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NaK-LUBRICATED JOURNAL AND THRUST BEARINGS FOR A PUMP-MOTOR ASSEMBLY IN SNAP-8

By

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AEROJET-GENERAL CORPORATION
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS 5-417
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TOPICAL REPORT

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Cleveland, Ohio

Martin J. Saari, Program Manager

SNAP-8 Program Office

FORWARD

The design and development effort described in this report was performed by Aerojet-General Corporation, Azusa, California, as part of the SNAP-8 Electrical Systems Contract being conducted within the Power Systems Department. The work was performed under NASA Contract NAS 5-417 with Mr. Martin J. Saari as NASA Program Manager and Dr. W. F. Banks as Aerojet-General Corporation Program Manager.

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ABSTRACT

The operational experience with SNAP-8 pump-motor assembly bearings lubricated with high temperature sodium-potassium alloy (NaK-78) is presented in this report. The bearings were segmented, hydrodynamic fluid film, tilting pad journal bearings and Kingsbury-type self-aligning thrust bearings. The experience included development tests and separate endurance tests of 3,000 and 10,000 hours utilizing a complete pump-motor assembly. Minor development problems were encountered, but the bearings completed the endurance tests satisfactorily with no measurable bearing wear. Inspection revealed minor scratches to the journal bearings which were attributed to oxides passing through the bearings or other contaminants in the NaK lubricant.

SUMMARY

Sodium-potassium (NaK) lubricated bearings were required for a pump-motor assembly with a design life of 10,000 hours continuous operation unattended at temperatures up to 600^{O}F. The selected bearing design consists of segmented, hydrodynamic fluid film, tilting-pad journal bearings and Kingsbury-type, self-aligning thrust bearings. The journal bearing has four pads, each pivoted on a ball with each pad covering an arc of 80 degrees. The thrust bearing has a thrust runner and six pads on each side of the runner with each pad pivoted on a ball segmented support ring. The support ring is positioned on a pivoted gimbal plate.

The bearings were first tested in oil, followed by testing as part of the assembled pump in water and in NaK. The oil test results indicated that both the journal bearings and the thrust bearings had adequate load capacity throughout the test range, and the minimum lubricant film thicknesses were close to the calculated values. There was no evidence of half frequency whirl or other bearing instabilities during the tests.

In the water tests, it was determined that a pad surface finish of 10 to 15 microinches, rms was required because an excessively fine surface finish, which was on the order of 2 to 4 microinches, was subject to "wringing", i.e., the pads would adhere to the rotating.

The bearing tests with NaK indicated that the self-aligning pivot on the thrust bearing pads allowed the pads to wedge against the thrust runner. This wedging increased the starting torque of the pump. The design was modified to restrict the circumferential travel of the pads by pinning the pivots and this appeared to have resolved this problem.

A pump-motor with the modified design was subjected to a 3,000-hour endurance test which included numerous starts and stops. After this test, the bearings were inspected and found to be in good condition. Initial subsequent testing caused damage to the journal bearings and pads due to the passage of foreign matter or oxides.

The same pump was put back into service with replacement journal bearing pads and a total test time of 13,913 hours was then accumulated. Experience from these tests indicated that the bearings were sensitive to contaminants in the NaK fluid. In several instances, the failure of the pump to start could have been directly attributed to the contaminant. The latter part of this test was for a total time of 10,362 hours including 786 starts. There were no apparent changes in the bearing performance during this test period. However, examination of the journal bearings after the tests were completed revealed some scratches on the running surfaces. The scratches were slight and probably would not affect the bearing performance for extended operation. The thrust bearings showed no wear or scratches as a result of the endurance tests.

The test experience has shown that conventional theory for hydrodynamic bearing design is satisfactory for use in the design of NaK-lubricated bearings. A bearing design life of much greater than 10,000 hours and 100 starts can be attained.

I. INTRODUCTION

SNAP-8 is a 35 to 90 kWe nuclear-electrical power conversion system for use in space. The system operates on a mercury Rankine cycle using NaK (eutectic sodium - potassium mixture) as heat-input and heat-rejection working fluids. The power conversion system is being developed by Aerojet-General Corporation for the National Aeronautics and Space Administration. The nuclear reactor is being developed for the Atomics Energy Commission by Atomics International.

The power conversion system (see Figure 1 for the 35 kWe system) uses mercury as a working fluid and is coupled to the reactor cooling loop by a heat exchanger (boiler) where the mercury is preheated, vaporized, and superheated. The superheated vapor drives a turbine-alternator assembly which develops the 400 Hz electrical output of the system. The saturated mercury vapor leaving the turbine passes through a condenser and then to a mercury pump to complete its cycle. Cooling for the condenser is provided by a second NaK loop which couples the condenser and space radiator where waste heat is rejected. Lubrication and cooling of the system is provided by a loop using an organic working fluid (polyphenyl ether).

The NaK pump-motor assembly (NaK PMA) was designed for pumping the liquid metal, eutectic mixture of sodium and potassium (NaK-78) in the SNAP-8 system. Two NaK PMA's are required; one in the reactor or primary NaK loop (PNL) and the other in the heat rejection loop (HRL) as shown in the schematic diagram, Figure 1. The original minimum life requirement was 10,000 hours without maintenance. However, a more recent SNAP-8 life requirement is for 5 years, minimum.

Journal and thrust bearings, lubricated with NaK, are utilized in the pump-motor assemblies (PMA). The bearings are designed to operate at temperatures up to 600°F and carry a load imposed by a conventional centrifugal pump and 3-phase, 400 hertz motor rotating at 6000 rpm. The journal bearings are of the segmented, hydrodynamic fluid film, tilting-pad type, and the thrust bearings are Kingsbury self-aligning type. Testing of the bearings

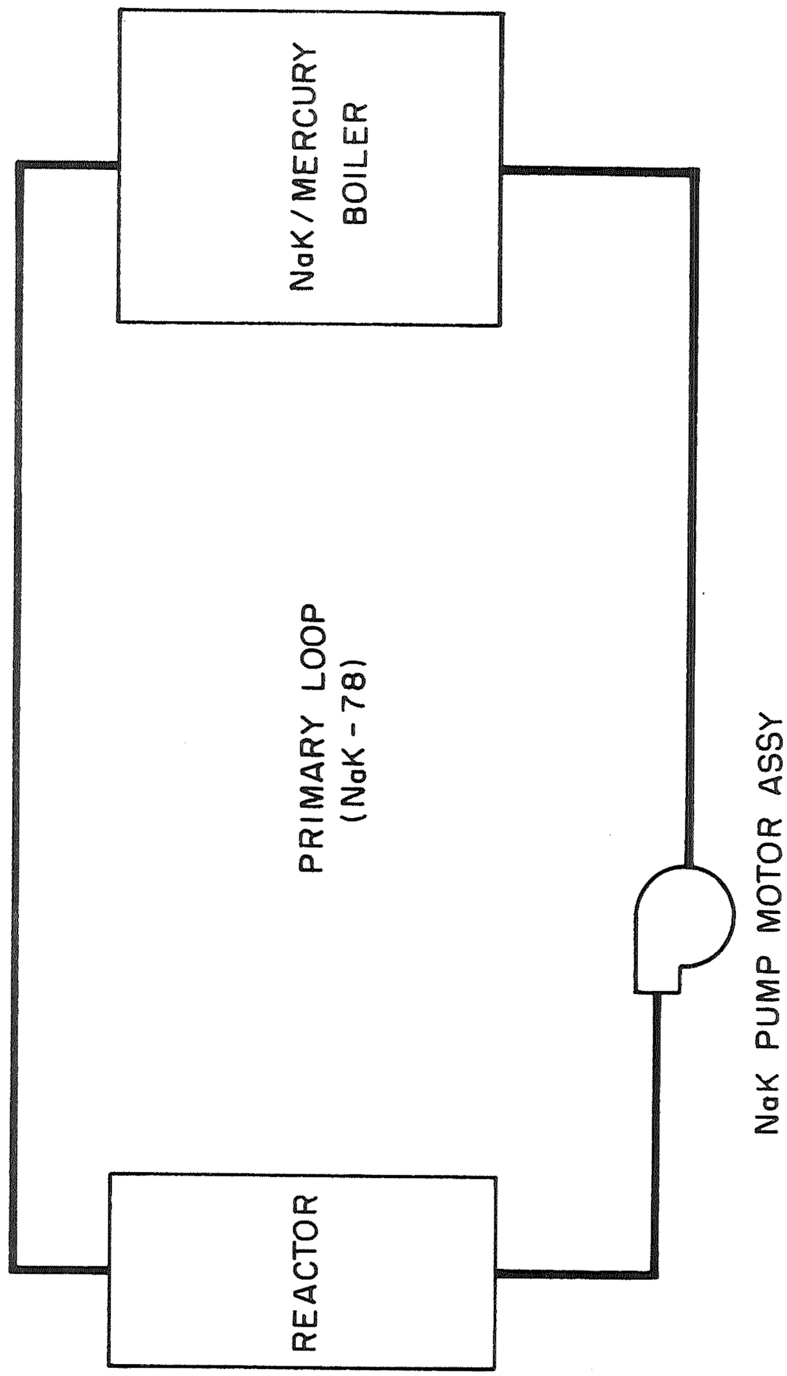


Figure 1 Simplified Schematic of SNAP-8
Primary NaK System

under actual operating conditions has been continuous since 1964. The longest test on a single set of journal bearings is 10,362 hours, and the longest test on a single set of thrust bearings is 13,913 hours.

This report documents the more significant developments during the testing through September 1970. Tests were initially conducted as bearing sets alone in oil during the bearing development phase. Later, testing in complete NaK pump-motor assemblies was conducted in water and then in NaK systems. The latter tests were primarily concerned with pump hydraulic performance and only concerned with bearing development from the standpoint of obtaining endurance life data.

The bearings have performed satisfactorily for the required design life of the SNAP-8 system, although on occasions some starting difficulties were experienced. These were either due to oxides or foreign matter interfering with the journal bearing operation, or to the geometry of the journal bearing pivot arrangement allowing jamming to occur between the bearing pad and journal due to a wedging effect. A recent study of the geometry of the these bearings and their pivot design revealed the possibility that jamming of the bearing shoes could occur due to the bearing pivot ball being able to move within its constraints thereby creating a wedging effect which tends to apply a large radial force between the bearing shoe and journal sleeve thus preventing the rotor from turning. Some minor changes were analyzed which restricted the motions of the ball pivot and these were evaluated by testing the PMA in water. Although the test results were not 100% conclusive, there were indications of resolution of the problem.

II. BEARING DESIGN AND DEVELOPMENT

A. NaK PUMP MOTOR ASSEMBLY DESCRIPTION (See Figure 2)

The NaK pump motor assembly (NaK PMA) incorporates the main centrifugal impeller, the canned motor armature, the thrust and journal bearings, and a recirculation pump impeller on a common shaft. The main impeller is used to pump NaK in the system loop, while the recirculation impeller circulates a small volume of NaK through the bearings, the motor cavity and through a heat exchanger, cold trap and filter system, thus cooling the motor and bearings, and assuring maximum cleanliness in the fluid being circulated through the bearing by cooling, trapping out and filtering the impurities and oxides in the NaK. Isolation of the closed recirculating NaK from the main loop NaK is achieved by means of a close clearance annulus between the main pump and the PMA bearings and motor cavity. Thus, the recirculating fluid is virtually uncontaminated by the main loop fluid, with any small amounts of oxides which migrate through the annulus being "trapped out" in the cold trap system.

The main centrifugal pump impeller is a semiopen type with back vanes to achieve overall assembly thrust balance. A double volute is used to minimize radial loading over a wide range of pumping conditions. The recirculation pump has a full-open vane impeller (see Reference 1).

The three-phase, 208 volt, 400-Hertz, 6000 rpm, squirrel-cage induction motor is hermetically sealed by use of Inconel cans on both the rotor and stator to prevent entry of NaK. The rotor shaft is supported on two tilting-pad journal-bearing assemblies of four pads per bearing. The thrust bearing consists of a rotating thrust "runner" with six pads on each side of the runner to support axial loads in either direction.

