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BALL BEARING DESIGN FOR THE SNAP-8 TURBOALTERNATOR AND MERCURY PUMP

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Azusa, California

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NASA Lewis Research Center
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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 at Aerojet-General Corporation, Azusa, California, as part of the SNAP-8 Electrical Generating System contract being conducted within the Power Systems Operations. The contributors to the technology employed in this application are too numerous to mention individually. Information was provided by Suppliers, Consultants, specialists of the National Aeronautics and Space Administration and the General Electric Company. The work was performed under NASA Contract NAS 5-417 with Martin J. Saari as program manager for NASA and Dr. W. F. Banks as program manager for Aerojet-General Corporation.

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ABSTRACT

This report covers the design and application of angular-contact, tool-steel ball bearings used in the turbine-alternator and the mercury pump of the SNAP-8 electrical generating system. These SNAP-8 rotating machines run on ball bearings with a design objective of 99.5% reliability for 10,000 hours of continuous operation. A summary of ball bearing operating experience through September 1970 is presented in this report. The bearing design selected for operation in the SNAP-8 system has exceeded the original life objectives, and has operated in excess of 23,000 hours. From the experience gained to date it is anticipated that bearing life of 5 years or greater is feasible.

SUMMARY

To stay within the "state-of-the-art", oil lubricated conventional ball bearings were selected to support the rotating parts of the mercury pump and turbine-alternator for the SNAP-8 Rankine-cycle nuclear space power system. The bearing design goal was to obtain a minimum life at rated conditions, of 10,000 hours with no maintenance and a 99.5 percent reliability.

Double spring preloaded, angular-contact ball bearings are used with one nonthrust low shoulder in the inner ring and with a one-piece outer ring piloted ball separator. The bearings are jet lubricated with a polyphenyl ether (4P3E) which was selected for its radiation resistance. Angular-contact bearings were selected for their high-load capacity and good radial stiffness. With axial preload, no internal looseness occurs under operating conditions and full rolling contact and optimum dynamic balance are maintained at all times. In the turbine-alternator and mercury pump designs the major emphasis was made to extend bearing life. The reliability and long-life requirements of the bearing dictated the use of vacuum melted materials. The bearings are made from triple consumable electrode vacuum melt (CEVM) M-50 tool steel with a specified hardness of 64 to 66 Rockwell C utilizing selectively matched balls and rings. Precise fatigue life calculations for the bearing design were not possible due to insufficient data for this material at these lightly-loaded conditions; however, calculations with conservative assumptions indicated satisfactory fatigue life.

The lubrication requirements for the mercury pump were experimentally determined. When tested under system operating conditions, a minimum of 60 lb/hr of lubricant was required to maintain bearing temperatures below design requirements. The maximum satisfactory bearing-lubricant flow was 200 lb/hr. Flows greater than this tended to flood the bearings, increasing pump power consumption by 0.25 horsepower.

The lubricant design flow for each turbine-alternator bearing and slinger arrangement is 200 lb/hr. Tests demonstrated that the bearing lubrication system adequately removed heat from the alternator rotor and heat generated

by seals and bearings. Bearing temperature rise above the incoming oil temperature was within acceptable limits.

Endurance tests of all the SNAP-8 bearings have exceeded original life objectives of 10,000 hours. One turbine-alternator bearing set has been run for 12,800 hours, a mercury pump bearing set for 12,455 hours. In a test of an alternator at rated load in an electrical test facility, a total of 23,130 hours was accomplished.

Examination of the bearings from these components show satisfactory condition with no indications of life limiting modes. The lubrication system was adequate and elastohydrodynamic film conditions exist within the bearings. Consequently, bearing life of at least five years (approximately 43,000 hr) is anticipated.

I. INTRODUCTION

SNAP-8 is a 35 to 90-kw nuclear-electric power conversion system for use in a space environment. The system operates on a mercury Rankine-cycle using a eutectic sodium-potassium mixture (NaK) as the heat-input and heat-rejection heat transfer fluid. 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 Atomic Energy Commission by Atomics International.

The Rankine cycle power conversion system (Figure 1) 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 which develops the electrical output of the system. The saturated mercury vapor leaving the turbine passes through a condenser from which the liquid mercury is pumped to the boiler to complete its cycle. Condenser cooling is provided by another pump-driven NaK loop which couples the condenser and space radiator. Polyphenyl ether, used as a lubricant-coolant for the system, is circulated in a fourth loop. This fluid lubricates the bearings in the turbine-alternator and the mercury pump.

The turbine-alternator was developed to meet the SNAP-8 power generating system requirements which includes the ability to start up and generate electrical power unattended in a space environment for 10,000 hours of continuous operation. This assembly is comprised of a turbine mounted overhung on ball bearings, the alternator rotor being straddle mounted on separate ball bearings with a flexible drive shaft coupling the turbine and alternator.

The mercury pump is an integrated assembly with drive motor and pump impeller mounted on the same shaft.

This report covers the design, development, and testing of the ball bearings used in the SNAP-8 turbine-alternator and mercury pump for the period between 1963 and August 1970. Aerojet-General Corporation, Azusa, designed and built the turbine and mercury pump. General Electric Company, Erie, Pennsylvania, designed and built the alternator. Ball bearing life was a

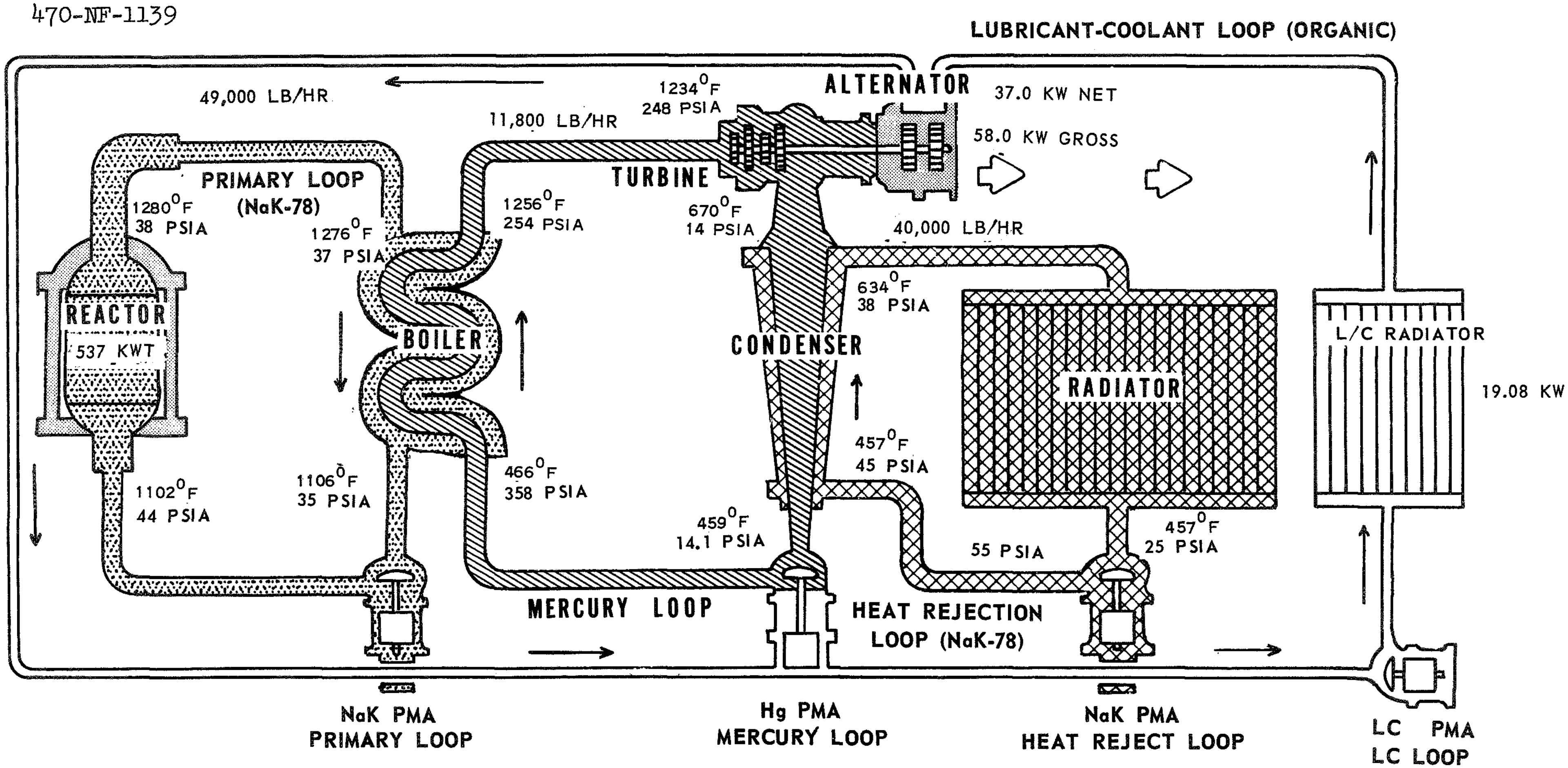

Figure 1. SNAP-8 System Schematic

major objective in designing the turbine-alternator and mercury pump. The designs incorporate features to minimize bearing loading thus extending bearing life. Selection of the system lubricant-coolant fluid was based on elevated temperature stability in a nuclear radiation environment.

II. BEARING DESIGN AND DEVELOPMENT

A. MERCURY LOOP ROTATING COMPONENTS

Of the five components in the SNAP-8 power conversion system that contain rotating parts; two - the turbine-alternator and the mercury pump - are in the Rankine-cycle mercury loop. Design and development of the bearings for these two components is the subject of this report. The following are brief descriptions of the turbine-alternator and mercury pump.

1. Turbine-Alternator

The turbine-alternator includes two major subassemblies, the turbine and the alternator, assembled and mounted as a single unit as shown in Figure 2. The turbine is mounted to the alternator which is supported at the combined machine center of gravity on trunnions which allows motion for thermal changes. A splined quill drive shaft connects the rotors of the turbine and alternator. Use of this drive shaft was specified to minimize the spline loading and reacting forces on the preloaded angular-contact ball bearings.

The turbine-alternator lubricant-coolant system consists of two main branches in parallel: one to carry bearing lubricant, and the other to circulate coolant to the space seal and the alternator stator. The lubricant branch is protected with a filter and valved so that it can be isolated during startup to prevent flooding of cavities.

The turbine is a four-stage, axial flow, impulse type mounted overhung on a ball-bearing-supported shaft rotating at 12,000 rpm. The mercury-vapor working fluid is separated from the lubricant by a system of seals and a cavity vented to space. Loss of mercury and lubricant to the space vacuum is minimized by screw type seals and molecular pumps. (See Figure 3)

The alternator (Figure 4) was designed to produce 60 kw output at 0.75 power factor (80 kva). It is a solid rotor, brushless, radial air gap, homopolar inductor unit that produces 120/208-volt, 400 Hz, three-phase power. The rotor is straddle mounted on two spring preloaded angular-contact ball bearings. Lubricant is supplied to each bearing by four jets.

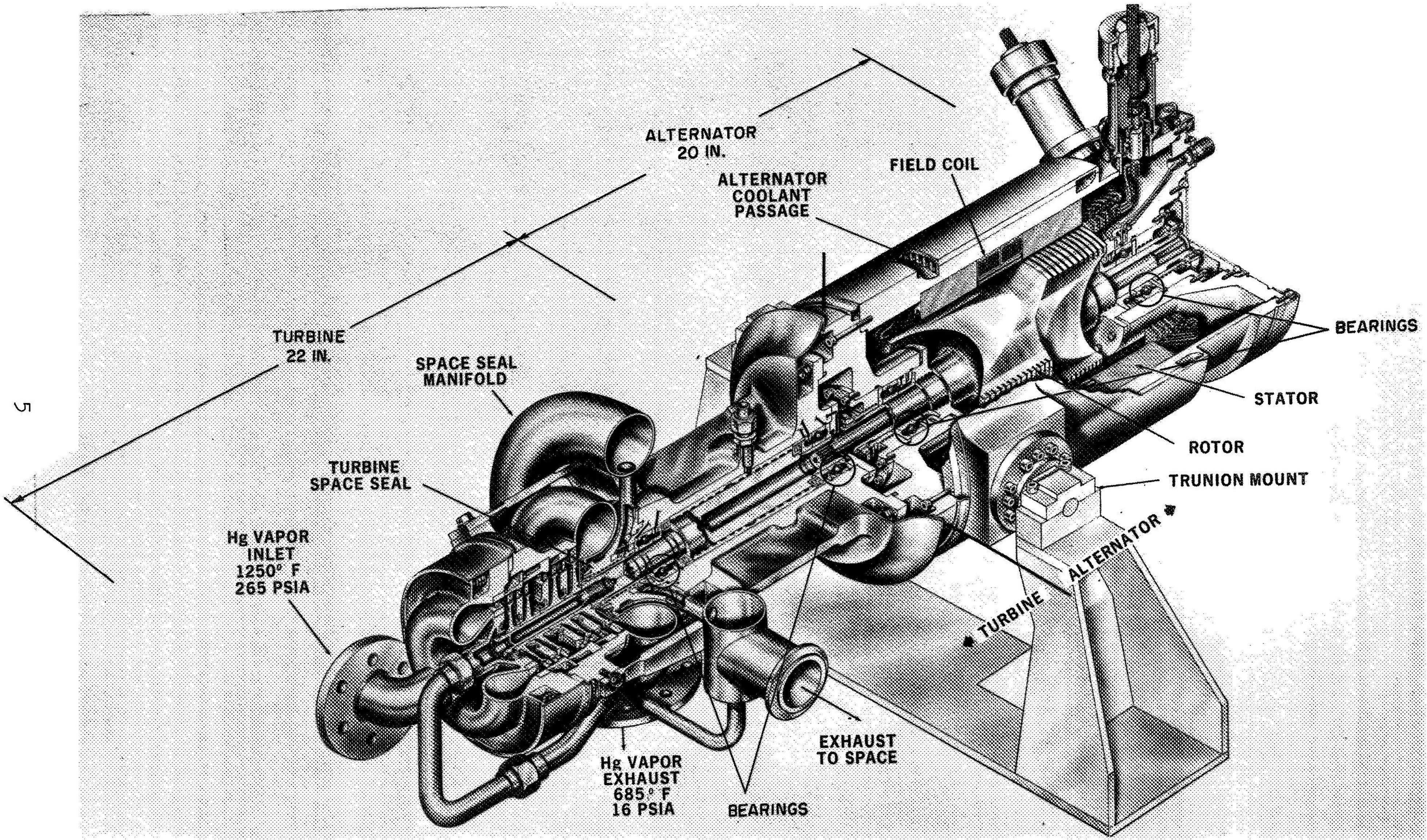

Figure 2. SNAP-8 Turbine-Alternator

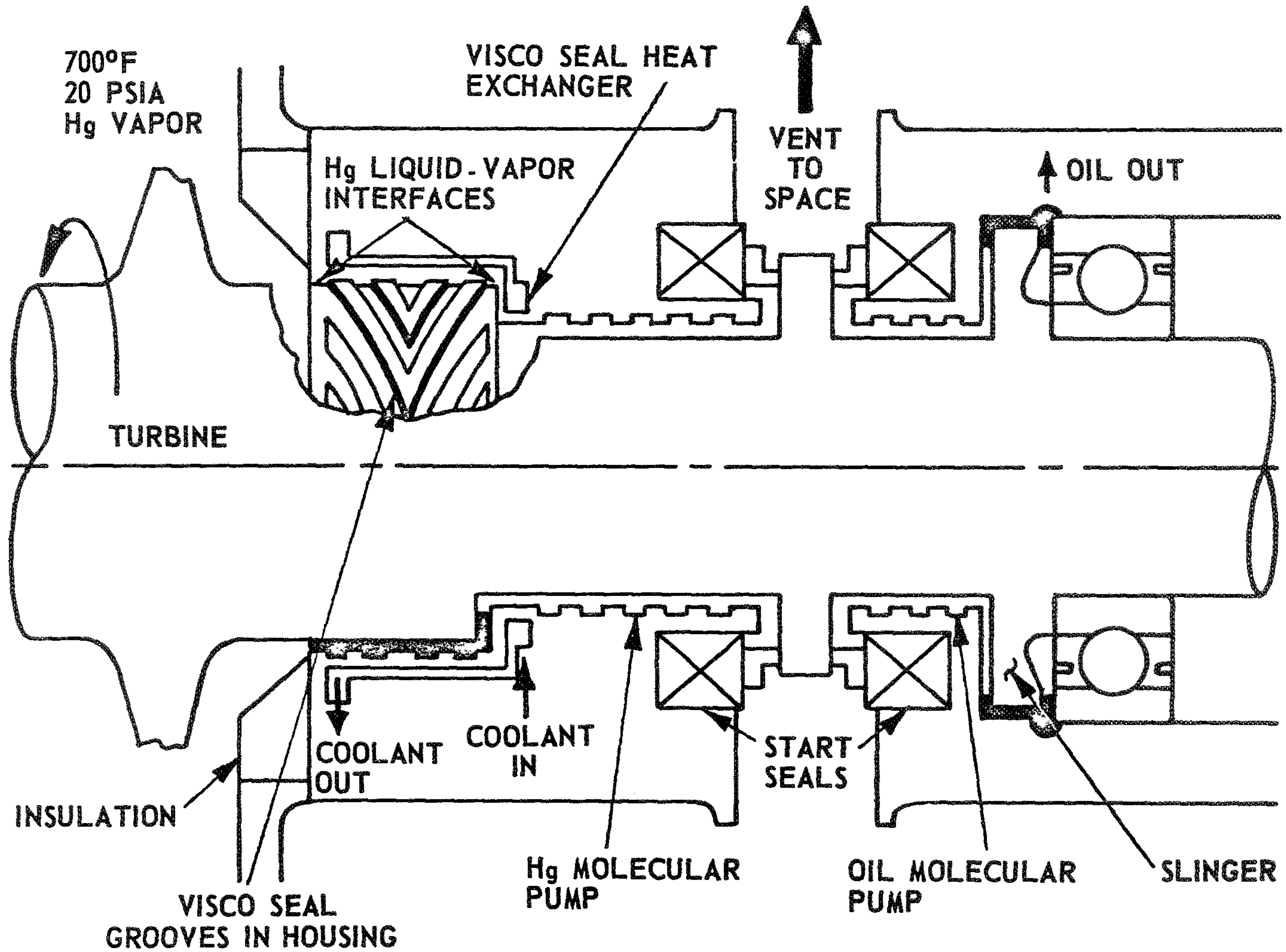

Figure 3. Turbine Dynamic Seal System

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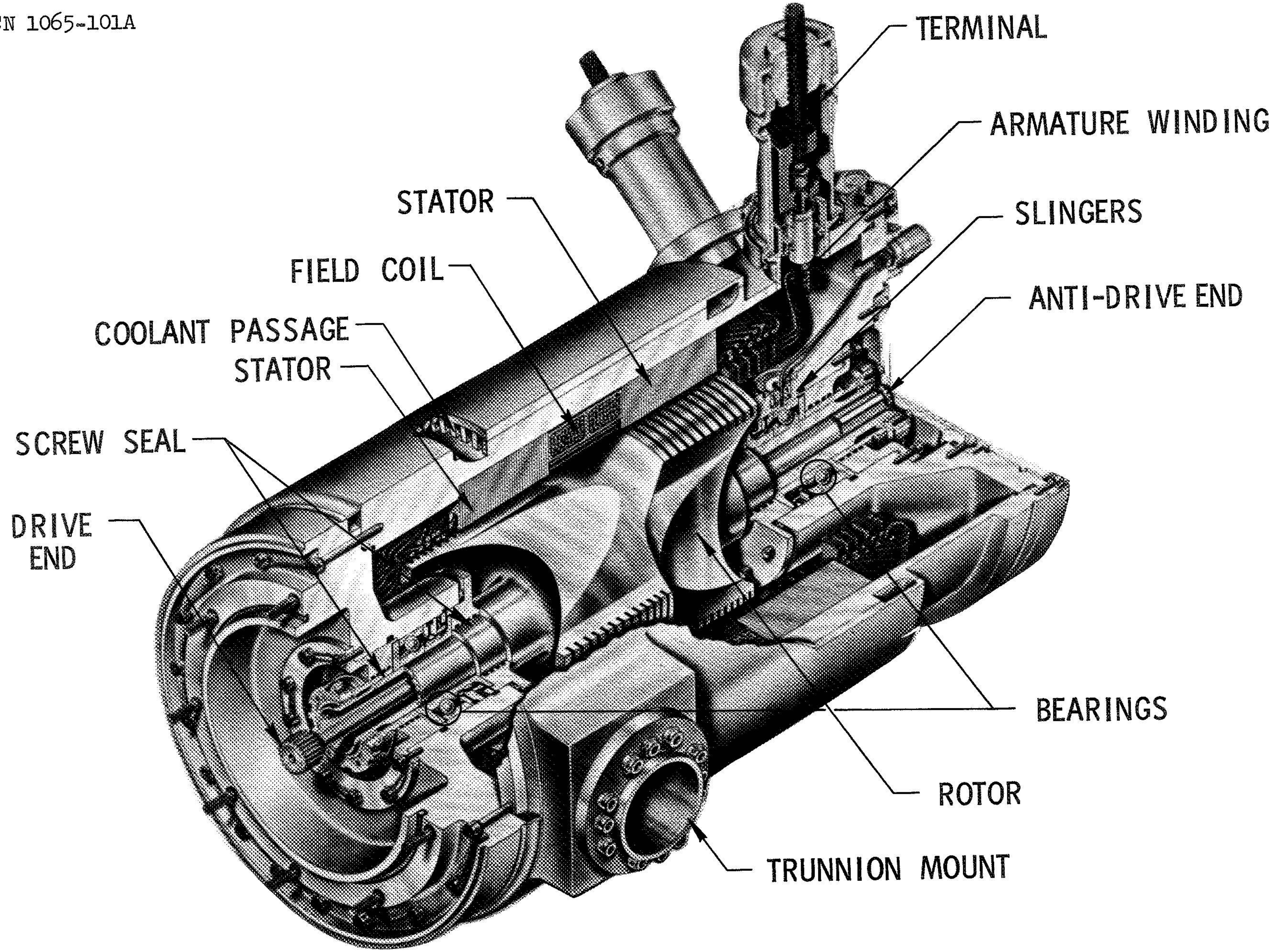


