LIQUID-MERCURY LUBRICATED HYDROSPHERE BEARINGS

By G. Y. Ono and D. C. Reemsnyder
Thompson Ramo Wooldridge, Inc.

ABSTRACT

The SNAP I power conversion system utilizes mercury as the working fluid in a Rankine closed-cycle heat engine in order to convert nuclear energy to electrical energy in space. Utilization of the working fluids as the bearing lubricant in the turbomachinery package eliminates the need for seals and provides a hermetically sealed package unit. The bearings were required to operate continuously and unattended for sixty days with a minimum power loss at 40,000 rpm and 400° F. The hydrosphere bearing, which is a mated spherical journal and hemispherical socket, has both thrust and radial load capacity, good dynamic stability, misalignment capability, and close control of clearances for other rotating components. Analytical expressions are presented for flow, torque, axial load capacity, and pressure distribution for the hydrosphere bearing. Experimental results of the bearing development program and a discussion of the liquid mercury bearing test rigs are included. Satisfactory bearing life capabilities have been exhibited in a 2510-hour endurance test of the system prototype test package with the 1/2-inch diameter hydrosphere bearings under simulated ground operating conditions.

INTRODUCTION

Advanced closed-cycle space power systems will utilize liquid metals, such as mercury, potassium, and rubidium, as the thermodynamic working fluid at elevated operating temperatures. Specific bearing requirements for long life and unattended operation in space environments with zerc leakage and maximum reliability dictate the use of the closed-cycle working fluid as the bearing lubricant. Several critical problems associated with the use of liquid metals as the bearing lubricant are indicated below:

(1) Corrosiveness or incompatibility of the lubricant with system materials, resulting in the contamination of the lubricant with its subsequent effect on bearing clearances and system integrity.

(2) Viscosities of the liquid metals at the elevated temperatures are low as compared to conventional lubricants at normal operating temperatures.

(3) In the case of liquid mercury, high density may result in turbulent bearing flow and substantial inertia effects.

(4) Non-wetting characteristics alter the velocity profile and prevent easy handling in the classical hydrodynamic equations.

Typical of Rankine cycle systems for space application is the 500-watt electrical output SNAP I turboelectric system which utilizes mercury as the working fluid. In SNAP I, which was initiated and developed under a contract with the Atomic Energy Commission, a radioisotope is used as the energy source to vaporize mercury in a boiler; turbomachinery extracts the useful energy from the vapor and converts it into electrical energy; the exhaust vapor is condensed by rejecting the waste thermal energy to space in a condenser-radiator. Current application of a portion of the system is being conducted under the auspices of WADD. Figure 1 shows the SNAP I turbomachinery package, which is a subminiature electric power generator containing a mercury vapor turbine, alternator, and

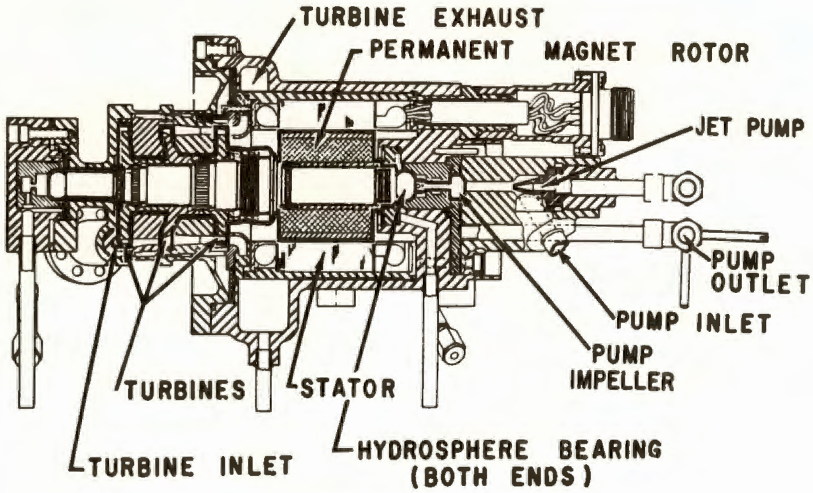

FIGURE 1. SNAP I TURBOMACHINERY PACKAGE

mercury pump for the Rankine cycle power conversion system. These components are mounted on a common shaft rotating at 40, 000 rpm and supported by mercury-lubricated hydrosphere bearings.

The hydrosphere bearing consists of a spherical journal rotating in a slightly larger diameter hemispherical socket with an integral flow restriction. Primary design considerations in the selection of the hydrosphere were reliability, low power loss, sufficient load capacity, and minimum lubricant flow requirements. The hydrosphere bearing offers combined thrust and radial load capacity, good dynamic stability, misalignment capability, and close control of clearances for other components. Analytical and empirical design procedures were evolved for the bearings and verified by experimental results. Two test rigs were designed, fabricated and used for the development testing of liquid-mercury-lubricated bearings.

The 1/2-inch diameter hydrosphere bearings were successfully incorporated in the turbomachinery package unit. The SNAP I power conversion system exceeded its original 60-day life specification in a system endurance test of 2510 hours, which was shut down on schedule without defect and provided a conclusive demonstration of mercury-lubricated hydrosphere bearing endurance capabilities.

