thermodynamic cycle fluids must not be contaminated with any foreign material (such as a lubricant).

COMBINED ROTATING UNIT (CRU) SHAFT CONFIGURATION

FIGURE 1

The cycle fluid contamination problem required that either the cycle fluid itself be used as a lubricant; that a high-speed, high-temperature zero leakage shaft seal be used to separate the lubricated from the unlubricated package areas; or that a dry (unlubricated) bearing system be developed.

The state of the art of high-speed, high-temperature dry rubbing contact materials precluded the use of a conventional type shaft seal or dry running bearings because of the short life expectancy as compared to the one year of continuous operation requirement.

Life expectancy also ruled out rolling contact type bearings either dry or lubricated with the secondary cycle fluid (mercury). The non-wettability of mercury makes it inadequate for the lubrication requirements of a rolling bearing. Furthermore its high density would cause excessive power loss and possible speed variations due to impingement of high momentum fluid on the moving parts of a rolling contact bearing.

It was apparent that a fluid support type bearing using liquid mercury was probably the most feasible for the SNAP II application. The following bearing types were analyzed:

1. The hydrostatic journal and thrust bearing.
2. The hydrodynamic journal and thrust bearing.
3. The hydrosphere.

Hydrostatic Bearings – Hydrostatic bearings derive their load support from a static fluid pressure produced by an external pumping source. The advantages of a hydrostatic bearing are:

1. High load support at low speeds.
2. Good load support with low viscosity fluids.

Disadvantages of the hydrostatic bearing are:

1. Reliance on external pressure source.
2. Sensitivity of hydraulic resistances.
3. Extreme sensitivity to alignment.
4. Sensitivity to manufacturing tolerances.
5. Sensitivity to thermal distortions.

Hydrodynamic Bearings - Hydrodynamic bearings derive their load support from dynamic fluid pressures produced within the bearing by the relative motion of the rotating and stationary members. Advantages of hydrodynamic type bearings are:

1. Requires external pressure source only to "get lubricant to the bearing".
2. Has good load capacity at high speeds.
3. Requires no hydraulic resistance system for control of load support.
4. Is relatively insensitive to normal manufacturing tolerances.
5. Is relatively insensitive to small thermal distortions.

Disadvantages of hydrodynamic type bearings are:

1. Lack of lubricant film at start-up.
2. Low load capacity at low speeds.
3. Alignment problems.
4. Tendency of lubricant film whirl at high speeds.

Hydrosphere Bearings - The hydrosphere is a spherical bearing which is capable of both radial and axial load support. Basically, radial load capacity is derived from hydrodynamic forces and axial load capacity from hydrostatic pressures. Advantages of the hydrosphere bearing are:

1. High axial load capacity.
2. Relative insensitivity to alignment.
3. Relatively low lubricant flow.

Disadvantages of the hydrosphere bearing are:

1. Introduction of lubricant at shaft ends makes overhung components difficult.
2. Complexity of hydraulic resistances.

3. Reliance on external pressure source.
4. Relatively low radial load capacity.
5. High fabrication costs.
6. Very close tolerances.
7. Sensitivity to thermal gradients in the axial direction.

Three major factors influenced the choice of a hydrodynamic journal-thrust combination over either the hydrostatic journal-thrust or the hydrosphere. First, a hydrostatic type bearing was deemed inadvisable because of its reliance on the package pump discharge pressure for load support. The negligible pump output pressure at speeds below 20,000 rpm make the hydrostatic bearing system inadequate for start-up, shut-down, and low speed operations. Furthermore the requirement of an intricate hydraulic resistance system could make the hydrostatic system somewhat unreliable when the operating temperatures and life requirement are considered.

The hydrosphere offers the definite advantage of alleviating many of the alignment problems inherent in the type package used in the SNAP II system. However, as shown in Figure 2, the hydrosphere is necessarily end fed. For this reason the relatively heavy overhung NaK pump

BASIC HYDROSPHERE BEARING CONFIGURATION

FIGURE 2

incorporated into the CRU does not easily lend itself to the basic hydrosphere bearing configuration. It can be seen from Figure 3 that an outboard shaft would require that an efficient high-pressure, high-speed shaft seal be incorporated into the hydrosphere system between the bearing supply and NaK pump.

FIGURE 3

Furthermore the weight of the NaK pump requires a minimum outboard shaft diameter of 0.5 inches to avoid critical speed problems. This would require a hydrosphere of such size that its power consumption, combined with the seal power losses, would be excessive.

The hydrodynamic journal bearing on the other hand is compatible with an overhung load as no high pressure seal is needed between bearing and NaK pump. This arises from the fact that the high pressure is introduced to the interior of the bearing instead of the ends and is expelled to a low pressure at the ends (see Figure 4). Since the NaK pump can operate in a mercury vapor atmosphere, the only requirement is to prevent drain cavities from becoming filled with liquid mercury.

The hydrodynamic bearing requires only a source of lubricant and this need not be at any elevated pressure. Therefore only very low supply pressures are required at start-up and for low speed operation. Higher pressures are required only at increased speeds to assure sufficient lubricant flow to remove generated heat.

JOURNAL BEARING WITH OVERHUNG LOAD

FIGURE 4

3.3 Design and Assumptions

3.3.1 Journal Bearing

The journal bearing design was based on standard short bearing theory. The major assumptions limiting the application of this theory are:

1. The hydrodynamic film flow is laminar.
2. The lubricant is Newtonian.
3. The inertia effects of the lubricant are negligible.
4. The effect of the circumferential pressure gradient on circumferential flow is negligible.
5. The journal is perfectly aligned in the bearing.