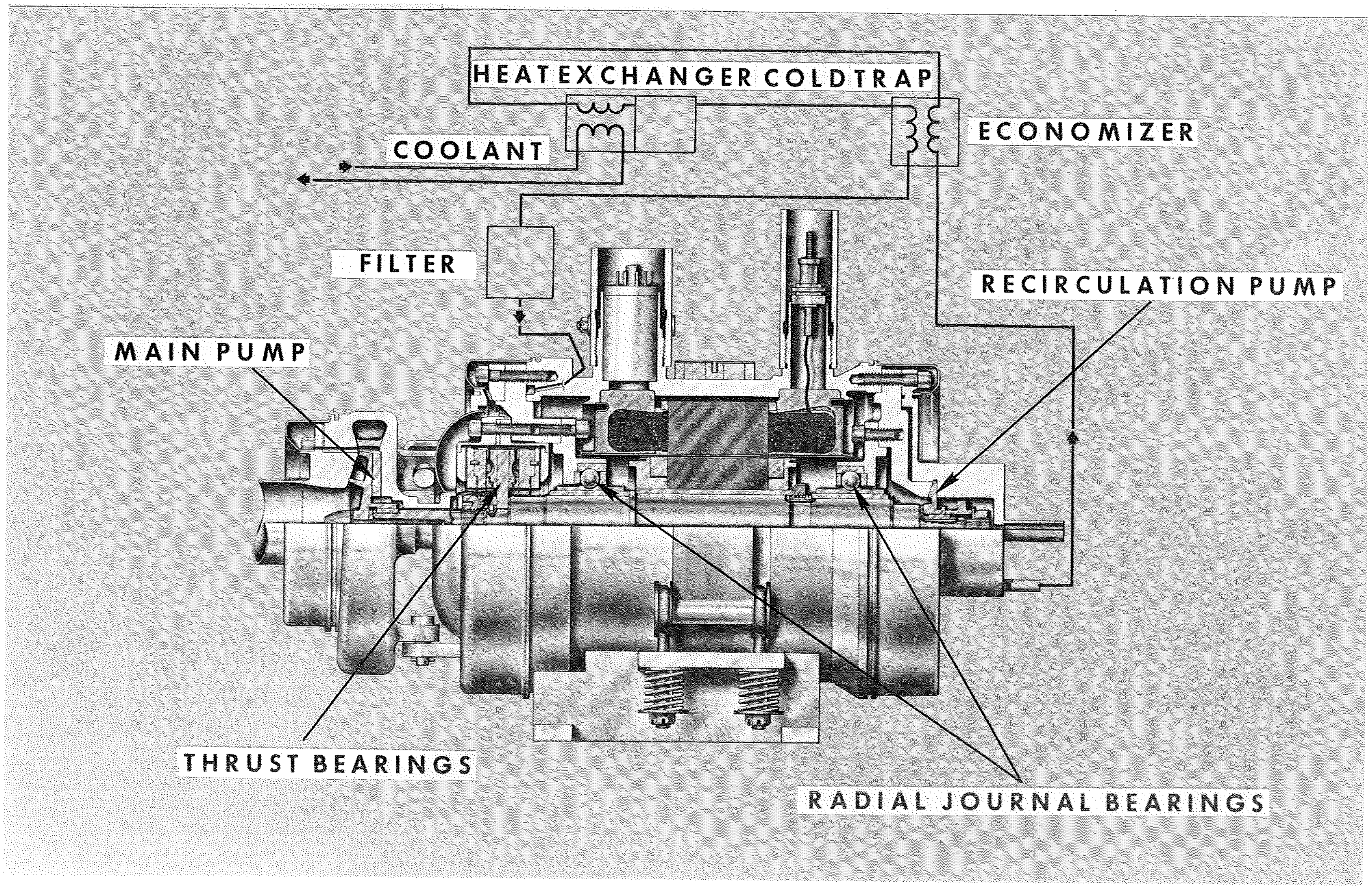


Figure 2. Cross Section Drawing of the NaK Pump-Motor Assembly

B. BEARING DESIGN

1. Requirements

The bearings are designed for a pump-motor assembly that is required to operate in space for 10,000 hours and 100 starts without maintenance or repair. The bearing lubricant is the pump-motor working fluid, a sodium-potassium alloy (NaK-78) which is cooled to a maximum temperature of 600°F. The specific bearing requirements are as follows:

Journal bearings:

Steady state load, lb	0 to 50
Rotating load, lb	0 to 5
Speed, rpm	6000
Minimum bearing stiffness @ 6000 rpm, lb/in	18,000

Thrust bearings:

Load, lb	0 to 80
Speed, rpm	6000

2. Design Procedure

Based on a review of the more successful conventional bearing design applications, it was determined that the best bearing for SNAP-8 pump-motor use would be the segmented, hydrodynamic fluid film, tilting-pad journal bearings with the thrust taken up by a Kingsbury-type, self-aligning bearing.

Factors to consider in tilting-pad bearing design are total load carried, shaft speed, shaft diameter, lubricant inlet temperature, desired lubricant temperature rise and lubricant viscosity. A critical requirement for hydrodynamic bearing design is the ability of the bearings to survive repeated start-ups when metal-to-metal contact could occur prior to the hydrodynamic film being formed. Viscosity characteristics of NaK are shown in Figure 3. After determining the total load carried and other specified characteristics, the number of pads and size of pads may then be determined, which will give the desired values of temperature rise and minimum film thickness. Another variable which requires determination is the location of the pivot. For rotation in one direction only, the optimum pivot location is generally 58 percent of the distance from the leading edge to the trailing edge of the pad. Use of this pivot point gives maximum load capacity and minimum friction.

a. Journal Bearings

In journal bearings, the power loss increases directly with bearing length and in proportion to the cube of the diameter. In thrust bearings, the power loss increases as the fourth power of the diameter, which indicates that an advantage exists with keeping bearing diameters as small as possible. (See Reference 2).

The tilting-pad journal bearing is an effective design for whirl-free operation under very light load conditions with low viscosity fluids. Each pad is able to tilt to assume the most effective working position. The pivot point is 58 percent of the pad length from the leading edge. (See Reference 3).

A method of analysis for journal tilting-pad bearings has been published by Boyd and Raimondi (see Reference 4). They found that eccentricity ratios at L/D = 0.3 (Length and Diameter of Bearing) are smaller for tilting-pad bearings than for a plain journal bearing. This results in a greater bearing stiffness which helps prevent whirl.

One of the basic design criteria for journal bearings is the operating temperature of the bearing. The operating temperature is related to the working viscosity of the lubricant which controls minimum fluid film thickness, rate of fluid flow through the bearing, and rate of heat production or power loss in the bearing.

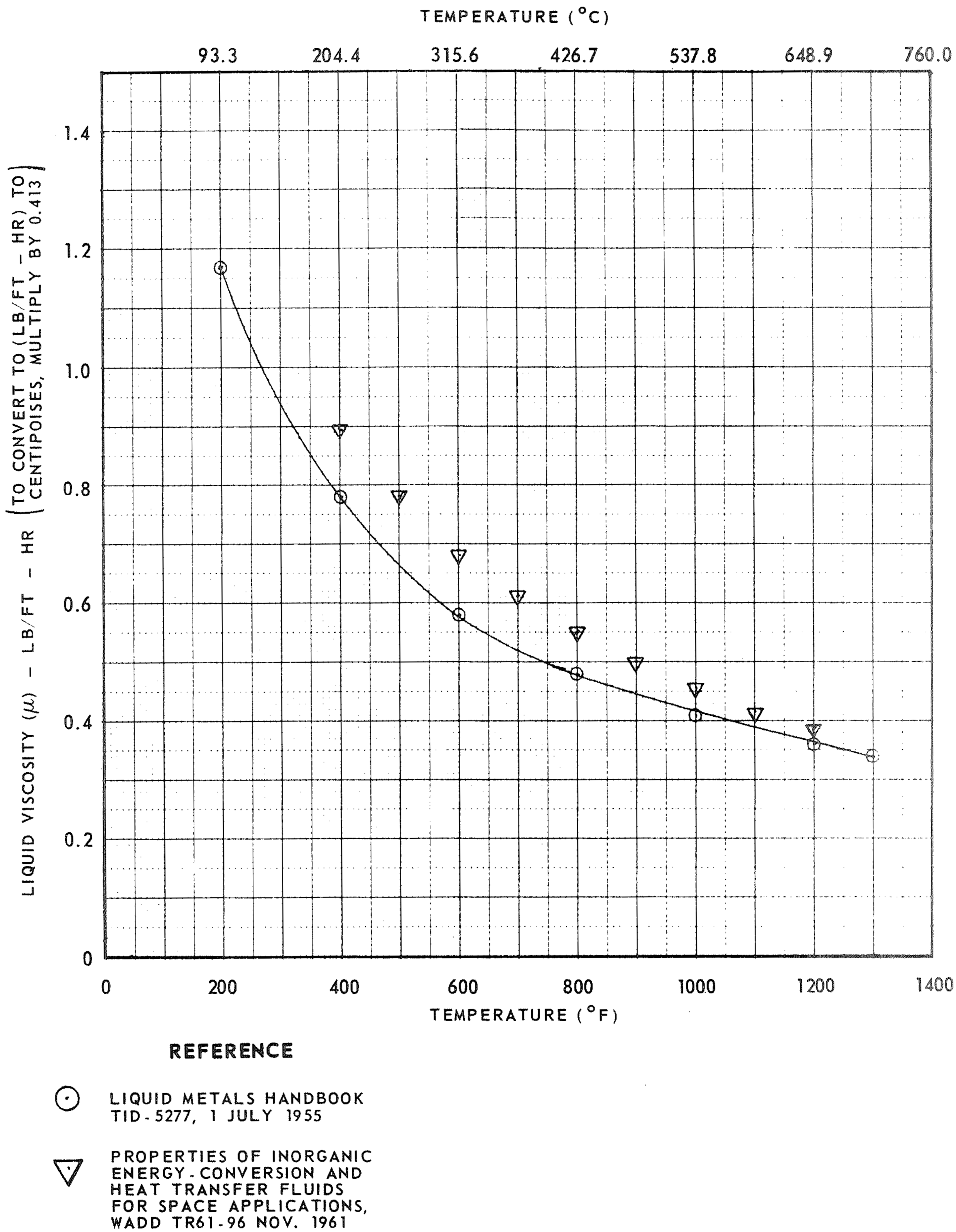


Figure 3. Absolute Viscosity of Liquid NaK (NaK-78)

The NaK PMA journal bearing uses four pads per bearing with each pad pivoted on a ball which is retained in a non-rotating separator. The cage or separator keeps the four balls equally spaced. This bearing was chosen because of its inherent stability and low frictional characteristics. It can tolerate small magnitudes of shaft misalignment and can be designed for high stiffness at zero load. Figure 4 shows a cross-section of the journal bearing assembly.

The performance of the four-pad journal bearings was calculated as a function of pad and bearing clearances for a unidirectional bearing load of 55 pounds (design load is 50 lb), NaK lubricant at $600^{o}F$ (viscosity 3.5×10^{-8} lb-sec/in.^{2}) and a pump speed of 6,000 rpm. For the range of clearances defined by the manufacturing tolerances that were specified in the design, the calculated film thickness is approximately 0.00035 inches. The bearings depend on hydrodynamic lubrication for low-friction characteristics. If a fluid-film bearing is operating in the hydrodynamic region, the bearing friction becomes a function of the viscosity of the lubricant rather than of the coefficient of sliding friction between bearing surfaces.

The load stiffness of the bearings is sensitive to bearing-to-shaft clearance, and to pad-to-gimbal clearance. Assuming the most severe conditions, the lowest value of load stiffness calculated is 19,000 pounds per inch.

The natural frequency of the journal pads was calculated to see if they would track shaft motions. The results show that, over the range of loads and clearances considered, the natural frequency range for the lower pads was 710 Hz to 3680 Hz and the natural frequency for the upper pads was 210 Hz to 1700 Hz. The pads will thus track the shaft motions without difficulty, and probably with no fluid film pad resonance in the operating speed range.