BEARING REQUIREMENTS AND SELECTION

Rankine space power conversion systems require turbomachinery bearings which allow continuous, unattended, and reliable operation. The bearings must provide a minimum parasitic torque when exposed to axial and radial loads imposed by the shaft-mounted equipment, and by thermal, acceleration, vibration, and gyroscopic loading. The bearings must also provide accurate radial and axial positioning of the shaft, sufficient allowance for reasonable amounts of lubricant contamination, and relatively constant bearing flow from the cold start to the stable running conditions.

Investigation of various bearing types was conducted to find the most promising bearing to satisfy the specification. Liquid lubricated hydrodynamic journal and thrust bearings were considered to have relatively high percentage power loss for such a low power output unit and would present the added complexity of incorporating journal and thrust bearing combinations. Gas- or vapor-lubricated hydrodynamic bearings would require excessive flow to insure thick film lubrication and would have very low load-carrying capacity. Due to the high speeds and light loads in zero gravity operation, liquid, vapor, and gas bearings may be prone to instability. Hydrostatic journal and thrust bearings increase design complexity, possess instability characteristics,and require high flows and alignment precision. Anti-friction bearings would have limited life and present formidable seal problems.

The hydrosphere bearing, shown in Figure 2, has both an axial and a radial load-carrying capacity due to hydrostatic and hydrodynamic pressure distribution. An additional radial load-carrying capacity arises from viscous centering forces. The fact that the hydrosphere is insensitive to misalignment is a distinct advantage since bearing loading due to warpage from residual housing strains and

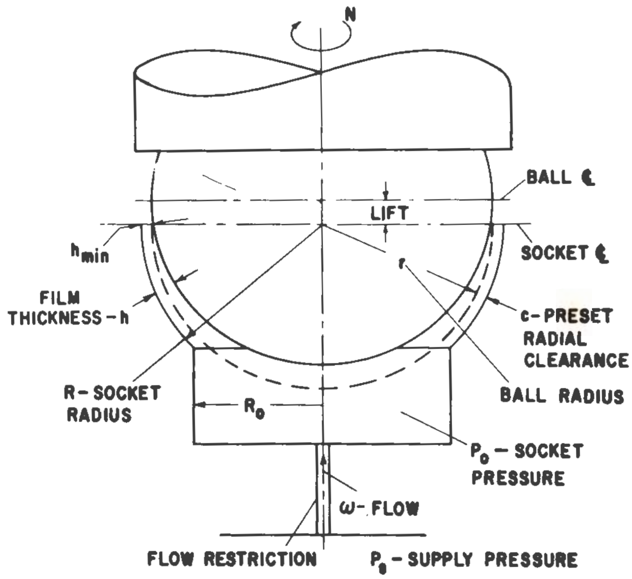

FIGURE 2. HYDROSPHERE BEARING ASSEMBLY

unequal thermal distribution should be avoided. The more conventional journal and thrust bearings are alignment-sensitive, whereas the hydrosphere does not require that the axis of rotation of the sphere be coincident with the geometric axis of the socket. A factor in the hydrosphere selection was its small radial clearance which tended toward high-speed stability. Inspection of the bearing cross section, shown in Figure 2, reveals small equatorial clearances, and axial movement of the shaft does not induce proportional changes in this radial clearance. The small radial clearance and low power loss of the hydrosphere permit a minimum flow rate for absorption of the heat generated by the viscous shear in the bearing.

A disadvantage of the hydrosphere is its precision three-dimensional fabrication requirements. In addition, like most pressurized hydrostatic bearings, unless there is some device to increase bearing socket pressure with increasing axial load the bearing will collapse, shutting off lubricant flow and resulting in bearing seizure. Such a device may be a restricting orifice in the bearing inlet line. The system then is essentially a dual-orifice system with flow being a function of lift; and a properly designed orifice in the inlet line will cause the bearing pressure to increase as the bearing sphere approaches the socket, thereby compensating for the load increase.
type lubricants were eliminated from consideration due to the high temperature, nuclear radiation environment, increased complexity due to required additional lubrication system, and the unavailability of positive rotating seals for 60-day operating capability. Utilization of the system working fluid, mercury, as the bearing lubricant dictated a development program for high-temperature mercury-lubricated bearings but eliminated the seal development.

HYDROSPHERE BEARING ANALYTICAL APPROACH

Background

Early work on the hydrosphere bearing was done by M. C. Shaw and C. D. Strang with conventional lubricants[1, 2]. Their attempts of mathematical analysis based on development of pressure along a mean streamline resulted in a solution which was not completely practical due to its complexity and its utilization of parameters which could not be determined experimentally. Consequently, most of their work was of an empirical nature. The reported tests were made with axially loaded bearings, although some mention was made of radial load tests.

As can be seen from Figure 2, the lubricant film is crescent-shaped in any plane through the axis of rotation, if the ball center is axially displaced outward from the socket center. Since this wedge shape is not in the direction of motion and since the density and viscosity of the fluid are assumed constant throughout the film, the manner in which hydrodynamic pressures are developed to support axial loads is not immediately apparent. A rigorous mathematical analysis of the hydrosphere is complicated by the three-dimensional film which cannot be simply described. Moreover, in the present application, the nonwetting and high-density characteristics of mercury cannot be easily handled in the classical hydrodynamic bearing equations. Because of these complexities and the necessity to formulate parametric relationships at an early date, analytical development was supplemented by experimental tests.

