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DESIGN AND DEVELOPMENT OF SNAP-8 MERCURY PUMP MOTOR ASSEMBLY

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TOPICAL REPORT

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FOREWORD

The design and development effort described in this report was performed by the Mechanical Systems Operations, 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. Acknowledgement is given Messers. H. O. Slone and A. Stromquist of NASA-Lewis Research Center for their guidance and assistance.

ABSTRACT

This report describes the design and development of a mercury pump and motor for use in the SNAP-8 Electrical Generating System while in a space environment. The pump, a combination jet-centrifugal type, is driven by a motor with oil-lubricated ball bearings. A dynamic, shaft seal-to-space is used to separate the mercury and the lubricant oil.

The pump has been under development since 1963. A total of 6 units have been built and operated for an accumulated testing time of 22,213 hours; the longest test time on one unit was 12,227 hours. The pump was considered to be successful and exceeded the specification requirements for a 10,000 hour life. The only complications encountered were in the seal-to-space. The pump development is currently being continued as of December, 1968.

The current goal is to extend the pump design life from 10,000 hours to approximately 40,000 hours. The potential problems that need to be resolved in order to extend the pump life include cavitation damage on the back side of the impeller hub and on the mercury Viscoseal.

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CONTENTS (cont.)

SUMMARY

A mercury pump-motor assembly was designed and developed as a boiler feed pump for the SNAP-8 nuclear turbo-electric space power system. The design was started in 1963, and the development is continuing at the present. The pump motor assembly is required to pump mercury at a temperature of approximately 500°F, continuously for 10,000 hours within the performance envelope. The pump motor assembly, except for the seal-to-space vent ports, is a self-contained, hermetically-sealed unit. The pump motor assembly, hereafter referred to as the "mercury pump", contains a centrifugal pump, low-leakage dynamic seals, and an induction motor mounted on a single shaft and supported by angular contact ball bearings. Polyphenyl ether (Mix 4P3E) is used as a motor and space-seal coolant and as a lubricant for the bearings. Integral with the assembly are a jet pump and lubricant-coolant valves with associated plumbing. The jet pump is used to increase the mercury pump suction performance during startup when the system pressure is low.

This report describes the work accomplished during the pump motor design and development conducted at Aerojet-General Corporation during the period from 1963 to 1968. The report includes the more significant details of the analysis and design including the highlights of the development progress up to December 1968.

During this period six different mercury pumps were tested for a cumulative test time of 21,213 hours. One pump was tested for 12,227 hours and was subjected to 109 startups during this test. The test results verified that the short-term performance of the mercury pump was approximately as designed. The mercury pump motor assembly efficiency was 12.7% with a jet pump hydraulic efficiency of 12.5% and a combined jet-centrifugal pump hydraulic efficiency of 29%.

SUMMARY (cont.)

The only significant problem encountered during development was the rupture of the bellows in the carbon face seal lift-off device. The shaft seals have a carbon face seal with a lift-off device to reduce wear during steady state operation. The lift-off device consisted of a bellows actuated with a high pressure fluid. Originally, the actuating pressure was supplied by mercury from the pump discharge. In this mode of operation, the "water-hammer" effect of the mercury ruptured the actuating bellows. Subsequently, 200 psi nitrogen was used to supply the actuating pressure. However, the problem was not completely solved, and there were still instances of bellows failure. A new lift-off device was designed, and is presently being developed for incorporation into the mercury pump design.

A mercury pump post-test analysis after the 12,227-hour test indicated that there were two problems which might limit the mercury pump life to less than 5 years. One was cavitation damage of the space seal parts and of the pump impeller hub next to the back vanes. The other problem was fouling of the visco pump grooves by mass transfer deposits. Solutions to these problems which are currently under study, should not require major changes in the pump design. The mercury pump is considered developed, since it has met the original performance and the 10,000 hour endurance requirements. Testing is currently being conducted on two mercury pumps which are operating within the power conversion system to achieve a five year life within specification performance limits.

I. INTRODUCTION

The systems for nuclear auxiliary power (SNAP) program was initiated by the National Aeronautics and Space Administration to develop a series of power systems for use in outer space. SNAP-8 was the designation of the program that was awarded to the Mechanical Systems Operations of the Aerojet-General Corporation under Contract NAS 5-417. The primary objective of the SNAP-8 Program was to develop a system which would convert heat from a nuclear reactor into 35 kw of useful electrical power. The components were to have a design life of 10,000 hours (unattended), and the design was to reflect the then-current technology.

The SNAP-8 system design utilizes a mercury Rankine-cycle for the conversion of heat into electrical energy as shown in Figure 1. A mercury pump is used to force liquid through a boiler which vaporizes the mercury and provides the force to drive the turbine. The turbine then drives the alternator which generates the electrical power. The turbine exhausts to a condenser and the liquid mercury then returns to the pump. The mercury condensate enters the pump at a flow of 1.8 gpm, an inlet temperature of 505°F and pressure of 10.5 psia. The mercury pressure increases to 498 psia at the pump outlet with a resulting head rise of 87 feet across the pump.

The primary design considerations of the pump were the selection of materials compatible with the hot mercury, the designing of a shaft sealing system which would prevent the mercury and the lubricant from mixing with one another or to leak out into space, and the designing of a set of components which would perform without failure for 10,000 hours. Evaluation of several tested pumps gave strong indications of an expected life considerably in excess of the 10,000 hours.