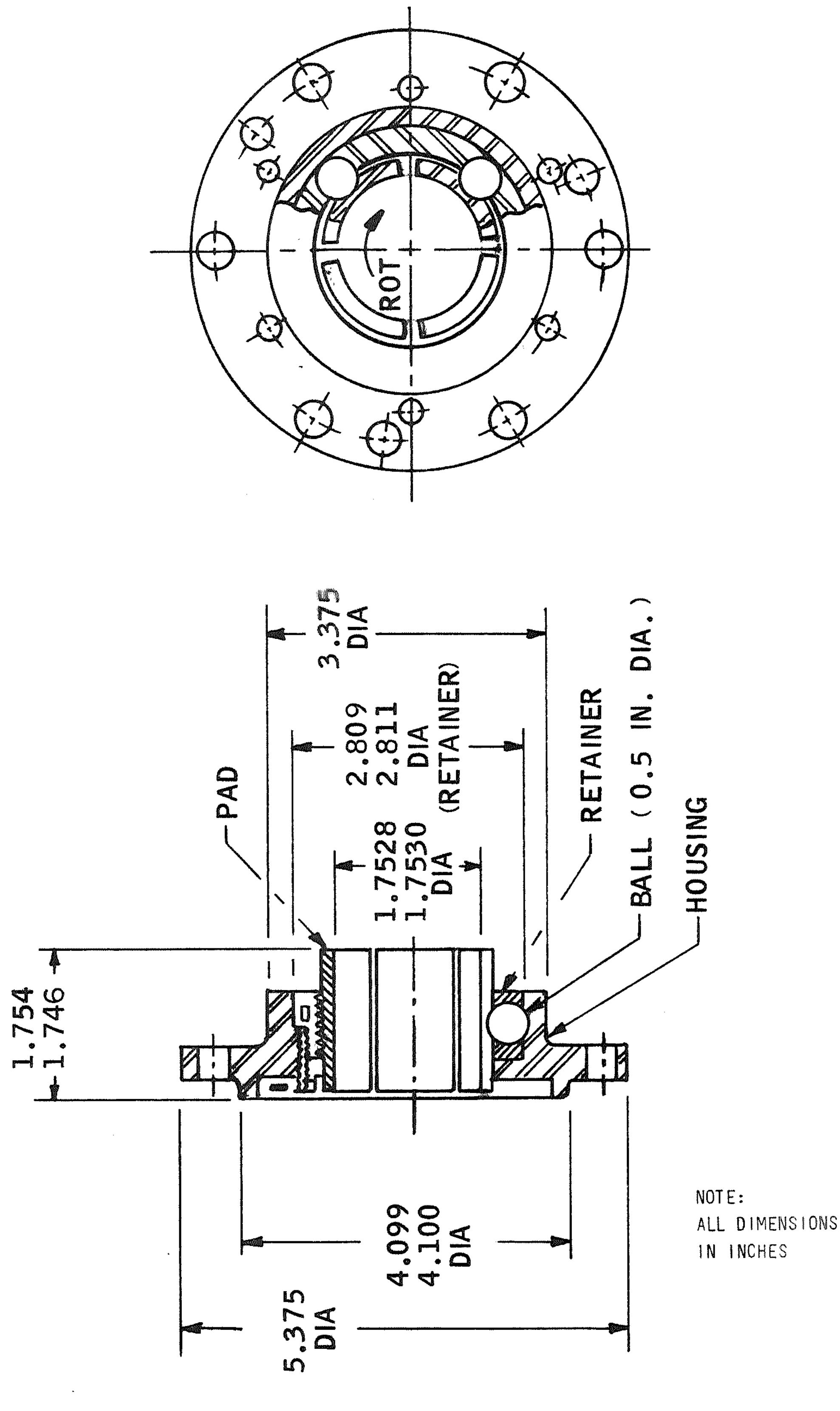


Figure 4. NaK Pump Journal Bearing Assembly

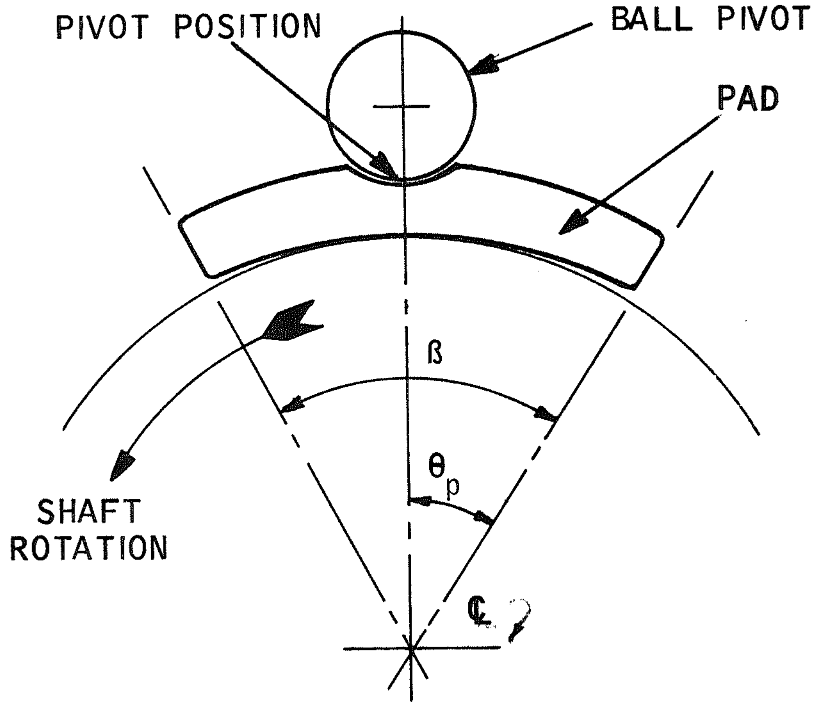
The pump-motor journal bearing dimensions and geometry are as follows:

Type	4 pad, tilting bearing pad
Length, inch	1.75
Bearing diameter, inch	1.75
β - Angular extent of pad, degree	80
$\frac{\theta_P}{\beta}$ - Angular distance from leading edge to pivot (See Figure 5A)	46° (58% of total angular extent of pad)
C_P - Pad radial clearance, inch	$0.0015\ ^{+0}_{-0.0002}$
C_B - Bearing radial clearance, inch	$0.0010\ ^{+0.0003}_{-0.0002}$

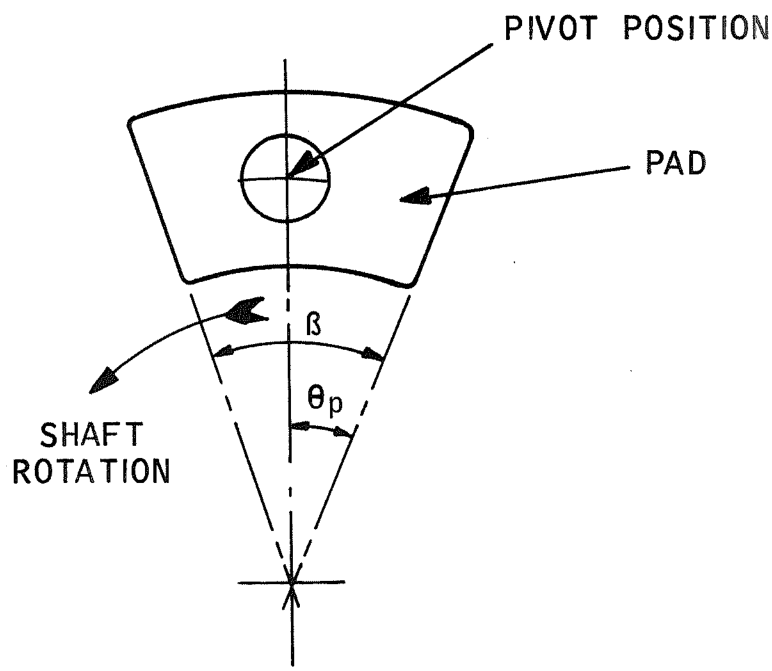
b. Thrust Bearings

The thrust bearings consist of a rotating thrust plate or "runner" with a thrust bearing assembly on each side of the runner to support axial loads in either direction. Each bearing consists of six pads pivoted on spherical segments (support ring) that in turn are supported by a support ring and a gimbal pivoted ring. The combination of tilting pads, spherical segments and pivoted gimbal rings allows the accommodation of shaft misalignment that may result from manufacturing tolerances and thermal distortion. The pads are flat and the thrust runner is slightly "coned" such that the apex is on the shaft centerline. This was done to prevent "wringing" of the flat pads to a flat thrust runner. Figure 6 shows a cross-section of the thrust bearing assembly.

With the load equally distributed between the pads, the thrust bearing will carry the maximum load at a minimum film thickness of 0.0004 inch and with a mean temperature rise of $6.4^{\circ}F$ across the lubricant film. If the maximum load difference between the pads due to the tolerances and deflections is assumed, then the most heavily loaded pair



(A.) JOURNAL BEARING PIVOT POSITION



(B.) THRUST BEARING PIVOT POSITION

Figure 5. Bearing Pivot Details (Not to Scale)

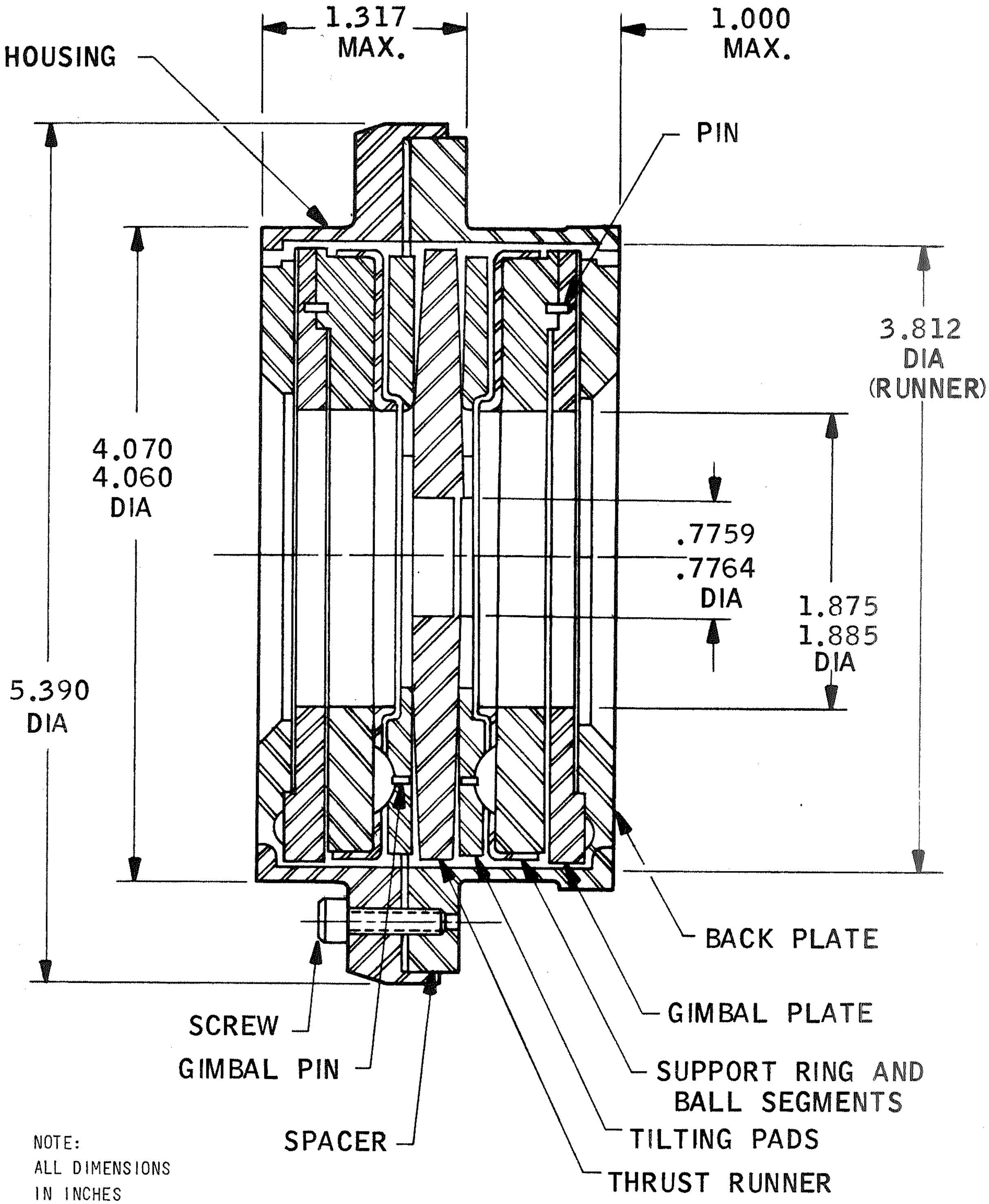


Figure 6. NaK Pump Thrust Bearing Assembly

of pads will have a minimum film thickness of 0.00031 inch with a temperature rise across the lubricant film of 10.7°F.