Parametric Relationships

Parametric relationships were obtained by considering only the axi-symmetric configuration of the hydrosphere, with the variable geometric parameter being the axial displacement of the ball out of the socket. Ball rotation effects were ignored, except in the analysis of bearing power. In addition, the classical assumptions of a wetting, Newtonian, constant-viscosity, low-density fluid were assumed. In this manner first-order parametric relationships were derived. By analogy to cylindrical journal bearings and by test results, these approximate relationships proved useful in design and in prediction of hydrosphere performance.

The axi-symmetric configuration greatly simplifies the mathematical description of the film, although it is valid only in the "no radial load" condition. The clearance space between the spherical ball and the slightly larger hemispherical socket constitutes the film as shown in Figure 2. The film geometry is a function not only of the difference in ball and socket radii, but of the axial displacement between their centers. Zero displacement is defined as the concentric position and only outward displacement is considered.

Pressure Distribution The axi-symmetric pressure distribution in the film was derived assuming laminar flow in the axial direction only The effects of ball rotation were ignored. With these assumptions the expression for film pressure was derived to be:

$$P - P_o = \frac{6\mu Q}{\pi R^3} (Kp - Kp_o) \qquad (1)$$

where

$$Kp - Kp_o = f \text{ (geometry, lift)}$$

Flow. Essentially, the hydrosphere is a variable-flow restriction coupled with a fixed restriction. Thus, lubricant flow is an important parameter directly related to pressure distribution, which, in turn, is related to axial load capacity. Rearrangement of Equation (1) and substitution of the conditions at the equator or discharge area yields:

$$Q = \frac{\pi R^3}{6\mu} (P_o - P_m) Kq \qquad (2)$$

where

$$Kq = f \text{ (geometry, lift)}$$

Axial Load Capacity. The axial load capacity of the axi-symmetric hydrosphere is obtained by integration of the pressure forces acting on the ball. By integrating the axial components of the pressure forces, the axial load capacity can be shown to be:

$$W_T = \pi R^2 (P_o - P_m) K_A \qquad (3)$$

where

$$K_A = f \text{ (geometry, lift)}$$

Consideration of the order of magnitudes for several terms of the equation for K_A for practical range of values show that $K_A \approx 1$ for hydrostatic operation. Some increase ($\approx 10\%$) in axial load capacity with speed has been observed experimentally.

Radial Load Capacity. To sustain a radial load, the hydrosphere must have a film pressure distribution which is unsymmetrical about its axis. Unfortunately, the mathematical description for the film thickness of a radially and axially displaced ball is extremely unwieldy. Even an expression for the simpler hydrostatic load capacity becomes practically underivable. The added consideration of ball rotation for the hydrodynamic case makes this solution formidable. The empirical relationship suggested by Shaw and Macks was therefore used. This expression has no analytical basis, but was suggested by their test observations.

$$W_R \leq 1/2 W_T \tag{4}$$

It was obvious that speed, lubricant temperature, surface finish, and socket pressure would affect radial load capacity, so that Equation (4) was used as a limit value rather than as a prediction of capacity.

Power Loss. In addition to its dual load capacity, low power loss was a major factor in the original selection of the hydrosphere. The low power loss was originally predicted from a simplified analysis. Early test results, however, showed higher power consumption and indicated the need for a more refined expression for bearing power loss. The derivation for this expression assumes the axi-symmetric case and considers only circumferential flow. The assumption is analogous to the Petroff condition in classical cylindrical journal bearing analysis. It should be noted that in the case of the axi-symmetric hydrosphere, power loss decreases with increasing lift (i. e. , increasing film thickness). Practically, this reduction in power loss may not be realized in a radially loaded bearing which will not be axi-symmetric. In practical journal bearings it can be shown that power loss may be insensitive to comparatively large changes in radial clearance, if all other conditions are held constant. However, as in cylindrical journal bearing analysis, the axi-symmetric case yields useable estimates of hydrosphere power loss.

Applying the assumptions previously outlined, the expression for power loss in the axi-symmetric case was derived to be:

$$H = 7.8 \times 10^{-3} \mu R^3 N^2 K_T \tag{5}$$

where

K_T = f (geometry, lift)