When considering the density of mercury and the operational speed of the CRU it became apparent that fluid inertia could not be neglected. The important consideration of high inertia fluids is that of turbulence. Preliminary analysis of the journal bearing for the SNAP II application indicated that the lubricant film would depart from the laminar region at between 8,000 and 16,000 rpm.

At speeds above this, bearing performance could no longer be expected to follow classic bearing theory.

The state of the art of turbulent flow analysis within eccentric rotating cylinders does not permit mathematical interpretation of hydrodynamic flow beyond the laminar flow region. However, experimental data from bearings operating with Reynolds numbers in excess of 1000 have been reported in the literature. Results of these tests indicate that noticeable increases in both load capacity and power absorption along with a decrease in flow occur when a bearing is operated at Reynolds numbers higher than 1000 (approximate transition point). Empirical formulae have been developed which allow approximation of turbulent film load capacity and power consumption. From these relationships, the journal bearing power consumption was estimated.

Although it was realized that high load capacity might be experienced in the SNAP II bearings because of the probability of film turbulence, design evaluation was based on short bearing theory using laminar flow. This was the result of the necessity of using a minimum 5/8 in. diameter journal to provide critical speed well above CRU operating speed.

This analysis showed that a 5/8 in. diameter $\times$ 0.5 L/D bearing could adequately support design launch loads if laminar flow were assumed. Furthermore, neglecting turbulence in load capacity calculations left a reasonable margin of safety for unanticipated overloads.

Prediction of journal lubricant flow was not feasible because the operating eccentricity ratio could not be accurately determined. Furthermore the difficulty of analyzing turbulent film flow made determination of a film flow coefficient impractical. However a minimum flow had to be set to guarantee adequate heat removal along with a maximum to satisfy system requirements. Therefore the inlet configuration and the running clearance was only tentatively set. Final determination of the clearance and inlet configuration was to be made after developmental tests of various bearing types of the same basic size (0.625 dia. $\times$ 0.5 L/D ratio). The tabulation below shows the results of the journal bearing design analysis:

Diameter	0.625 inches
Length	0.312 inches
Dia. clearance	.001 inches
Operating speed	40,000 rpm

Eccentricity ratio	
at 25 lb load and 40,000 rpm	0.5 (approx.)
Power consumption	200 watts (approx.)
Maximum lubricant flow	5 ppm
Minimum lubricant flow	3 ppm

3.3.2 Thrust Bearing

Because there is possible axial loading in either direction, two thrust bearings are required in the CRU. To minimize the thermal expansion problem over the long shaft length, the thrust bearings were placed back-to-back at one end. To minimize the film power loss, the bearing outside diameter was set at 0.625 inch, the same as the journal bearing, with a 2 to 1 outside-to-inside diameter ratio.

Choice of the thrust bearing involved the following types:

1. Tapered land
2. Step
3. Tilting pad
4. And hydrodynamic - pocket

The difference in performance between the tapered land and step types in the size and operating range of the SNAP II application is not significant. While the tilting pad is a variable tapered land type but with the complication of an adjustment mechanism, the hydrodynamic-pocket type offers a 30-50 percent lower film power loss than the other three for the same load capacity and was picked for this reason. Figure 5 shows details of the bearing selected. Operation involves entry of the lubricant from the inlet holes, hydrodynamic pressure build-up across the step portion, trapping of the pressurized lubricant in the pockets and leakage of the lubricant across the dam. The running clearance adjusts itself until flow into and out of the pockets is equal. The bearing can be thought of as being comprised of a hydrodynamic portion (see Figure 5) and a hydrostatic portion. The reduction in power loss, compared with the other types of thrust bearings, comes from the lower loss over the large pocket clearances. Some of the more pertinent points regarding the thrust bearing are as follows:

1. Running Clearance - The film thickness for a single bearing varies from .0002 to .0008 inches depending on load. This sets the total axial clearance of a pair of bearings at .0005 -

HYDRODYNAMIC POCKET-TYPE THRUST BEARING

FIGURE 5

.0010 inches. These clearances require precision machining as regards parallelism between surfaces and surface finish. This does not imply, however, that machining will be a problem but rather that the care involved in precision work must be taken.

2. Flow Rate - Total flow is expected to be in the 2 to 12 ppm range which falls within the specification. Here, as in the journal bearings, a minimum flow rate will be required because of the lubricant temperature rise from film friction. Furthermore, the total flow rate to a pair of bearings remains fairly constant as axial load varies. The total axial clearance of .0010 above was set so as to avoid flow rates over 12 ppm per pair of bearings.
3. Power Loss - The thrust bearing analysis based on laminar film conditions results in a power loss of approximately 20 watts. But the prevalence of turbulent film conditions and resulting modification to the power loss calculations increases the value to approximately 80 to 100 watts at 40,000 rpm. Two interesting comments may be made regarding these turbulent conditions:
 - a. As film thickness increases, the modification factor increases the power loss at a greater rate than the increase in film thickness decreases it. The net effect is to increase power loss as the clearance increases.
 - b. If this is the case, then the expected reduction in power loss over the pockets may instead show up as an increase in power loss.

Below is a tabulation of the important parameters resulting from the thrust bearing design analysis.

Outside diameter	0.625 inches
Inside diameter	0.312 inches
Total axial clearance	.0005 -.0010 inches
Operating speed	40,000 rpm
Minimum film thickness at design speed and 15 lb load	200 microinches
Power consumption	100 watts (approximate)
Mercury flow	2-12 ppm

4.0 TEST APPARATUS

Bearing tests were conducted in three different test rigs:

1. The GTB Rig
2. The CUR Rig
3. The FRR Rig

The following description will acquaint the reader with the parametric data available from bearing tests conducted in these rigs.