The basic mercury pump design is a single shaft machine consisting of a centrifugal impeller type pump with a jet pump at the inlet to boost the pump suction pressure during startup and during steady state operation to enhance the impeller pump suction performance (Figure 2). The pump is

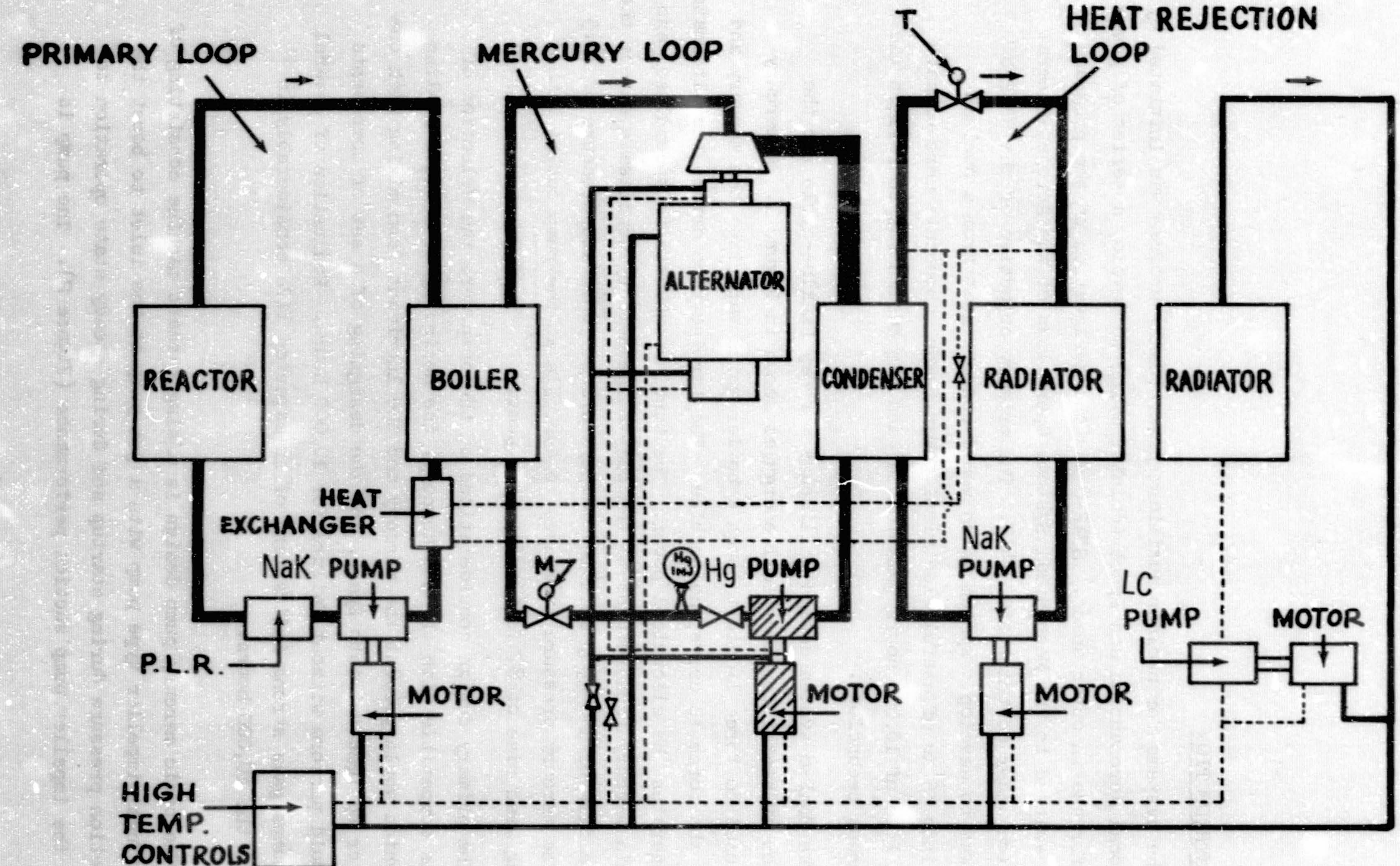


Figure 1. SNAP-8 System Loop Schematic Showing Location (Crosshatching) of the Mercury Pump-Motor Assembly