Figure 4. SNAP-8 Alternator

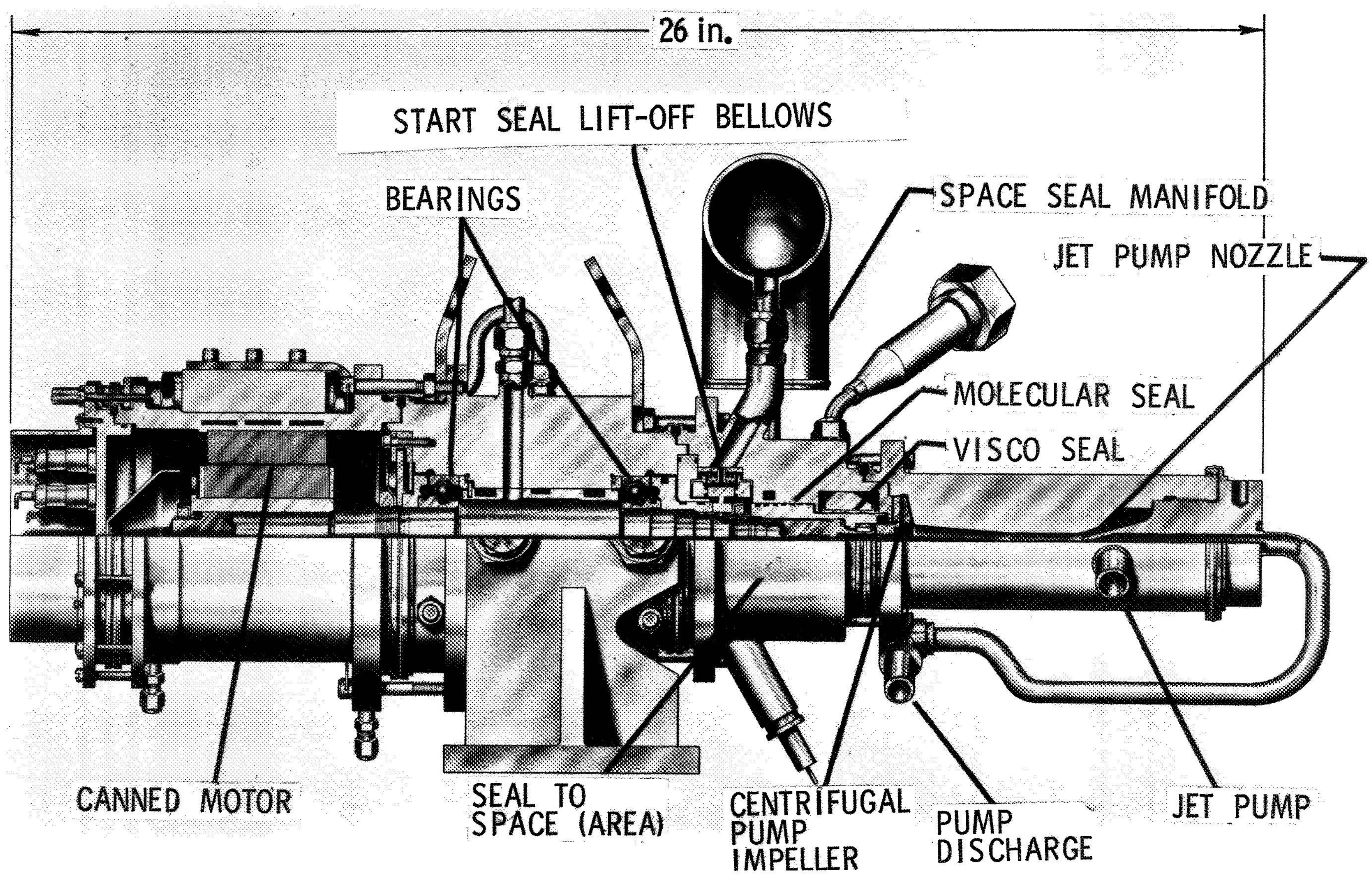


Figure 5. SNAP-8 Mercury Pump

Slingers on each side of both bearings remove the lubricant from the unit to prevent cavity flooding. A dynamic screw seal system on the shaft directs lubricant vapors to the slinger liquid-vapor interface, and prevents leakage to the rotor cavity. See Reference 1 for a detailed description of this alternator.

2. Mercury Pump

The mercury pump operates at 7800 rpm with 400 Hz, 120/208-volt, three-phase input power. The unit is an integrated assembly with the drive motor and pump impeller mounted on the same shaft (See Figure 5). Passages in the motor housing circulate the system lubricant-coolant fluid to cool the motor stators. This fluid is isolated from the pump working fluid (mercury) by a series of static and dynamic seals vented to space, similar to those in the turbine-alternator as shown in Figure 3.

The pump shaft is supported on preloaded angular-contact ball bearings which are jet lubricated with the system organic fluid. Slingers are employed to return the lubricant to the lube-coolant pump. In addition, these slingers prevent liquid flooding of the motor rotor cavity and lubricant leakage to space.

A combined jet and centrifugal pump is included, the centrifugal impeller being a semi-open vane type. Impeller back vanes are used to improve the assembly thrust balance reducing bearing loading and to reduce the pressure at the dynamic screw seals.

The three-phase, 208 volt, 400 Hz, 7800 rpm squirrel-caged induction motor operates in an environment of lubricating fluid vapor. A cooling jacket incorporated into the motor housing cools the motor stator. See Reference 2 for a detailed description of this mercury pump.

B. BEARING DESIGN AND APPLICATION

The design philosophy in SNAP-8 ball bearing development was to take advantage of the most sophisticated technology within the state-of-the-art. This section of the report discusses the particular SNAP-8 operating requirements and conditions for ball bearings and explains the criteria used in bearing design.

1. Environmental Conditions

The environmental conditions specified for the SNAP-8 system are given in Table I.

Table I. Environmental Conditions, SNAP-8 System

Environment	Storage and Transportation	Launch (Non Operating)	Space Operation
Temperature	-40°F to +160°F	-40°F to +160°F	200°F to 400°F
Life	2-yr minimum	Transient	$>$10,000-hr*
Atmospheric Pressure	Sea level to 50,000 ft	Sea level	Zero
Shock	**	15 g's in each direction along 3 mutually perpendicular axes (18 total shocks)	**
Random Vibration	**	20 to 100 Hz at 3 dB per octave increase 100 to 600 Hz at $.4g^2/H2$ 600 to 2000 Hz at 6 dB per octave decrease for 3 minutes in ea of three mutually perpendicular axis	**
Nuclear radiation - Capable of withstanding total integrated radiation: a. fast neutrons 5×10^{12} nvt, total in 10,000 hr. b. gammas 5×10^{7} rads c. total in 10,000 hr.			
Note: * This life goal has been exceeded by the ball bearings. An increased goal of 5 years is now the objective for the SNAP-8 system. ** Applicable to total system only.			

2. System Loads

The designs resulted in the bearing loads and operating conditions shown in Table II.

Table II. Bearing Design Loads and Operating Conditions

Operating Conditions	Turbine	Alternator	Mercury Pump
Speed, rpm	12,000	12,000	7,800
Mechanical unbalance load, lb	2.50	1.43	7.5
Magnetic unbalance load, lb	0	70 lb 1/2 sine wave max.	30
Spline thrust load, lb	16	16	None
Rotor weight, lb	20	70	15
Hydrostatic axial unbalance, lb	0 ± 20	0	63 ± 30
Spring preload, lb	60	60	55
Spline radial load, lb	Negligible	Negligible	None
Design axial load, lb	100	76	110
Design radial load, lb	50	70 lb peak sine wave	45
Design equivalent load, lb	159	154	165

3. <u>Bearing Load Analysis</u>

a. Unbalance Loads

The balancing tolerances used for SNAP-8 rotating machinery result in relatively insignificant loads at operating speeds. The operating speeds are well below the first critical speed (first natural frequency of the rotor) and the rotors are sufficiently rigid to minimize deflections due to the residual imbalance loads. However, due to construction features all the SNAP-8 rotor assemblies must be disassembled after balancing and reassembled in the housings. This introduces a possible shift from the measured balance. Careful assembly checks are utilized to minimize the final assembly unbalance.

Extended operation of these units may produce balance changes due to nonsymmetrical creep, mass transfer deposits and erosion damage. Provision for these changes was not made when calculating the bearing loads, however, the normal post operation inspections have indicated no significant changes in loads due to these factors that would effect bearing life.

b. Magnetic Forces

Magnetic loads result from the variations in alternator or motor radial air gap and affect the magnetic reluctance and flux density. A large radial air gap results in decreased radial bearing forces caused by eccentricity of the rotor.

The magnetic force at each end of the alternator varies with eccentricity. When the rotor is parallel but eccentric, the force is unidirectional relative to the stator. The forces on the opposite shaft ends are out of phase with each other. If the axis of the rotor is misaligned with respect to the stator such that the points of minimum eccentricity are 90 degrees apart, the forces are axial but in-phase.

In the mercury pump rotor, eccentricities will also produce magnetic forces proportional to the eccentricity. The arrangement of poles in the same plane and circumferentially distributed about the rotor permits simplification in assuming a single unidirectional force.

c. Contact Angle

Angular contact bearings which carry both radial and axial loads normally operate with a nonradial line of contact between rings and balls. Figures 6 and 7 show the relationship of contact angle to axial load. The basic contact angle of a bearing is determined by geometry including radial play and race curvature relative to ball size. The SNAP-8 bearings nominally have a 16^{o} contact angle which may vary from 14^{o} - 18^{o} by manufacturing tolerances. At operating conditions the inner ring is normally hotter than the outer ring and the temperature difference results in a reduction of radial play tending to reduce the contact angle.

The selection of design contact is made by balancing the requirements for radial and axial load capacity. Initial larger angles will support larger axial loads at the same stress level. Too large a contact angle for the operating speed produces noncircular ball paths and spinning of the balls resulting in excessive wear. Increasing the contact angle results in stiffening the bearing in the axial direction and softening the spring rate in the radial direction thus tending to reduce the critical speed of the rotor system. (See Reference 3).

d. Centrifugal Loads

When the bearing balls rotate around the inner ring, a centrifugal force is applied by each ball on the outer ring. This amounts to 12 pounds per ball for the turbine-alternator bearing and 3.6 pounds per ball for the mercury pump bearing, producing a force which causes a change in the angle of contact between the balls and the rings. The contact angle with respect to the outer ring decreases and with respect to inner ring increases as speed is increased. This produces a greater load on the outer ring and larger stresses in the ball and outer ring.

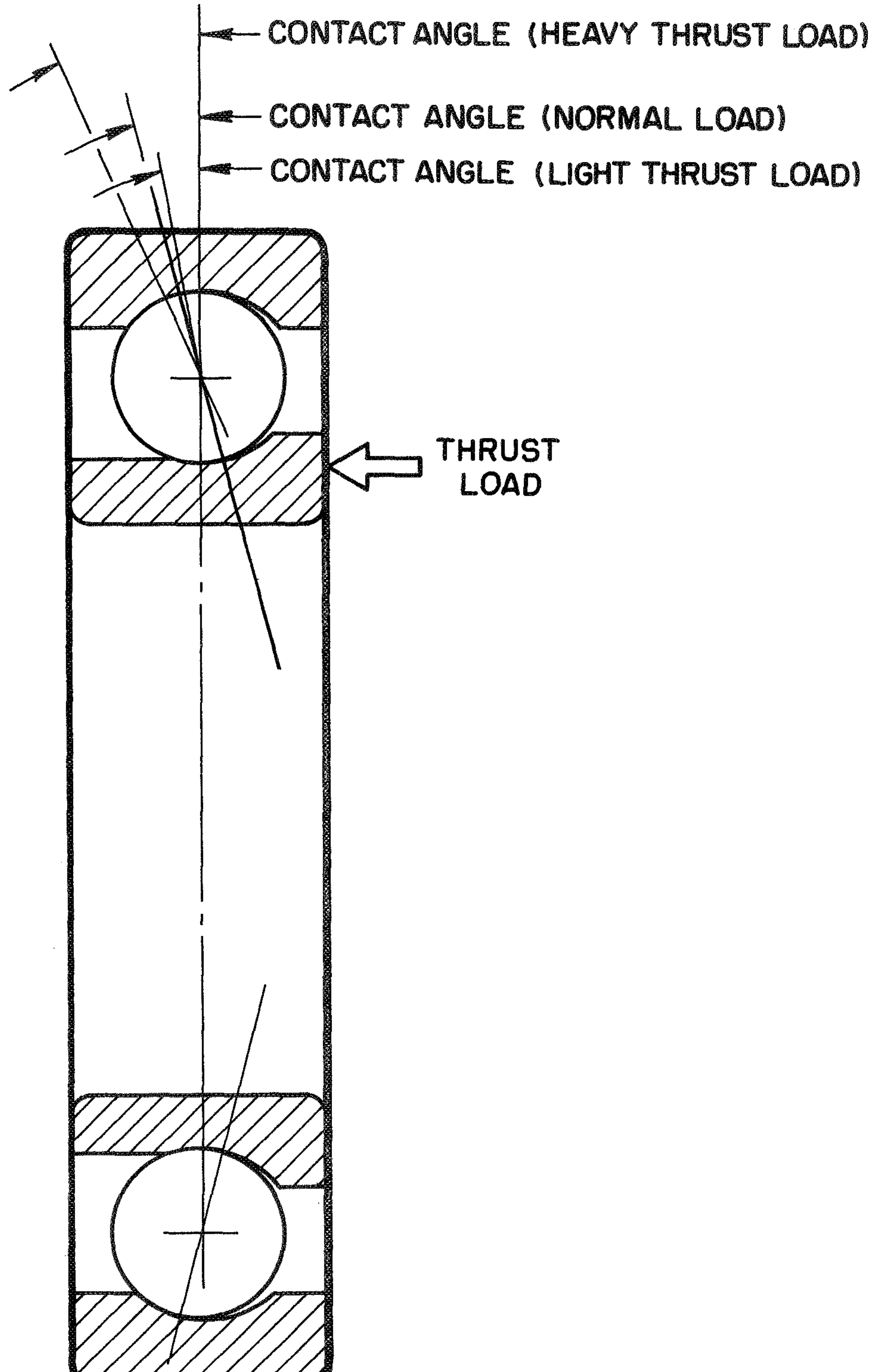


Figure 6. Effect of Load on Contact Angle

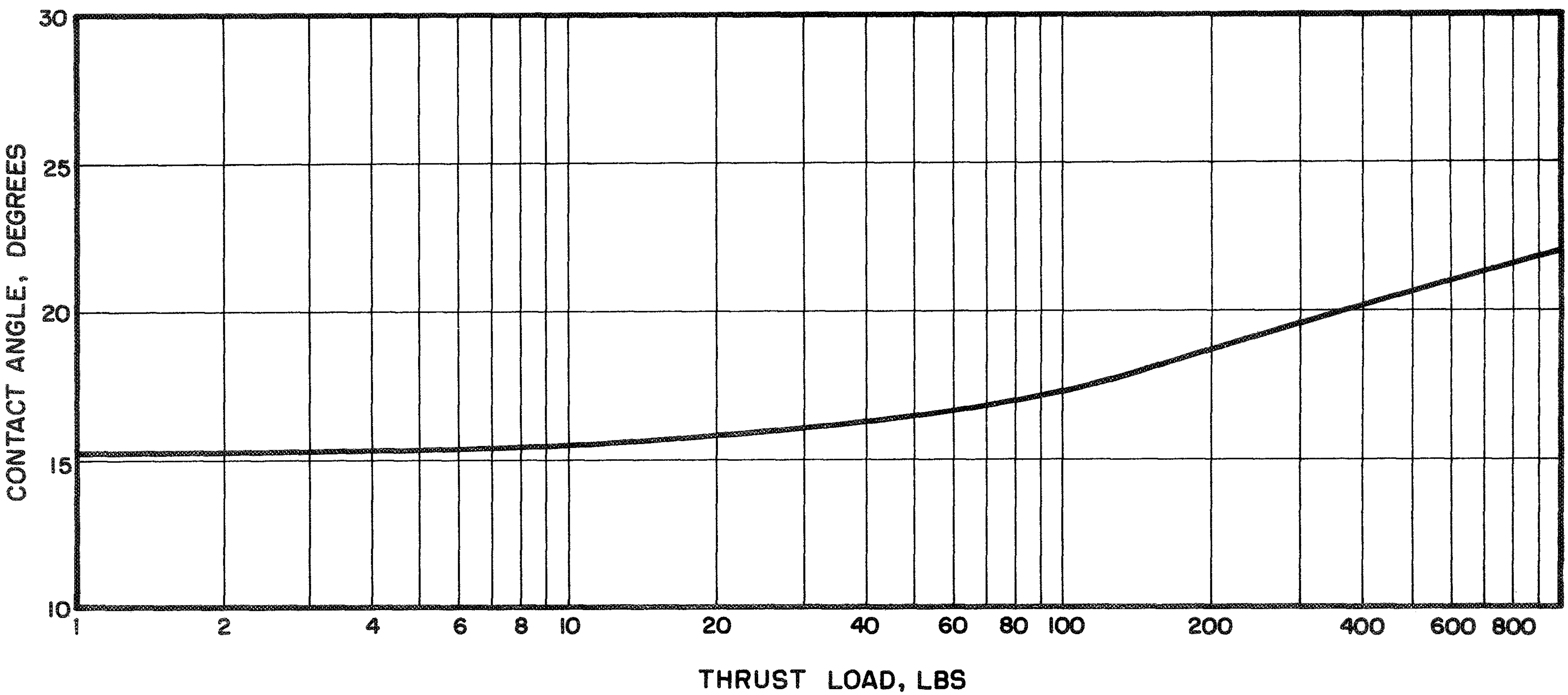


Figure 7. Contact Angle Versus Thrust Load for SNAP-8 Turbine-Alternator Bearings

The change in contact angle between the balls and the inner ring may increase or decrease stresses, depending on the nature of the load. The stresses resulting from radial load components increase as the contact angle increases. The stresses resulting from axial load components decrease as the contact angle decreases. The computed life calculations performed for the alternator included the effects of centrifugal loads. The calculations performed for the mercury pump and the turbine neglected centrifugal load effects since they were determined to be too small to affect bearing life.

e. Axial Loads

The rotors in SNAP-8 rotating machines must have closed axial positioning to operate satisfactorily. The following end play is provided in the axial stack to permit the differential expansion of rotor and stator without deforming the bearings by large loads.