The thrust bearing dimensions and geometry are as follows:

Type	6 pad, tilting-pad bearing with supported on gimbal plates
Outside bearing diameter, inch	3.75
Inside bearing diameter, inch	1.875
β - Angular extent of pad, degree	50
$\frac{\theta_P}{\beta}$ - Position of pivot (Ratio of angular distance from pivot to pad leading edge/β (See Figure 5B))	.58
Tolerance stackup, pivot & pad thickness	+0 -0.00015 inch
Runner taper	.0022 inch/inch

3. Materials Selection

The selection of a material for the bearings must take into account the problems of starting the hydrodynamic bearings. At start and during low speed operation, a metal-to-metal contact frequently exists due to the lack of hydrodynamic forces. This means that to minimize galling, the bearing materials must be compatible for both dry operation and hot NaK lubricant operation.

The high-temperature molybdenum based steel (M-2) selected for the SNAP-8 pump-motor journal bearings contains sufficient percentages of chromium, molybdenum, tungsten, and vanadium for use to about 900°F. However, oxidation resistance becomes marginal at this temperature. The pads are made of the M-2 steel and the sleeves are made of tungsten T-1 tool steel. The radial bearing ball pivots were made of M-2 tool steel.

The thrust bearing pads, runner, gimbal plate, support ring and ball segments are made of Carboloy 44A. Carbide parts are characterized by a hardened surface and a soft core. The carburized materials may be less susceptible to catastrophic failure, where shock and high vibrational loads are present, because of the softer ductile inner core.

In summary the bearing materials selected are as follows:

Journal bearings:

Sleeves, balls, retainer	Tungsten T-1
Pads	Moly-tung M-2

Thrust bearings:

Pads, runner, gimbal plate	Carboloy C44A
Support ring and ball segments	Carboloy C44A
Back plate	Tungsten T-1

C. PROCUREMENT, INSPECTION, AND ASSEMBLY REQUIREMENTS

1. Journal Bearing Procurement

Journal bearing pads were fabricated from M-2 alloy tool steel with the chemical composition of the material in accordance with the requirements of Table I.

Table I - Journal Pad Composition (M-2)

Element	Minimum (percent)	Maximum (percent)
Carbon	0.80	0.85
Manganese	0.20	0.35
Phosphorus		0.03
Sulphur		0.03
Silicon	0.20	0.40
Chromium	3.90	4.40
Molybdenum	4.75	5.25
Vanadium	1.75	2.05
Nickel		0.10
Cobalt		0.25
Tungsten	6.0	6.75
Copper		0.10
Iron	Balance	Balance

The inspection and procurement requirements for the journal bearing assembly are as indicated in Table II with attributes tested on a sampling basis.

The remainder of the journal bearing assembly, other than the pads, are fabricated from T-1 alloy tool steel. The chemical composition of the material for these parts, which include the sleeves, balls, retainer and housing (ring), are in accordance with the requirements of Table III.

Table II - Journal Pad Inspection Requirements

<table>
<tr><th>Attribute</th><th>Requirement</th><th>Method</th></tr>
<tr><td>Heat Treatment</td><td>$>$ Rockwell C 65</td><td>Preheat to $1440^\circ \pm 10^\circ$F for 20 to 30 minutes. Austenitize at $2200^\circ \pm 10^\circ$F for 8 to 10 minutes, then air quench to room ambient temperature.</td></tr>
<tr><td>Annealed Hardness</td><td>$<$ Brinell 250
$<$ Rockwell B 100</td><td>For forgings and hot rolled stock.
For bar stock (1/2 inch round or less)</td></tr>
<tr><td>Grain Size</td><td></td><td>Less than or equivalent to ASTM #5.</td></tr>
<tr><td>Microstructure</td><td>Hemogeneous</td><td>Constituents evenly distributed and free from excessive segregation detrimental to material condition.</td></tr>
<tr><td>Inclusions</td><td>$<$ 4</td><td>Lengths of 1/32 per specimen</td></tr>
<tr><td>Micro-inclusions</td><td>not to exceed JK Limits</td><td rowspan="4">Per method A of ASTM-E 45. For A, B, C $\leq$ 3 of 1.5 A and B and $\leq$ 5 of 1.0 field. For D not greater than 3 of 1.5 and 2 of 1.0 fields. One field each of 1.0 for A, B, C, or D.</td></tr>
<tr><td>Worst Field</td><td>A   B   C   D</td></tr>
<tr><td>Thin</td><td>1.5   1.5   1.0   1.5</td></tr>
<tr><td>Thick</td><td>1.0   1.0   1.0   1.0</td></tr>
<tr><td>Decarburization</td><td>None</td><td>Surface hardness not be to lower than the interior.</td></tr>
<tr><td>Workmanship</td><td>Visual</td><td>Uniform in quality and condition, clean and free of defects.</td></tr>
<tr><td>Metallography</td><td>100 X magnification</td><td>For microscopic examinations</td></tr>
<tr><td>Macroscopic</td><td></td><td>Per ASTM-A317</td></tr>
<tr><td>Magnetic Particle</td><td></td><td>Per MIL-I-6868</td></tr>
</table>

Table III - Journal Bearing Parts Composition (T-1)

Element	Minimum (percent)	Maximum (percent)
Carbon	0.70	0.76
Manganese	0.20	0.40
Phosphorus		0.03
Sulphur		0.03
Silicon	0.25	0.35
Chromium	3.90	4.25
Molybdenum	0.50	0.90
Vanadium	1.00	1.15
Nickel		0.10
Cobalt		0.25
Tungsten	17.50	18.50
Copper		0.10
Iron	Balance	Balance

The inspection and procurement requirements for the journal bearing parts are as indicated previously in Table II except for heat treating to Rockwell C 64 by:

- Pre-heating to $1650^{\circ}F \pm 10^{\circ}F$ for 20 to 25 minutes.
- Austenitize at $2325^{\circ}F \pm 10^{\circ}F$ for 8 to 10 minutes and air quench to room ambient temperature.

2. Thrust Bearing Procurement

The thrust bearing pads and runner are fabricated from C44A cobalt-bonded cemented tungsten carbide. The back plate and housing are fabricated from T-1 tool steel alloy. The chemical composition of the carbide (C44A) is in accordance with the following requirements:

Element	Percent
Tungsten carbide	93-1/2 to 95%
Cobalt	5% to 6-1/2%

The inspection and procurement requirements for the thrust bearing are as follows:

Attribute	Method
Density	At least 14.9 glem^3 per ASTM-B311-58.
Finish	Sintered condition and free of chips or spalling at the edges or surfaces and no laminations.
Hardness	Using Rockwell hardness tester with the "A" scale check for hardness of at least R_c A90 or determine equivalent reading on a Shore Scleroscope.
Penetrant	Per MIL-L-6866

III. PERFORMANCE TESTS

A. PRELIMINARY BEARING TESTS

Preliminary tests were conducted on the bearing designs to ensure operating stability and load carrying ability. The tests were conducted with a silicone oil with a viscosity comparable to that of NaK at $600^{\circ}F$ (0.65 centistokes).

1. Journal Bearing Tests

The journal bearing tests were performed with a special test fixture. The test bearing was enclosed in a chamber which could be flooded with oil. The bearing loads were applied through attachments on the housing, and the applied forces were measured with spring scales. Five inductance probes were used to measure the position of the bearings with respect to the shaft. The bearing eccentricity ratio, the attitude angle of the bearings, and the oil film thickness could be determined from the probe measurements.

Loads of 6 to 50 pounds were imposed at shaft speeds of 1200 to 8000 rpm. The test results indicated that the bearings performed according to expectations. The bearings had adequate load capacity throughout the test range, the attitude angle was nominally zero, and the minimum film thicknesses were close to the calculated values.

No whirl or instabilities were encountered during these tests. The probe on the trailing edge of one of the pads showed that the pads tracked the motions of the shaft accurately. The flutter data indicated that the amplitudes increased with load and decreased with speed, and the only flutter of any magnitude was 0.0018 inch at a load of 50 pounds and a speed of 3000 rpm. On the basis of this data, it was felt that flutter would not cause difficulties under normal speed conditions.

2. Thrust Bearing Tests

The thrust bearing tests were performed on a fixture which permitted the bearings to be tested at variable speeds and variable loads. Thrust loads were applied to the bearings using a level system and by application of dead weight. An oil circulation system was used, and the oil temperatures were monitored at the bearing inlet, between the bearing pads, and at the bearing outlet. Two of the pads were instrumented to measure the running film thickness. The instrumentation consisted of three inductance type proximity probes, with one on each of the two pads. The instrumentation resolution was set at 0.0002 inch.

The test results indicated that the measured film thickness was close to the calculated film thickness, and was within the resolution of the instrumentation for the maximum load and maximum speed condition. The tests indicated that the bearings carried the maximum applied load without difficulty.

During the tests it was noted that one of the pads tended to stick and would not lift until the shaft speed had increased to over 3000 rpm. Bench tests showed that this apparent "sticking" was due to the extra fine finish on the pads. As a result of these tests, the pads were made with a rougher surface.

3. Fretting and Wear Tests on Pivots in NaK

One of the problems in running tilting-pad bearings over an extended period of time is the possibility of bearing pivot failure. This type of failure is usually caused by fretting and wear, and if serious enough, can impair the operation of the bearings by increasing the bearing clearances. Operation of bearings in NaK tends to promote bare metal contact by removing all surface films, therefore, the probability of failure due to the wear of the pivots is increased. Fretting and wear tests on the bearing pivots provided information for the bearing design regarding this problem.

The test fixture for this test contained six ball and socket combinations immersed in NaK. The ball pivots were loaded by the use of calibrated springs. The desired pivot motion was provided by a reciprocal plate and the motion characteristics were measured to ensure good simulation.

Tests were conducted on C44A Tungsten carbide and the T-1 and M-2 tool steel combinations. The C44A Tungsten carbide was found to be superior to the tool steel, however the test results showed that there would be no significant wear in 100 hours.

B. ENDURANCE AND STARTING TESTS FOR COMPLETE PUMP ASSEMBLY

Pumps were first tested in water and included sets of bearings as described previously. The results showed that the starting torques required were much higher than expected. Since there had been problems with wringing of the thrust bearings during the thrust bearing tests, it was assumed that the same problem existed with the water tests. When the bearings finish was further roughened to 18 to 24 microinches, rms from the original 2 to 4 microinch finish, the starting torque was reduced by about 50 percent.

Endurance testing of complete NaK pumps was conducted in component test loops with NaK as the working fluid, and in SNAP-8 system test facilities also with NaK as the working fluid. A total of 56,367 operating hours was achieved to February 1970 with 3,362 start cycles on 15 NaK pumps. All of this experience was at approximately rated conditions for the complete pump assembly, and the bearings used in these tests were in accordance with the designs described in this report. The most extensive operating times achieved on individual bearing sets within the pumps tested were 10,362 hours and 786 start cycles for the radial bearings and 13,913 hours and 838 starts for the thrust bearings. The following describes the experienced gained during endurance testing as specifically related to bearing operation.

The first complete pump assembly tested in NaK in the component test loop operated for 52 hours with 15 starts. No unusual incidents were noted during this test run, but the sixteenth start was not successful.

After the pump was removed from the test loop and cleaned, it was found that a torque of 150 to 170 inch-pounds was required to turn the rotor. Slow rotation of the rotor by hand indicated that the torque would increase with time; but if the rotation was hand rotated faster (approximately 30 rpm), the torque would stay at about 5 inch-pounds. Upon disassembly, the bearings were found to be in good condition. Conclusions reached at the time were that a combination of oxide contamination and the wringing of the bearings caused the high drag.

Analysis indicated that the individual thrust bearing pads were wedging between the thrust runner and the gimbal sockets when the pads grabbed onto the runner because of wringing or because of the presence of oxides between the pads and the runner. Design modifications were made to minimize the grabbing effect and the wedging. The modification consisted of making the rotating thrust runner into a coned surface to present a line contract with the pads and also pinning the gimbal joints on the pivots. (See Figures 7 and 8).

During the next series of tests, the modified bearings were tested extensively for startup performance. In a series of 100 startups over a running duration of 244 hours, no problems were detected. An endurance test was then started in the component test loop operating the pump at rated conditions and a temperature of 1100°F at the pump inlet.