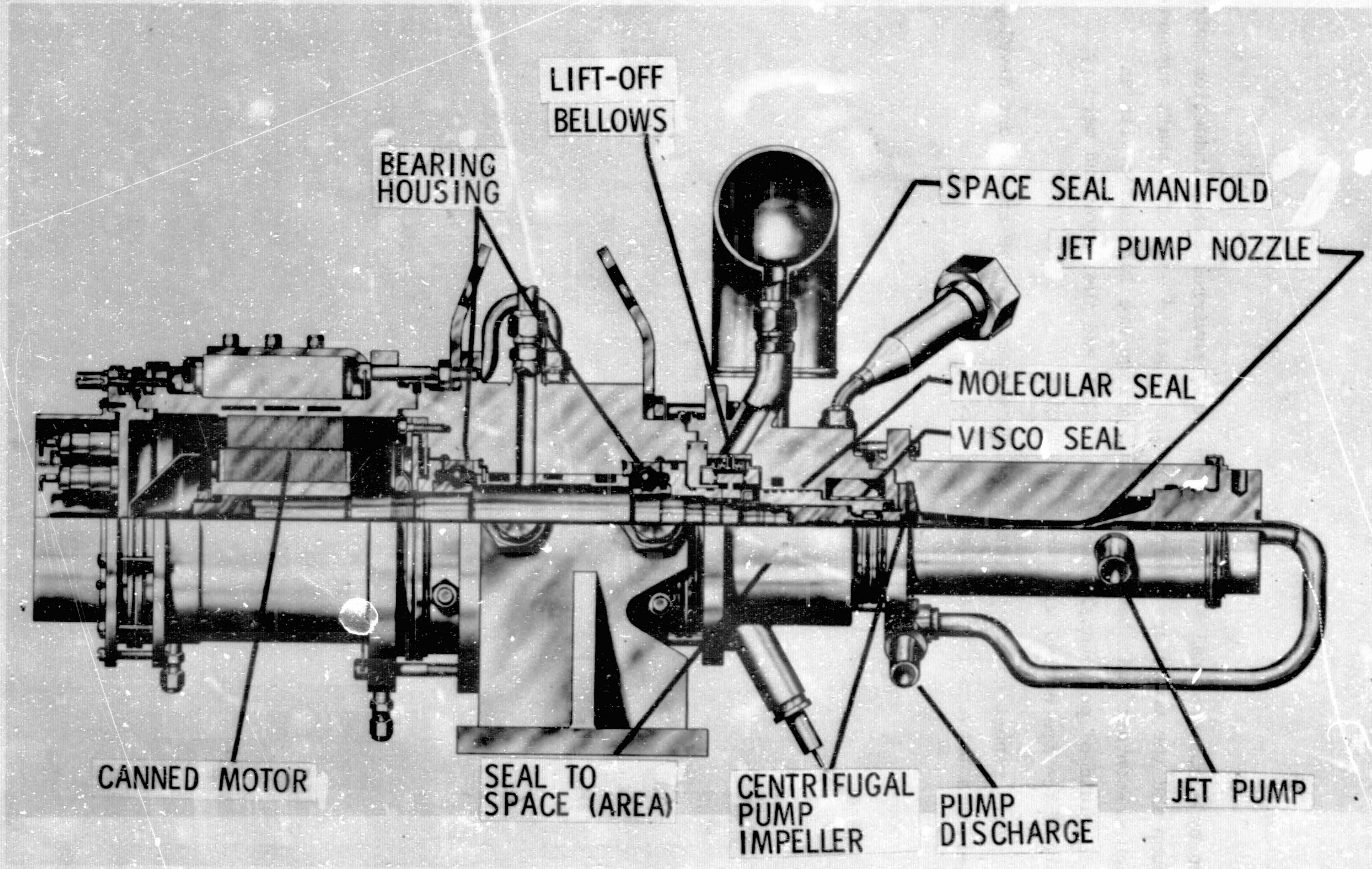


Figure 2. Cutaway Drawing Showing Design Details of the SNAP-8 Mercury Pump-Motor Assembly

driven by a conventional 400 Hz, three-phase, squirrel-cage, induction motor. The pump and the motor are overhung on the opposite ends of a shaft supported by angular-contact ball bearings. The bearings are lubricated with an organic fluid of polyphenyl ether (Mix-4P3E), and the motor and mercury seal to space are cooled with the same fluid.

This report covers the design of the pump and the subsequent development efforts between 1963 and December 1968.

II. MERCURY PUMP DESIGN

The general philosophy used in the design of the pump was one of conservatism. Where possible, design margin was used so that maximum flexibility of operation could be obtained. A mercury pump design for long life led to the selection of a centrifugal-impeller type pump. This type was selected to avoid the life-limiting wear associated with pumps of the positive-displacement type. A centrifugal pump, unlike a positive-displacement pump, contains no basic rubbing parts in the pumping elements. Another centrifugal pump advantage is its adaptability to slight changes in performance requirements. New requirements can be imposed without necessitating changes in the basic centrifugal pump design or operating conditions. The mercury pump design was required to be within the current technology (1963). The following were the nominal performance requirements on which the mercury pump design was based:

Characteristic	
Service Fluid	mercury
Flow, gal/min	1.8
Head rise across pump, ft	87
Suction (inlet) temperature, $^{\circ}F$	505
Minimum Net Positive Suction Head (NPSH) available (during start transient at 40% rated flow), ft	0.14
Minimum efficiency (Hydraulic output/ Electrical input), %	11
Life, hours	10,000

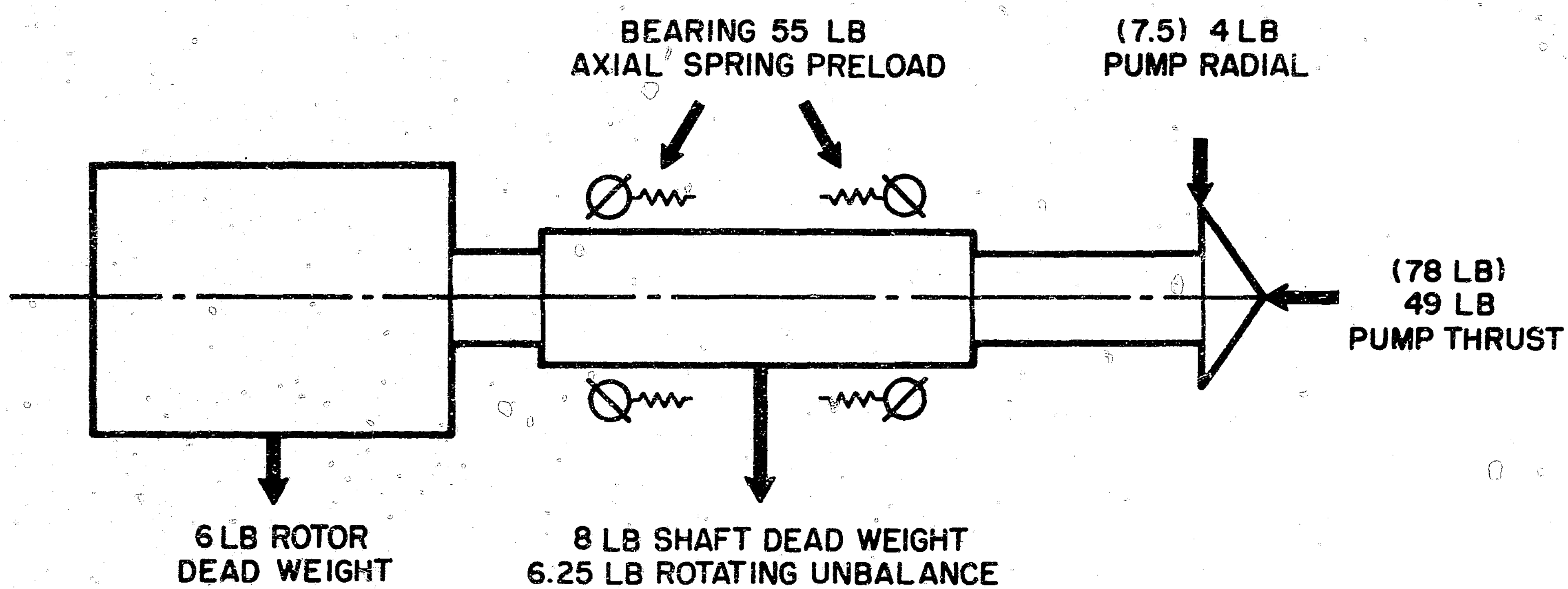
Materials in direct contact with mercury were selected for their resistance to attack by hot liquid mercury as well as for their strength and creep properties at the pump operating temperatures. The materials of construction of the more important elements of the pump and motor are as shown in Table I. A comparison of mercury pump efficiencies is shown in Table II. The mercury pump loads are as shown in Figure 3.