Machine	Axial Gap or End Play (in.)
Turbine	0.004 to 0.006
Mercury pump	0.004 to 0.010
Alternator	0.010 to 0.012

Contact between the races and the balls is always positively maintained by the preload springs. The application of preload is shown in Figures 8, 9, and 10. The selection of axial preload values is based on several requirements. First, the balls must not become unloaded. When the axial load is zero, radial loads are carried by fewer than half of the balls. When no load is carried by a ball, it is free to skid, spin, slide and decelerate until it becomes loaded again. This produces wear on the balls, races, and the separator.

A secondary purpose of axial preload is to restrain the balls from spinning due to gyroscopic moments. The turbine and alternator gyroscopic moment is 0.38 lb/in. Unrestrained spinning of balls is considered deleterious to long life. The resistance to spinning is a function of the coefficient of friction and the load. However, the coefficient of friction varies so widely that spinning is possible and even likely when the uneven loading effects resulting from radial loads are considered. According to Palmgren methods (Reference 4), a preload of 154 pounds would be needed to ensure the total absence of spinning in the turbine bearing. This amount of preload would adversely affect the contact angle, raise gyroscopic forces, and require even more preload. The calculated fatigue life would be affected to a marked degree by this relatively large load.

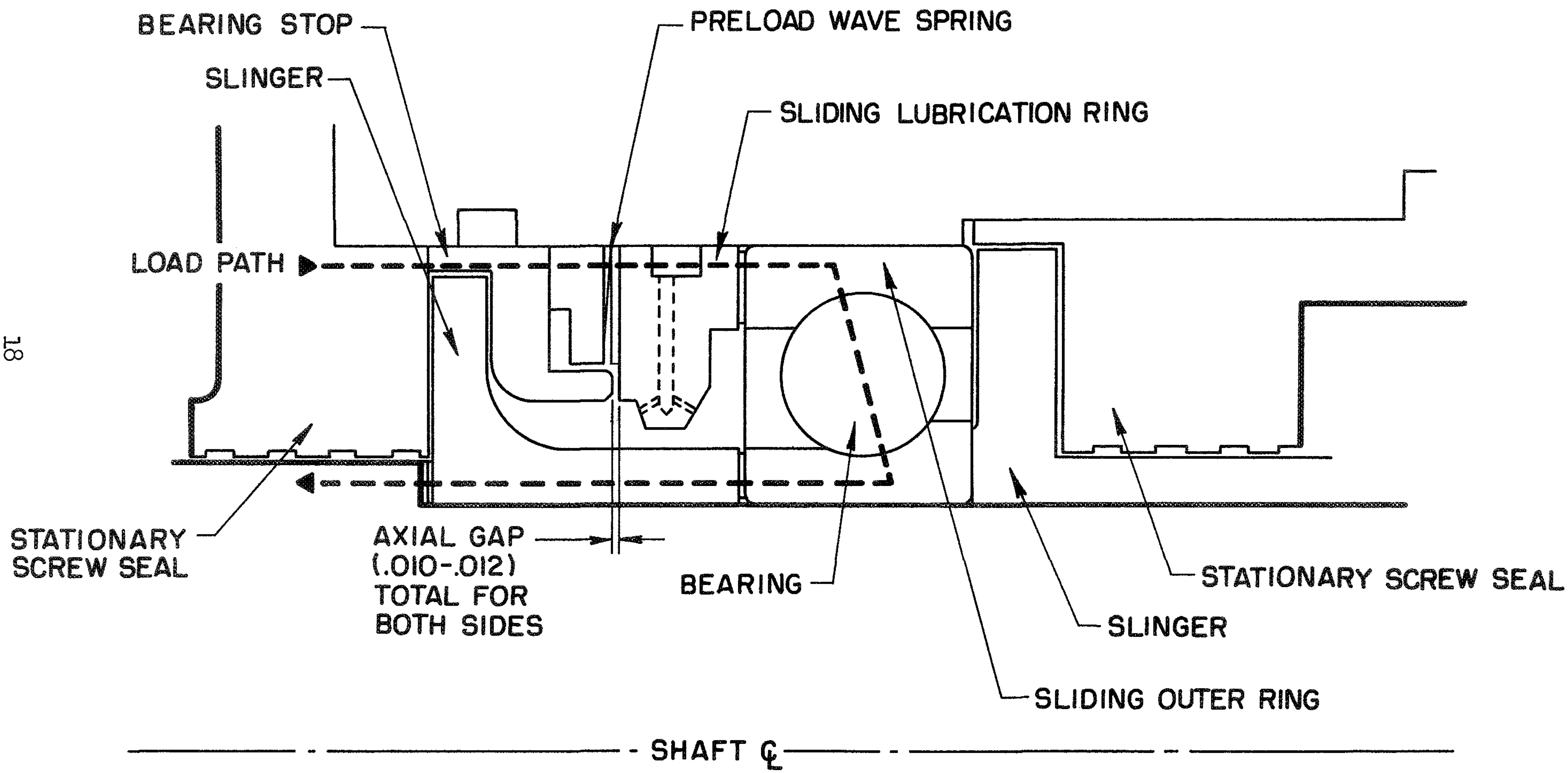


Figure 8. Alternator Preload Spring System

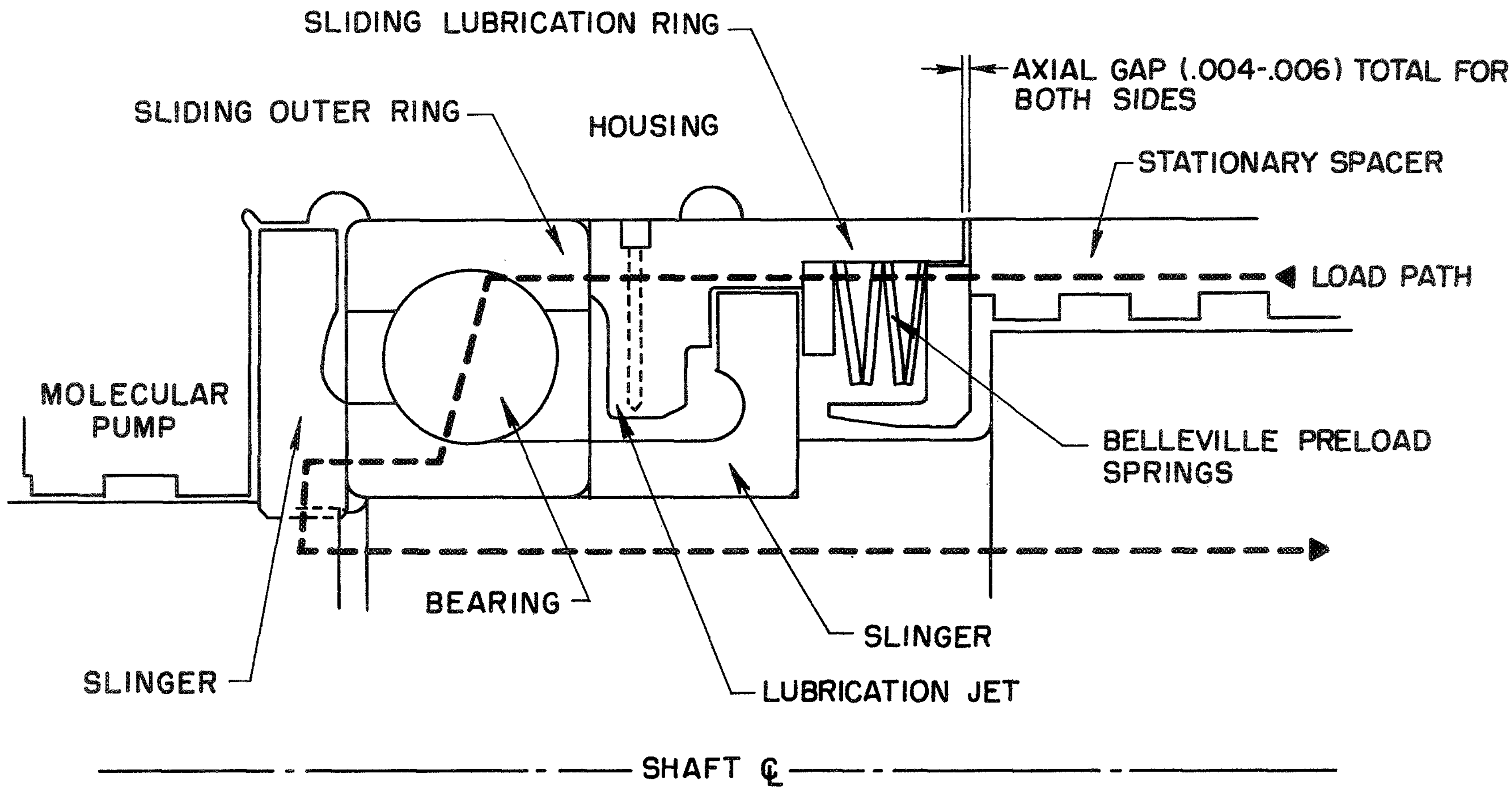


Figure 9. Turbine Preload Spring System

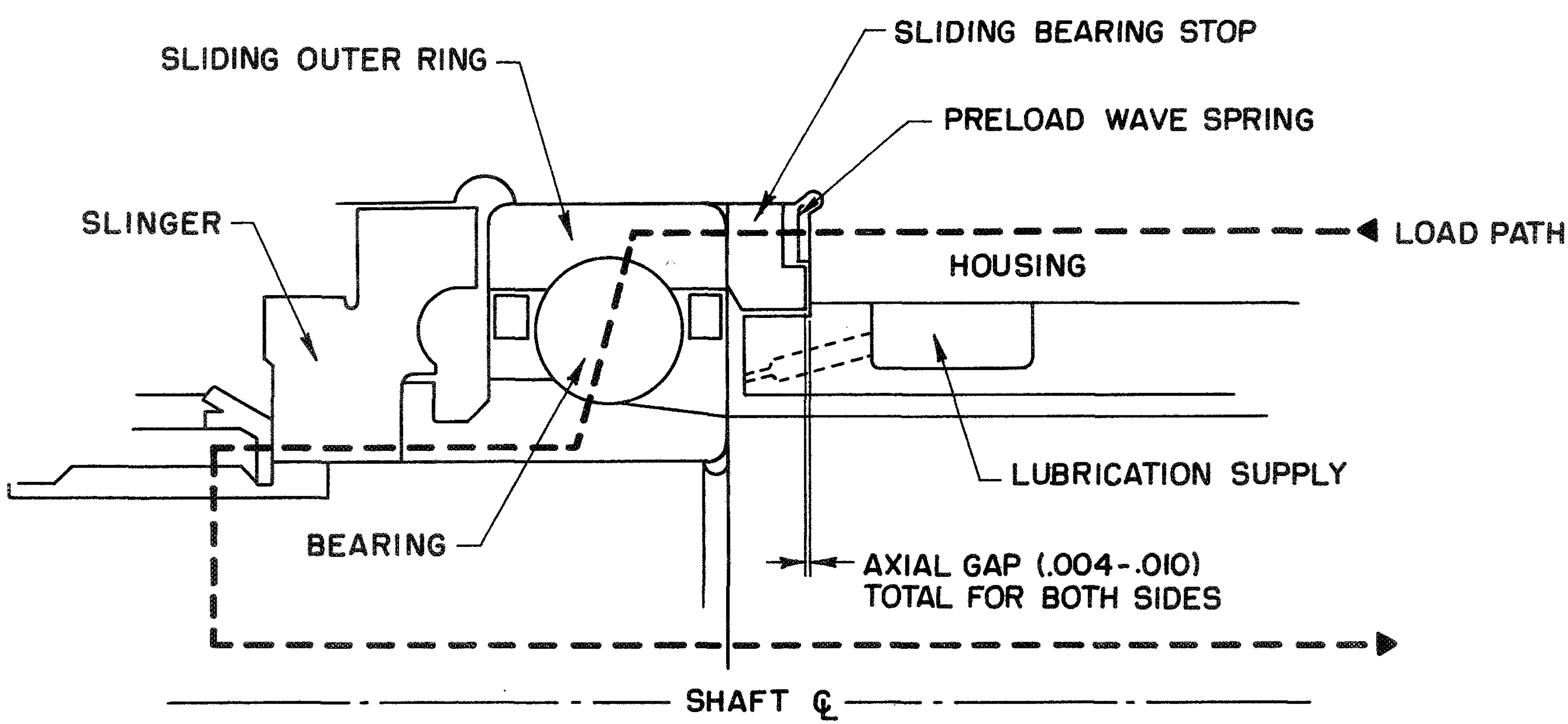


Figure 10. Mercury Pump Preload Spring System

A reduced value of 60 pounds for the turbine and alternator preload was chosen based on the relatively light applied loads and long life requirements. The preload value of 55 pounds for the mercury pump was considered sufficient to prevent gyroscopic spinning.

No indications of ball spinning have been found in post operative inspection of the SNAP-8 bearings, thus suggesting that the selected preloads were adequate.

f. Radial Stiffness

The spring constant of the bearing is a variable influenced by many factors. When a bearing has internal clearance in the direction of the applied force, the spring constant is zero until the looseness is taken up. The preload spring is used to eliminate this looseness. The direction of the applied load, the ratio of radial to axial load, the magnitude of the spring load, and the geometry of the bearing all influence the spring constant.

After the amount of the preload was established by the other requirements, such as bearing fatigue load limits, the bearing as a spring system was studied to determine (1) spring constant acceptability for axial positioning of the shaft under load, (2) the radial control of the rotor when loaded, and (3) the critical speed of the assembly. The SNAP-8 machines were designed to operate below their first critical speed. Therefore, spring constant values were established to prevent lowering the critical speed too greatly by excessively softening the mounting system.

An increase in axial load on the bearings above the preload values will increase the spring constant in both the axial and radial directions. An increase in the radial loads applied to the bearings decreases the spring constant which is shown by Figures 11 and 12.

g. Speed Parameters

Experience is used to evaluate the general practicality of operating bearings at high speeds. The common speed parameter is the DN value which is the product of the bore in millimeters (D) and the speed of

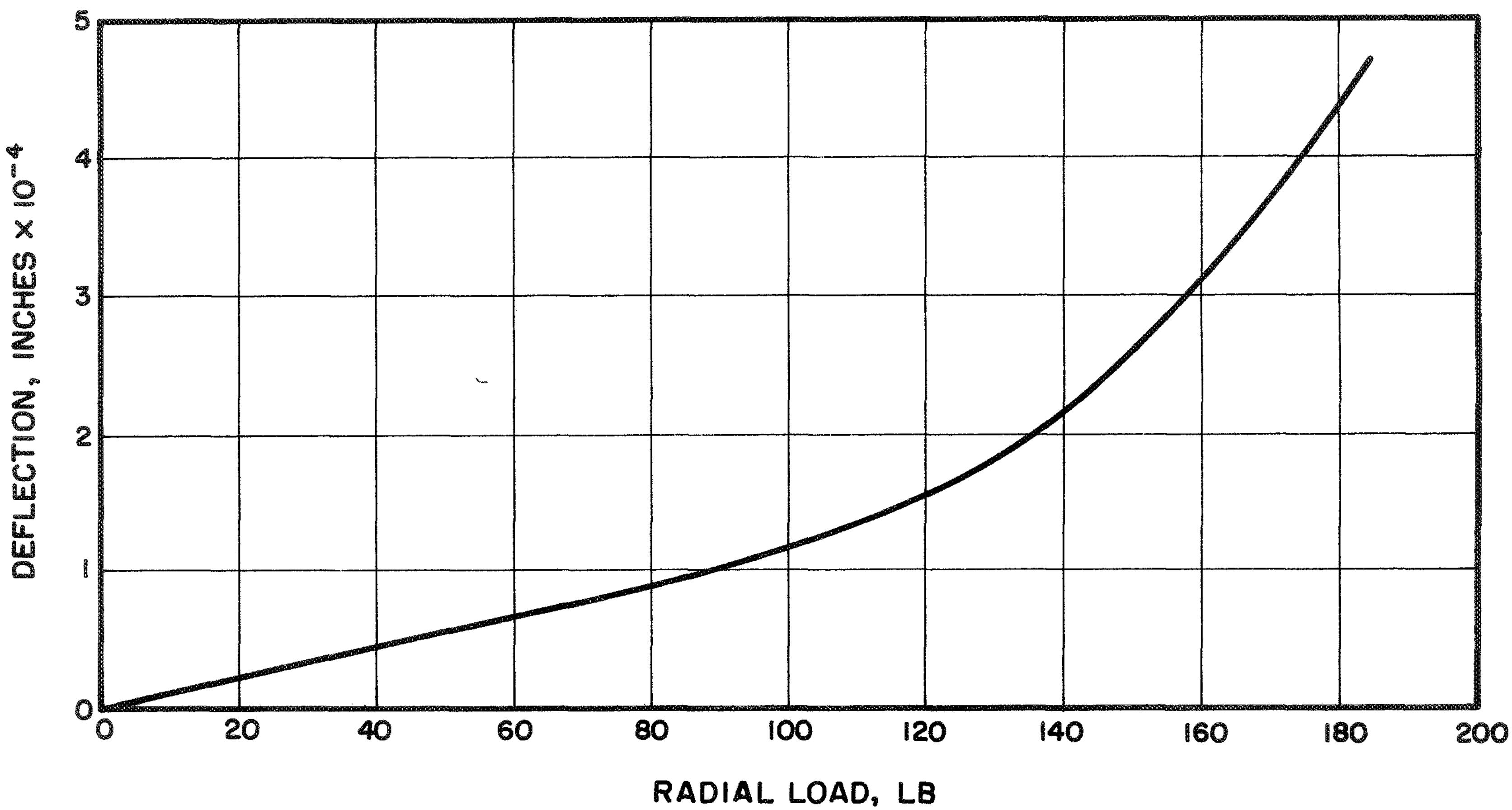


Figure 11. Radial Deflection Versus Radial Load for Turbine Bearing with 60 Pounds Preload