4.1 GTB Test Rig (Thrust Bearing Test Rig)

The GTB was designed to obtain parametric data on single face thrust bearings and is schematically shown in Figure 6. The design of the rig is such that a variety of thrust bearing types could be tested. Drive was by a high speed motor mounted in a cradle which was free to move in a direction parallel to the axis of rotation. Thrust load to the test bearing was accomplished by means of a calibrated spring load behind the motor cradle at the motor center line.

The test rig was instrumented to obtain the following data:-

1. Speed
2. Power consumption (from calibrated motor drive)
3. Lubricant film thickness (by measuring motor movement in the axial direction)
4. Bearing lube flow
5. Lubricant supply pressure
6. Lubricant temperature rise
7. Thrust load

4.2 CUR Test Rig (Journal Bearing Feasibility Rig)

The CUR test rig was fabricated to test the feasibility of the journal bearing concept. This rig consisted of a calibrated high speed motor to which a short 5/8 inch OD journal was attached (see Figure 7). The bearing was hung on the journal with known weights attached for radial load application. Data obtainable from this rig included:

1. Speed
2. Power consumption
3. Bearing lube flow
4. Lubricant supply pressure
5. Radial load

GTB TEST RIG

FIGURE 6

FIGURE 7

4.3 FRR Test Rig (Free Running Bearing Test Rig)

The FRR was designed to simulate the combined bearing construction of the final application. Into this rig two journal bearings and a double face thrust bearing were incorporated as shown in Figure 8. The drive to rotate the bearings was a Terry-type turbine, driven with shop air. Incorporated in the test fixture were load pads pressurized by shop air for application of radial and axial loads. A photograph of the FRR and a portion of the control panels are shown in Figures 9 and 10. A photograph of the FRR shaft and bearing assemblies is shown in Figure 11. This test was instrumented to obtain the following data:

1. Speed
2. Individual bearing lube flows
3. Individual bearing supply pressures
4. Vibration
5. Radial load
6. Axial load
7. Power consumption (from speed decay data)

FRR TEST RIG

FIGURE 8

FREE RUNNING BEARING TEST RIG

FREE RUNNING MERCURY BEARING TEST RIG

FREE RUNNING RIG SHAFT AND BEARING SUPPORTS

5.0 BEARING DEVELOPMENT PROGRAM

After the preliminary design evaluation was completed a development test program was initiated in which various bearing operating characteristics were investigated. The object of this program was to develop the best possible bearing configuration for the SNAP II application consistent with CRU package design requirements and schedules. Paralleling this effort was an investigation of bearing characteristics in SNAP II system tests for possible problem areas and correlation with component bearing tests. The following bearing types were tested during the development program:

1. Hole fed journal
2. Axial groove fed journal
3. Tangential groove fed journal
4. Annular or circumferential groove journal
5. Load-pad journal
6. Single face step-pocket thrust
7. Double face step-pocket thrust

With the exception of the hole fed journal, all the bearing types tested demonstrated design load and speed capacity with proper adjustment of clearance. Two problem areas were solved in development. First, obtaining sufficient lubricant flow through the bearing to permit adequate heat removal and thereby avoid film vaporization; and secondly, deriving a bearing inlet configuration and clearance combination which permitted stable operation of the journal bearing at high speeds with low radial load application.

5.1 Journal Bearings

5.1.1 Hole, Axial Groove and Tangential Thrust Journal Bearings

Discussion of Design: The hole fed bearing shown in Figure 12 was the basic journal resulting from the design evaluation discussed in Section 3.3.1. Preliminary testing of this bearing in the CUR Test Rig demonstrated the presence of lubricant film turbulence which was manifested in high power consumption and low lube flows as compared to that which might be expected with a laminar lube film.

The increased power consumption resulting from film turbulence along with the high ambient temperatures of the SNAP II package required that the lubricant film flow be at least 3 ppm to avoid film vaporization. A flow of this magnitude was not realized in preliminary tests of the hole fed bearing.

HOLE FED JOURNAL BEARING

FIGURE 12

One possible solution to this situation was to decrease the bearing diameter and thus reduce the operating Reynolds number. But to do this was not consistent with package design requirements of a minimum $5/8$ inch diameter bearing required for shaft stiffness. To increase the flow of the $5/8$ inch diameter $\times$ $5/16$ inch long bearing, two possibilities were available: First, increase the bearing clearance to increase the end leakage; and secondly, change the inlet configuration to improve the discharge coefficient at the bearing inlet. Disadvantages of increasing the bearing clearance are that the Reynolds number is increased, thereby increasing film power^{*}; the load capacity is decreased; and the increased possibility of fluid film whirl. Problems associated with changing the inlet configuration are that the bearing surface is disturbed with the possibility of making the bearing more sensitive to load direction and reducing rotating radial load capacity.

A combination of these two solutions was accomplished by incorporating two different inlet configurations into the basic $5/8$ inch OD $\times$ $5/16$ long bearing and operating them with varying clearances to determine performance.

The first modified inlet was the axial groove pictured in Figure 13. The purpose of this was to decrease the pressure drop at the inlet and to introduce the lubricant over the entire width of the bearing.

AXIAL GROOVE JOURNAL BEARING

FIGURE 13

The second modification of the basic journal was the incorporation of the tangential inlet as shown in Figure 14. The purpose of this was twofold. First, by introducing the mercury lubricant tangent to the clearance space no direction change at the inlet was involved. This could be significant because of the high mass density of mercury and resulting high energy loss with changing direction. Secondly, by introducing the lubricant tangent to the clearance space, more effective viscous pumping by the rotating shaft into the loaded area was expected.

TANGENTIAL INLET JOURNAL BEARING

FIGURE 14

Discussion of Test Results

The axial groove and tangential inlet bearings were tested at speeds up to 40,000 rpm with loads to 25 pounds using room temperature mercury lubricant. Diametral clearances of .0010, .0012, .0014, .0020 inches were tested.