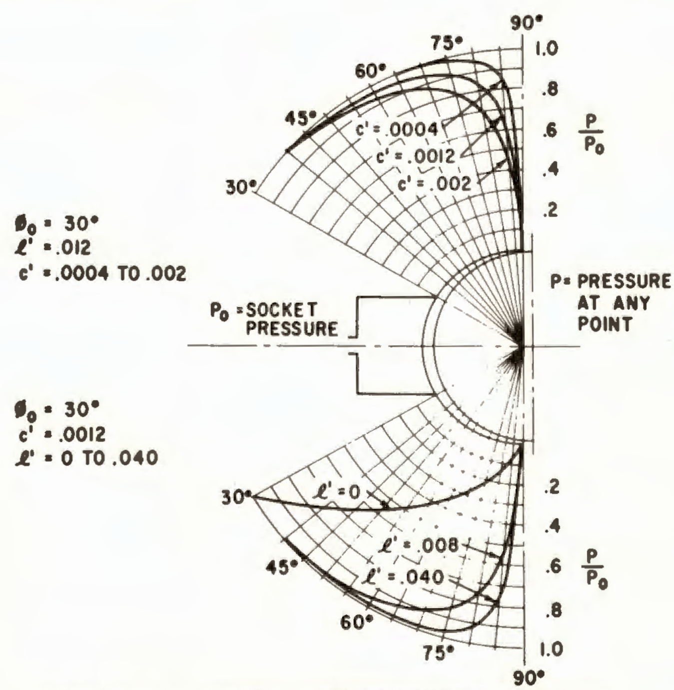

FIGURE 3. TYPICAL HYDROSPHERE BEARING FILM PRESSURE DISTRIBUTION

Design Curves. Figure 3 shows typical pressure distribution curves obtained from Equation (1). It may be noted that the major portion of the pressure deop in the bearing occurs within 15° of the equator. The upper curves are for a constant axial lift ratio and varying initial radial clearance ratios.

As the radial clearance is decreased, the major portion of the pressure drop is taken closer to the equator due to the greater decrease in effective flow area at the equator. The lower curve is based on a constant radial clearance ratio and the axial lift ratio is varied. Increasing axial lift tends to move the pressure distribution curve closer to the equator, since the minimum effective flow area is at the equator and relatively larger clearances exist in the inlet portion of the bearing.

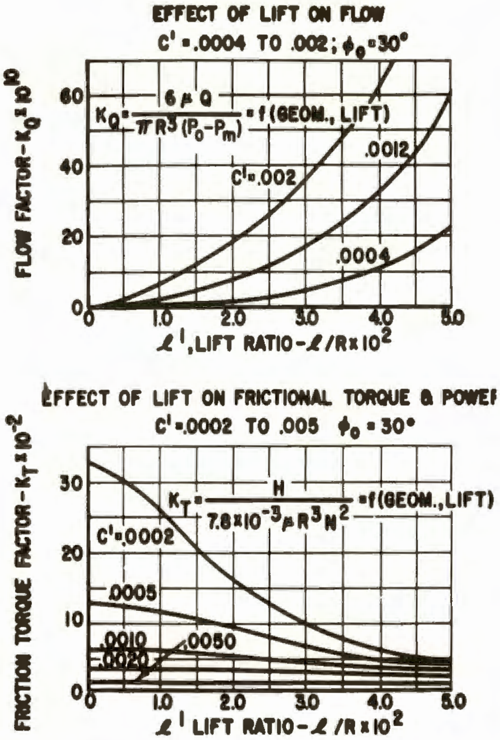

FIGURE 4. HYDROSPHERE BEARING, NON-DIMENSIONAL FLOW AND TORQUE FACTORS

Figure 4 shows the non-dimensional flow and frictional torque factors for the hydrosphere bearings. These factors are complicated functions of bearing geometry and lift. Both factors are plotted versus lift ratio at various radial clearance ratios.

HYDROSPHERE BEARING EXPERIMENTAL CHARACTERISTICS

Initial development of liquid-mercury lubricated hydrosphere bearings involved sockets having grooves in the axial plane and mating perfectly with the spherical journal. Radial load tests showed that the grooves limited capacity to values less than the SNAP I requirements. The present SNAP I hydrosphere bearing configuration consists of a loosely-fitting ball in a nongrooved socket with an integral flow restriction to control socket pressure. Table 1 shows the preliminary design objectives and the current experimental performance of the SNAP I hydrosphere bearings. Figure 5 is a photograph of a dual-bearing test shaft with integral balls and the hydrosphere bearing sockets after 104 hours of liquid mercury operation. Experimental testing of the SNAP I hydrosphere bearings resulted in the determination of the following characteristics.

Flow.

The flow of mercury lubricant to the bearing was controlled by flow restrictions upstream of the bearings and by the axial and radial clearance of the bearings. From Figure 6 it may be noted that total bearing flow is relatively independent of speed. Sufficient lubricant flow must be maintained to remove heat due to the viscous losses in the bearing, as the resultant heat rise may add to the thermal expansion problem and the likelihood of film cavitation. In order to satisfy the system and the bearings the total lubricant flow must be from 6. 0 to 10. 0 lb/min.