TABLE I MERCURY PUMP MOTOR ASSEMBLY LIST OF MAJOR MATERIALS

Pump Element	Material	
Shaft	AISI 4340	Alloy Steel
Dynamic Seals	AISI 4340	Alloy Steel
Bearings	M50-CEVM (Triple Vacuum Melt)	Tool Steel
Bearing Housing	9Cr-1Mo	Alloy Steel
Pump Housing	9Cr-1Mo	Alloy Steel
Motor Housing	9Cr-1Mo	Alloy Steel
Pump Jet Nozzle	9Cr-1Mo	Alloy Steel
Pump Suction and Discharge Lines	9Cr-1Mo	Alloy Steel
Pump Impeller	9Cr-1Mo	Alloy Steel
Motor Stator Laminations	M19	Silicon Steel
Motor Rotor Laminations	M19	Silicon Steel
Rotor Conductor Bars	Silver	
Motor Stator Windings	Copper	
Motor Insulation System	Polyimide "ML"	Organic

TABLE II MERCURY PUMP EFFICIENCIES

	Efficiency, %			
	Jet Pump	Cent. Pump	Jet-Cent. Pump	Pump Motor Assembly
Required	-	-	-	11
Expected	15	41	22	10.6
Achieved	12.5	49	29	12.7



PUMP LOADS ARE COMPUTED FROM TESTED PRESSURE PROFILES
(VALUES IN PARENTHESIS ARE ANALYTICAL)

Figure 3. Mercury Pump Loads

A. CENTRIFUGAL PUMP

An analysis of the mercury pump requirements indicated that the centrifugal pump should be designed for as high a rotating speed as possible to obtain maximum efficiency. The maximum speed that could be used, however, was dependent upon motor and bearing limitations. The centrifugal pump was, therefore, optimized with respect to the motor and bearing designs. A 7,800 rpm centrifugal-pump speed was selected on the basis of the motor's 400 Hz input frequency.

1. Impeller Design

Reference 1 describes the standard industry approach in the design of a centrifugal-pump impeller for commercial pumps. Utilizing this approach, the generalized parameter specific speed was used to classify impeller pumps by their hydraulic characteristics. "Specific speed" (N_s) is defined as follows:

$$N_s = \frac{NQ^{1/2}}{H^{3/4}}$$

where N = pump speed in revolutions-per-minute

Q = pump flow in gallons-per-minute

and H = pump head rise in feet.

For a 7,800 rpm centrifugal-pump design speed, a 1.8 gallon-per-minute design flow, and an 87 foot (design) head rise, the centrifugal pump's specific speed would be 522.

Hydraulic design and manufacturing considerations were used to determine the geometry of the impeller's passages. Flow transitions were selected which minimized friction losses and maintained maximum fluid control. On the basis of empirical data, an impeller design with four front vanes was selected, and the front-vane parameters were as follows:

Front-Vane Parameters

Impeller diameter, inches	2.0
Impeller vane height at discharge, inch	0.1
Impeller inlet diameter, inch	0.75
Impeller vane height at inlet, inch	0.22
Front-vane inlet angle, degrees	15
Front-vane discharge angle, degrees	25

To reduce the axial thrust on the bearings and the pressure on the dynamic shaft seal, back vanes were also included in the impeller design. Using empirical data the optimized back-vane parameters were as follows:

Back-Vane Parameters

Number of back vanes	32
Back-vane height, inch	0.10
Back-vane clearance, inch	0.10
Back-vane length, inches	1.22

2. Housing Design

The height-to-front vane clearance relationship required that the impeller vane height be at least 0.1 inch at the impeller discharge. To accommodate this vane height, the volute housing was designed for partial emission (1/3 Flowing Section). A full admission design would have required 0.03 inch vane height and 0.015 inch front vane clearance. Performance with such a design would be less predictable.

Characteristic

Admission, degrees	120
Inlet angle, degrees	8.9
Inlet velocity, feet per second	29.1

3. Performance

a. Hydraulic Efficiency

For the specified pump suction conditions, calculations showed the hydraulic design to be adequate.

b. Suction Performance Estimate

An estimate of the centrifugal-pump's suction performance was made by examining the local flow patterns at the various impeller inlet areas. The analysis indicated that the centrifugal-pump should be free of cavitation at suction specific speeds of up to 9,000. Since the design suction specific speed was only 6,000, there was sufficient margin for any unexpected perturbations.

$$\text{Suction Specific Speed} = \frac{NQ^{1/2}}{(NPSH)^{3/4}}$$

c. Impeller-Loading Determination

The radial loading on the impeller was determined by examining the pressure distribution around the impeller periphery. The following radial and axial loading on the impeller was calculated.

Flow (% Design)	Radial Load (Pounds)	Axial Load (Pounds)
0	29	55
50	13	53
100	7.5	50
130	12	45

The axial loading on the impeller was to a large extent neutralized by the use of back vanes. Analysis of the pressure distribution on the pump housing at various flows showed that the highest axial loading to be expected under any normal operation condition is 55 pounds, at nominal design conditions, with a loading of 110 pounds only at startup.

B. JET PUMP

The purpose of the jet pump was to provide adequate suction pressure to the centrifugal pump to prevent cavitation in the centrifugal pump during startup and extended operation. Since cavitation damage was a critical factor in the performance of the pump during starting and for long duration operation, more emphasis was placed upon producing a noncavitating pump than a high performance pump.

The jet pump design parameters were selected on the basis of the centrifugal pump requirements and reliability considerations. To ensure that the centrifugal pump would not cavitate, the jet pump was designed to produce sufficient head so that the centrifugal pump would not be required to operate at suction specific speeds of greater than 6000 under any startup or steady-state conditions. The optimum design for the jet pump required a rather small discharge nozzle area for the jet. It was expected that such a small nozzle discharge area would be susceptible to clogging if the mercury fluid should become contaminated. The nozzle was therefore sized for maximum reliability with a resulting compromise in jet pump performance.

A jet pump's maximum efficiency is usually about 30%, but compromises for minimum nozzle size and low Reynolds Numbers reduced the predicted efficiency to about 15%.