Flow calibrations at 40,000 rpm (with 25 pounds applied load) are plotted in Figure 15. These curves show little significant difference in flow performance between the axial groove and tangential inlet bearings.

Indicator recordings of the bearing surface profiles (accurate to less than 10μ inches) showed a discontinuity in the surface adjacent to the inlet port. This discontinuity took the form of a valley in the surface and was caused by the constant wheel pressure grinding process removing a relatively constant volume of material when the inlet area was contacted. This valley allowed high leakage at the inlet area which was probably a significant portion of the bearing flow and the primary contributor to the inconsistencies in flow characteristics between bearings. For this reason, it is felt that little of the total bearing flow contributed to film cooling as most was lost at the inlet.

Examination of Figure 15 reveals that a clearance of .002 inches (diametral) is necessary to obtain the desirable bearing flow of 3ppm at approximately 150 psig. A clearance of this magnitude on a 5/8 inch diameter bearing is somewhat large although not intolerable. As might be expected, some evidence of instability or film whirl was noted with axial groove and tangential inlet bearings operating at .002 inches diametral clearance.

Figure 16 shows the variations in flow calibrations with varying radial loads for the axial and tangential inlet groove bearings. The width of these flow bands make these bearings undesirable from a system control standpoint, if the CRU is to operate through a 10 g launch phase. Should it be desirable to start the CRU in orbit and thus avoid the 10 g operating requirement, the wide flow band with varying radial load would not be a problem.

Results of speed decay tests in the FRR verified the calculated power consumption of 200 watts per journal bearing at 40,000 rpm. A significant variance in power with changes in radial load and clearance was not experienced. This was to be expected since film turbulence accounted for the major contribution to power loss, and the inertia forces do not change significantly with slight variations in film thickness

FLOW CALIBRATIONS OF AXIAL AND TANGENTIAL
INLET GROOVE BEARINGS AT 40,000 RPM WITH
2.5 LBS RADIAL LOAD PER BEARING

FIGURE 15

FIGURE 16

(or clearance) when the total clearance space (circumferentially) is considered.

A synopsis of the test results for these bearings is included in Section 5.3.

5.1.2 Annular Groove Journal Bearing

Discussion of Design

The annular groove bearing was designed to overcome some of the inherent problems of the axial groove and tangential inlet bearings. Of immediate concern was the wide variations in bearing flow, should the CRU be required to be operational during launch. Furthermore, it was desirable to decrease the bearing clearance to insure a safer load capacity and to reduce the possibility of lube film whirl. The advantages of an annular or circumferential groove bearing over the plane bearing are:

1. Smaller flow variations with eccentricity (radial load).
2. Inlet configuration imposes no radial load on the journal due to supply pressure.
3. Higher pressures are available on the unloaded side of the bearing thereby producing a more complete lube film.
4. More efficient film cooling is possible since a continuous supply of high pressure lubricant is available around the bearing.
5. Insensitivity to load direction.

Disadvantages of an annular groove bearing are:

1. Reduced load capacity for the same effective length.
2. Increased power consumption for the same load capacity as the plane bearing.
3. Possibility of viscous pumping in the supply annulus.

For the SNAP II application, the annular groove bearing was designed with a $5/8$ inch diameter and the active length set at 0.346 inches, such that the same load capacity as in the $5/8 \times 5/16$ plane bearing (based on laminar flow theory) would be achieved. In order that the effect of annular pumping could be evaluated, one bearing was designed with four protrusions or dams in the supply annulus (to interrupt circumferential flow) while the other bearing contained an uninterrupted supply annulus.

Both of these bearings (see Figure 17) were assembled into the same test unit (FRR) for analysis.

FIGURE 17

Discussion of Test Results

Annular groove bearings with diametric clearances of .0010, .0013 and .0015 inches were tested at speeds to 40,000 rpm and radial loads to 37 pounds. These tests indicated that annular groove fed bearings could be successfully operated beyond the design maximum speed and load with operating clearances from .0013 to .0015 inches and supply pressures under 150 psig. With smaller clearances, the bearing pressure drop became excessive and operation at speeds over 30,000 rpm could not be accomplished due to inadequate lubricant flows.

Figure 18 shows the range of flow calibrations with radial load as compared to the tangential inlet bearing. This curve clearly demonstrates the advantage of the annular groove bearing when considering flow variations with radial load. It should be noted, however, that this flow range is somewhat dependent on the roundness of the bearing bore. That is, as the bearing becomes more elliptical, the range of flow with radial load increases.

Investigations of the relative motion between the shaft and bearing during

COMPARISON OF TANGENTIAL INLET
AND ANNULAR GROOVE JOURNAL
BEARING FLOW CHARACTERISTICS WITH
RADIAL LOAD AT 40,000 RPM

FIGURE 18

operation revealed that the annular groove journal bearing was hydrodynamic at speeds above 150 rpm (with 2.5 lb radial load) and that lubricant film whirl existed over most of the design operating range. The externally applied radial load required to suppress whirl is plotted in Figure 19 for the axial groove, tangential inlet and annular groove bearing. This curve clearly shows the less desirable stability characteristics of the annular groove bearing.

Comparisons of bearing performance data between the bearing with the interrupted annulus and the bearing with a non-interrupted annulus, revealed that there were no significant differences in flow characteristics. It was therefore concluded that either no annular pumping existed or that if it did, it was not detrimental to bearing performance.

Power consumption data indicated annular groove bearing power loss to be approximately 200 watts at 40,000 rpm.

A synopsis of the test results for the annular groove bearing is included in Section 5.3.