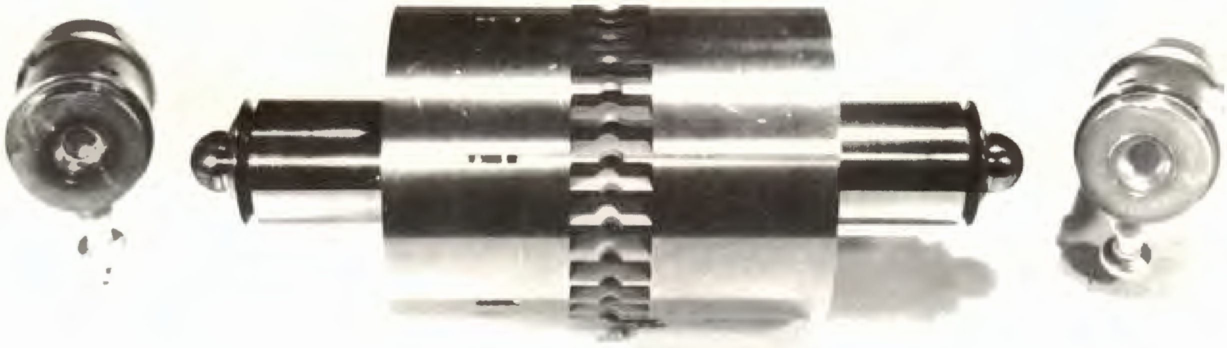

FIGURE 5. DUAL HYDROSPHERE BEARING SHAFT AND SOCKETS AFTER 104 HOURS OF LIQUID MERCURY OPERATION

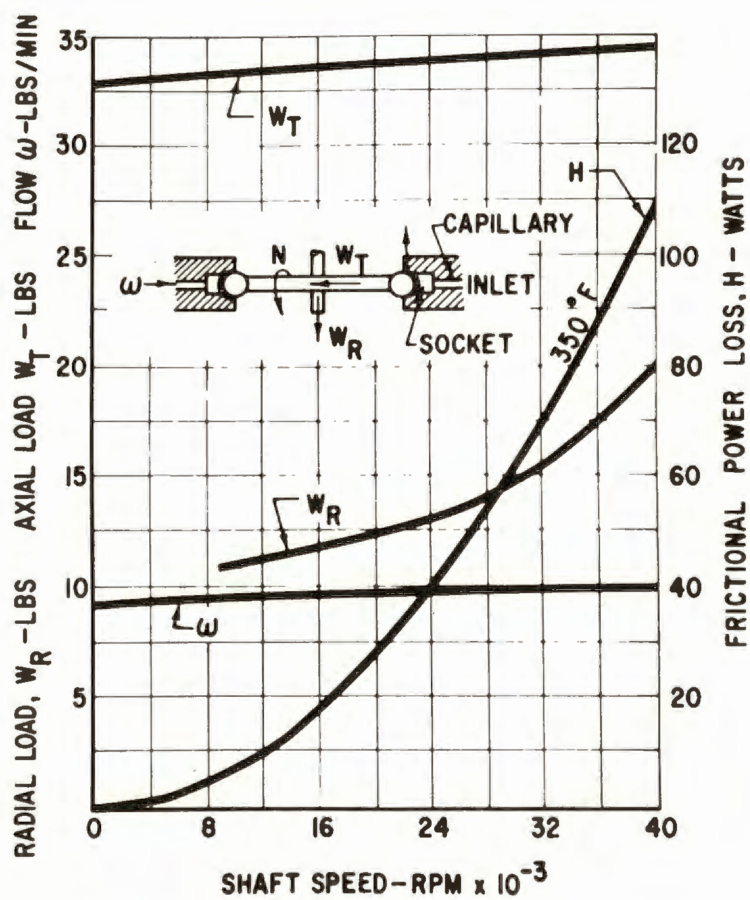

FIGURE 6. COMPOSITE 1/2-INCH DIAMETER HYDROSPHERE BEARING PERFORMANCE, LIQUID-MERCURY LUBRICATION

TABLE 1. SNAP I HYDROSPHERE BEARING DESIGN PERFORMANCE

Parameter	Design Objective	Experimental Performance
Bearing Diameter, in.	--	1/2
Lubricant	Liquid Mercury	Liquid Mercury
Speed, rpm	40,000	Up to 50,000
Power Absorption (2 bearings), watts	100	110
Flow (2 bearings), lb/min	10	10
Lubricant supply temperature, °F	350	350
Lubricant supply pressure, psia	250	250
Lubricant drain pressure, psia	2	2
Life, hr	1440*	2510
Load Capacity:		
Thrust, lb	20	32
Radial, lb	2.5 (operating) 20 (launch)	25
Dynamic Operation	Stable	Stable

*60 days unattended

Flow Restrictions.

To accommodate changing axial loads, a flow restriction was designed integral with the hydrosphere bearing, upstream of the socket. In the preliminary component and prototype test packages, the flow to the bearings was controlled by fixed flow restrictions located between the reservoir and the bearings. Since the incorporation of the jet-centrifugal pump outboard of the alternator bearing complicated the thrust capacity and leakage path of the bearing, a redesign of the flow restriction was necessitated. Analysis indicated that the best solution was an annulus between the pump shaft and the bearing housing. The interdependence of the bearing inlet flow and pump cavity pressure, which is a function of speed, prohibited dynamic annulus calibration of a pump-bearing unit. Therefore, tests were conducted with a unit having an extended shaft, but without a pump. From these tests, conducted at 40,000 rpm, it has been determined that the dynamic calibration of this annulus is 30% less than the static calibration.

Power Loss.

The most reliable power loss data was obtained from spin-down tests of the dual bearing test rig and package units. Bearing power loss is obtained by determining the rotor deceleration power from speed decay data and subtracting the calculated rotor windage power. A typical power loss curve for 1/2-inch diameter dual hydrosphere bearings is shown in Figure 6.