The hydraulic and physical design parameters for the jet pump were as follows:

<u>Hydraulic and Physical Design Parameters</u>

Ratio of drive head to suction head	200
Ratio of through-flow to jet-flow	1.29
Ratio of jet-pump head to centrifugal-pump head	0.11
Suction flow velocity, ft/sec	4.73
Jet discharge velocity, ft/sec	64.7
Jet pump discharge head , ft	6.6
Reynold's Number for drive jet	6.5×10^5
Jet nozzle diameter, inch	0.093
Mixing section diameter, inch	0.404

C. SHAFT SEALS

A dynamic sealing system is used to prevent the intermixing of the mercury working fluid and the organic bearing lubricant. A vent-to-space cavity is provided between the two sealing systems so that the small fluid leakages are ported to space. Additionally, since the dynamic sealing systems are only effective when the shaft is rotating above 6,000 rpm, two carbon face seals are employed, one on the mercury side and one on the oil side. The carbon face seals provide sealing when the pump is stationary and during start-up and shut-down operations. During normal operation, these carbon face seals are lifted away from the rotating mating surface to minimize contact wear.

1. Mercury Shaft-Seal

The mercury shaft-seal consists of a visco-pump, a molecular pump, and a carbon face-seal in series. The visco-pump is basically a screw pump, with a rotating helical channel in a close-fitting housing, which generates a pressure differential balancing the back-vane hub pressure of the centrifugal pump's impeller (see Reference 2). This pressure balance produces a liquid-vapor interface. Molecules evaporating from the interface are restricted from flowing to space by the adjacent molecular pump. The molecular pump, similar in appearance to the visco-pump, is used to return the gas particles to the liquid interface by the process of absorption and remittance following collision with the rotating helix and the housing (see Reference 3).

a. Mercury Visco-pump

For the visco-pump to function as an effective dynamic seal, a cold, stable liquid-vapor interface is required. By cooling the interface, the vapor pressure is reduced. A heat exchanger is contained within the seal housing which uses the SNAP-8 organic fluid (Mix-4p3E) as a cooling medium. The purpose is to lower the temperature at the liquid/vapor interface of the seal in order to lower the vapor pressure, and hence reduce leakages to acceptable values. The following were the optimized visco-pump system parameters:

Visco-pump System Parameters

Coolant inlet temperature, °F	210
Coolant flow, pounds-per-hour	2600
Mercury liquid/vapor interface temp., °F	300
Visco-pump geometry:	
Diameter, inches	1.25
Length, inches	2.50
Radial clearance, inch	0.003
Helix angle, degrees	15
Groove width, inch	0.083
Land width, inch	0.083
Groove depth, inch	0.0125
Number of groove starts	6
Power consumption, horsepower	0.25
Pressure generation, psi/inch	75

b. Mercury Molecular Pump

The mercury molecular pump was designed to keep the mercury vapor leakage below five pounds in 10,000 hours for a mercury liquid/vapor interface temperature of 300°F or lower. The optimized molecular pump design parameters are as follows:

Seal Geometry

Diameter, inches	1.625
Length, inches	1.42
Radial clearance, inch	0.003
Helix angle, degrees	2.25
Groove width, inch	0.10
Land width, inch	0.10
Groove depth, inch	variable from 0.010 to 0.060
Number of groove starts	1
Expected mercury leakage, lb/10,000-hr	1.76

c. Bellows-Type Carbon Face Seal

The bellows-type carbon face seal is used to seal the pump when the pump is not rotating. The seal would not last 10,000 hours if it was allowed to rub continuously during pump operation, therefore, a lift-off device was incorporated into the sealing system to lift the seal when the pump speed reached 6,000 rpm. The lift-off device originally consisted of a bellows which is ported to the discharge of the mercury pump. As the pump discharge pressure increases beyond 200 psia, the lift-off device lifts the carbon face seal from the running ring. The process is reversed when the pump is shut down. The maximum power consumption expected for the face seal is less than 0.2 horsepower with the seals engaged.

2. Lubricant Seal

The lubricant seal consists of a slinger and a molecular pump in series as a dynamic seal, and a carbon face seal for a static seal. The slinger is used to develop a stable liquid/vapor interface, and the molecular pump is used to restrict the leakage of vapor molecules. The function of the carbon face seal is to seal when the dynamic seal is not capable of performing the sealing function as described for the mercury shaft seal.

a. Lubricant Slinger

The slinger is a smooth disk with a return groove in the housing at the outer periphery for returning the lubricant to the lubrication system. It was assembled with an axial clearance of 0.019 inch and a radial clearance of 0.010 inch and generated a pressure of 5 psia while maintaining a stable interface. The expected power consumption is 0.35 horsepower.

b. Lubricant Molecular Pump

The molecular pump minimizes the leakage of the organic fluid (vapor) past the shaft. The expected leakage rate of the optimized design is about 0.08 pound per 10,000 hours.

The geometry of the molecular pump is as follows:

Geometry of Molecular Pump

Diameter, inches	1.7
Length, inch	0.85
Radial clearance, inch	0.003
Helix angle, degrees	4.25
Groove width, inch	0.18
Land width, inch	0.22
Groove depth, inch	0.0575
Number of groove starts	1

c. Lubricant Carbon Face Seal

The carbon face seal on the oil side of the shaft is the same as the one on the mercury side. It is also actuated by the mercury pump discharge pressure to lift off when a speed of 6,000 rpm is attained.

D. LUBRICATION SYSTEM

The requirement to use conventional technology dictated the use of conventional ball bearings. Because the bearings were required to operate 10,000 hours without maintenance, an active, flowing lubrication system was required. The lubrication system incorporated an injection system and scavenging device. An injection system was required to supply only as much lubricant as needed. A scavenging device was required to scavenge the bearing cavities faster than the inward flow of the required amount of lubricant to prevent flooding.

1. Bearing-Lubricating Injector

The lubricant injector was designed to provide 400 pounds per hour lubricant to the two bearings. This rate of lubricant flow provides abundant lubrication and also 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 consisted of an injection ring with six equally distributed 0.04 inch diameter holes which direct lubricant toward the inner race of the bearings.

2. Bearing-Lubricant Slingers

Two lubricant slingers are used, which are opposite to the lubricant injection rings in order to scavenge the bearing cavities of polyphenyl ether lubricant. To prevent bearing flooding, each slinger is designed to pump against a 5 psia discharge pressure which is a requirement for the lubricant-coolant pump, and at a flow which is much greater than the specified bearing flow of 200 pounds-per-hour. The selected slinger design has smooth faces. Nominal assembled axial clearances of 0.019 inch, and radial clearances of 0.010 inch are obtained. The calculated power consumption for each slinger is 0.14 horsepower.