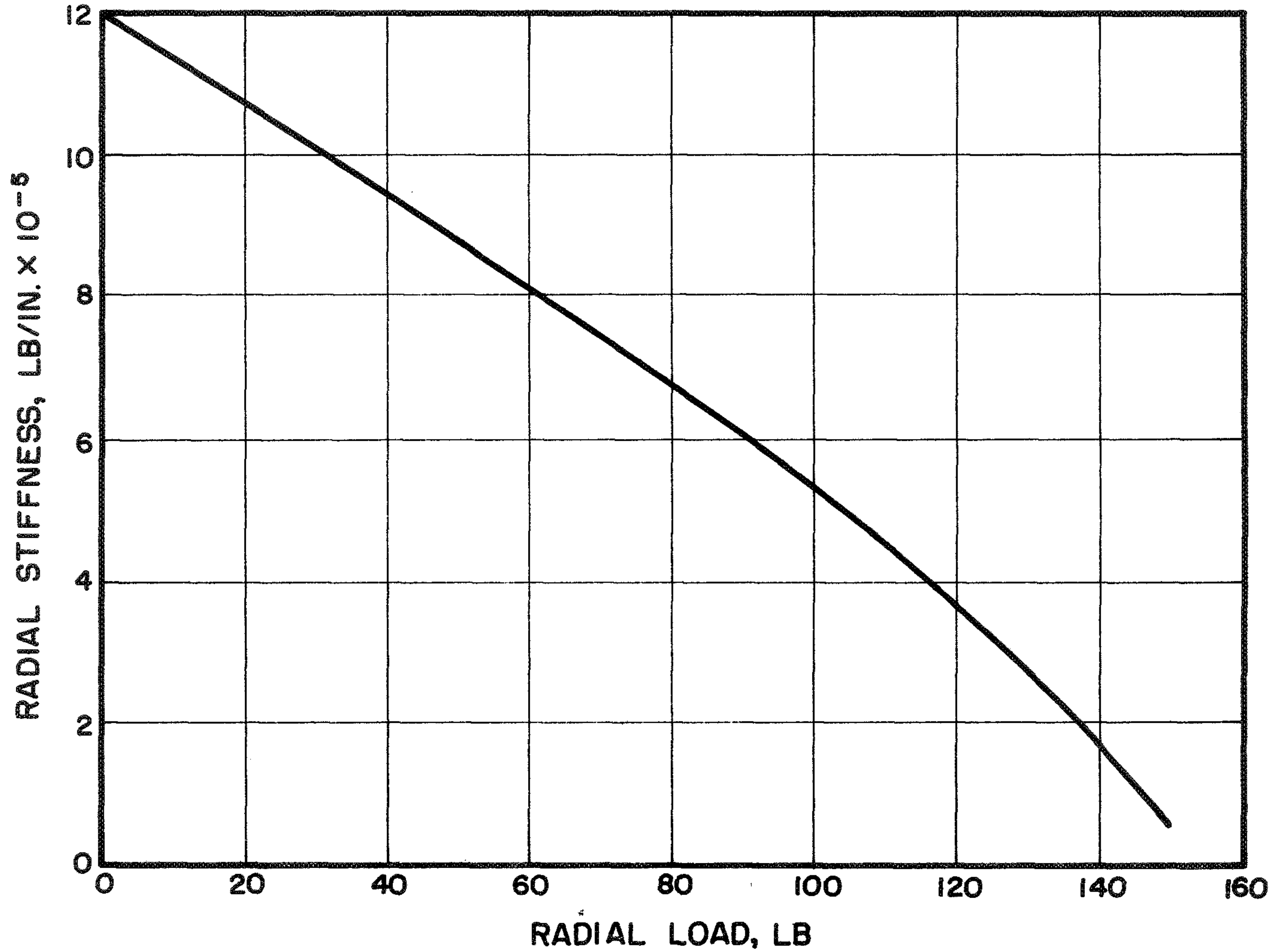


Figure 12. Turbine Bearing Radial Stiffness Versus Radial Load with 60 Pound Preload

inner ring rotation in revolutions per minute (N). The values for the SNAP-8 bearings are:

Turbine: 40 mm x 12,000 rpm = 480,000 DN

Mercury Pump: 35 mm x 7800 rom = 273,000 DN

These values are in the normal operating range for angular-contact bearings. Avoiding high DN values reduces problems in lubrication, separator performance, and ball control in the raceways. For comparison, DN values as high as 1,500,000 are practical with special techniques such as are used in the SNAP-8 bearings.

h. Bearing Size

A light series 208 bearing (40 millimeter bore) was selected for the SNAP-8 turbine and alternator. The 208 size was selected based on load-carrying capacity, shaft dynamics and optimized seal configurations (See Figure 13).

For the pump design, a light series 207 bearing (35 millimeter bore) was selected using the same reasoning as for the turbine-alternator.

The pump bearing design is generally the same configuration as the turbine bearing design (See Figure 14). To facilitate pump assembly and disassembly, a ball retaining separator and an extended inner ring with a pulling groove were employed. The ball retaining separator makes it possible to place the outer ring and balls in the housing before the inner ring is pressed onto the shaft.

i. Bearing Fatigue

Exposure to moisture, dirt, or heavy local loads which deform the balls and raceways, limit bearing life. However, if the bearings are effectively protected, well lubricated, and properly handled, only one life-limiting source remains - the fatigue of the material due to repeated stresses in the contact areas. The effect of this fatigue is flaking, which starts as a crack and develops into a spalled area on one of the load-carrying surfaces.

1. TYPE - ROLLING ELEMENT ANGULAR CONTACT BALL BEARING

2. BEARING INTERNAL GEOMETRY
   a. Contact Angle - 16° (Nominal)
   b. Race Curvatures: Inner - 52%, Outer - 53%
   c. Number of Balls - 13
   d. Size of Balls - 15/32 in.

3. SEPARATOR - OUTER LAND RIDING
   a. Material - Iron Silicon Bronze
   b. Diametral Clearance - 0.020 in.
   c. Ball Pocket Clearance - 0.025 in.

4. MATERIALS
   a. Rings - CEVM, AISI M-50 (Single Heat)
      Hardness - Rockwell C 63 to 65
   b. Balls - CEVM, AISI M-50 (Single Heat)
      Hardness - Rockwell C 64 to 66

5. BEARING TOLERANCES
   a. Per ABEC Class 7 With Exceptions
      (1) Balls - Grade 5
      (2) Inner Race Curvature $\pm$ 0.001 in.
      (3) Outer Race Curvature $\pm$ 0.002 in.
   b. Marked for Selective Assembly

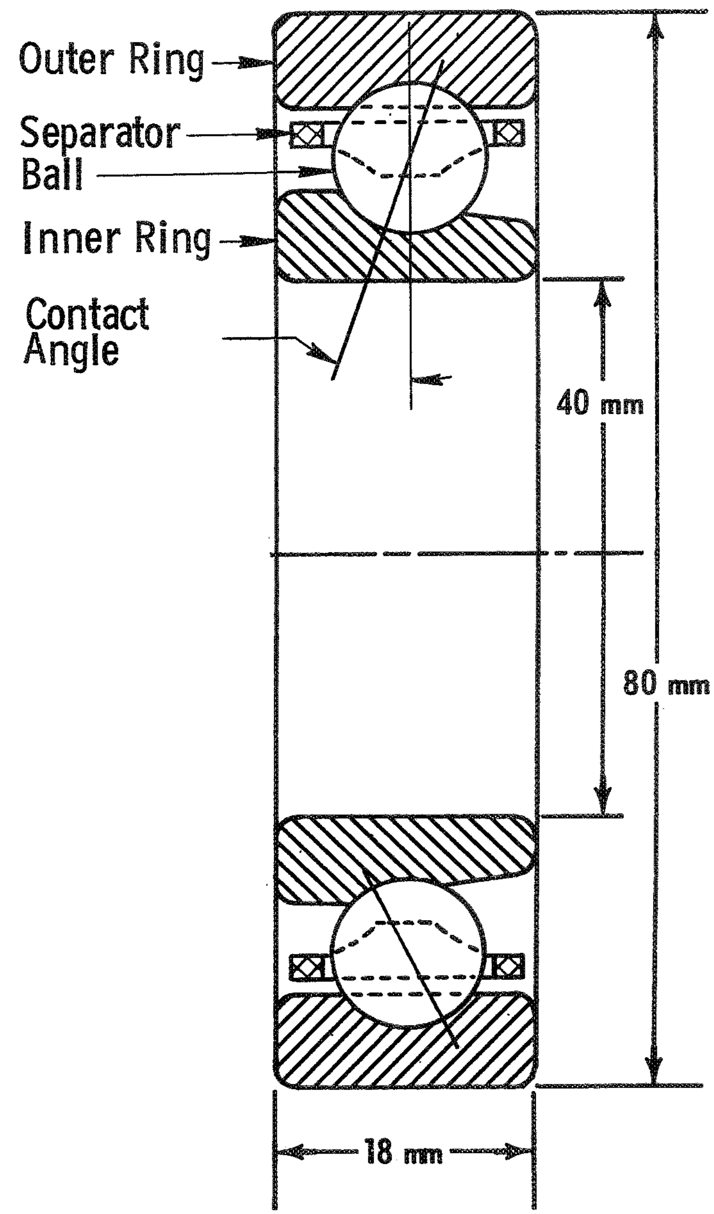


Figure 13. Turbine-Alternator Bearing Design

1. TYPE - ROLLING ELEMENT ANGULAR CONTACT BALL BEARING

2. BEARING INTERNAL GEOMETRY
   a. Contact Angle - 16^{0} (Nominal)
   b. Race Curvatures: Inner - 52%, Outer - 53%
   c. Number of Balls - 12
   d. Size of Balls - 7/16 in.

3. SEPARATOR - OUTER LAND RIDING
   a. Material - Iron Silicon Bronze
   b. Diametral Clearance - 0.020 in.
   c. Ball Pocket Clearance - 0.031 in.

4. MATERIALS
   a. Rings - CEVM, AISI M-50 (Single Heat)
      Hardness - Rockwell C63 to 65
   b. Balls - CEVM, AISI M-59 (Single Heat)
      Hardness - Rockwell C64 to 66

5. BEARING TOLERANCES
   a. Per ABEC Class 7 with Exceptions
      (1) Balls - Grade 5
      (2) Inner Race Curvature ± 0.001 in.
      (3) Outer Race Curvature ± 0.002 in.
   b. Marked for Selective Assembly

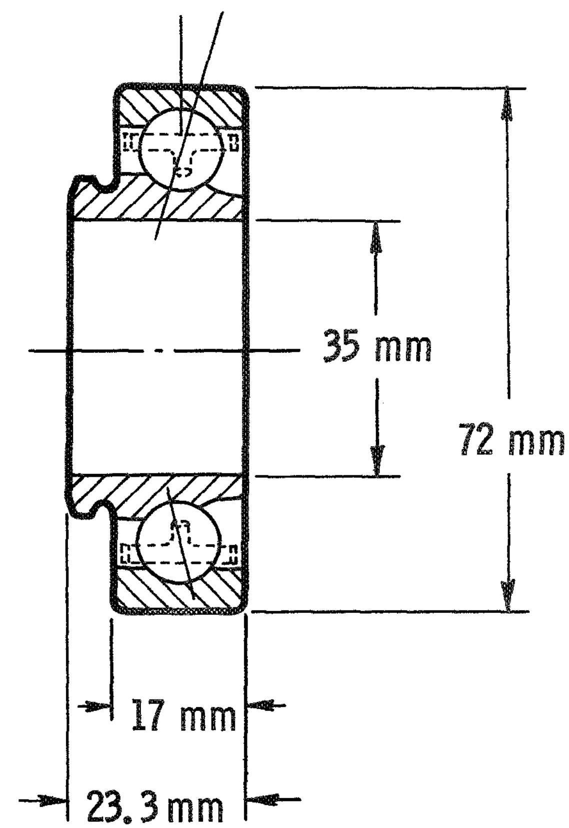


Figure 14. Mercury Pump Bearing Design

j. Load-Life Calculations

The conventional method of engineering bearing applications is to use formulas based on contact stress patterns and to relate empirical fatigue data to the application at hand. Statistically, the method is satisfactory for commercial applications where an acceptable criterion is 90% of units exceeding the design life. This criterion, called the "L_{10}" life of the bearings, is approximately one fourth the average life of the bearings. This L_{10} life can be related to the bearing dynamic-load capacity, the load at which 90% of a group of bearings can survive one-million inner-race revolutions: $L_{10}\ \text{life} = \dfrac{\text{Dynamic Load Capacity}}{\text{Bearing Equivalent Load}}$. The equivalent load is a load in either the radial or axial direction, or a combined radial and axial load. Formulas for calculating equivalent load are given in AFBMA* standards (Reference 5). The dynamic capacity is the rating of the given bearing.

The empirical data are modified for SNAP-8 applications due to the difference in bearing fabrication and application. Some of the factors which can significantly improve the life predicted by generally accepted rating systems are material with reduced impurities, greater precision due to better than Class 7 tolerances, greater hardness, optimized lubrication method, better finish on ball and rings, and improved inspection methods.

The positive effect of the foregoing differences has been established and quantitative improvements can be estimated when they are individually varied. However, the combined effect of multiple improvement factors has not previously been established, and statistical fatigue testing of bearings under SNAP-8 conditions has not yet been possible. To demonstrate bearing usage for 10,000 hours at 99.5% reliability would require a great number of data points for accuracy. The Weibull plot or fatigue failure distribution curve normally used in bearing work has been shown to be very

* AFBMA is the Anti-Friction Bearing Manufacturers Association

conservative in the high-reliability range, but is sufficiently accurate in the normal range of bearing application (see Tallian, Reference 6). A correction factor developed by Tallian for lightly loaded bearings was employed for SNAP-8 fatigue calculations.

To complete the life calculations, an assignment of a modified dynamic capacity was made. For the turbine, this was 26% greater than the standard capacity as obtained by Palmgren methods (Reference 2).

	Palmgren Capacity	AFBMA Capacity	Modified M-50 Capacity*
Turbine-alternator, lb	6140	6250	7750
Mercury pump, lb	5200	5220	6550

Life Calculations for Turbine Bearings: (See Figure 15 for Calculated Load vs Life Curve)

Standard life (10% failure rate) -

$$L_{10} = \frac{\text{Capacity}^3}{\text{Load}} \times 10^6 = \frac{7750^3}{159} \times 10^6 = 115{,}500 \times 10^6 \text{ revolutions}$$

$$L_{10} = \frac{115{,}500 \times 10^6}{12{,}000 \times 60} = 160{,}000 \text{ hours}$$

99.5% Reliability life (0.5% failure rate) -

$$L_{005} = 0.103\ L_{10} = 0.103 \times 160{,}000 = 16{,}500 \text{ hours using standard failure distribution}$$

$$L_{005} = 0.149 \times L_{10} = 0.149 \times 160{,}000 = 24{,}000 \text{ hours using Tallian failure distribution}$$

* Used in load-life calculations.

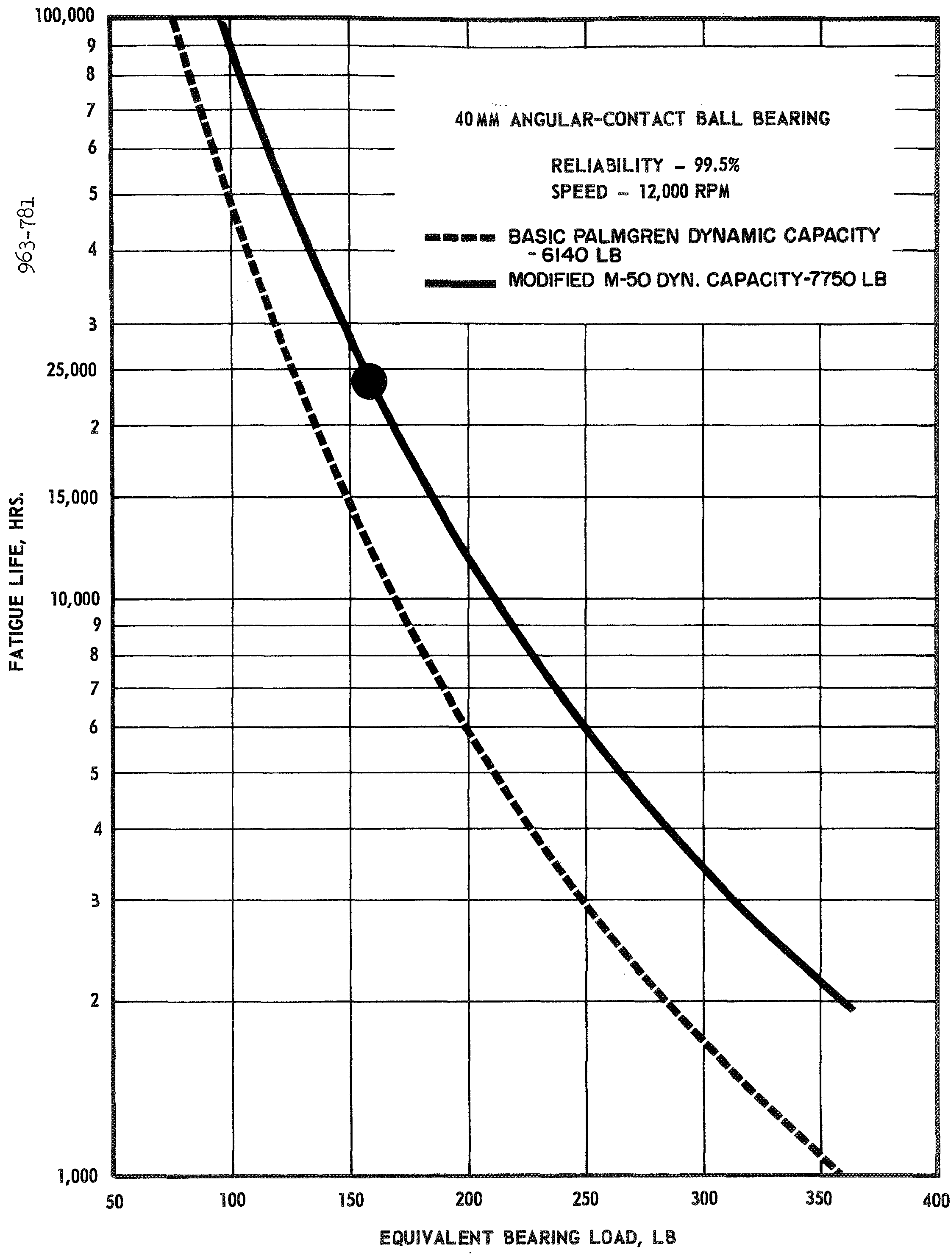


Figure 15. Bearing Fatigue Life

Life Calculations for Alternator Bearings:

Standard life (10% failure rate) -

$$L_{10} = \frac{\text{Capacity}^3}{\text{Load}} \times 10^6 = \frac{7750^3}{154} \times 10^6 = 127{,}000 \times 10^6 \text{ revolutions}$$

$$L_{10} = \frac{127{,}000 \times 10^6}{12{,}000 \times 60} = 176{,}000 \text{ hours}$$

99.5% Reliability life (0.5% failure rate) -

$$L_{005} = 0.103\ L_{10} = 0.103 \times 176{,}000 = 18{,}100 \text{ hours using standard failure distribution}$$

$$L_{005} = 0.149 \times L_{10} = 0.149 \times 176{,}000 = 26{,}200 \text{ hours using Tallian failure distribution}$$

Life Calculations for Mercury Pump Bearings:

Standard life (10% failure rate) -

$$L_{10} = \frac{\text{Capacity}^3}{\text{Load}} \times 10^6 = \frac{6550^3}{165} \times 10^6 = 62{,}500 \times 10^6 \text{ revolutions}$$

$$L_{10} = \frac{62{,}500 \times 10^6}{7800 \text{ rpm} \times 60} = 133{,}700 \text{ hours}$$

99.5% Reliability life (0.5% failure rate) -

$$L_{005} = 0.103 \times L_{10} = 0.103 \times 133{,}700 = 13{,}800 \text{ hours using standard failure distribution}$$

$$L_{005} = 0.149 \times L_{10} = 0.149 \times 133{,}700 = 19{,}900 \text{ hours using Tallian failure distribution}$$

k. Static Loads

When a bearing at rest is loaded, there is a limit to the load it can endure without suffering excessive permanent deformation. This load limit is called the static capacity. The static capacity is dependent on bearing size, the degree of conformity between rolling elements and race, and material characteristics. The maximum permanent deformation of a rolling element or race at a contact which can be sustained without significantly degrading bearing performance is 0.0001 times the diameter of the rolling element. Equations for static capacity are given in AFBMA Section 9, Reference 3. These values are 5000 lb and 3090 lb for the turbine-alternator and mercury pump bearings respectively.

C. BEARING TYPE SELECTION

The basic bearing characteristics and features considered in the selection of the best bearing for a particular design are load capacity, materials, production features, separators, high-speed applications, and lubrication. All of these features are discussed in the basic design considerations for the SNAP-8 system rotating components. The selected bearing design characteristics and tolerance requirements for the turbine-alternator and mercury pump are presented in Table III.