The motor input torque was measured before, during and after the 3000 hour run for startup evaluation. The startup tests were made with both hot and cold motors and with applied frequencies of 60 Hz. A total of 97 startups were made. The results indicated that a certain amount of wedging of the thrust bearings still occurred. Normal startup torque was 11 to 20 inch-pounds, and this would increase to as high as 80 inch-pounds.

In a series of 60 startups made after the endurance tests, the starting torque results duplicated those values obtained before and during the long testing. The pump was then disassembled for examination. The

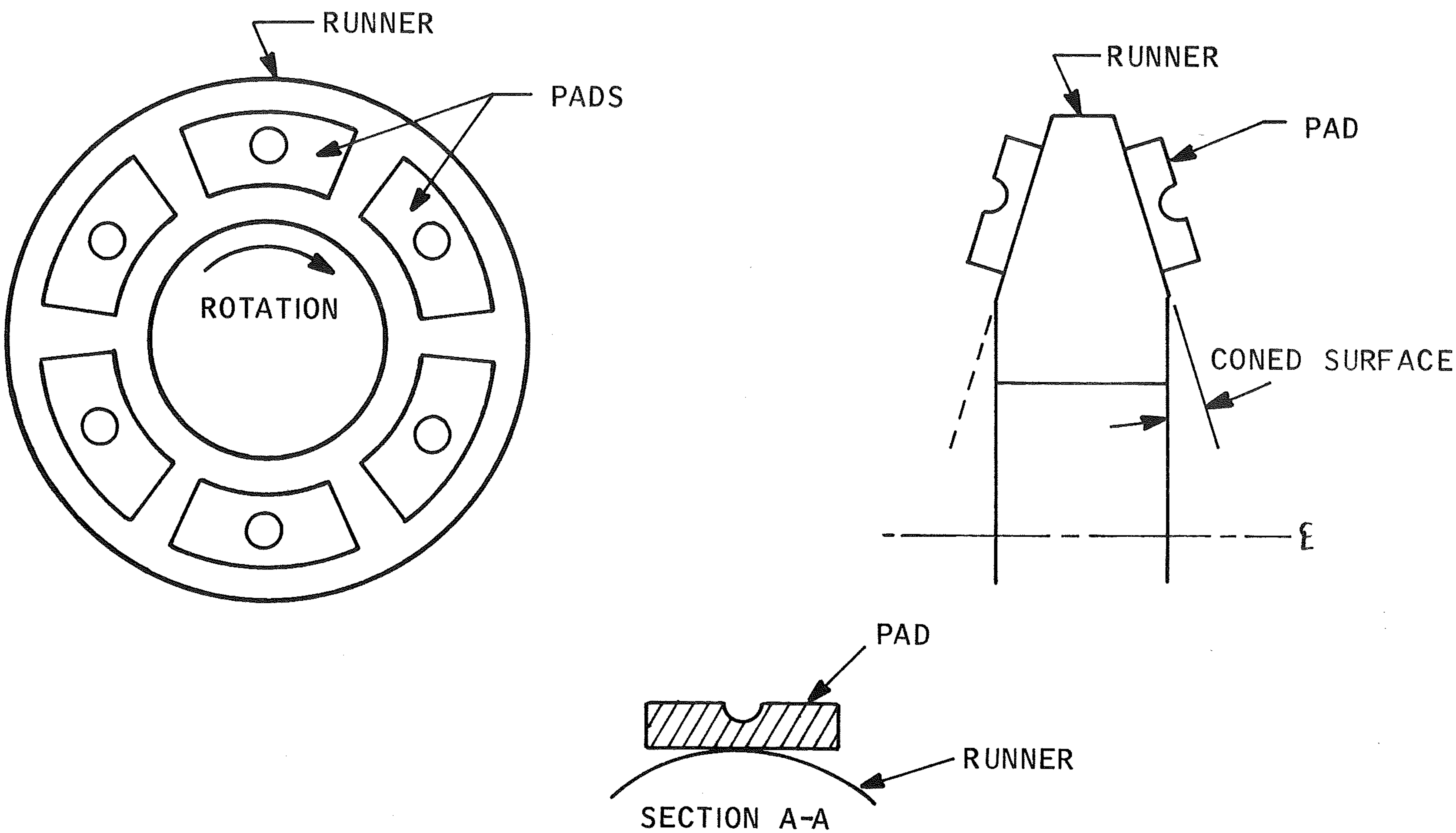


Figure 7. SNAP-8 NaK Pump - Modifications to Thrust Bearing Runner by Coning

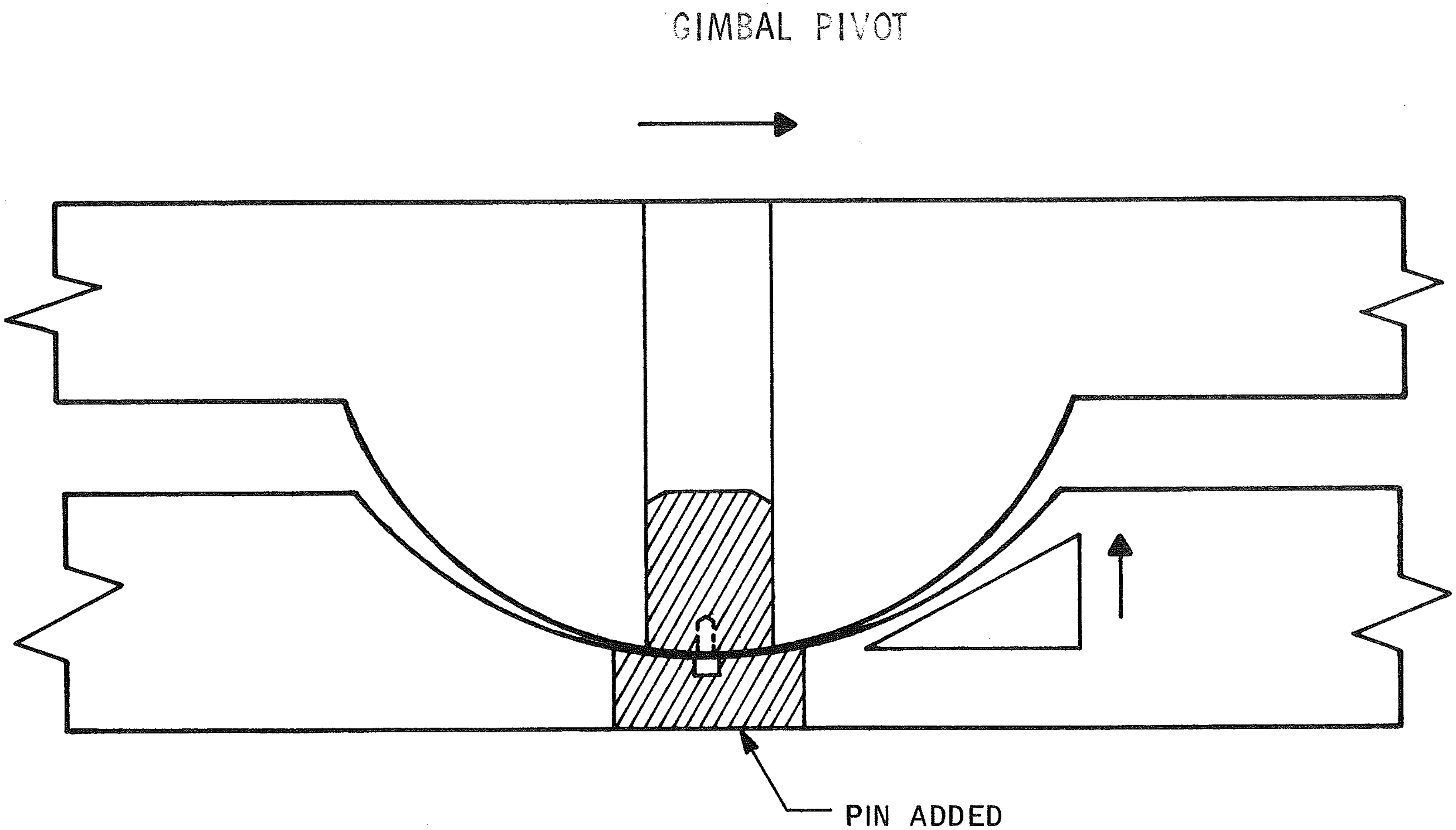


Figure 8. SNAP-8 NaK Pump - Modifications to Thrust Bearing by Pinning Gimbal Pivot

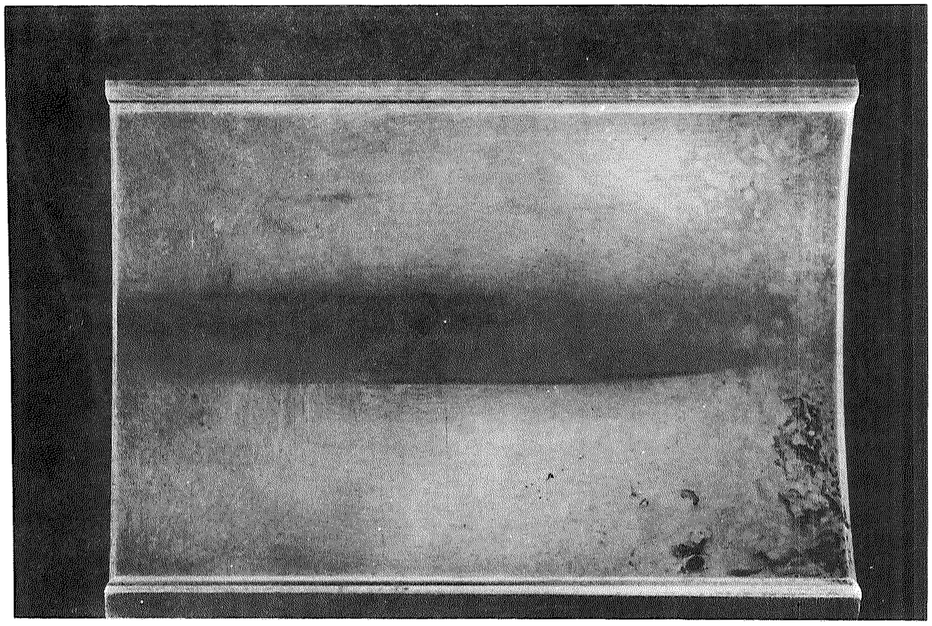
bearings were closely inspected after the tests and the thrust bearing pad was found to be in excellent condition with no evidence of wear. There were blemishes and stains on some of the journal bearing pads. (See Figure 9).

The pump was returned for further operation, however, following some problems with facility operation which it was believed had damaged the pump, the pump was again removed for examination. It was found that the journal bearings were heavily damaged by the passage of oxides or foreign matter between the pads and journals. It was believed that the debris was caused by filling the NaK loop without adequate cleanliness being maintained. The pads and journals of this pump were replaced and the pump returned for further endurance testing in the same facility. No changes to the thrust bearings were made. The unit completed 10,363 hours of operation with 786 starts after which it was decontaminated, disassembled and inspected.

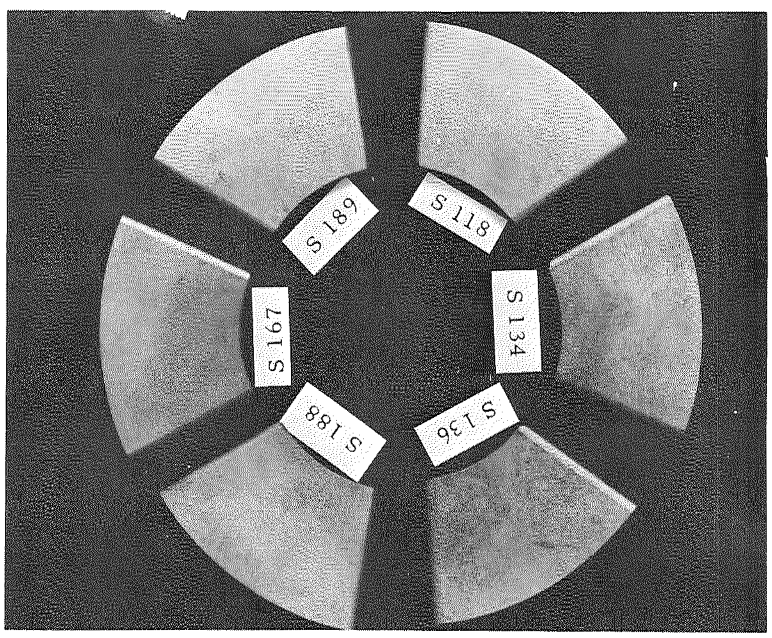
All of the rotating parts and bearing components were dye penetrant inspected and found to be free of cracks. Some surfaces were discolored and some were covered by a thin layer of deposit.