Thrust Load.

The thrust load capacity of a dual hydrosphere system has been shown experimentally to be equal to the difference in socket pressures times the projected area of the ball. For two hydrosphere bearings of the same diameter the maximum thrust capacity is dependent upon socket pressures which are, in turn, dictated by the choice of supply pressure and flow restrictions. The requirements of the SNAP I system resulted in a bearing whose demonstrated thrust capacity is in the range of 25-35 lb. Since hydrosphere thrust load capacity is principally hydrostatic, it is relatively independent of speed. The addition of an extended shaft on which the system pump was mounted, outboard of the bearing, complicated the simple relationship, although the thrust capacity can still be adequately determined by the difference of the product of the respective cross-sectional areas and bearing socket pressure.

Radial Load.

The destructive characteristics of radial load capacity tests and the shortage of time and test specimens precluded absolute determination of the radial load. However, tests indicated 1/2-inch diameter dual-bearing radial load capacities to 25 pounds. Other tests indicate that the bearing is also capable of hydrostatic radial load support.

Lubricant Temperature Rise.

The monitoring of lubricant temperature rise was used in determining bearing power loss through a heat balance analysis. However, it was soon discovered that the complexity of the package housings and shaft resulted in thermal gradients which were difficult to determine, and this method was disregarded. Temperature rise data was employed only to insure that the discharge temperature of the bearing was below the vaporization point.

System Contamination.

The hydrosphere system is sensitive to clearance changes since a reduction in clearance of the flow restrictions or the bearings while running results in a corresponding reduction in load capacity. Therefore, the amount of system contaminants which might result in clearance changes should be minimized.

Bearing Life.

Bearing life at design operating conditions is one of the most important considerations of the SNAP I system. An endurance run of 2510 hours was completed with the prototype turbomachinery unit in the mercury test facility, and the test was terminated with a planned shutdown without encountering any hydrosphere bearing malfunction.

Dual Hydrosphere Bearing Characteristics.

The operation of a rotating shaft supported by hydrosphere bearings on either end imposes several unique problems. To simplify the discussion several assumptions will be made:

(1) Both bearing configurations are the same (equal diameters, initial radial clearance, inlet hole, etc.).

(2) Both inlet flow restrictions are the same.

(3) Constant bearing supply and drain pressure.

(4) Equilibrium or steady-state conditions.

Figure 7 shows the hydrostatic 1/2-inch diameter dual bearing performance (pressures, flows, and axial loads) vs. axial clearance for a preset total axial clearance of 0.008 inch. To establish equilibrium with the dual bearing system a force balance must exist. Since the flow restrictions and the projected areas of the balls are the same, the shaft will be centered and the bearing socket pressures will be approximately equal with no applied axial load for either hydrostatic or hydrodynamic operation.

If an axial load is applied to the shaft, the shaft will move toward the loaded bearing, reducing the loaded bearing effective flow area and increasing the socket pressure. Flow through the unloaded bearing will increase and the socket pressure will decrease. The shaft will reach a new equilibrium position toward the loaded bearing when the applied thrust load is roughly equal to the difference in socket pressure times the projected area of the ball. Additional applied axial loads will move the shaft closer to the loaded bearing, increasing the socket pressure and decreasing the flow. The limit of axial load capacity occurs as the socket pressure approaches the supply pressure, lubricant flow approaches zero, and the bearings are operating practically dry with minimum clearances. As axial loads are applied toward the loaded bearing, the unloaded bearing clearance and flow increase and the socket

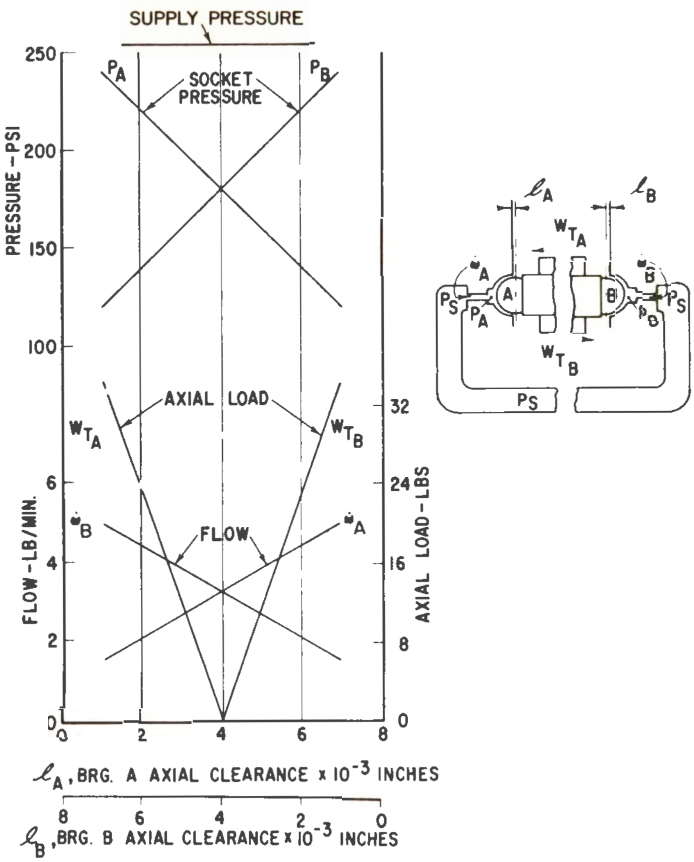

FIGURE 7. TYPICAL DUAL HYDROSPHERE BEARING CHARACTERISTICS

pressure decreases. The lower socket pressure and increased clearances of the unloaded bearing reduce the radial load capacity of this bearing. Variations in effective flow areas of the bearing or of the flow restriction, and changes in supply or drain pressures, would complicate the above relationships.