1. Type of Bearing

Basic decisions were made affecting bearing type selection at the beginning of the bearing design program. Ball rather than hydrodynamic bearings were to be used and a suitable lubricant provided. The reason for this selection was the use of ball bearings would minimize the longer development normally associated with hydrodynamic bearings for reliable use in a zero gravity environment.

Angular-contact ball bearings were chosen in preference to other types of bearings which were suitable for turbomachinery applications. They provide more capacity and stiffness than similarly sized Conrad-type (or deep groove) bearings, and can be preloaded to reduce wear and provide closer rotor dynamic control. Cylindrical roller bearings cannot be radially

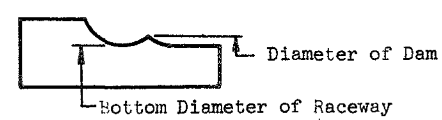
Table III. Bearing Design Characteristics and Tolerances Requirements

Element	Characteristic	Turbine Alternator Bearing	Mercury Pump
Assembled Bearing (Reference Section 3 of AFBM Std)	Contact Angle	16° ± 2° (Reference)	16° ± 2° (Reference)
	Internal Diametral Clearance	0.0016 to 0.0021 inches under 5-1/2 lb reversing load	0.0015 to 0.0020 inches under 5-1/2 lb reversing load
	Separator Diametral Clearance to Guiding Lands in Outer Ring	0.016 to 0.024 inches	0.014 to 0.021 inches
	Ball to Separator Pocket Diametral Clearance	0.030 to 0.032 inches	0.028 to 0.030 inches
	Low Shoulder Dam Diameter on Inner Ring	(* See Below)	None - bearing to be fully separable
Balls (Reference Section 10 of AFBM Std for Grade 5 Balls)	Number per bearing	13	12
	Nominal Ball Diameter	15/32 inches	7/16 inches
	Ball Grade	AFBMA Std. Gr. 5 1/	AFBMA Std. Gr. 5 1/
	Ball Surface Roughness (Micro-inches)	0.7 RMS, maximum	0.7 RMS, maximum
Inner Ring	Bore	1.5746 to 1.5748 inches	1.3778 to 1.3780 inches
	Width	0.708 to 0.707 inches	0.9193 to 0.9183 inches
	Raceway Curvature (Radius)	0.244 ± 0.001 inches	0.228 ± 0.001 inches
	Radial Runout, raceway	0.0001 inch, maximum	0.0001 inch, maximum
	Width Variation	0.0001 inch, maximum	0.0001 inch, maximum
	Reference Side Runout with Bore	0.00015 inch, maximum	0.00015 inch, maximum
	Raceway Runout with Ref. Side	0.00015 inch, maximum	0.00015 inch, maximum
	Roundness of Bore	0.0001 inch, maximum	0.0001 inch, maximum
	Bore Taper	0.0001 inch/inch, maximum	0.0001 inch/inch, maximum
	Raceway Finish (Microinches)	4 RMS, maximum	4 RMS, maximum
Outer Ring	Outside Diameter	3.1496 to 3.1494 inches	2.8346 to 2.8344
	Width	0.708 to 0.707 inches	0.6698 to 0.6678
	Raceway Curvature (Radius)	0.248 ± 0.002 inches	0.232 ± 0.002 inches
	Radial Runout, raceway	0.00015 inch, maximum	0.00015 inch, maximum
	Width Variation	0.0001 inch, maximum	0.0001 inch, maximum
	Outside Cylindrical Surface Runout with Ref. Side	0.0015 inch, maximum	0.00015 inch, maximum
	Raceway Runout with Ref. Side	0.0002 inch, maximum	0.0002 inch, maximum
	O.D. Taper	0.0001 inch/inch, maximum	0.0001 inch/inch, maximum
	Raceway Finish (Microinches)	4 RMS, maximum	4 RMS, maximum
Separator	Roundness of O.D.	0.002 inch, maximum	0.002 inch, maximum
	Taper of O.D.	0.001 inch/inch, maximum	0.001 inch/inch, maximum
	Finish of O.D.	32 RMS inches, maximum	32 RMS inches, maximum
	Pocket Size Variation	0.002 inch, maximum	
	Pocket Axial Position Variation	0.002 inch, maximum	0.002 inch, maximum
	Pocket Finish at Pitch Diameter of Bearing	32 RMS, maximum	32 RMS, maximum
	Inside Diameter Concentricity with O.D.	0.002 inches TIR, maximum	0.002 inches TIR, maximum
	Pocket to Pocket Circumferential Distance Variation	0.005 inches, maximum	0.005 inches, maximum

1/ Note: Tolerances for AFBMA Gr. 5 balls are as follows:

Ball basic dia. tol.	±0.00005 inch
Ball out-of-roundness:	
Dia. tol. per ball:	0.000005 inch
"V" block out-of-roundness tol.	0.000005 inch
Allowable dia. variation per bearing	0.00001 inch

* The bottom diameter of the raceway plus the bearing internal diametrical clearance plus 2.0 to 2.5 X 10^{-3} inches.



spring preloaded, and at high speeds and temperatures provide only limited axial capacity. Tapered roller bearings would be subject to speed-associated problems which would reduce reliability.

Angular-contact ball bearings are designed for use where radial and thrust loads are combined, and where precise shaft location is required. This particular bearing was selected as the best basic type for SNAP-8 high-speed operation, for long life and high reliability. It provides a maximum ball complement for large bearing load capacity and high radial stiffness and permits the use of a one-piece lightweight maximum-strength ball separator. The low, nonthrust shoulder is on the inner ring and the separator is piloted on both lands of the outer ring. This design provides sufficient outer ring surface to employ radial slingers directly against bearing shoulders. With axial preloading, the bearing operates without internal clearance, permitting optimum rotor assembly dynamic balance and giving maximum control against eccentricity which would adversely affect the close-clearance dynamic shaft seals.

2. Separator Selection

The use of an angular-contact bearing accommodates a one-piece separator for lightweight construction and for maximum strength with more reliability. The separator, or cage, maintains the proper distance between the rolling elements.

Since the preponderance of modern designs for one-piece separators are land-riding, and relatively simple to manufacture and inspect, this method was selected. It was further decided that the separators should ride on (or be piloted by) the inner lands of the outer-ring since this provides; (a) a more open inner ring for better cooling by oil-jet lubrication, (b) a more positive oil flow due to centrifugal effects between the separator and the guiding lands, and (c) no reaction on the separator from impinging oil jets. With the low race shoulder located on the inner ring, it was possible to use both lands to control the separator; this develops more stability in separator hydrodynamics with a larger length-to-diameter ratio to resist cocking.

The balls drive the separator and the guiding lands produce a viscous drag in this configuration. In order to drive the separator with tangential forces, the ball-pocket bores are cylindrical with the point of contact between ball and pocket occurring at the pitch line of the balls. This ensures full fluid-film development to reduce separator pocket and ball wear, and does not restrict the flow of lubricant through the bearing.

3. Bearing Fits and Clearances

Bearings must be carefully fitted to the shafts and housings in which they are used and must be constructed for optimum performance. Inner rings must fit the shaft tightly enough to minimize eccentricities and provide maximum heat transfer, yet not be so tight as to lose the bearing internal clearance necessary for load distribution and avoidance of radial overloads. The initial fits must permit practical installation and disassembly of the bearing by use of reasonable force or thermal expansions. Shaft material and bearing material coefficients of expansion, moduli of elasticity, and sections must be matched over the entire range of bearing operation including startup and shut-down periods when cooling may not be available.

Outer rings must be fitted closely, provide ease of assembly, freedom to slide during operation, yet have enough control to provide correct rotor radial positioning. Self jamming or axial installation forces that expand the ring against housing and necessitate excessive additional mounting force must be avoided. Fits for shafts and bearings were selected as shown in Table IV.

D. MATERIAL SELECTION

1. Bearing Material Selection

S.A.E. 52100 steel has been commonly used in the bearing industry since 1920. This is a high-carbon chromium steel which exhibits hardness loss at temperatures above 350°F. In addition, metallurgical transformations can occur which adversely affect geometry. Therefore, a more stable material for use at SNAP-8 operating temperatures was required.

Table IV. Bearing Fits and Clearances

	Inner Ring on Shaft (in.)	Outer Ring In Housing (in.)
Turbine Drive End		
Room temperature	.0003T to .0001L	.0001L to .0007L
Operating temperature	.00045T to .00005T	.0004L to .0010L
Turbine Antidrive End		
Room temperature	.0003T to .0001L	.0001L to .0007L
Operating temperature	.0003T to .0001L	.0001L to .0007L
Alternator, Both Ends (G.E.)		
Room temperature	.0001T to .0007T	.0001L to .0007L
Operating temperature	.0000 to .0006T	.0003L to .0009L
Mercury Pump, Both Ends		
Room temperature	.0000 to .0004T	.0001L to .0005L
Operating temperature	.0001T to .0005T	.0002L to .0006L

T = tight

L = loose

The design requirements for bearing materials are as follows:

- Temperature:

  Operating: 200° to 400°F

  Storage: -40° to 160°F
- Dimensional and hardness stability at operating conditions
- Producibility, availability, and experience
- Cleanliness of materials to limit points of stress concentration

M-50 tool steel has been widely used for intermediate temperature applications from 350° to 600°F, and industry experience with this material is extensive. In addition, ball bearing tests comparing statistical lives with a number of material vacuum remelt cycles had been performed using M-50 steel. Vacuum remelting can provide improved trace element control which has been correlated with improved performance.

Consumable-electrode vacuum melt (CEVM) is an improved method of processing bearing steel. This process uses electrodes made from a primary air-melted heat which are melted by an electric-arc process. The remelt product solidifies in a water-cooled copper mold under vacuum to produce a more consistent high-quality steel. It is possible with the remelt technique to produce material with a lower inclusion content, particularly those inclusions which are more injurious, such as oxides, silicates and aluminates.

Hardness of material must be considered in the selection of bearings. Temperature at which the bearing will operate must also be considered, since hardness decreases with increasing temperatures. Heat treatment of bearing material will uniformly harden and temper the finished product equivalent to 63 to 66 Rockwell C. It is the design intent that balls have a greater hardness than the rings.

The SNAP-8 system bearings selected are made of M-50 tool steel manufactured by the vacuum remelt process, uniformly hardened and triple tempered to 64 through 66 on the Rockwell C scale.

2. Separator Material Selection

In conventional bearing use, separators are either metallic or nonmetallic, stamped from low carbon steel, or machined from either iron-silicon bronze or a lead brass. Precision bearings are usually equipped with a machined separator made of a copper alloy. The separator material selected for the bearings of the SNAP-8 turbine-alternator and mercury pump is iron-silicon bronze for strength and the minimizing of ball wear.

E. LUBRICATION AND COOLING SYSTEM

The lubricant-coolant loop in the SNAP-8 electrical generating system carries a polyphenyl ether (Mix-4P3E) which flows through cooling passages in the turbine-alternator and mercury pump and serves as a lubricant to cool the alternator stator and pump motor windings. The necessary cooling for the space seal system in both components is also provided by the Mix-4P3E. The lubricant-coolant loop contains its own pump and radiator.

A schematic of the lubricant-coolant loop in the SNAP-8 test configuration is shown in Figure 16.

1. Lubrication System Flow Control

The lubricant and coolant portions of the loop are separated at inlet and return lines by solenoid valves (See Figure 16). This permits the machines to start without cavity flooding. Fluid return from the bearings is accomplished with slinger pumps dependent on shaft rotation for effective capacity. Lubricant flow to the bearing is initiated at approximately 75% of normal operating speed. See Table V for details of the selected jet sizes and lubricant flows.

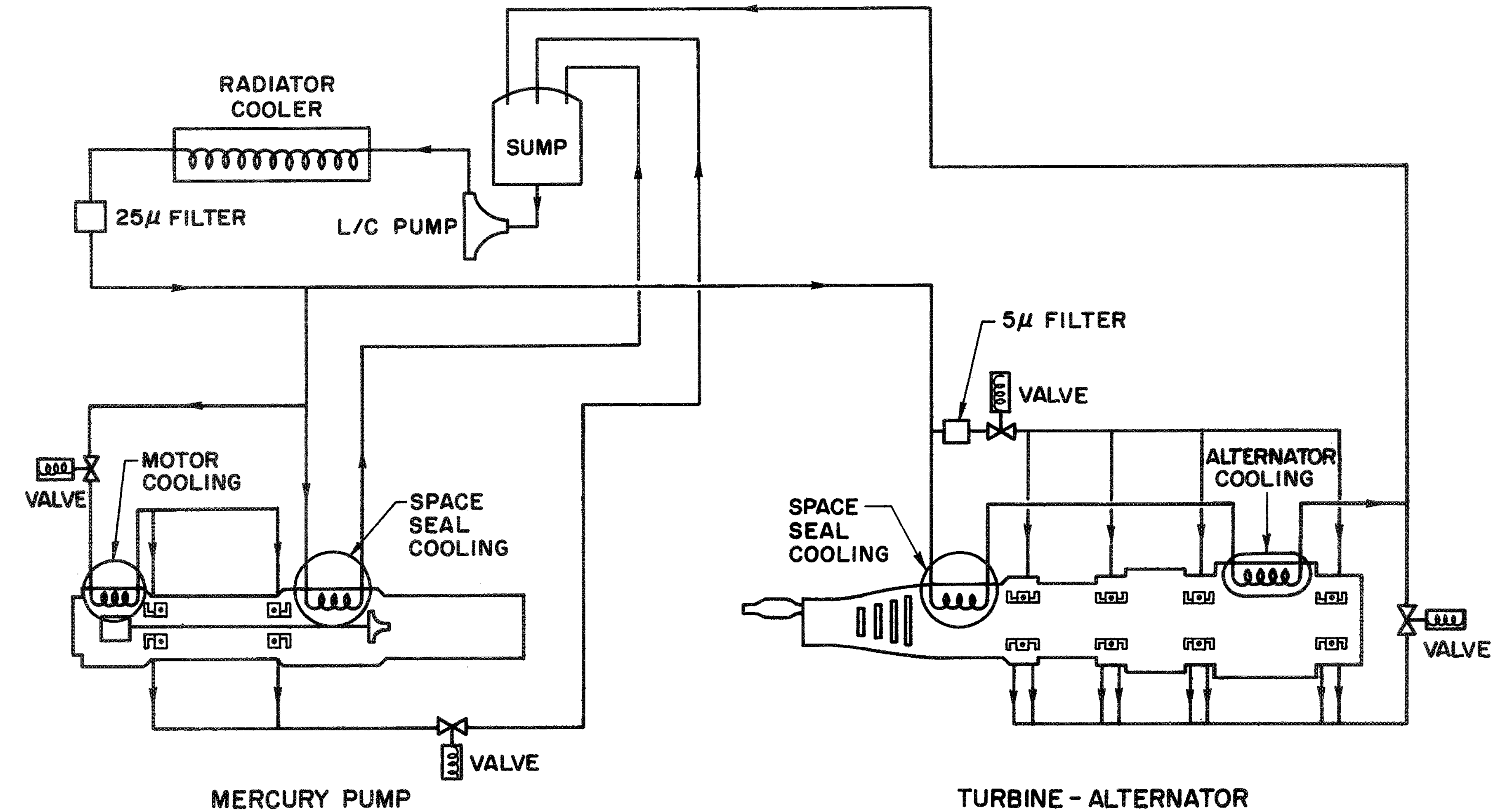


Figure 16. SNAP-8 System Lubricant-Coolant Loop Flow Schematic

Table V. Bearing Lubricant Flow

Flow Requirements	Turbine	Alternator	Mercury Pump
Total flow, lb/hr bearing	200	200	100
Pressure drop across, jets, psi	15	15	1.0
Number of jets,			
Towards bearing	3	4	6
Towards slinger	--	2	--
Diameter of jets			
Towards bearings, in.	0.040	0.022	0.040
Towards slingers, in.	--	0.028	--

2. Turbine-Alternator Lubrication

The turbine-alternator uses mercury vapor as the turbine working fluid, and Mix-4P3E as the lubricant-coolant. It is essential to keep these two fluids separated because even a small amount of Mix-4P3E in the mercury loop will adversely affect boiler performance and seriously inhibit the production of superheated mercury vapor. The turbine is provided with a seal vented to space as a barrier to prevent intermixing (See Figure 3). The space seal uses visco pumps and slingers to dynamically form a liquid barrier and provides separation of the two fluids. Molecular pumps retard boil-off from the liquid surface. Small amounts of fluids which evaporate and thereby succeed in crossing the dynamic barrier are vented to space.

Fluid is injected to the bearings through multiple jets directed at the low shoulder of the inner rings. The flow through the bearings is aided by the natural pumping action of the bearing. Tests for the turbine and alternator indicate that more than 80% of the fluid passes through the bearing, and the remainder is removed by the slinger on the oil inlet side.

The alternator bearings are lubricated by multi-jet injection of the Mix-4P3E shown diagramatically in Figure 17. Scavenging slinger seals ensure nonflooding operation and return the lubricant to the suction side of the lubricant-coolant loop. Lubrication of all the bearing elements, maximum bearing through flow, and effective cooling are achieved by injecting the lubricant towards the low, nonthrust shoulder of the bearing.

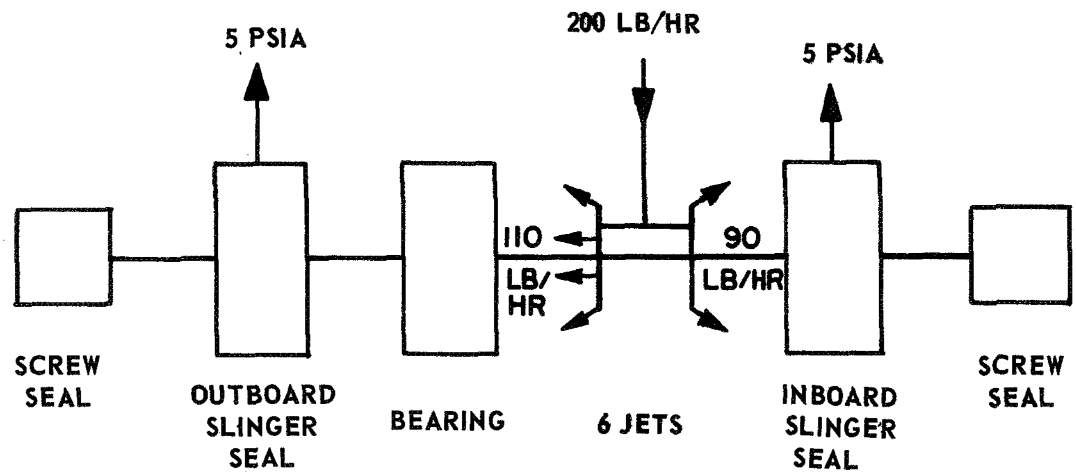


Figure 17. Bearing and Seal System Schematic for SNAP-8 Alternator

3. Mercury Pump Lubrication

The mercury pump lubrication system incorporates an injection mechanism and scavenging device to prevent flooding. Injector rings provide lubricant at a rate of 100 lb/hr to each bearing. This rate assures adequate lubrication and keeps the bearing temperature below 250°F. To ensure reliability, each lubricant injection hole was made large enough to minimize possible fouling. The selected design consists of an injection ring with six equally distributed 0.040 inch diameter holes which direct lubricant toward the inner race of the bearings

Two slingers are positioned opposite the lubricant injection rings to scavenge the bearing cavities of Mix 4P3E polyphenyl ether. To prevent bearing flooding, each slinger is designed to pump against a 5 psia back pressure in the discharge line.

4. Heat Management

In SNAP-8 bearing applications, only a small amount of lubricant is actually required for bearing lubrication. However, cooling the bearings to remove friction heat requires a greater lubricant flow, and cooling the rotors, housings, and space seals places the greatest demand on the lubricant supply.

Complete thermal analyses were performed for the turbine-alternator and mercury pump. A thermal map shows the temperature distributions in the components that can be used to locate heat paths. The turbine hot-end bearing and slingers remove heat from the shaft. Each alternator bearing and the associated slingers remove heat from the rotor. The significant alternator temperatures are shown in Figure 18. The mercury pump bearings and slingers carry heat from the rotor in the pump motor and from the mercury heated parts.