EXTERNAL APPLIED RADIAL LOAD
REQUIRED TO SUPPRESS WHIRL

FIGURE 19

5.1.3 Load-Pad Type Bearing

Discussion of Design

Although the lube film whirl inherent in the plane and annular groove type bearings was not serious enough to induce any bearing damage, development of a non-whirling bearing was desirable from a reliability and life standpoint.

The load-pad bearing pictured in Figure 20 was designed to accomplish this end and incorporated some of the advantages of annular groove while maintaining the static loading feature of the tangential inlet and axial groove bearing. Here the lubricant was introduced to a circumferential groove or annulus in the lower half of the bearing where it provided lubrication and cooling. From the groove, the fluid was pumped into a load-pad in the upper half of the bearing where load was provided by both the lubricant static pressure (due to inlet pressure plus viscous pumping) and the velocity pressure available from damming the flow at the extreme end of the pad.

LOAD - PAD TYPE JOURNAL BEARING

FIGURE 20

Discussion of Test Results

Tests of the load-pad-type bearing with a diametral clearance of .0010 inches to speeds in excess of 40,000 rpm, demonstrated load capacity to the design maximum of 25 pounds per bearing. Furthermore, no lube film whirl was encountered at speeds to 50,000 rpm with negligible applied loads. Figure 21 shows a representative range of flow calibrations with varying radial loads. As would be expected, the range is somewhat greater than that encountered in the annular groove bearing because of the reduced surface area and resultant higher leakage.

It will be noticed that the supply pressure required to produce 3 ppm lube flow is approximately 40 psi less than the annular groove bearing and 120 psi less than the axial groove bearing. To increase the required supply pressure would necessitate a clearance reduction below .0010 inches (diametral). Such a reduction is not considered consistent with the high temperature design requirements of the CRU. For this reason, a flow restriction will be required between the mercury condensate pump and the journal bearings in the CRU if a high pressure pump is used.

Power consumption of the pad type journal bearings did not vary significantly from that of the other journal bearing types tested. A synopsis of test results is included as Section 5.3 of this report.

RANGE OF FLOW CHARACTERISTICS WITH
RADIAL LOAD FOR LOAD-PAD TYPE
JOURNAL BEARING

FIGURE 21

5.2 Thrust Bearing

Two thrust bearing configurations were tested during this program. One was a single faced bearing (taking load in one direction only), and the other a double faced bearing (taking load in two directions). Both had thrust faces of identical design and were a result of the design evaluation discussed in Section 3.2.2.

5.2.1 Discussion of Test Results

The purpose of the single face thrust bearing test was to determine thrust bearing operating characteristics prior to testing the journal and thrust bearings together in the FRR. The single thrust bearing tests conducted in the GTB rig were, in a sense, feasibility type tests and were performed concurrently with journal bearing feasibility tests in the CUR rig. The single face thrust bearing incorporated into the GTB rig was of such design that changes in bearing design could be made with minimum expense.

The thrust bearing tested in the GTB rig demonstrated load capacities in excess of thirteen pounds at 40,000 rpm. Hydrodynamic action was indicated at approximately 10,000 rpm with an applied thrust load of thirteen pounds. With lower applied loads, hydrodynamic action was indicated at correspondingly lower speeds. Power absorption tests of this bearing verified calculated power approximations (which assumed turbulent film flow) of 100 watts at 40,000 rpm.

Following successful results of preliminary tests of the single face thrust bearing in the GTB rig, a double face thrust bearing of identical design was incorporated into the FRR (Free Running Test Rig). A photograph of this bearing is shown in Figure 22. This bearing was operated with various total axial clearances from .0005 to .0010 inches at speeds to 50,000 rpm and loads from zero to fifteen pounds. During this time the bearing had undergone more than seventy starts and stops with a minimum of three pounds applied thrust load with only slight signs of wear resulting (See Figure 23). Tests of this bearing design with applied loads of twenty pounds at 40,000 RPM showed definite signs of wear at the bearing OD, clearly indicating that the maximum load capacity of the thrust bearing probably does not exceed the design objective of fifteen pounds.

Tests of the thrust bearing with a total axial clearance of .0007-.0008 inches with axial loads varying from three to fifteen pounds, showed that the flow remained fairly constant with load as is shown

DOUBLE FACE THRUST BEARING

THRUST BEARING AFTER 51 HOURS OF TESTING IN FRR

in Figure 24. Figure 25 shows representative thrust bearing flow calibrations operating at 3-5 lbs axial load with various total axial clearances. This curve clearly shows the wide variance in flow characteristics with small changes in total axial clearance. Although the differences in clearances tested was slight, the percentage difference of the total was significant, thus a sizable variation in flow may be expected. Control of the total axial clearance to the tolerances required for proper flow control is difficult due to the squareness and tolerance stack-up of the parts involved. For this reason, an external flow control may be necessary to assure maintenance of the system pressure level should thrust bearing clearances approach .001 inches (TAC) in the CRU.

5.3 Synopsis of Test Results

From the data obtained during the development test program, design calculations and assumptions were verified. Although some important parametric data was not obtainable due to test rig limitations, test data from many different tests and test rigs at least give reasonable approximations of the parameters sought. The following is a generalized tabulation of the significant thrust bearing test parameters:

Thrust Bearing

Lubricant	Liquid mercury
Lubricant supply temperature	75° - 400°F
Operating speed	40,000 rpm
Power consumption	100-125 watts
Total axial clearances tested	.0007 - .0010 inches
Mean lube 150 psig supply pressure	8.6 ppm
% flow variation at 150 psig over above clearance range	79%
Maximum load capacity	15 pounds
Approximately lube film thickness at 15 pounds load	.0002 inches

Significant journal bearing test parameters are tabulated in Table 1.