Hydrosphere Bearings in Test Packages.

Incorporation of the hydrosphere bearings in subsystem and system test packages followed the successful development of the hydrosphere bearings in component tests. Extended shaft tests were necessary to obtain a satisfactory configuration and bearing characteristics prior to incorporation of the jet-centrifugal pump. These tests were followed by subsystem tests of the hydrosphere bearing-jet centrifugal pump combination in order to simulate final package configuration and to insure satisfactory bearing and pump operation while coupled. Integration of the bearings in the system test packages represented the culmination of a successful hydrosphere bearing development program.

Materials Development.

Several bearing materials and platings were evaluated in actual hydrosphere bearing operation during the experimental development program. Additional anti-scoring tests of a roller on a flat plate with liquid mercury as a lubricant were conducted by New Departure Division of General Motors to determine the relative anti-scoring qualities of ten promising materials pairs selected by TRW. Mercury materials evaluation capsule tests were also conducted to check mercury compatibility with the bearing materials pair.

The majority of the experimental development bearings consisted of a tungsten alloy high-speed tool steel as a base material for both the ball and socket. A tungsten-titanium carbide combination appeared superior to the tool steel combination for wear and scoring resistance, although the carbides exhibited brittleness and poor resistance to mercury corrosion. The final materials selection of a high-speed tool steel ball and socket was based on wear and scoring resistance, mercury compatibility, equal thermal coefficients of expansion, and several thousand hours of experimental high-speed tool steel bearing performance.

Hydrosphere Bearing Fabrication and Inspection.

Specialized processes were developed for the fabrication, inspection, and assembly of the hydrosphere bearing due to its unique characteristics. The incorporation of the ball on the shaft prevented utilization of conventional machining techniques. Both the bearing and the socket were rough machined in the soft condition, heat treated, and then finish machined. The bearing socket was then lapped with a cast iron ball which was machined to size. The hydrosphere ball was machined and final lapped to the diameter which provides the desired radial clearance between the ball and socket. This diameter was checked with a dial indicator micrometer which was indexed with a master ball. Before and after each test, the hydrosphere bearing ball and socket were visually and dimensionally inspected and a profile of the socket was obtained on the Indiron recorder at distances of 1/16, 1/8, and 1/4 inches from the equator.

BEARING TEST FACILITIES

The original bearing test rig was basically a hydrosphere bearing feasibility rig capable of rotating the bearing at 40,000 rpm and applying axial loads while measuring bearing torque.

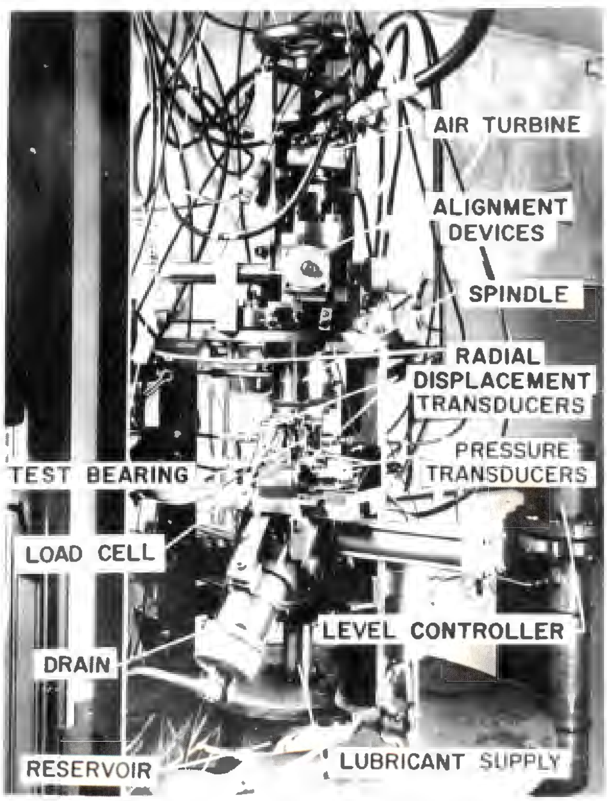

FIGURE 8. SINGLE HYDROSPHERE BEARING TEST RIG

A single bearing test rig, as shown in Figure 8, was designed to obtain parametric data on various bearing configurations, using mercury as a lubricant. This rig features an air turbine drive, a pressurized mercury reservoir, means for loading the bearing both axially and radially, and a complement of instrumentation for measuring temperatures, pressures, power losses, flows, torque, axial and radial loads, and axial and radial displacement. This rig can test axial bearing loads of about 30 pounds and radial bearing loads up to 20 pounds. The axial lift and radial clearance of the bearing are obtained from proximity transducers.