5. Lubricant Filtration

To improve bearing reliability and prevent plugging of the jets, the bearing system is provided with a 5-micron filter. The filter elements selected are constructed of Type 304 stainless steel "Dutch" weave cloth pleated around a cylindrical form. The 5-micron rating has an absolute

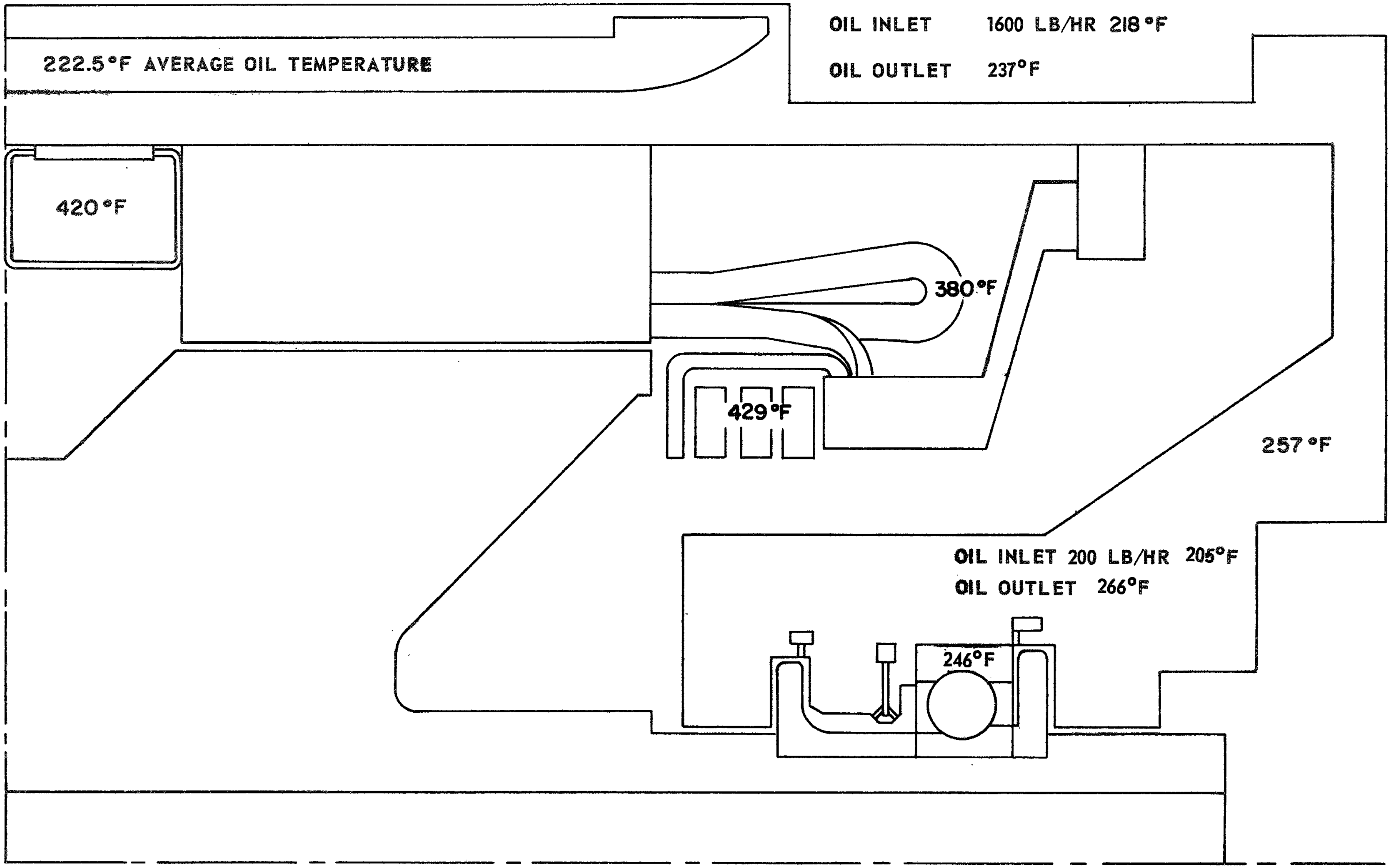


Figure 18. Significant Alternator Temperatures

particle stoppage of 18 microns. The coolant portion of the lubricant-coolant system has an additional 25-micron filter. This large filter collects gross particulates, since the entire flow of the system (6800 lb/hr) is directed through the coarser filter. The bearing lubricant filter element flow is 800 lb/hr in the turbine-alternator and 200 lb/hr in the mercury pump.

To date, operating experience has demonstrated that the 5-micron filter has sufficient capacity. Particulate matter collection has been a small fraction of the filter capacity and excessive pressure drops have not occurred.

6. Lubricant Selection

The SNAP-8 lubricant-coolant fluid must meet a number of requirements in terms of its working characteristics; these are: thermal stability, nuclear stability, high heat-transfer coefficient, high specific heat, suitable viscosity at 200 to 400°F, noncorrosive to common engineering materials, good lubricity, and low vapor pressure. Mix-4P3E was selected as the best of the available fluids to meet these requirements, mainly because of its ability to withstand nuclear radiation. The term "Mix-4P3E" is an abbreviated chemical nomenclature for a mixture of six isomers of chemical bis (phenoxyphenyl) ether. An isomer is a compound having the same number and kind of atoms in the molecule, but the atoms have a different orientation.

This fluid falls into the class of polynuclear aromatics which are known to be the most radiation resistant, thermally and oxidatively stable fluids currently available. It is compatible with metallic construction materials at high temperature, but it reacts with most soft gasket materials except Vitron A and Teflon.

The general properties of Mix-4P3E are listed in Table VI. Fluid characteristics are shown in Figure 19.

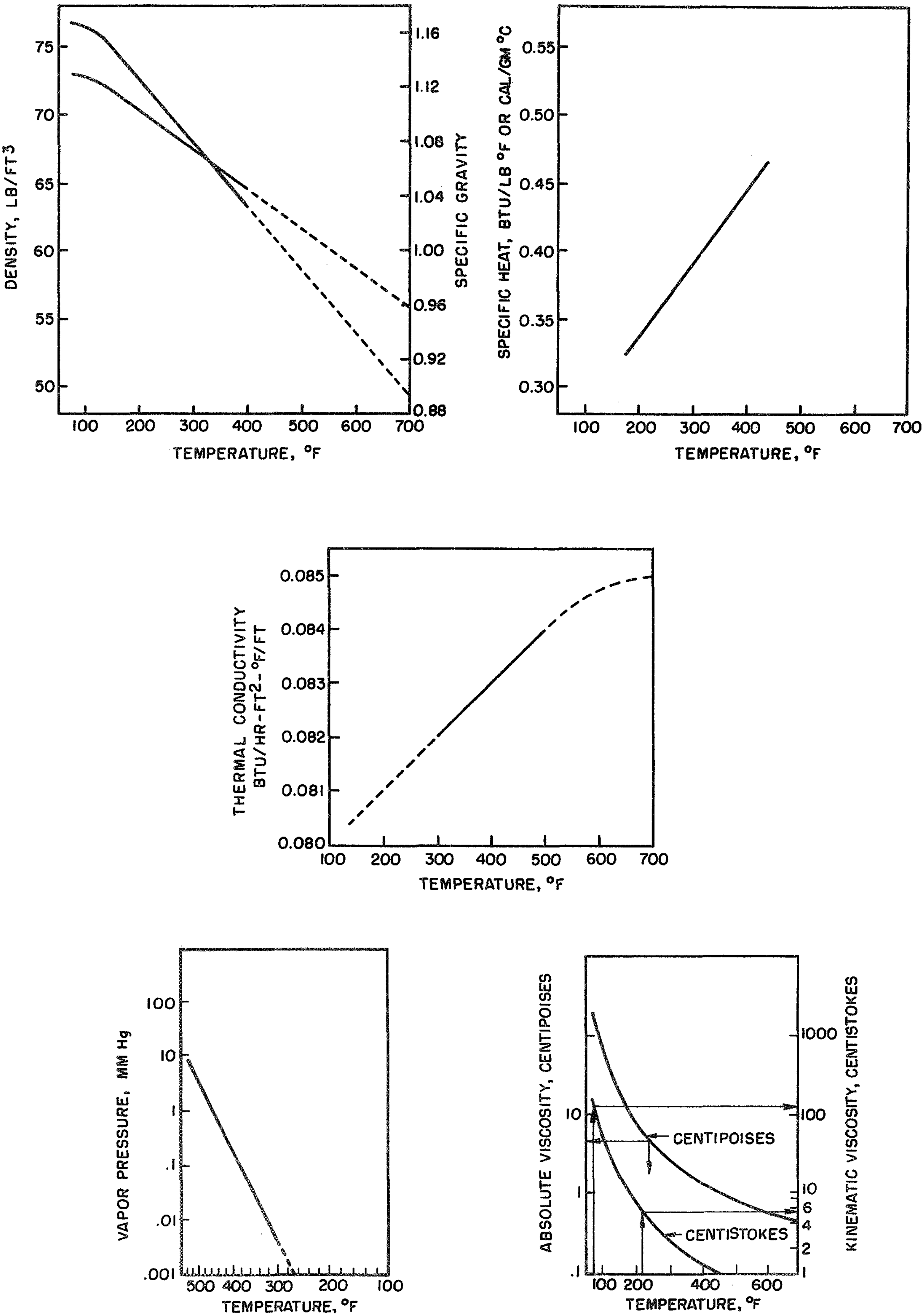


Figure 19. Mix 4P3E Characteristics

Table VI. General Properties of Mix-4P3E Polyphenyl Ether

Property	Value
Atomic Weight, AMU	354
Pour Point, $^{\circ}F$	25
Boiling Point, $^{\circ}F$	830
Thermal Decomposition Point, $^{\circ}F$	835
Density, lb/gal at $100^{\circ}F$	9.8
Density, lb/ft^3 at $100^{\circ}F$	73.0
Density, lb/ft^3 at $300^{\circ}F$	67.4
Fire Point, $^{\circ}F$	525
Flash Point, $^{\circ}F$	465
Refractive Index, at $77^{\circ}F$	1.622
Spontaneous Ignition Temperature, $^{\circ}F$	1000
Dielectric Strength, volts/mil	300
Head Height Pressure Equivalent, inches/psi at $100^{\circ}F$	23.6
Relative Volumetric Thermal Expansion Coefficient, $1/^{\circ}F$	3.052×10^{-4} at 50° to $250^{\circ}F$ 4.253×10^{-4} at 50° to $550^{\circ}F$ 4.742×10^{-4} at 50° to $700^{\circ}F$
Color	Slightly yellow
Appearance	Transparent

The calculated decomposition of Mix-4P3E is as follows:

Temp., °F	Decomposition in 10,000 Hr of Static Isotherm Exposure, percent
400	.001
500	.034
600	1.55
700	12.5
800	36.2

F. BEARING PROCUREMENT

1. Procurement Requirements

The first SNAP-8 bearings purchased were manufactured using M-50 single vacuum melt steel. The second group was fabricated from M-50 triple vacuum melt steel. The first group of bearings were not run long enough to have fatigue problems so comparison data are not presented. All of the long run testing was accomplished with M-50 triple vacuum melt steel bearings.

The requirements for procurement and inspection of M-50 tool steel bars for use in the SNAP-8 system are presented in Tables VII and VIII. The chemical composition and gas content of the material is in accordance with the requirements of Table VII. The inspection attributes and requirements are presented in Table VIII.

The bearing industry is highly standardized. Fine methods have been developed for producing a statistically good part. Since the SNAP-8 requirements were for a relatively small number of parts requiring special processing, special tolerances and very high reliability, it was necessary to customize the quality program. Superior quality control techniques were attained as the result of a combined effort between the purchaser and the manufacturer. The SNAP-8 component bearing characteristics specified for manufacturing inspection quality control are presented in Table IX. The sampling plan for the hardness test is presented in Table X.

Table VII. Chemical Composition and Gas Content of M-50 Tool Steel

Element (Alloy)	Proportion (%)	Accuracy tolerance allowable (%)(1)
Carbon	0.77 to 0.85	± 5
Manganese	0.20 to 0.30	± 5
Silicon	0.15 to 0.25	± 5
Chromium	3.75 to 4.25	± 2
Molybdenum	4.00 to 4.50	± 2
Vanadium	0.90 to 1.10	± 2
Nickel	0.00 to 0.07	± 2
Copper	0.00 to 0.08	± 5
Iron	Balance	± 10

Element (Trace)	Maximum (%)(2)
Aluminum	0.01
Arsenic	0.01
Cobalt	0.25
Columbium	0.10
Lead	0.0005
Magnesium	0.01
Phosphorus	0.01
Sulfur	0.01
Tin	0.05
Titanium	0.30
Tungsten	0.25

Element (Gas)	Maximum (%)(3)
Hydrogen	0.001
Nitrogen	0.005
Oxygen	0.005

Notes: (1) Allowable accuracy tolerance of the chemical analysis value
(2) Analysis accuracy tolerance for trace element: $\pm 10\%$
(3) Analysis accuracy tolerance for gas element: $\pm 15\%$

Table VIII. Bearing Material Inspection Attributes

Inspection Attributes	Requirements or Methods
Visual Inspection (all materials)	Bars free of heat checks, impressed scale, flakes, bursts, seams, laps, excessive segregation of carbides (banding or other defects).
Chemical Analysis (one sample)	In accordance with requirements of Table VII using STD-151, spectrochemical analysis for elements and vacuum fusion for gases.
Hardness Test (one sample)	Not less than 63 Rockwell C when heat-treated from annealed stock hardness Brinell 202 and 245.
Grain Size (hardness sample)	ASTM 5 or finer when heat-treated (molten metal, one charge from one furnace)
Metallographic Inspection (one sample)	Homogeneous structure with constituents evenly distributed, free from ingotism, excessive impurities and segregation. No decarburization permitted on material.
Micro-inclusion, Count (a sample)	Per Method A of STD ASTM-E45, not to exceed the Jernkontoret (J-K)* limits.
Macroscopic Inspection (cross-section sample)	On all parts for dimensions, identification, workmanship, cleanliness, and surface defects.
Magnetic Particle Inspection	Inner and outer rings of each bearing.
Ultrasonic Test (all materials)	Per MIL-STD-271 for forgings and wrought bars.
Macrostructure	Cleanliness per macro-etch test, equal to or better than macrograph A-2, B-2, C-2 of MIL-STD-430. Type D defects are unacceptable.
Defects	AMS-2300 maximum average frequency rating 0.50 and maximum average severity rating 1.00
Forging	See Table X
Heat Treatment	See Table X

*Swedish Ironmasters Association

Table IX. Hardness Test Sampling Plan

Sampling For Rings			Sampling For Balls		
Lot Size	Sample Size	Acceptance Level	Lot Size	Sample Size	Acceptance Level
2 to 15	2	No Defects	26 to 90	5	No Defects Permitted
16 to 25	3		91 to 150	8	
26 to 90	5		151 to 500	13	
91 to 150	8		501 to 1200	20	

Table X. Bearing Inspection Characteristics and Methods

Characteristic	Inspection Method	Gage Accuracy or Smallest Graduation
Magnetic Particle Insp.	Per MIL-I-6868	
Visual Inspect.	At least 10X magnification	
BALLS - Heat No.:	C31944 Per MIL-H-6875	
Visual Inspect.	10X to 30X using Optical Methods	
Ball Diameter	Basic	
Sphericity (Out-of- roundness)	.000005 in.	
Diameter Variation per Bearing	±.000005 in.	
Surface Finish	Using Optical Methods	
SEPARATORS		
Material - Silicon Iron Bronze	Per AMS4616	
Visual Inspect.	3X Inspection Glass	
Outside Dia.	SMI (Standard Measuring)	.0001
Inside Dia.	SMI (Standard Instruments)	.0001
Width	SMI	.0001
Ball Pockets (Nom.)	Plug Gage	.0003
Surface Finish - OD and Ball Rocket	Profilometer	
ASSEMBLED BEARING		
Visual Inspect.	3X Insp. Glass	
Ring Flushness	Standout Gage	.00005
Radial Play w/5.5 lb gage Iced	Radial Play Gage	.0001
Initial Contact Angle	Determine using axial thrust load not to exceed 5 lb manually or with fixture	

Allowable test equipment error no greater than ±20%

Characteristic	Inspection Method	Gage Accuracy or Smallest Graduation
OUTER RING -		
Outside Dia. (OD)	Indicator with stand	.00002
Out of Roundness of OD		
Taper of OD		
Width	Indicator with stand	.0001
Width Variation		
Race Parallelism	Indicator with stand	.0001
Race Curvature (Radius)	Ball/Pt. Comp	.00005
Radial Run-Out, Raceway	Concentric Gage	.00005
Raceway Run-Out with Reference Side	Surface Plate Hght, Gage w/Ind.	.0001
Reference Side Run-Out with OD	Surface Plate Hght, Gage w/Ind.	.0001
Shoulder Dia.	Ind. w/stand	.0001
Surface Finish of Race	Using Optical Methods	.0001
Surface Finish of OD	Using Optical Methods	
Surface Finish of Land	Using Optical Methods	
Magnetic Particle	Per MIL-L-6868	
Visual Inspect.	At least 10X magnification	
INNER RING -		
Bore Dia. (ID)	Air Gage	.00002
Out of Roundness of ID		
Taper of ID	Air Gage	.00002
Width	Indicator with stand	.0001
Width Variation	Indicator with stand	.0001
Race Parallelism	Indicator with stand	.0001
Radial Run-Out, Raceway	Concentric Gage	.00005
Raceway Run-Out with Reference Side	Surface Plate Hght. Gage w/Inc.	.0001
Reference Side Run Out with Bore	Bench Centers Hgt. Gage w/Ind.	.0001
High Shoulder Dia.	Indicator with stand	.0001
Low Shoulder Dia. (at face)	Indicator with stand	.0001
Surface Finish of Race	Using Optical Methods	
Surface Finish of Races	Using Optical Methods	
Surface Finish of Bore	Using Optical Methods	
Surface Finish of Land	Using Optical Methods	
Raceway Curvature (Radius)	Ball/Pt. Comp.	.0001

III. BEARING PERFORMANCE

Service testing of the bearings for the SNAP-8 rotating machinery required stringent controls and procedures to assure the validity and accuracy of the data. The inspection data recorded and the inspection methods used were as previously indicated on Tables III and X.

Upon each disassembly inspection, the bearings were reinspected for mounting dimensions, contact angle radial play and weight of the elements. The bearings were cleaned and any particulate matter obtained was examined to determine the origin. The ball tracks and other wear surfaces were studied for adverse operating conditions such as quality of lubrication, preload friction, mounting alignment, operating temperatures and unbalanced loads.

A. INITIAL COMPONENT TESTS

1. The alternator bearing and seal system was first evaluated in preprototype machines which were tested by the vendor. Inadvertent operation with cold Mix-4P3E lubricant resulted in a nonprototype bearing failure due to excessive viscous drag. Later tests employing normally heated oil proved satisfactory. The tests confirmed the lubrication and bearing system as satisfactory with respect to seal performance and friction. See Table XI.

The bearing and seal performance of the turbine was first evaluated by running the assembled turbine-alternator with hot nitrogen gas as a working fluid. Using the alternator previously stand calibrated as a load, the instrumented nitrogen working fluid enabled the turbine bearing and seal losses to be determined with accuracy. The results agreed with design predictions as to friction losses and lubrication slinger performance. Upon inspection, the bearings were found to be in excellent condition and no design changes were indicated.