3. Motor-Cavity Scavenge Slinger

Initially it was intended to provide a carbon seal and lift-off device between the motor end bearing and the motor winding cavity to prevent leakage of Mix-4P3E into the cavity when the unit is not operating, or at lower speeds when the dynamic seal system is not effective. The lift-off seal system would be similar to that described previously (1C). Consideration was given to the fact that leakage into the cavity might occur due to an ineffective seal, and the Mix-4P3E could only be removed from the motor by either draining the fluid overboard, which would have required additional valving, or by boiling-off the fluid by motor thermal conditions. A thermal analysis under flooded motor conditions was performed, and this indicated that the winding temperature would increase by approximately 100°F due to added power consumption of 0.5 horsepower and viscous drag effects. This effect is undesirable from an insulation life standpoint and the introduction of drain valves introduces other complications. Therefore, a motor cavity scavenge slinger was incorporated in the design. This had the capability of purging the rotor "air" gap of fluid and return it to the lubricant-coolant system. The maximum power for a flooded rotor and slinger was calculated to be 0.9 horsepower, but this falls off rapidly as scavenging is accomplished (see Figure 4).

E. BEARING SYSTEM

The bearing design is a single row, angular contact ball bearing, with a low shoulder in the inner ring, and with an outer ring-piloted ball separator of one-piece construction. The bearing design goal was to obtain a minimum life of 10,000 hours with 99.5% reliability.

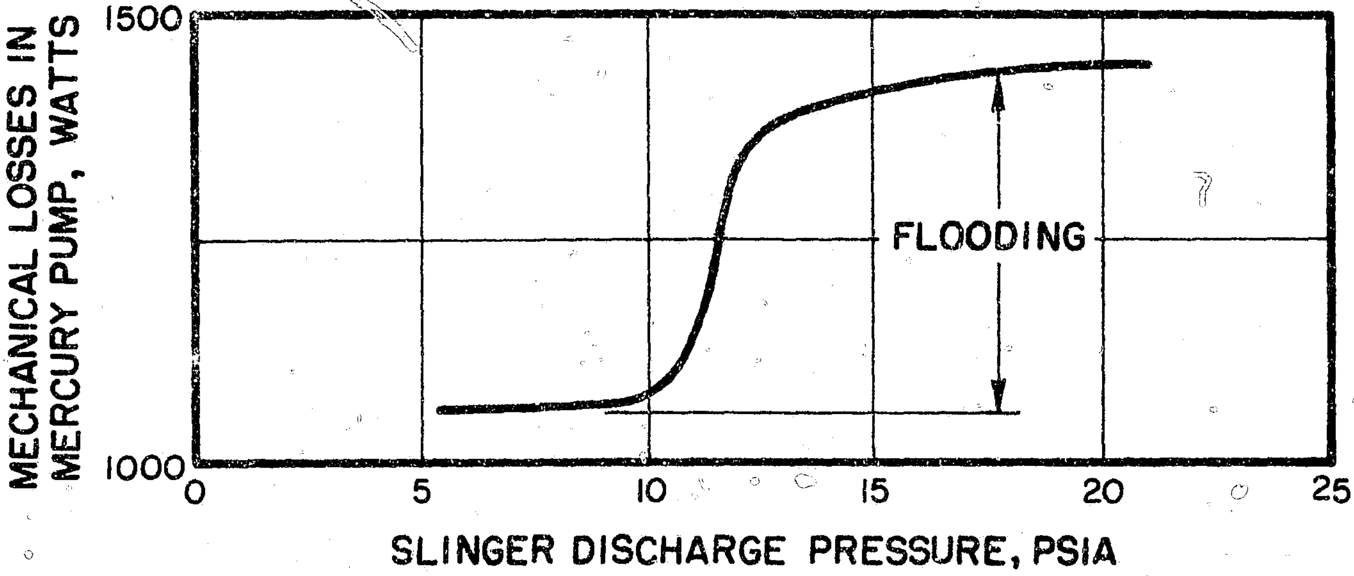


Figure 4. Effect of Lubricant Discharge Pressure on Mercury Pump Performance (at 210°F Oil Inlet Temperature)