There was no measurable wear or damage to the thrust bearing parts. (See Figure 10). There were some burnish-like marks on the runner and pads apparently caused by passage of some debris through the bearing of lower hardness than that of the bearings. The pivot ball segments and the gimbal pivots showed no wear. (See Figure 10). This thrust bearing had accumulated over 13,913 hours operation and 838 starts.

The journal bearings showed light scratches but were in excellent condition. The pivot, journal and typical pads are shown in Figure 11. Some slight deviations recorded in pad bowing is shown by Table IV, but these are most probably inspection discrepancies and not true distortion.



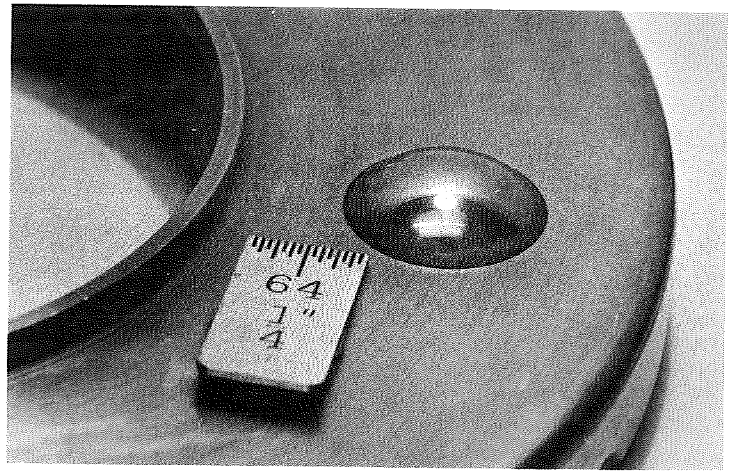
Journal Bearing



Thrust Bearing

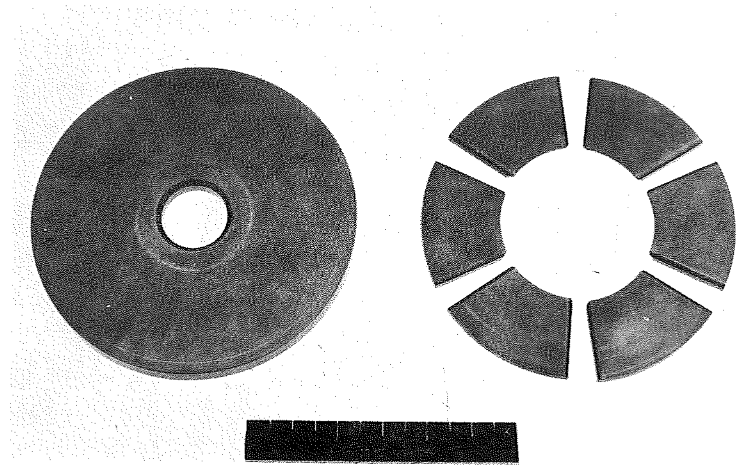
Figure 9. NaK Pump Bearing Pads After 3000 Hours of Testing (97 Starts)

768-496



Ball Pivot

768-510



Thrust Plate and Pads

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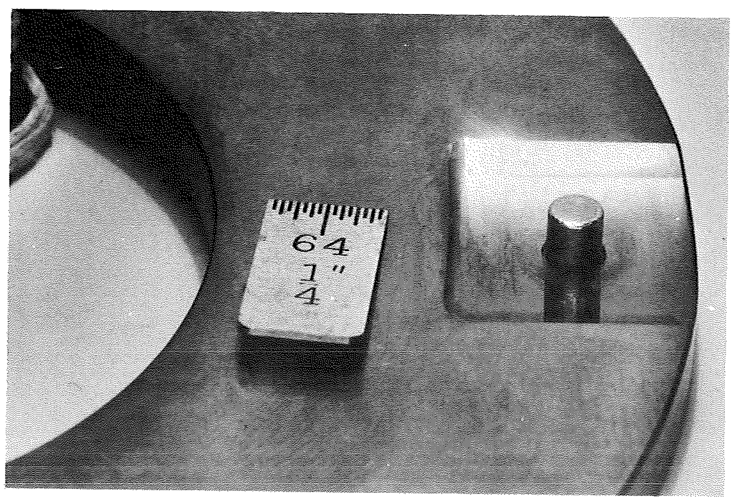

Gimbal Plate Pivot

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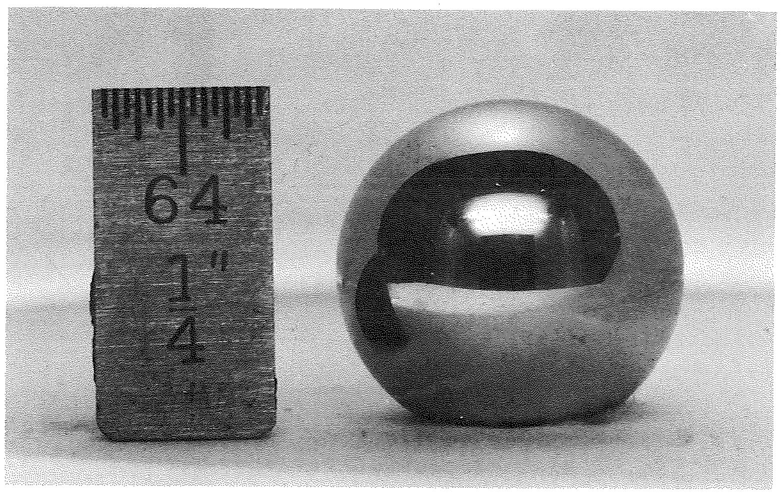
Housing

Figure 10. Condition of NaK Pump Thrust Bearing Components After 13,913 Hours of Testing

768-519



Pivot Ball

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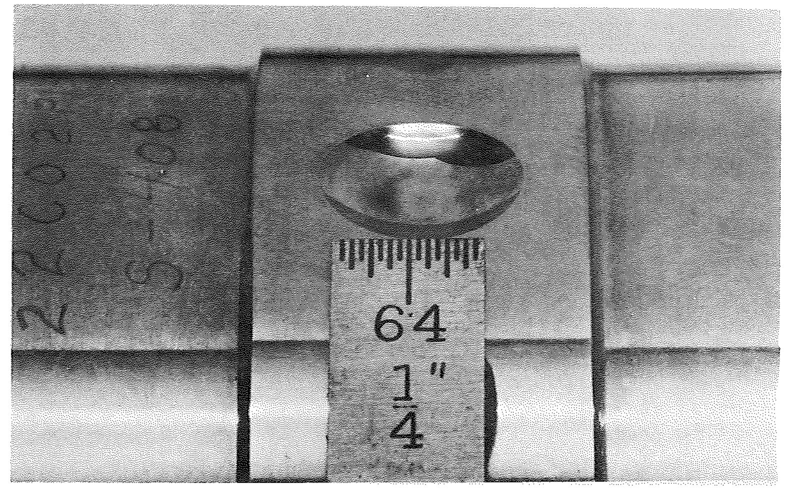

Ball Pivot Socket

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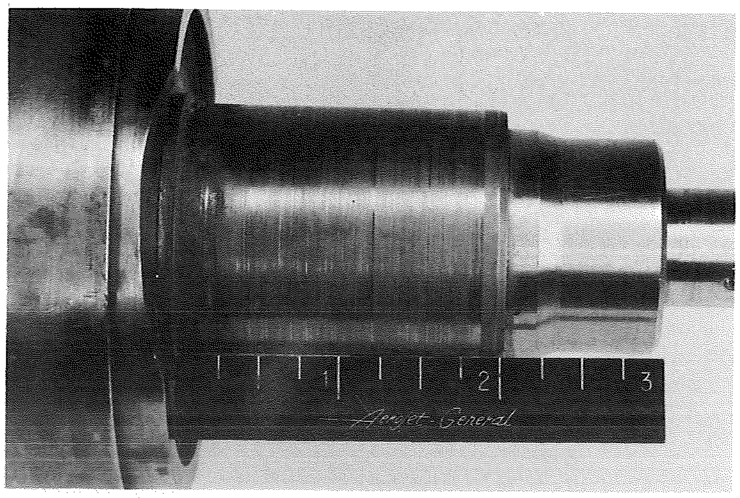

Shaft

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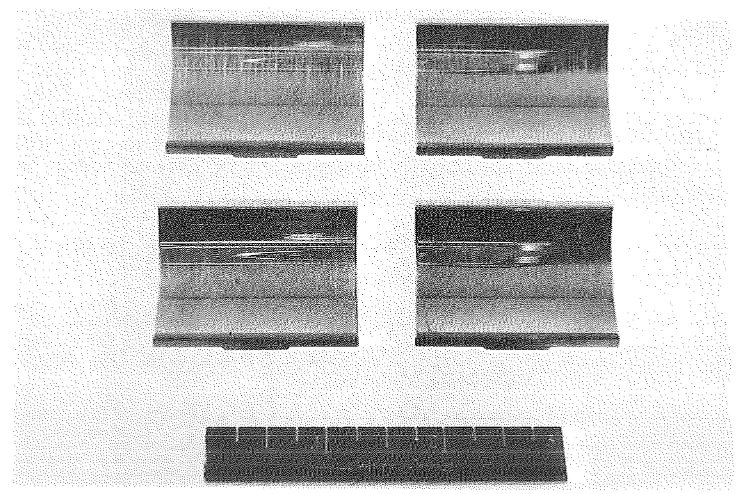


Bearing Pads

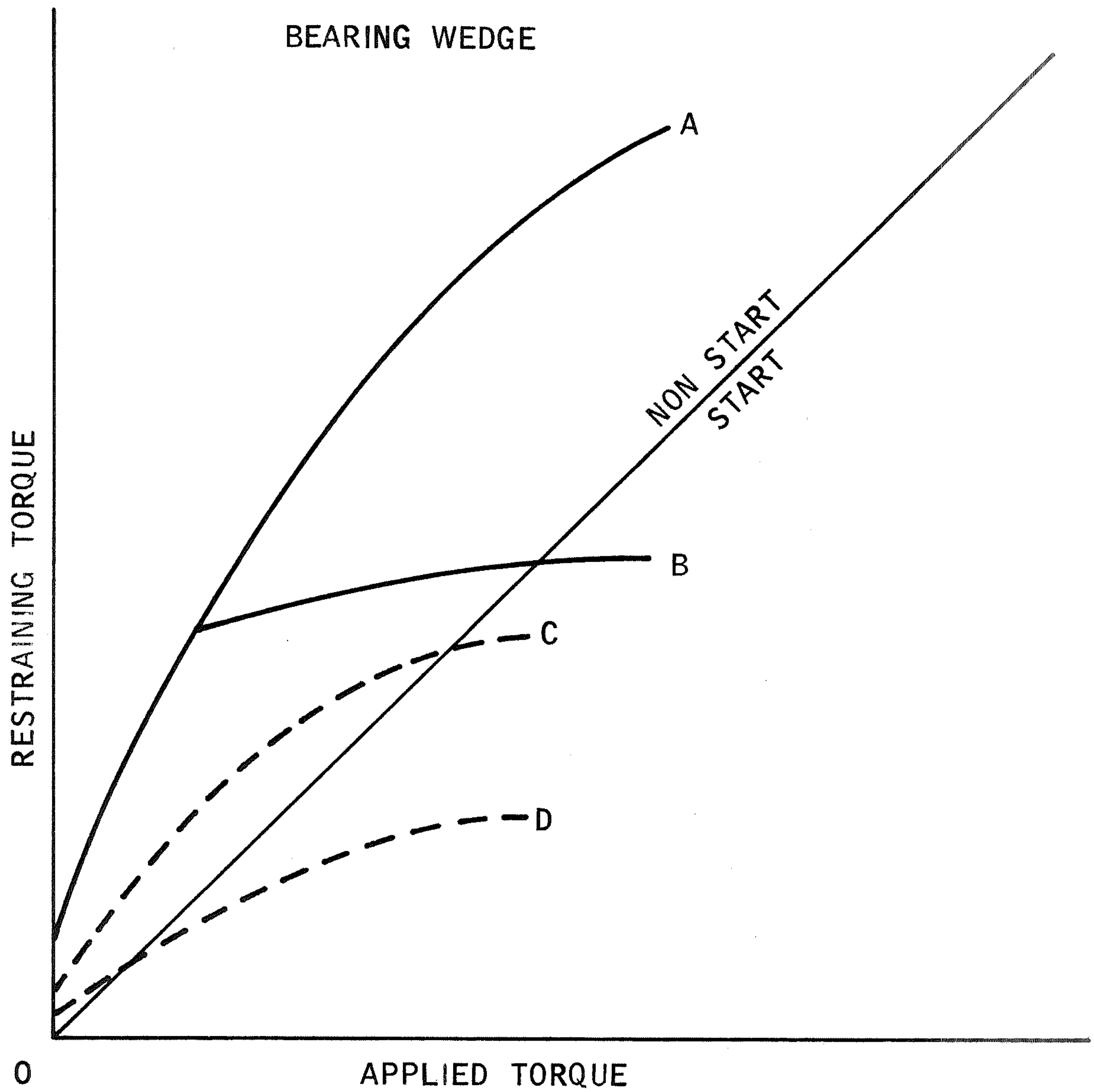
Figure 11. Condition of NaK Pump Journal Bearing Components After 10,363 Hours of Testing