When the prototype alternator, tested with the first hot mercury operated turbine, was inspected, the bearings were found to have sustained minor damage from contamination from the parts adjacent to the bearings. This problem was resolved by hand working these parts and recleaning the units. Improved clean room procedures were initiated for alternator assembly. Later inspections revealed no further significant damage from manufacturing contamination. Operation of

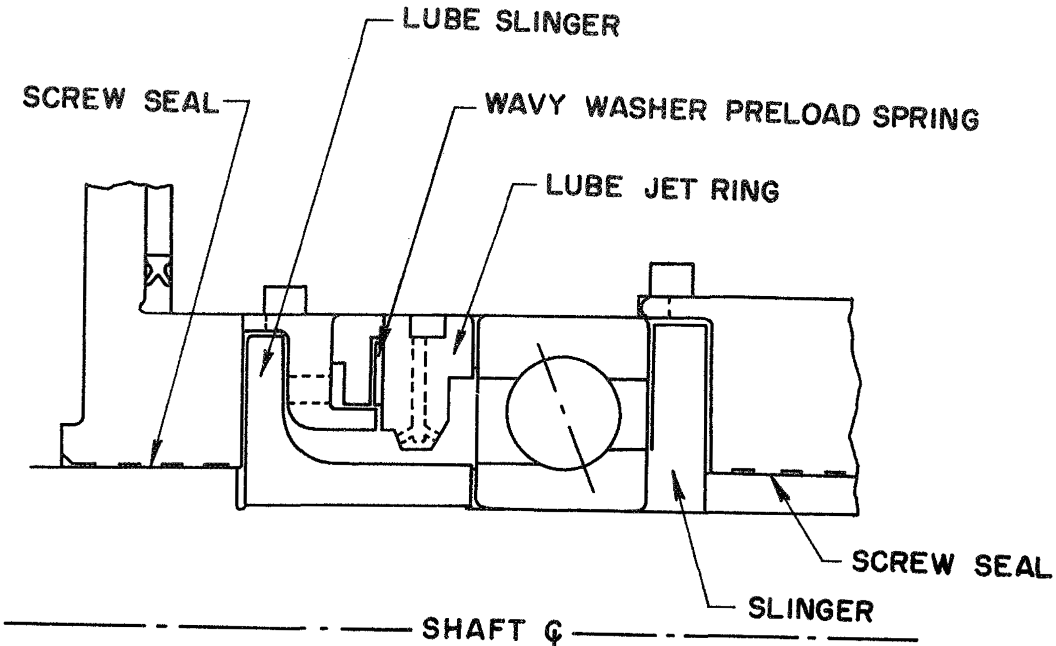
Table XI. Comparison of Thermal Requirements with Test Results for the Alternator

Item	Specification	Test*	
Cooling Oil	Polyphenyl ether	Polyphenyl ether	
Flow, gpm	2.84	2.84	
Inlet pressure, psia	-	21.9	
temperature, °F	205°F	205°F	
Outlet pressure, psia	-	8.3	
temperature, °F	-	230°F	
ΔP, psid	14 psi max.	13.6	
		**DE	***ADE
Bearing			
Inlet flow, gpm	0.35	0.338	0.320
pressure, psia	20 psia ± 1 psi	20.2	20.2
temperature, °F		196	198
Outlet flow - inboard, gpm		0.155	0.138
flow - outboard, gpm		0.175	0.176
pressure - inboard, psia		6.7	6.5
pressure - outboard, psia		6.7	6.7
temperature - inboard, °F		225	248
temperature - outboard, °F		259	258
Bearing cavity pressure, psia	260°F	0.8	0.8

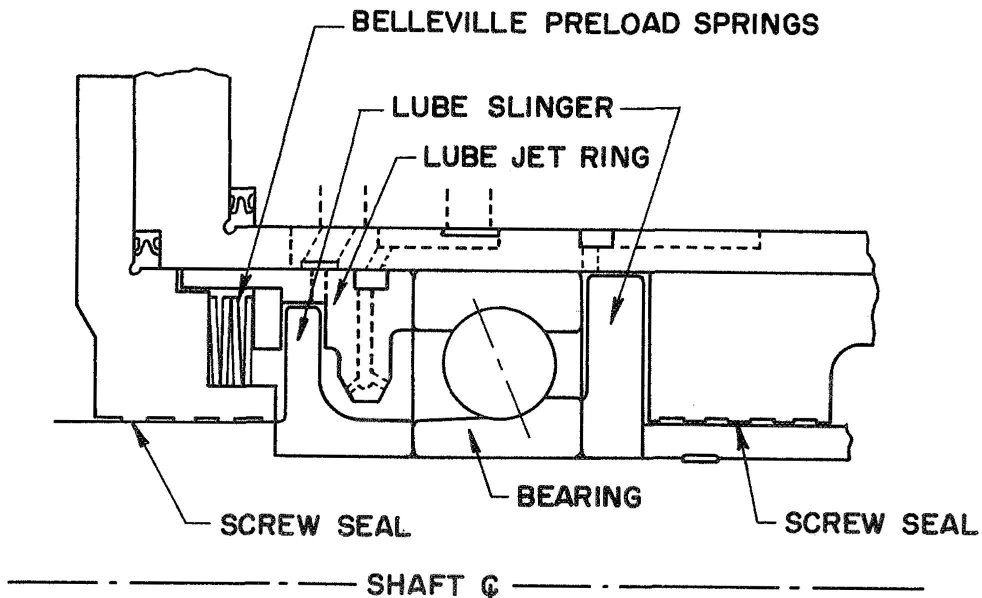
*Based on Prototype Serial Number 481489.

**DE = Drive End.

***ADE = Antidrive End.



Current Design



Proposed Design

Figure 20. Alternator Bearing Spring Preload System

turbine-alternators in the hot-mercury vapor test facility resulted in two alternator failures which were attributed to a seizure of the rotor in the screw seal elements.

Analyses of these failures led to the conclusion that the problem was due to a loss of preload in the alternator bearing system whereby the sliding parts had hung up in the housing. The consequent increase in radial play allowed the rotor to orbit and run against the stationary seal elements. These design deficiencies led to a design modification effort to improve the system which included replacing the present wavy sporing preload washers with belleville sporings and increasing the length-to-diameter ratio to permit better sliding. A comparison between the present design and proposed new design is shown by Figure 20.

The mercury pump was first run in a mercury pump test loop. No bearing problems were uncovered during operation or in the post test examination. Bearing seal losses were determined from operation without mercury flow. Test results agreed with design predictions (See Figure 21).

B. COMPONENT ENDURANCE TESTS

1. Bearings - Turbine-Alternator

The final SNAP-8 turbine-alternator was assembled for endurance testing in May 1967. The first series of test runs totaling 2133 hours was concluded on November 16, 1967. The turbine-alternator was then disassembled, inspected, reassembled, and returned to test with the same bearing set. The second series brought the accumulated total running time to 10,823 hours. The condition of some typical balls from this turbine is shown by Figure 22. The planned shut-down was made to permit examination of the complete turbine-alternator and to assess the potential life. The bearing characteristics were checked for this unit and are presented in Tables XII and XIII. The inspection data is compared with two previous data points at zero hours and 2123 hours of operation. Additional testing raised the total unit time to 12,800 hours with satisfactory operation but no final inspection was made due to termination of the SNAP-8 Contract.

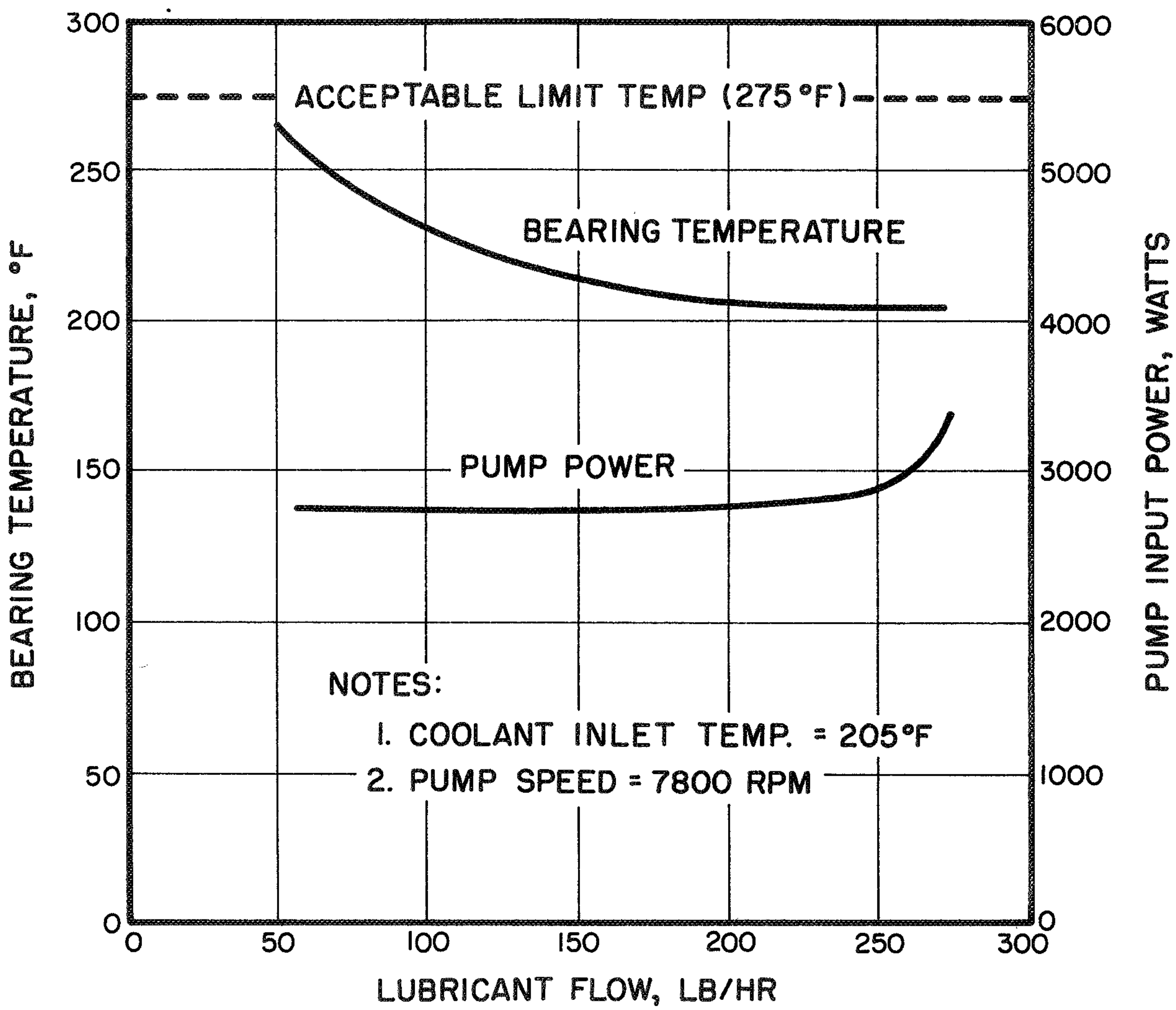


Figure 21. Mercury Pump Bearing Lubrication Flow Optimization

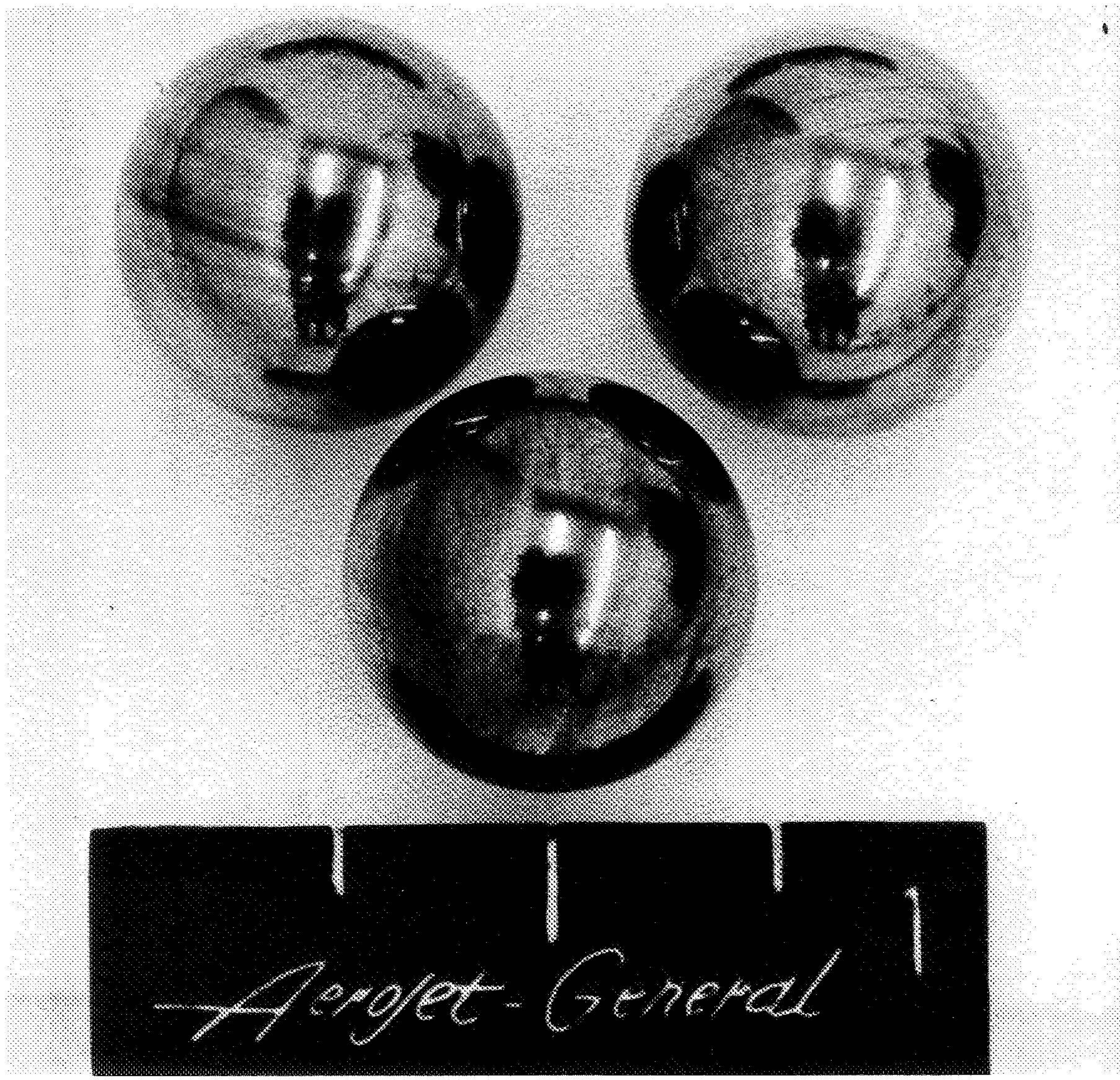


Figure 22. SNAP-8 Turbine Alternator Assembly, Unit 5/4 P/N 096800-5
After 10,822 Hours Operation
Turbine End Bearing P/N 095355, S/N A-52
Three Typical Balls

Table XII. Turbine Bearing Characteristics After Endurance Tests

Turbine End (Bearing S/N A52)

Service Time, hours	0	2,123	10,823
Diametrical Clearance, in.	0.00165	0.0016	.0016
Contact Angle, degrees	15.9°	N/A	15.47
Ring Flushness, in.	0.0001	0.0005	0.0008
Inside Diameter, in.	1.57465	1.57468	1.57468
Outside Diameter, in.	3.14948	3.1494	3.1494
Ball Roundness, micro-inches	5	Not Taken	7-13

Alternator End (Bearing S/N A41)

Diametrical Clearance, in.	0.00195	0.0018	0.0017
Contact Angle, degrees	15.3°	N/A	16.1
Ring Flushness, in.	0.0001	0.0004	0.00042
Inside Diameter, in.	1.57475	1.57475	1.57472
Outside Diameter, in.	3.14948	3.1494	3.1494
Ball Roundness, micro-inches	5	Not Taken	5-25*

N/A = Not Available

* = High value was due to a local scratch (not ovality).

Table XIII. Alternator Bearing Characteristics After Endurance Tests

<u>Antidrive End</u> (Bearing S/N A39)				
Service Time, hours	0	2123	10,823	
Diametrical Clearance, in.	0.00175	0.0017	0.0017	
Contact Angle, degrees	15.1°	N/A	14.73°	
Ring Flushness, in.	0.00005	0.0005	0.00022	
Inside Diameter, in.	1.57480	Not Taken	1.57480	
Outside Diameter, in.	3.14947	Not Taken	3.149527	
Ball Roundness, micro-inches	5	Not Taken	3-23	
<u>Drive End</u> (Bearing S/N A42)				
Diametrical Clearance, in.	0.0017	0.0017	0.0016	
Contact Angle, degrees	15.1°	N/A	15.47°	
Ring Flushness, in.	0.0001	0.0005	0.0004	
Inside Diameter, in.	1.57470	Not Taken	1.574680	
				1.574690
Outside Diameter, in.	3.14945	Not Taken	3.14949	
Ball Roundness, micro-inches	5	Not Taken	5-12	

N/A = Not Available

The evidence showed that the lubrication for all the bearings was good. The bearings were free of dirt, pits, and abrasive wear. The bearings showed no signs of having over-heated, and the varnish and stains noted (typical of parts operated in Mix-4P3E) were very minor compared to the amount necessary to cause functional difficulties in operation.

No difficulties were encountered with the turbine bearings mounting system. The bearings were free from misalignment, large unbalance loads, and creeping of races on the shaft and in the housing. The ball path positions indicated proper functioning of the spring preload system at all times. No indication of operation at low contact angles was observed.

The alternator mounting system had been previously judged as requiring revision. No evidence of mounting system damage was found in the bearings. However, the hangup of sliding parts which resulted in limited end play of 0.006-inch on disassembly versus 0.013-inch on assembly buildup indicated poor alignment in bearing mounting. In this case, the permanent set of one lobe of the wavy washer preload spring could have cocked and jammed the bearing to limit rotor end play. Since the ball tracks in the raceway were located in the normal position, it appeared that the raceway hangup must have occurred during the handling associated with turbine-alternator removal from the loop and disassembly.

The out-of-roundness of several balls was the only significant factor affecting bearing life. Unequal load distribution due to out-of-round balls will accelerate fatigue failures in highly stressed elements. Most of the balls inspected were suitable for extended running. The worst ball was out-of-round by 23 micron-inches. The original tolerance was 5 micro-inches. The out-of-roundness was caused by wear against the edges of the ball separator. Minor improvement of the separator design is expected to alleviate the problem.

The differences in tabularized contact angles before and after testing are attributed to different inspection methods at the vendor and at Aerojet, and are not considered to be of significance. The small changes in inner ring bores are either due to ring growth or inspection discrepancies. In either case the bearing operation has not been affected.

2. <u>Bearing - Mercury Pump</u>

The mercury pump was run in the liquid mercury facility for 12,227 hours of operation with 109 starts. The mercury pump was disassembled after endurance tests and the bearings examined. The bearings appeared to be darkened with varnish or "cooked" lubricant stains. No significant amount of debris was found in the bearing assembly. The bearings were disassembled and checked. The characteristics are presented in Table XIV.

The bearings from this unit were in good condition and suitable for extended service. Wear on rolling and rubbing surfaces was extremely slight. However, some detracting evidence was noted. Minor brinelling was attributed to mishandling during assembly. Also, bearing elements were coated with a light stain or varnish which, if allowed to progress sufficiently could impair heat transfer and possibly prevent sliding of the bearing outer rings causing loss of preload. Figure 23 shows the bearing condition after endurance testing.

The excellent condition of rolling and sliding contact surfaces demonstrated the suitability of the Mix-4P3E lubricant for this type of ball bearing application.

The heat stains and stickiness of some of the balls indicated high shutdown temperatures or soak back. This occurs when the pump is shut down and the lubricant flow is interrupted allowing the heat of the stator and rotor to reach the bearings. The pattern of heat stains appeared to be formed from a meniscus of lubricant drying between the rolling elements. In addition, the outer ring had a dried out "bathtub-ring" pattern in an area which should be completely flooded by the discharge slinger during lubricated operation. The reduction in measured diametral clearance is associated with inner ring growth and this would have the effect of decreasing the contact angle also.