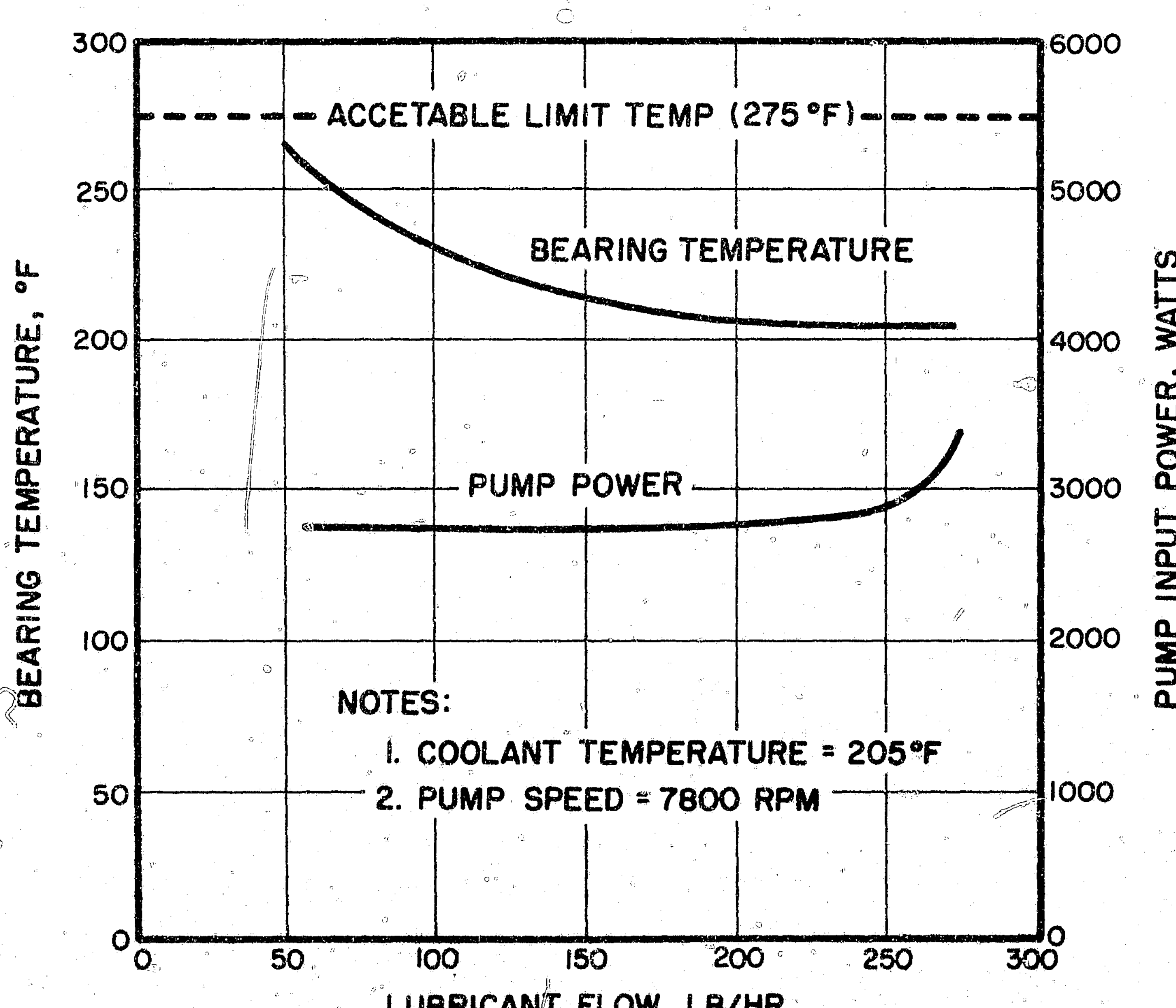


Figure 5. Bearing Lubricant Flow Optimization for Mercury Pump

The bearing design was based on the following conditions:

Characteristics for Bearing Design

Speed, rpm	7800
Loads:	
Axial, lb	55 (110 lb at start)
Radial, lb	45 (65 lb at start)
Equivalent, lb	165
Maximum acceleration, rpm/sec	800
Lubricant:	(polyphenyl ether Mix 4P3E)
Pressure, psia	1.0
Flow, lb/hr	200

Angular contact bearings were selected for their high load capacity and good radial stiffness. With axial preload there is no internal looseness under operating conditions. Full-rolling contact and optimum dynamic balance can be maintained at all times. The contact angle was established by the radial play requirements and the race curvature. The tolerances were chosen for long bearing life and high reliability. Selective assembly techniques were used to prevent sliding fit between the outer races and the housing. A one-piece iron-silicon bronze cage design was selected for maximum strength and light weight construction. The cage was also designed for maximum lubricant flow. The detailed description of the bearings is shown in Table III.

The expected bearing life may be estimated by assigning life increments for improvements in this bearing over a conventional bearing. The more important bearing life parameters are as follows:

Bearing Life Parameters	
Load rating, lb	6,560
Fatigue life with 98.5% reliability, hours	40,000
Power consumption, horsepower	0.2 (140 watts)
Shaft loading, see Figure 3	

Table III. Mercury Pump-Motor Assembly Bearing Design

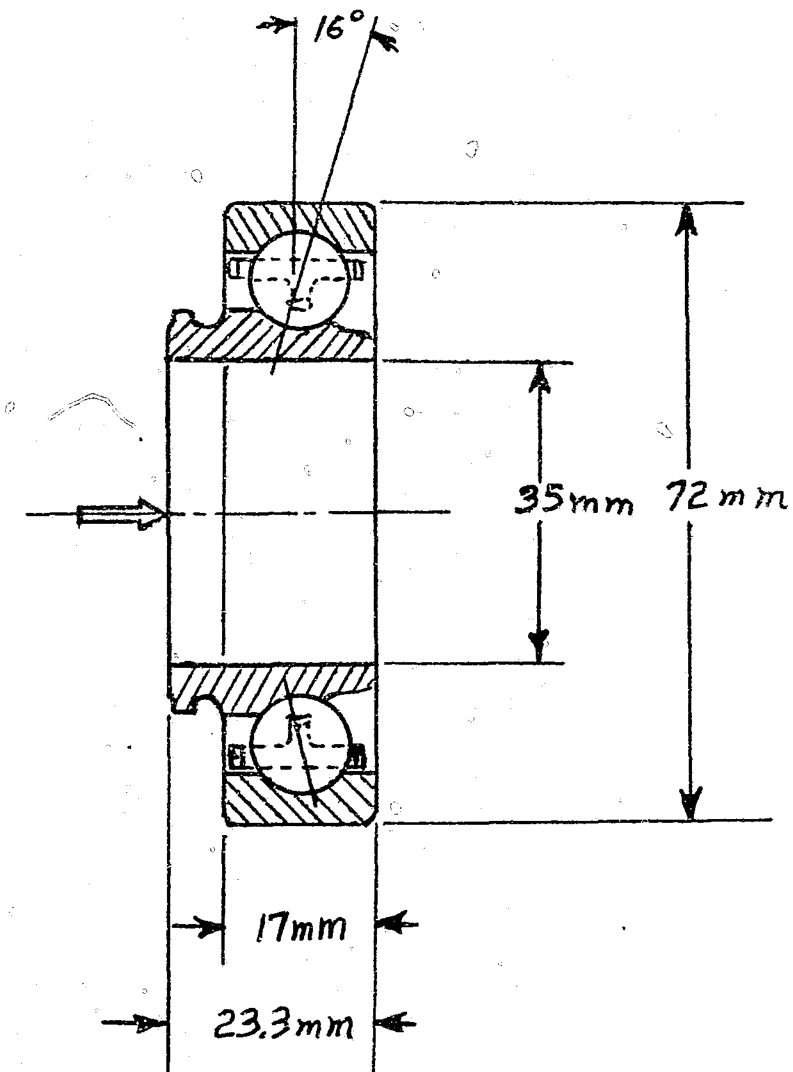
1. TYPE - ROLLING ELEMENT ANGULAR CONTACT BALL BEARING

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

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

4. MATERIALS
   a. Races - 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 and:
      (1) Balls - Grade 5
      (2) Inner Race Curvature ± 0.001 in.
      (3) Outer Race Curvature ∓ 0.002 in.
   b. Marked for Selective Assembly