The dual bearing test rig shown in Figure 9 was designed to test a dual bearing shaft under simulated package conditions. This rig features a Terry-type air turbine drive, an air preheater, a pressurized mercury reservoir, and a means for applying various axial and radial loads simultaneously and individually to the bearings. This rig is instrumented to measure pressures, flows,

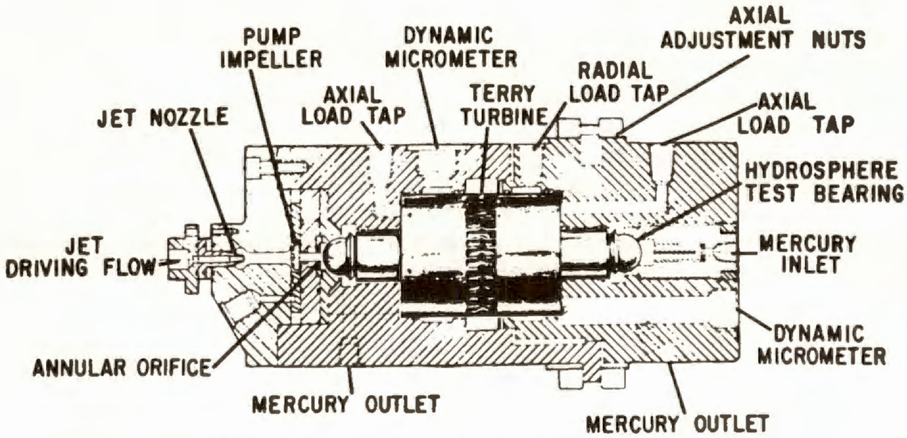

FIGURE 9. DUAL BEARING TEST FIXTURE

temperatures, speed, and axial and radial displacements with dynamic micrometers and an oscilloscope. Pressure transducers and a six-channel Brush recorder are utilized to record sudden changes in pressure, and an audioscope is used to determine bearing scoring through changes in noise level. The dual bearing test rig has provisions for applying external axial loads up to 60 pounds and radial loads up to 50 pounds in both an upward and downward direction. This rig can attain speeds in excess of 60, 000 rpm, can supply mercury to the bearings at 300° F, and can be used to conduct extended endurance tests.

CONCLUSIONS

Analytical and experimental SNAP I bearing development proved the capabilities of the 1/2-inch diameter mercury-lubricated hydrosphere bearings to satisfy the SNAP I bearing specifications. The following conclusions were obtained concerning hydrosphere bearing design and performance:

(1) Axial and radial loads of the magnitude encountered in the SNAP I application can be sustained by the 1/2-inch diameter dual hydrosphere bearings.

(2) Power losses as determined by a more refined analysis and by test data are higher than originally predicted but are within the specification.

(3) Dynamic stability of the lubricant film has been demonstrated consistently.

(4) Flow rates are reasonable and are relatively independent of speed.

(5) Small angular misalignment of the shaft can be tolerated without affecting performance.

(6) The bearing is sensitive to small clearance changes which may arise from differential thermal expansion, from build-up of lubricant contaminant or from inadequate precision of fabrication. These effects, however, can be controlled.

(7) Some hydrostatic support of radial loads has been demonstrated. This effect permits lifting the shaft prior to rotation if sufficient supply pressure is available.

(8) Axial grooves in the socket tend to reduce radial load-carrying capacity.

(9) Contact stresses at nominal radial loads are high due to line contact, but can be reduced to tolerable levels by proper design of the socket equator.

(10) High-speed tool steel was selected as the final bearing material for both the ball and socket due to wear and scoring resistance, mercury compatibility, equal thermal coefficients of expansion, and extended experimental bearing life.

(11) Endurance capability of the mercury-lubricated hydrosphere bearings was demonstrated in a 2510-hour life test of the system prototype test package.

NOMENCLATURE

r = radius of ball, inches

R = radius of socket, inches

c = radial clearance, inches

c' = c/R, clearance ratio

h = film thickness, inches

ℓ = axial displacement of ball center, inches

ℓ' = ℓ/R, lift ratio

R_o = inlet radius, inches

θ_o = inlet angle, degrees

$\dot{w}$ = weight flow, pounds/minute

W_R = radial load capacity, pounds

W_T = thrust load capacity, pounds

H = frictional power, watts

μ = absolute viscosity, lb sec/in^2

K_p = film pressure factor

K_q = flow factor

K_A = axial load factor

K_T = power loss factor

p = film pressure, psig

p_o = socket inlet pressure, psig

N = shaft speed, rpm

Q = volume flow, gpm

LIST OF REFERENCES

1. Shaw, M. C. and Macks, E. F., Analysis of Lubrication of Bearings McGraw-Hill, 1949.

2. Shaw, M. C. and Strang, C. D., "The Hydrosphere - A New Hydrodynamic Bearing," ASME Journal of Applied Mechanics, June 1948.

3. Goodzeit, C. L., "Compatibility of Metals in Bearing Contact," ASME Transactions, 58MD9, April 1958.

4. Fuller, D. D., Theory and Practice of Lubrication for Engineers, John Wiley and Sons, 1956.