The growth of the inner rings (0.0002 in.) and the tempering of all the rings (from 63.5 to 62.5 R_C) are conditions found in many long-duration, high-temperature bearings. Metallurgical data indicate that the material should not temper below 700°F and retained austenite should be insignificant in these parts. However, the transformation of any retained

Table XIV. Mercury Pump Bearing Characteristics After Endurance Tests

	Motor End		Pump End	
Service Time, hours	0	12,227	0	12,227
Dimensions				
Inside Diameter, in.	1.37788	1.37810	1.37783	1.37800
Outside Diameter, in.	2.83449	2.8345	2.83449	2.83452
Width, in.	0.6689	0.6689	0.6693	0.6690
Diametrical Clearance, in.	0.0018	0.0016	0.0016	0.0014
Ring Flushness, in.	0.0001	0.0002	0.00005	0.0003
Ball Roundness (Ave.), in. (TIR)	5×10^{-6}	10×10^{-6}	5×10^{-6}	10×10^{-6}
Contact Angle, degrees	16.0	16.5	14.75	13.0
Ring Hardness, R_C				
Outer Ring	64	62.5	63.5	62.5
Inner Ring	64	62.5	63.5	62.5

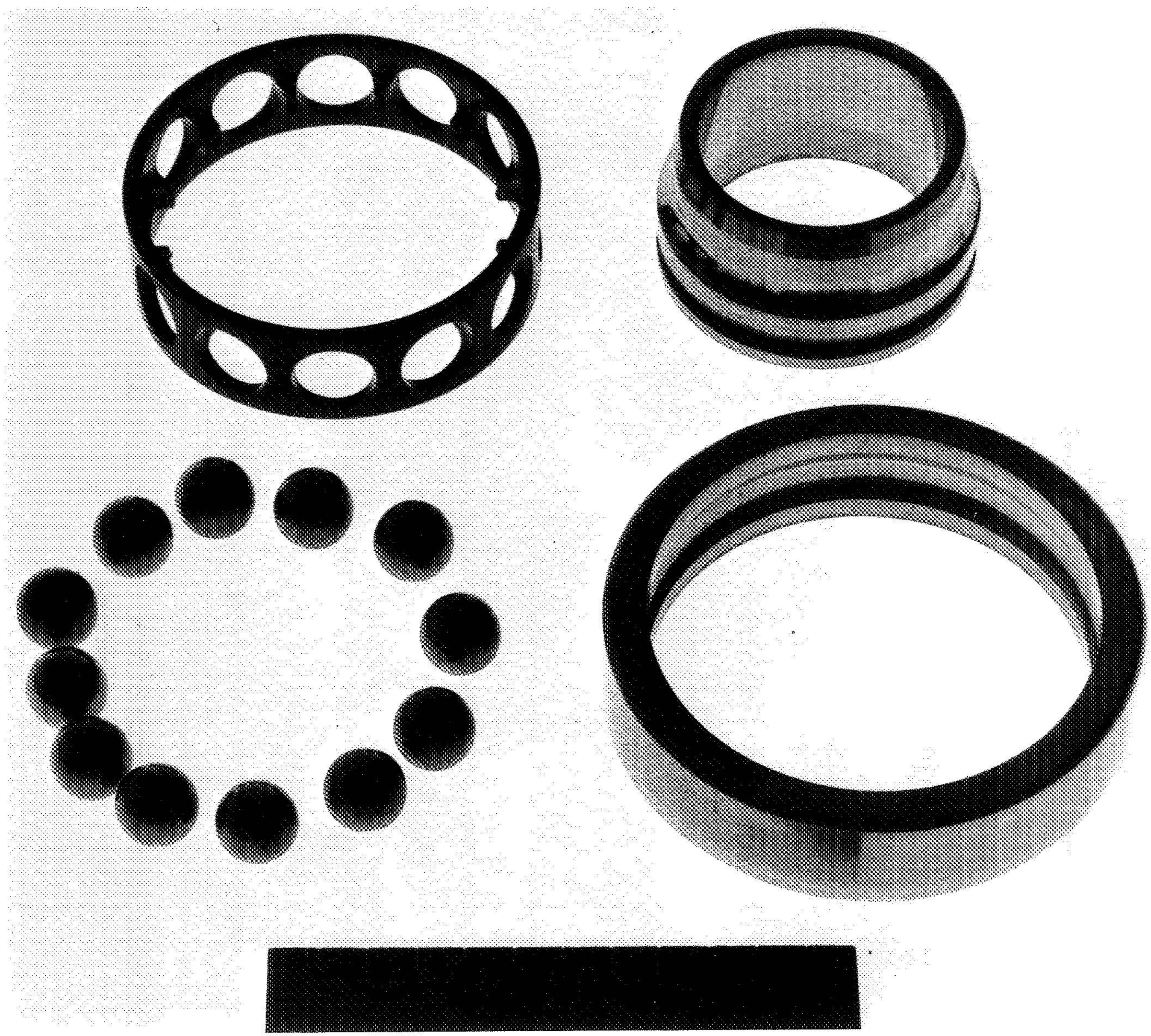


Figure 23. Motor End Bearings After 12,227.5 Hours Operation

austenite would tend to increase the ring size. In addition, tempering would tend to increase the ring size. Also during operation, temperature differentials between shaft and bearing will increase the interference fit. This will produce hoop stresses in the inner ring of approximately 7500 psi for 0.005-inch interference. Since the dimensional change is less than .01%, there is the possibility of creep being the cause for growth even though it is not normally considered to occur at these stresses in a very hard material.

3. Bearings - Alternator

The alternator was also endurance tested in an Electrical Component Test Facility (ECTF) without the turbine-hot-end and employing a motor-gear box drive system. The drive adapter employed was a turbine bearing housing and shaft. The unit was first installed in 1966. After 2553 hours an inspection was made, at which time the drive end bearings, the drive end shaft components, and the adjacent drive adapter parts were found to be corroded. This was attributed to the ingestion of moisture laden air into the unit during long shutdown periods. The unit was reassembled with a new drive end bearing in the alternator and a new inboard drive adapter bearing. The other bearings had minor corrosion damage only. A nitrogen cover gas system was incorporated to provide protection for the alternator during long periods of down time.

After 12,679 hours, the drive adapter only was inspected and the outboard bearing was found to have become pitted at the site of the heaviest corrosion spot on one of the balls. This bearing was replaced. The other adapter elements were satisfactory and reassembled.

The unit was returned to test. At 23,130 total hours the test was terminated due to rotor seizure at the drive and inboard screw seal. Sludge which was deposited at the oil slinger discharge groove in the housing prevented the bearing outer ring from sliding and following the rotor. The increase in diametral play permitted rotor orbiting resulting in sufficient bearing and rotor deflections to permit the rotor to contact the stationary seal. This failure was similar to those previously experienced and described in Section III, A1 (Also See Figure 20.)

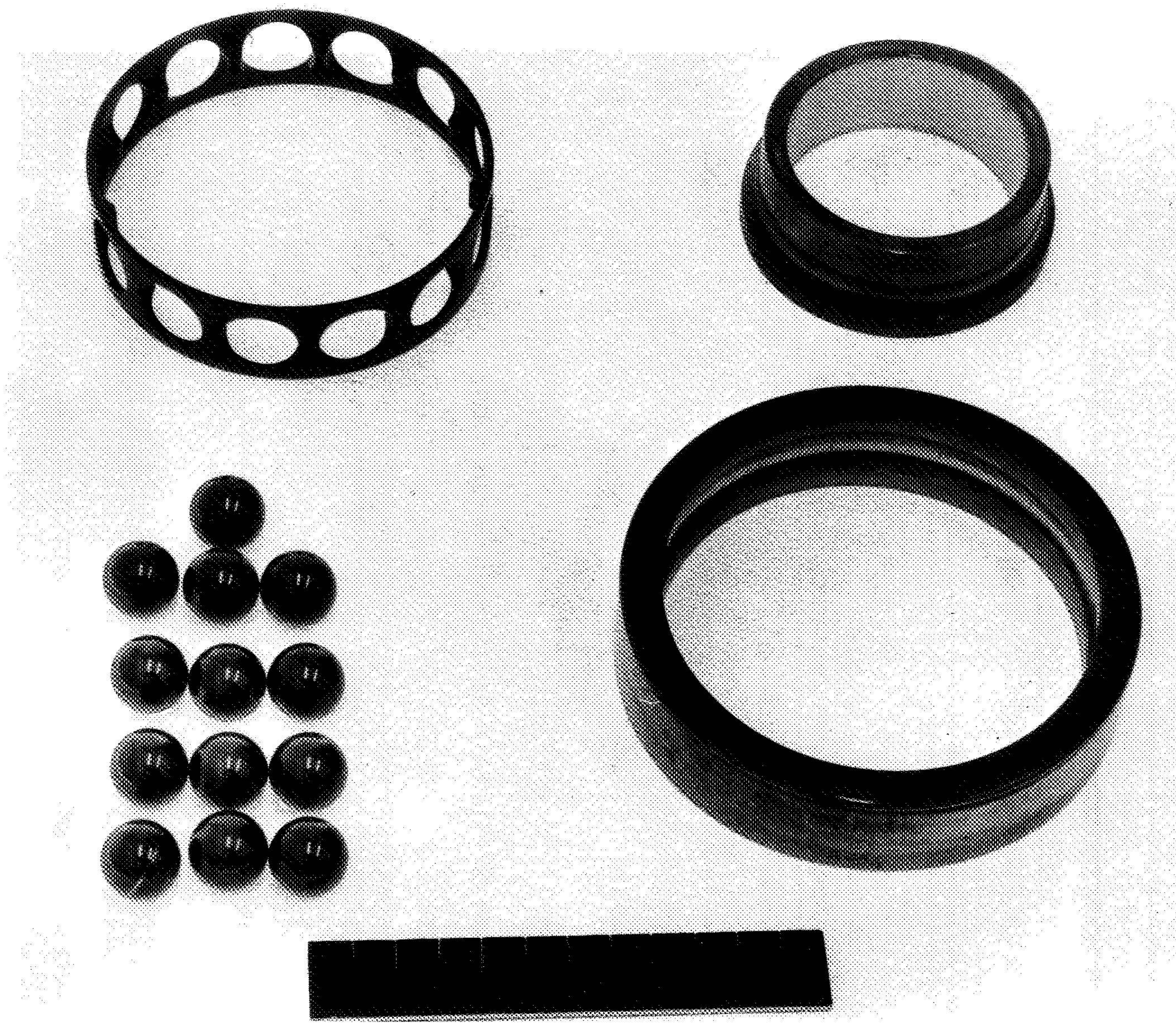


Figure 24. SNAP-8 Alternator P/N 094069 S/N 481490 after 23,130 Hours in ECTF
Ball Bearing P/N 095355 S/N A-54

The drive and bearing which was adjacent to the seizure area had some damage from debris but was fully functional with respect to extended usage.

Other than the light damage caused by the inadvertent shut-down and consequent debris entering the drive-end bearing, the bearings were judged to be in excellent condition and capable of considerable extended service. Figure 24 shows the typical condition of the bearings following this endurance test.

Table XV shows a comparison between principle bearing measurements taken at the end of the tests compared to earlier data. The major differences are in an apparent tempering of the rings with slight reductions in hardness.

As described under B.2, this condition is not uncommon in long-duration high temperature bearings. The tempering does not appear to have had any affect on the bearing wear life.

A summary of all major ball bearing test experience is shown in Table XVI.

Table XV. Electrical Component Test Facility Bearing Data

Alternator Drive End Bearing, S/N A54

Unit Service Time, hours	0	2553	23130
Diametrical Play	.00165		
Ball Diameter	.47068	.47055	.47044
Ring Flushness	.0005		.00015
Contact Angle	15° 53'		15° 30'
Outer Ring Hardness	63.6 Rc		62.3 Rc
Inner Ring Hardness	64 Rc		61.7 Rc
Outer Ring O.D.	3.14942		3.14950
Inner Ring Bore	1.57470		1.57470

Alternator Anti-Drive End Bearing, S/N A48

Unit Service Time, hours	Bearing installed at 2553 hours after original was rust damaged.	2553	23130
Diametrical Play		.00165	.0017
Ball Diameter		.47068	.47047
Ring Flushness		.0001	(.0004)
Contact Angle		14° 53'	15° 50'
Outer Ring Hardness		64 Rc	62.7 Rc
Inner Ring Hardness		63.6 Rc	62.4 Rc
Outer Ring O.D.		3.14951	3.14950/3.14957
Inner Ring Bore		1,57470	1.57474

Table XVI. SNAP-8 Ball Bearing Service History

Serial Number	Component	Position	Total Time (hr)	Last Inspection (hr)	Condition at Last Inspection	Remarks
A-41	Turbine	Drive end	12,200	10,823	Good	Minor loss of radial play and increase in stick out
A-52	Turbine	Antidrive	12,200	10,823	Good	Same as above
A-42	Alternator	Drive end	12,200	10,823	Good	Same as above
A-39	Alternator	Antidrive end	12,200	10,823	Good	Same as above
A-35	Alternator	Drive end	1,500	0	New	-
A-31	Alternator	Antidrive	1,500	0	New	-
A-54	Alternator	Drive end	18,500	0	New	-
A-48	Alternator	Antidrive end	21,000	2,500	Good	Minor rust damage
A-49	Turbine	Drive end	1,400	1,400	Excellent	-
A-46	Turbine	Antidrive end	1,400	1,400	Fair	Cracked wheel caused heavy unbalance load
A-2	Hg Pump	Pump end	12,455	12,227	Good	-
A-10	Hg Pump	Motor end	12,455	12,227	Good	-
A-9	Hg Pump	Motor end	2,250	2,250	Fair	Damage by debris from visco pump failure
A-7	Hg Pump	Pump end	2,250	2,250	Fair	Same as above
A-3	Hg Pump	Motor end	9,700	8,252	Good	-
A-14	Hg Pump	Pump end	8,252	8,252	Fair	Scratched during assembly

IV. RECOMMENDATIONS FOR EXTENDED LIFE IMPROVEMENTS

The SNAP-8 turbine and mercury pump bearings meet the initial design objectives. However, consideration should be given to increasing the contact angle to 20-25° on bearings that take the heavy thrust loads for both turbine and mercury pump. The present bearing, with 16° contact angle, would be used in the low thrust load positions in the mercury pump and turbine. The additional thrust capacity can extend the calculated life of the mercury pump bearings to 5 years.

The effect of increasing the contact angle, adding an additional bearing, and increasing the load would be to raise the dynamic load capacity above the present loading conditions. However, the net actual load would still be less than the original design load. The calculated load capacity of the bearing would drop 5.5%, but the 26.5% reduction in new design load versus original design load would more than offset this. Bearing calculated life would increase by a factor of 2.13 times the original design value.

In the event that loading is increased or the life requirement extended, the bearing fatigue life can be increased by incorporating the following improved bearing techniques:

- Bearing material fiber orientation - use preferred orientation of fiber flow in bearing ring.
- Surface pre-stressing of bearing, mechanically or by heat treatment (Marstressing).
- Use of ausforming process (strain induced carbide precipitation).
- Modify separator design to reduce incidence of ball scratching and wear.

Although not directly concerned with bearing design, the spring preload system in the alternator has shown deficiencies which also should be corrected in accordance with Figure 20, as discussed in Sections III-A1 and B1.

V. CONCLUSIONS

The objective of mounting the SNAP-8 turbine alternator and mercury pump on conventional ball bearings has been successfully achieved. Secondary bearing mounting system failures have been experienced when the operational condition exceeded the design condition, but the conservative design has prevented gross bearing damage and therefore aided in identifying the problems as they occurred.

In all cases the bearings used for endurance testing of the turbine-alternator and mercury pumps have demonstrated a sound design selection with no evidence of impending fatigue damage. Additionally, the absence of critical wear on the separator, balls, and raceways is indicative of good and adequate lubrication with good elastohydrodynamic operation. Even though measurements show relatively light wear, a life projection was made based on the maximum ball out-of-roundness (See Figure 25). This shows that based on an allowable ball out-of-roundness of 50 millionths of an inch life projections of 5 years or greater can be expected.

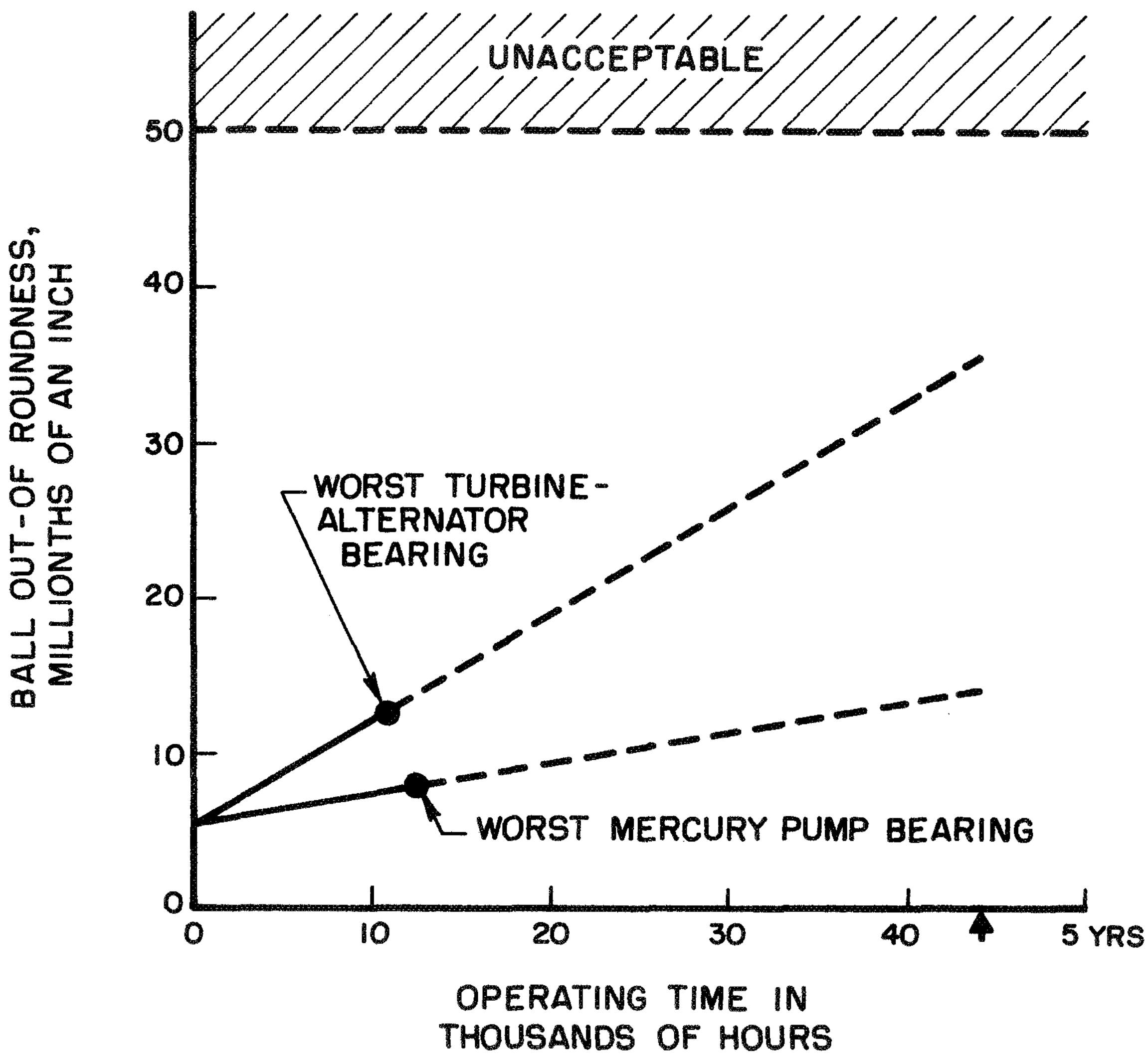


Figure 25. Projected Life of SNAP-8 Rolling Contact Bearings
Based on Ball Wear

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