C. EVALUATION OF TEST EXPERIENCE

During initial tests, when starting difficulties first occurred in NaK, the bearings were marked by solid particles and/or metal-to-metal contact. In addition, damage was sustained by pump impeller and housing and the rotor and stator cans. Units, which would not start at one time, became startable after a hot soak. This indicated that solid particles wedged between rotating and stationary parts and/or a substance bonded rotating and stationary parts together and had subsequently dissolved, or clearances had changed (opened up) and allowed rotation.

Figure 12 shows a diagram of starting torque versus restraining torque for the thrust bearings with and without pinned pivots and with journal bearings with high and low coefficients of friction. The thrust bearing in the initial NaK PMA's locked up as shown by line O-A in Figure 12. Adding pins to the gimbal pivots changed the characteristics to those shown in line O-B. The journal bearing has the characteristics of line O-D when the friction coefficient is low, and line O-C when the friction coefficient is high. The intersections of lines O-B and O-C with the nonstart-start line will vary with (1) the configuration of individual parts as fabricated, (2) thermal expansion of parts, and (3) dimensional changes due to wear or surface deposits. The friction coefficient will vary as affected by temperature and by build-up or removal of oxides or other substances from the surfaces. A curve of the average starting torque as a function of test time for the endurance test pump is shown by Figure 13.

There has been no start difficulties in the water tests after the initial surface roughening, but there has been start problems in NaK loops. The water loop of course has no tendency to remove the surface oxides of the base metals of which the bearings are made. Also, the rubbing surfaces of the PMA for water testing were coated with cocoa butter, and the test runs were of a short duration in the water loop.



FOR FRICTION COEFFICIENT HIGH ENOUGH TO CAUSE LOCKING
(INABILITY OF PUMP TO ROTATE)

O-A THRUST BEARING WITHOUT PINS IN GIMBAL PIVOTS
O-B THRUST BEARING WITH PINS IN GIMBAL PIVOTS
O-C JOURNAL BEARING WITH HIGH FRICTION COEFFICIENT
O-D JOURNAL BEARING WITH LOW FRICTION COEFFICIENT

Figure 12. SNAP-8 NaK Pump Bearing Starting Effects

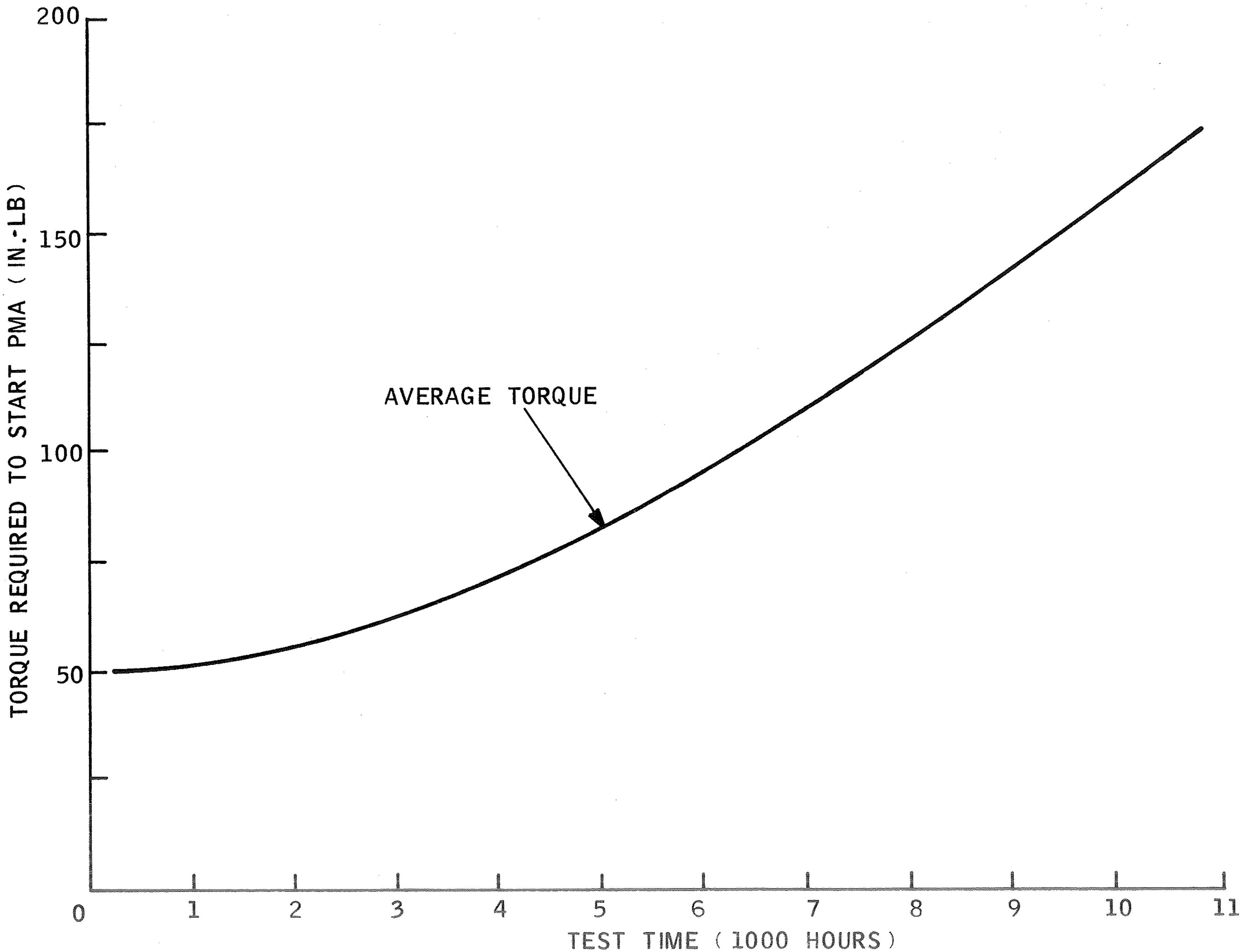


Figure 13. Torque Required to Start NaK as a Function of Test Time

Table IV - Dimensional Stability of Journal Bearing Pads
After 10,363 Hours of Testing and 786 Starts

MOTOR END BEARING	CONDITION	MAX. DIMENSIONAL DEVIATION, INCH
S/N 348	Convex bow in middle	0.00015
S/N 350	Good	-
S/N 351	Bowed and twisted	0.00020
S/N 349	Pad curvature decreased	0.00005
PUMP END BEARING		
S/N 405	Good	-
S/N 403	Good	-
S/N 393	Concave bow in middle	0.00004
S/N 408	Good	-

A review of the factors affecting the ability of the pump to start was made as follows:

1. Wringing

Wringing is used to describe the result when two very smooth and flat items, like measuring blocks, are squeezed together. The result is a tight locking of the parts held together by atmospheric pressure. Wringing has the effect of dragging the pads along a long radius ball contact which made an effective low-angle wedge to totally lock up the NaK pump shaft. It was noted that relapping to a four microinch, rms finish reduced but did not eliminate the wringing effect. Roughening the finish decreased the action fairly well on the journal bearings but on the thrust bearings further rework was necessary. The thrust pads originally had a flat surface mating with the flat surface of the running ring or thrust plate. To provide a line contact the running ring was coned, thus avoiding the flat plate wringing effect. Wringing was then eliminated.

2. Wedging - Gimbal Plates

The original thrust bearing with modifications as above, still tended to lock up the NaK pump shaft during preliminary testing in water. This was determined to be from the action of the relatively long radius cylindrical gimbal plate pivots whose long radius provides a shallow angle wedge with slight rotation of the gimbal plate.

The thrust bearing had to be assembled with a small clearance (0.002 - 0.004 in.) to limit the shaft end play and for pad no-load stability. Consequently, only a small force or drag from the pads could be compounded into a locked shaft. In a horizontal plane, the pads fell to a minimum clearance condition. The effect was eliminated by providing centrally located drive pins which prevent sliding movement in the gimbal pivots.

3. Wedging - Pad Ball Pivots

A wedging action is created when the radial bearing pads are swiveled by the ball operating in the shallow socket in the pad. The resulting wedging action can be severe if the clearance allowance is sufficient or if the ball to pad friction is large. The resultant large loads either lock the shaft or cause damage to the bearings in startup. This is shown in Figure 14.

The thrust bearing material has not shown any sign of damage, but the radial bearings have been galled and scratched in some instances. The clearance may be limited on the thrust bearing by mis-assembly, debris behind ball segments or at pads, gravitational dropping of pads and gimbal plates in the assembly. This situation may also occur in operation with unusually high bearing temperatures as compared to the housing. With the material coefficient of expansions involved, it is more likely that the thrust bearing clearances would increase with temperature effects. On the radial bearings, the clearance is set by manufacturing, checked, and rechecked for debris behind balls or on the pad. Hydrodynamic film, gravitational location of the pads (located at 45° points from center line), the act of galling, and a hot shaft on bearing parts in comparison to the housing, may provide the small clearance condition.

4. NaK Cleaning of Metal Oxides

Metal oxides on bearing surfaces normally provide the hard surface to reduce friction and tendency to gall from metal-to-metal contact. NaK removes this oxide thereby cleaning the metal surfaces. The metal sliding friction factor in NaK has been tested and is approximately 1.0 to 1.5. When a NaK pump is put into service, the first start has always been successful. It is presumed that no system or assembly debris is present and the bearing probably still has an oxide coated surface.