5.4 Start-Stop Tests

Because mercury does not wet the bearing surfaces, boundary lubrication is not available at start up and shut down. Therefore, from a lubrication standpoint, start-ups, and shut downs are essentially dry.

Although many starts and stops have been accomplished during performance testing of the various bearing configurations, it was desirable to demonstrate start-stop capabilities in a separate test under simulated starting load conditions.

THRUST BEARING FLOW RANGE WITH AXIAL
LOADS VARIED FROM 3 TO 15 POUNDS

(TAC = .0007 - .0008)

FIGURE 24

FIGURE 25

TABLE I
JOURNAL BEARING TYPE

<u>Lubricant</u>	<u>Hole Hg</u>	<u>Axial Groove Hg</u>	<u>Tangential Inlet Groove Hg</u>	<u>Annular Hg</u>	<u>Load Pad Hg</u>
Lube Supply Temp. (° F)	75	75-400	75-400	75	75
Operating Speed (rpm)	10,000	40,000	40,000	40,000	40,000
Power Consumption (watts)	—	180-220	180-220	180-220	180-220
Diametral Clearances Tested (inches)	.0010	.0010, .0012 .0014, .0020	.0012, .0014 .0020	.0010, .0013 .0015	.0010, .0005
Average Lube Flow at 2.5 pounds Applied Radial Load (ppm)	< 1.0 at 300 psig	2.0 at 150 psig (Cd = .0020)	1.1 at 150 psig (Cd = .0014)	3.0 at 150 psig (Cd = .0015)	2.6 at 100 psig (Cd = .0010)
Average Lube Flow at 25 Pound Applied Radial Load (ppm)	—	7.4 at 150 psig (Cd = .0020)	3.6 at 150 psig (Cd = .0014)	4.2 at 150 psig (Cd = .0015)	3.8 at 150 psig (Cd = .0010)
Maximum Load Tester (lbs)	2.5	25	25	35	25

Start-stop tests were conducted on both the tangential inlet and axial groove journal bearings in the FRR. In this test, twenty-six starts were accomplished with applied radial loads from 2.5 to 20 lbs per bearing and three to five pounds thrust load. Inspection of these bearings following start-stop testing revealed no signs of galling and only slight signs of wear. These were due to edge loading resulting from slight misalignment.

To further demonstrate the start-stop capabilities of the SNAP II bearing system, the load pad type journal bearings and thrust bearing was subjected to over 120 starts and stops with ten pounds applied radial load to each journal and three pounds to the thrust bearing. This loading valve was considered to be the maximum that the bearings would be subjected to in CRU operation. During this test the bearing flows were maintained at approximately the design minimum of 3 lb/min per journal and 6 lb/min thrust bearing flow. For testing convenience, the maximum shaft speed was limited to 20,000 rpm. This limitation, however, did not detract from the validity of test results as continuous recording speed traces showed that the journal bearings supported the load hydrodynamically at approximately 1000 rpm. Therefore operation at speeds between 0 and 1000 rpm was the only range where bearing damage would occur.

Figures 26 and 27 show one of the bearings following performance and start-stop testing. It can be seen that only slight wear from misalignment was experienced and was not serious enough to affect bearing performance. It should be pointed out that these journal bearings were operated with an "out of squareness" relationship of .0002 and .0003 inches TIR with the bearing centerline (defined by the bearing diameters at the extreme edges). The maximum allowable CRU bearing misalignment is .0002 inches TIR. The purpose of testing with this extreme misalignment was to demonstrate the ability of the bearings to operate under severe start-stop conditions without damage.

5.5 Package Tests

To date five of the SNAP II CRU's have been operated. Data regarding bearing parameters has generally confirmed bearing design calculations and component test data. Of particular significance is the effect of corrosion products in the mercury which has been generated in the high temperature mercury system. It was found that these products would clog up the bearing clearance spaces if mercury were allowed to flow through the bearings with the shaft static, but as soon as rotation was imparted to the shaft, the products that had collected were washed away and would not accumulate again as long as rotation of the bearings was maintained.

FRR VII - "B" END JOURNAL FOLLOWING START-STOP TESTS

FIGURE 26

FRR VII - "B" END BEARING FOLLOWING START-STOP TESTS

FIGURE 27

One of the CRU's tested was operated continuously for 484 hours with hot mercury (410°F) lubricant. During this test no bearing performance decay or effects of corrosion products was encountered. Inspection of the bearings following this test revealed no signs of wear on one journal bearing although the other journal had been destroyed as a result of inadvertent overspeed operation. The wear encountered on the thrust bearing was impossible to analyze due to the unknown effects caused by the journal failure (as this journal was integral with the thrust bearing). Neither of the bearings showed any signs of mercury corrosion products except for insignificant amounts found in the pockets of the thrust bearings. These products were located in areas and in such small amounts that performance would not be affected. Analysis of flow paths and pressure distributions across the thrust bearing, and microscopic examinations of test specimens indicated that the build up of corrosion products in critical bearing areas is improbable.

5.6 Bearing Failure Analysis

A minimum of bearing failures were encountered during the development test program. Although none were attributed to basic bearing design, their occurrence did point to some of the limitations and danger zones of the SNAP-II bearing system. For the purposes of this report, these failures will be discussed relative to their cause. Failures encountered during the test program were attributed to the following basic areas:

1. Material and assembly
2. Overload and vibration
3. Stability
4. Inadequate lubrication

5.6.1 Material and Assembly

Two journal bearing failures occurred early in the test program in the FRR. The first one was caused by a crack in the journal bearing resulting in seizure at approximately 38,000 RPM. Microscopic examination of the cracked bearing after failure revealed that either an inclusion in the material or local stress had caused a material failure in which a piece of the bearing surface was removed and forced into the clearance space resulting in seizure. A photograph of this bearing after seizure appears in Figures 28 and 29.