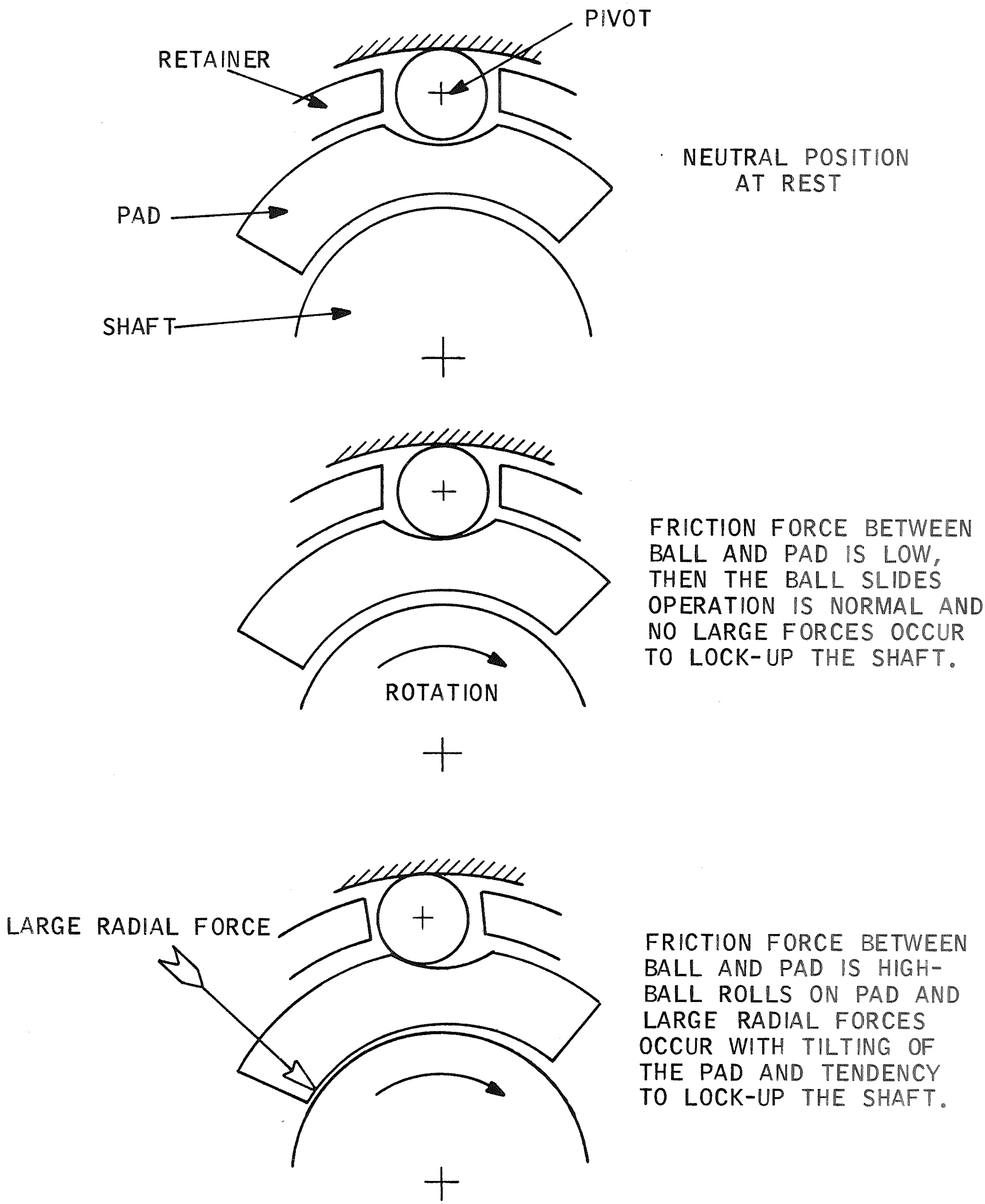


Figure 14. Bearing Pivot Wedging Action

5. Debris

Debris may be either in the form of hard particles or soft materials. The recirculation loop is designed to remove both the relatively soft NaK oxide and soft or hard material particles. The thrust bearings were not scratched in service, although burnishing on the surface shows the passage of some hard debris. The radial bearing will frequently show scratches indicating that a few hard particles may produce more particles from the radial pads to compound the scratching effect. NaK oxide particles are not sufficiently hard to scratch the pad material; but, with the wedging effect, a soft material thickness reduces the bearing clearance for a low wedge angle and then the resistance created in passing the pad surface will, in turn, produce large bearing loads. In startup after some run time, debris may force the pad and journal surface together under load, and will probably create the conditions most likely to cause galling.

6. NaK Oxide

NaK oxide in particle form or in solution presents a problem to the bearings. The particles act as any other debris, though to scratch the radial pads, the hardness must be above 63 Rockwell C. The soft pump impellers readily scratch. As an oxide in solution near the saturation point, the NaK oxide presents its most serious problems. The wringing and wedging effects will be aggravated in the presence of NaK oxide material through loss of clearance.

Some further analysis was made of the wedge angle for the radial bearing operation, and the variation of wedge angle with the clearance of the ball pivot in its retaining pocket. It was determined that the clearances in the retaining pocket are such that the pivot ball can roll from side to side, and the looseness in combination with the socket of the tilting pad can allow the pad to act at wedge angles, which when the ball to pad friction is large tends to apply large loads on the pads. An experimental test was made on a complete pump assembly with a modified radial pivot design. This was accomplished by eliminating the side clearance of the pivot balls in the pocket. Rotor break-away torque values were compared with the same pump assembly and bearings, but assembled in the normal manner, i.e., with nominal pivot ball clearances.

The break-away torques were measured in the forward and reverse rotations, with and without the thrust bearing assembly and in three different attitudes of the pump assembly. They were as follows:

- Shaft horizontal and the 4 journal pads at approximately 45° from the vertical.
- Shaft horizontal with the 4 journal pads on the vertical and horizontal centerline.
- Shaft axis vertical.

In the test with shaft-axis vertical and without the thrust bearing the shaft was supported on a ball. The results of this testing are shown in Table V.

The test results show that with the modified configuration, the rotor torque is slightly higher without the thrust bearing (Condition A) and about the same with the thrust bearing (Condition B) when the torque is suddenly applied. However, under Condition C, with the normal bearing "Case 1", it is seen that the torque for initial rotation is increased greatly with angular displacement through a small angle when the torque is slowly applied. The torque increases until break-away occurs at a much higher value. In the same test, the modified configuration shows similar torque values were obtained under Conditions A and B with no increase with slow angular displacement. This confirms that there is a tendency for the rotor to lock-up with the loose pivot ball design when the rotor is turned slowly.

Table V - Manual Break-Away Torques

Condition	Manual Break-Away Torque (Ounce-Inches)					
	Horizontal				Vertical	
	Normal		Rotated 45°			
	Fwd.	Rev.	Fwd.	Rev.	Fwd.	Rev.
A. WITHOUT THRUST BEARING						
Case 1 - Standard	26	26	26	26	4.5	4.5
Case 2 - Modified	40	40	40	35	7.5	7.5
B. WITH THRUST BEARING						
Case 1 - Standard	60	60	60	60	95	95
Case 2 - Modified	60	65	60	60	93	93
C. WITH THRUST BEARING Slowly pulled into Maximum Torque						
Case 1 - Standard - Min	96	720	Test not conducted		320	840
Max	1680	1500			1160	1200
Case 2 - Modified	60	65	60	60	93	93

IV. CONCLUSIONS

Extensive bearing test experience has been gained by operating the bearings in the pump-motor in the SNAP-8 component test facility and also in the 35 kWe power conversion system. A total of 56,367 hours of test time has been accumulated on 15 pumps. The experience gained has confirmed that hydrodynamic bearings can operate successfully in NaK and that the conventional theory for hydrodynamic bearings operating in NaK is applicable. The importance of cleanliness in successful operation of the NaK pump bearings has been emphasized.

The results of the startup tests were varied. Torques higher than anticipated were required to start the pump on occasions and then in the next startup the torque would be normal. These changes in torque were attributed to the wedging effect of the bearings. The startup torque also showed a marked tendency to increase as a function of test time. Initially, the average torque required to start the pump was about 50 inch-pounds. Toward the end of the test, the torque had increased to over 150 inch-pounds. (See Figure 12). There are two possible explanations for this increase. One is that the bearing surfaces had been supercleaned by the action of the NaK lubricant resulting in an increase in coefficient of friction with continued testing. The other explanation is that the test system had become progressively contaminated with oxide and debris causing the bearings to lock up during startup. The scratches on the bearing pads after the test indicate that the most likely cause was contamination of the NaK in the bearings.

The radial bearings were in good condition following the endurance test for 10,363 hours and 786 starts with only minor scratches to the pad and journal. The thrust bearing parts were unblemished except for a few burnish marks on the pad. The thrust runner and gimbal pivots showed no sign of wear or fatigue. The scratches and burnish marks did not affect the steady-state performance of the bearings.

A check of the dimensional stability of the journal bearing pads was made against the dimensions on the drawings. It was concluded that the pads which showed small dimensional discrepancies may have been manufactured that way originally when less accurate inspection methods were employed. The examination of the geometry of the radial bearings with resultant wedge angles being dependent on clearances appears to give a pointer as to probabilities for totally eliminating the lock-up of the rotor due to jamming of the radial bearings. This was confirmed to some extent by the comparitive tests where a pump was torque tested with the modified pivot design and then with the identical pump without the modified pivot design. A sketch showing the detailed modification to the pivot as recommended is shown in Figure 15. It should be noted that the only testing carried out with this design change was in water and extensive testing in hot NaK would be necessary before it could be assumed that the design change had totally alleviated the tendency for the radial bearings to lock-up.

An obvious conclusion also to be made from the experience gained in this development is that the radial bearings would be much less prone to damage by debris particles if they were made of a harder material. Since the NaK pump tungsten carbide thrust bearings presumably digested the same debris as the tool steel radial bearings with no evidence of surface scratching, the recommendation is that the radial bearing journal sleeve be changed to a flame plated tungsten carbide surface, and that the radial pads be fabricated from tungsten carbide.

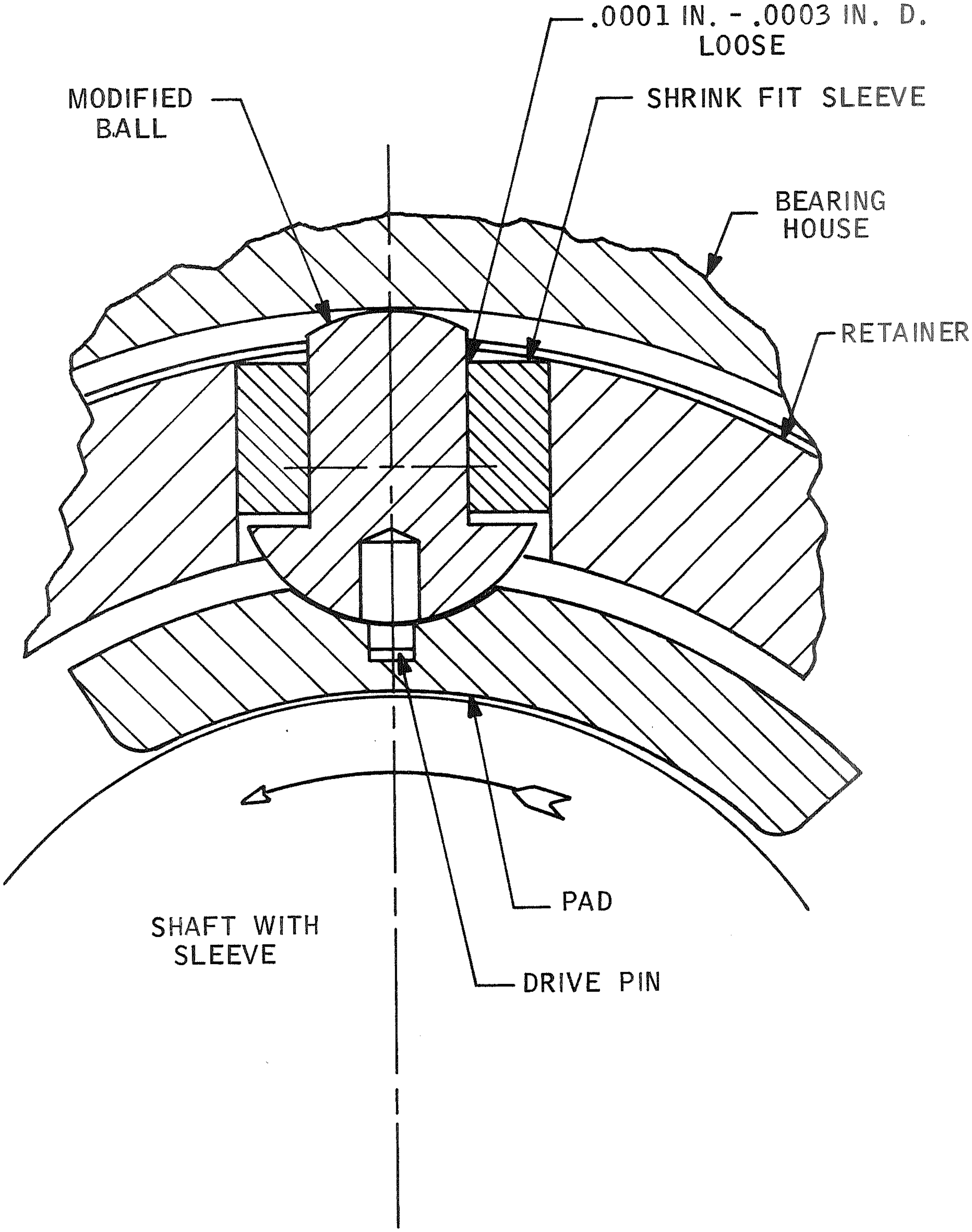


Figure 15. Modified Pivot Detail for SNAP-8 Pump Journal Bearings

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