The second failure was caused by the presence of grinding grit within the lube supply passages of the journal bearing which became entrained in the lube stream. Because of the method of fabrication of the journal bearings, it is necessary to finish grind the inside diameter of the bearing after it is pressed into the support housing. It

FRR 1 - "A" END BEARING FOLLOWING FAILURE

FRR 1 - "A" END JOURNAL FOLLOWING FAILURE

FIGURE 29

is impractical to prevent the grinding coolant and grit from entering the internal lube passage of the housing. Only adequate cleaning after grinding can safely remove this grit.

To prevent the recurrence of failures of this type, the material specification was changed to insure a more homogenous material; assembly procedures were revised to reduce the possibility of over stress in the bearing, and the cleaning procedures improved to provide better cleaning of internal passages.

5.6.2 Overload and Vibration

One bearing failure occurring in the FRR has been attributed to vibration and overload. This occurred during a test of the annular groove bearing in which the length had been reduced from the original design of 0.346 inches (effective) to 0.280 inches (effective). During this test the shaft became bent causing unbalance and severe vibrations. In an effort to reduce the amplitude of these vibrations an excessive radial load was applied and failure resulted. This failure cannot be attributed to a design limitation but to adverse test conditions.

The failure of two of the CRU's involved journal bearings. In both instances failure occurred while operating the units at or near critical speed where severe shaft vibrations caused excessive rotating overloads. Redesign of the shaft was required to rectify this situation.

5.6.3 Inadequate Lubrication

Excessive surface wear occurred on the single face thrust bearing tested in the GTB rig. This wear was sufficient to remove a major portion of the thrust bearing steps and significantly reduce the load capacity (see Figures 30 and 31). Analysis of the wear and test data revealed that inadequate lubrication during starting and stopping caused the wear. This bearing had undergone over forty starts and stops with loads up to thirteen pounds without lubrication.

5.7 Bearing Flow Regulation

The minimum required lubricant flow through a bearing is dictated by the quantity of fluid pumped hydrodynamically by the rotating bearing and the mass flow necessary for removal of the generated heat, such that the fluid temperature rise does not exceed vaporization. On the other hand, the actual lubricant flow is determined by the supply pressure available and the bearing pressure drop characteristics.

SINGLE FACE THRUST (STATIONARY MEMBER) BEARING FOLLOWING FAILURE

SINGLE THRUST BEARING (ROTATING MEMBER) FOLLOWING FAILURE

To satisfy hydrodynamic and temperature rise requirements, the journal bearing flow was minimized at 3 ppm (per bearing) and the thrust bearing at 6 ppm. From a bearing standpoint, flows in excess of this value are not detrimental to bearing performance. However, from a system standpoint, a total flow (to the three bearings) in excess of 20 ppm cannot be tolerated because of the danger of reducing the system pressure level and the possibility of cavitation in the mercury pump resulting from an excessive flow rate.

It was found that by proper adjustment of clearances in the axial and tangential inlet groove journal bearings, the desirable flow rate could be obtained at a predetermined supply pressure, but with considerable difficulty due to the effect of bearing surface irregularities resulting from manufacture. Furthermore, the range of flow with varying radial load was excessive for the application.

To improve the bearing flow characteristics, the annular groove bearing was built and tests showed it possessed the desirable flow characteristics for the application. The disadvantages of the annular groove bearing was its tendency toward instability under light loads.

As a solution to the instability problem, the load-pad type bearing was developed. This bearing had flow characteristics over a wider range than did the annular groove bearing, as was to be expected due to the inherently greater leakage.

To further compound the problem of total bearing flow, the thrust bearing pressure drop showed extreme sensitivity to axial clearance as is shown in Figure 25.

The most direct solution to the problem of maintaining the system pressure level while providing adequate lubricant flow is to select bearings with pressure requirements below system pressure and provide a pressure reducing device in the lubricant supply line.

A flow sensitive regulator was chosen over a fixed orifice because of greater versatility and the fact the fixed orifice would have a smaller hole diameter with an associated clogging tendency.

It is proposed that three regulators be used: one for each journal bearing (of identical design), and one for the thrust bearing. These regulators will be capable of maintaining the journal bearing flow at 3-5 ppm and the thrust bearing flow at 6-10 ppm with a mercury pump discharge pressure of 250-280 psi. The regulator downstream pressure (bearing supply pressure)

would then vary from approximately 50 to 225 psi, depending upon the bearing supply pressure required to maintain the forementioned flow.

In the interest of reliability, simplicity was a prime consideration in the design of the bearing flow regulators. A variable orifice system actuated by a metered pressure differential was selected. Figure 32 shows the operation of the bearing flow regulators. The flow through the metering

SCHEMATIC OF BEARING FLOW REGULATOR

FIGURE 32

orifices in the valve plunger defines the pressure differential ($P_1 - P_2$) across the plunger which in turn sets the regulating orifice pressure drop. As the downstream pressure decreases, the flow through the regulator and thus through the metering orifices tends to increase thus increasing the pressure differential ($P_1 - P_2$). The resulting force moves the plunger to the right, closing the regulating orifice and thus maintaining the flow at the design valve.

As can be seen from Figure 32, the only moving parts are the plunger and the spring. These are of such design that the presence of contaminants in the fluid would not impair their operation. Should contaminants build up around the regulating orifice, operation would not be impaired because of the self-compensating feature of the double orifice system.

To maintain simplicity and eliminate sliding surfaces, the valve is not pressure balanced. Therefore, the area of the plunger exposed to the downstream pressure at the regulating orifice will experience variations in pressure loading which will tend to reduce the accuracy of regulation.

Fortunately, the load-deflection characteristics of the Belleville spring can be chosen to compensate for these variations.

Figures 33 and 34 shows the development type flow regulator used for test purposes. This unit is of screwed construction with an "O" ring seal. The final units to be used in the SNAP II system will be of welded construction.

5.7.1 Test Results

Preliminary tests have been conducted on thrust and journal bearing regulators (the two regulators differ only in the size of the regulating and metering orifices.) Representative flow calibrations at 200 psig supply pressure to the regulator (simulated pump discharge pressure) are shown in Figures 35 and 36. As can be seen from these curves, the flow range over a simulated bearing pressure range of 25-175 psig remains well within specification for the thrust bearing. The journal bearing flow is somewhat high, although still within specification. A slight adjustment in the metering orifice diameter will bring this curve closer to middle range.

DEVELOPMENT TYPE BEARING FLOW REGULATOR

DEVELOPMENT TYPE BEARING FLOW REGULATOR
(DISASSEMBLED)

JOURNAL BEARING FLOW REGULATOR CALIBRATION

FIGURE 35

THRUST BEARING FLOW REGULATOR CALIBRATION

FIGURE 36

6.0 CONCLUSIONS

From the design and test results obtained during this program the following conclusions have been drawn:

1. The lubricant film at 40,000 rpm in both the journal and thrust bearing is turbulent. Turbulence in the lube film was demonstrated by its effect on lube flow, power consumption, and load capacity.
2. The thrust and journal bearing power consumption of 100 watts and 200 watts (at 40,000 rpm) respectively, agrees favorably with that predicted based upon turbulent flow approximation.
3. The load capacity of the turbulent journal bearing is in excess of that predicted (25 lb) by short bearing theory based on laminar lube film conditions.
4. The maximum thrust bearing load capacity of fifteen pounds agrees with the calculated capacity based on standard design procedures.
5. The following generalized observations have been made to qualitatively describe the performance of the five journal bearing types.

Hole Fed Bearing - The hole fed journal bearing (shown in Figure 12) was not suitable for the high speed SNAP II application because of the excessive lubricant pressure drop across the bearing at speeds near 10,000 rpm.

Axial and Tangential Inlet Groove Bearings - These two bearings showed similar performance characteristics and therefore will be classified together. The design load and speed conditions of twenty-five pounds at 40,000 rpm was demonstrated and the minimum lube flow of 3 ppm was achieved with a diametral clearance of .002 inches.

The high range or variation in flow with changing radial load makes these bearings somewhat undesirable from the operation during launch standpoint as this would require that the CRU be subjected to radial load variations from 0 to 10 g's. At the higher loads the bearing flow would be excessive, reducing the system pressure.

Although adequate lubricant flow for cooling purposes was achieved, the discontinuity or valley in the bearing surface at the lubricant inlet, caused by the finish grinding process, probably creates sufficient leakage at that point to cause a major portion of the total flow to be ineffective for heat transfer.

Another disadvantage of these bearings was their tendency to whirl at high speeds. Although no bearing damage has been evidenced as a result of instability, its

presence should be avoided if possible. Whirl, in this case, was primarily due to the large clearances necessary to obtain the minimum lubricant flow rate.

Annular Groove Bearing - The load capacity of this bearing was demonstrated to be in excess of the twenty-five pound requirement at 40,000 rpm. Furthermore, its geometric configuration allows for a rotating radial load capacity equal to its unidirectional load capacity.

With a diametral clearance of .0015 inches, the minimum required flow rate was obtained. The band in flow characteristics with changing radial load was narrow enough to meet system requirements when high radial loads were applied, thus making this bearing desirable from an operational launch standpoint.

The annular groove type bearing is felt to have cooling characteristics superior to the tangential and axial groove bearings because of the availability of cool lubricant at any point on the circumference of the bearing. In this case leakage from the bearing occurs at all low pressure regions instead of being localized as is the case with the tangential and axial groove bearings.

The disadvantage of this bearing was its tendency to whirl at high speeds even with applied loads approaching the design maximum. Although the whirl encountered in the annular groove bearing was more severe than in the tangential and axial groove bearings, no bearing damage was experienced.

Load-Pad Bearing - The load-pad bearing (shown in Figure 20) has a demonstrated load capacity in excess of the 25 pounds design load at design speed. With a diametral clearance of .0010 inches, no film whirl was encountered at speeds to 50,000 rpm and with negligible applied loads.

Adequate cooling flow was available at pressures below 150 psig. The total lubricant flow in the pad bearing is as effective for cooling purposes as that in the annular groove bearing as the circulation of lubricant within the internal grooving of the two bearings is similar.

The range of flow with radial load was higher than experienced with the annular groove bearing. This, combined with inconsistent flow characteristics between different bearings of the same type, makes pressure-flow matching within the system difficult.

6. Analysis of the flow characteristics of the journal and thrust bearings tested indicates that flow pressure relationships are not only dependent on the bearing type and clearance, but also upon surface profiles within the manufacturing tolerances which provide leakage paths near the same magnitude as that provided by clearance. Furthermore, bearing squareness relationships within the manu-

facturing tolerances provide another significant flow variable. Therefore prediction of flow characteristics accurate enough for system integration of the pump and bearings is not feasible.

7. The most realistic solution to the bearing-pump pressure integration problem: is the incorporation of a flow regulation device in the bearing lubrication supply lines. Such a device has been designed and tested with excellent results.

7.0 RECOMMENDATIONS

1. Since it is not feasible to match the journal and thrust bearing flow characteristics to that of the system condensate return pump, it is recommended that a flow controlling device be incorporated to control the bearing lubricant flows within a range consistent with bearing cooling requirements and the system mercury pump output capabilities.
2. It is recommended that the bearing life capabilities be investigated in an endurance test in which the system operating parameters are simulated as much as is economically and technically practical.

SNAP II

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