F. MOTOR DESIGN

The motor design was selected on the basis of an optimization between motor efficiency and pump efficiency. A preliminary study indicated that a conventional induction motor would be best for the application. For induction motors using 400 Hertz, 3-phase power, the highest efficiency is obtained by motors using six or eight poles. Since the mercury pump is also most efficient at the highest speed obtainable, a 400 Hertz, 3-phase induction six-pole motor was selected. To meet the performance requirements listed in Table III, a motor of the squirrel-cage induction type was selected. The gross rotor length was 2.75 inches and the diameter 3.25 inches. Liquid coolant, Mix-4P3E (at 210°F), flows through the outer stator periphery at a rate of 200 to 400 pounds-per-hour to keep the hot spot temperature below 400°F. An organic insulation system (polyimide "ML") was selected based on expected operating temperatures and hermetically-sealed housings were included to minimize outgassing in the vacuum environment.

The electrical losses in Table IV result in an expected motor electrical efficiency of 88% for a lagging power factor of 0.77.

TABLE IV. MOTOR PERFORMANCE AND ELECTRICAL LOSSES

Motor Performance Requirements	
Synchronous speed, rpm	8000
Power supply	3-phase, 208 volts (line-to-line), 400 Hz
Rated rotor developed power, hp	3.43
Speed at rated output, rpm (minimum)	7800
Minimum efficiency, %	86
Power factor, minimum	0.65
Motor Electrical Losses	
Stator I^2R and core losses, kw	0.196
Stator iron, kw	0.078
Rotor, kw	0.094
Total Electrical Losses, kw	0.368

III. MERCURY PUMP TESTING AND DEVELOPMENT

The individual mercury pump components and the complete mercury pump motor assembly were evaluated in a test program. The motor, the jet pump, the lubrication system, and the sealing system were first tested independently. After the individual component performance was determined, the performance of the complete pump motor assembly was evaluated in two mercury loops which simulated the SNAP-8 environmental and operational requirements in space. Simulation was achieved in the mercury loops by pumping mercury heated to 600°F, and by providing a vacuum system at the mercury pump seal-exhaust port. A complete coolant and lubrication system was also provided.

The mercury pump's short-term and endurance capabilities were evaluated in the mercury loops. The hydraulic and electrical performance was determined including the head-capacity characteristics, the temperature and pressure distributions, and the jet pump and mercury pump motor assembly efficiencies. The mercury pump motor assembly was incorporated in the complete power conversion system and tested to permit an evaluation of the compatibility and effect on performance while operating with all the system components under actual service conditions. System testing also allowed a more thorough analysis of transient system effects on the mercury pump during system startup and shutdown.

A. COMPONENT TESTS

1. Motor Insulation Tests

A polyimide-insulated motor was submerged in polyphenyl ether and operated at temperatures of 250° and 300°F for 20,000 hours. This test was conducted to determine the compatibility of the lubricant-coolant on the insulation. No perceptible insulation degradation was observed.

2. Bearing System Tests

The bearing system was tested independently to determine the minimum lubrication requirements necessary to prevent life-limiting bearing overheating. Lubricant flow as low as 50 pounds-per-hour held the bearing temperature to approximately 262°F, which was below the acceptable 275°F level (see Figure 5). When tested in the hot mercury loop as a complete pump-motor assembly, a minimum of 60 pounds-per-hour of lubricant was required to maintain bearing temperature below 262°F.

The maximum bearing-lubricant flow was found to be 260 pounds-per-hour (see Figure 5). Flows greater than 260 pounds-per-hour tended to flood the bearings, increasing bearing power consumption by 0.25 horsepower. Consistent with pre-test calculations, the bearing-scavenging slingers prevented bearing flooding for slinger discharge pressures below 10 psia. Since the system will probably never encounter pressures above 10 psia, then no flooding is expected to occur.

3. Dynamic Seal Tests

The optimum dynamic mercury seal design was derived by individually testing the visco-pump, the slinger, and the molecular pump. Various screw pump type seals and slinger designs and running clearances were evaluated. The best performing dynamic elements were then combined for integrated seal testing with the following results:

Characteristic	
Visco-pump pressure generation, psi/inch	75
Visco-pump power consumption, hp	0.25
Slinger power consumption, hp	0.35
Average measured mercury leakage, lb (2 to 200 hour testing durations)	<0.2

Both the visco-pump and the slinger maintained stable liquid-vapor interfaces which is necessary for good sealing. The visco-pump/molecular pump combination performed as expected. No precise leakage measurements were made, but a careful recording of the mercury found in various test system traps indicated that the leakage was low, and probably near the estimated leakage of 1.76 pounds per 10,000 hours.

4. Jet Pump Tests

Development testing of the jet pump was conducted in water and later in mercury for the purpose of determining jet pump performance related to orifice size, and the location of jet nozzle to centrifugal impeller. The water and mercury tests showed satisfactory correlation. With either liquid, the jet pump efficiency was approximately 12.5%, or slightly lower than expected (see Figure 6 for jet pump performance). This slightly lower-than-expected test efficiency was compensated by the jet pump's ability to supply a net positive suction head of 7 feet to the centrifugal pump inlet. This net positive suction head was sufficient to suppress centrifugal pump cavitation under normal operation conditions.

The jet nozzle diameter and jet to centrifugal pump length were selected based on these test results.

B. MERCURY PUMP PERFORMANCE UNDER SIMULATED SYSTEM CONDITIONS

1. Mercury Pump Short-Term Performance

The mercury pump's short-term performance was determined in simulated system tests in the liquid mercury loop test facility. The mercury pump's efficiency, temperature and pressure distributions, and its performance for normal and low suction pressures were determined.

a. Efficiency

The jet-centrifugal pump's efficiency was significantly higher than expected. While conservative calculations predicted a 22% jet-centrifugal hydraulic efficiency (ignoring seal, bearing, and motor losses), the test results indicated a 29% hydraulic efficiency as shown in Table V. This higher than expected test performance is attributed to the selection of conservative head coefficients for the centrifugal pump due to the lack of background experience with small mercury pumps.

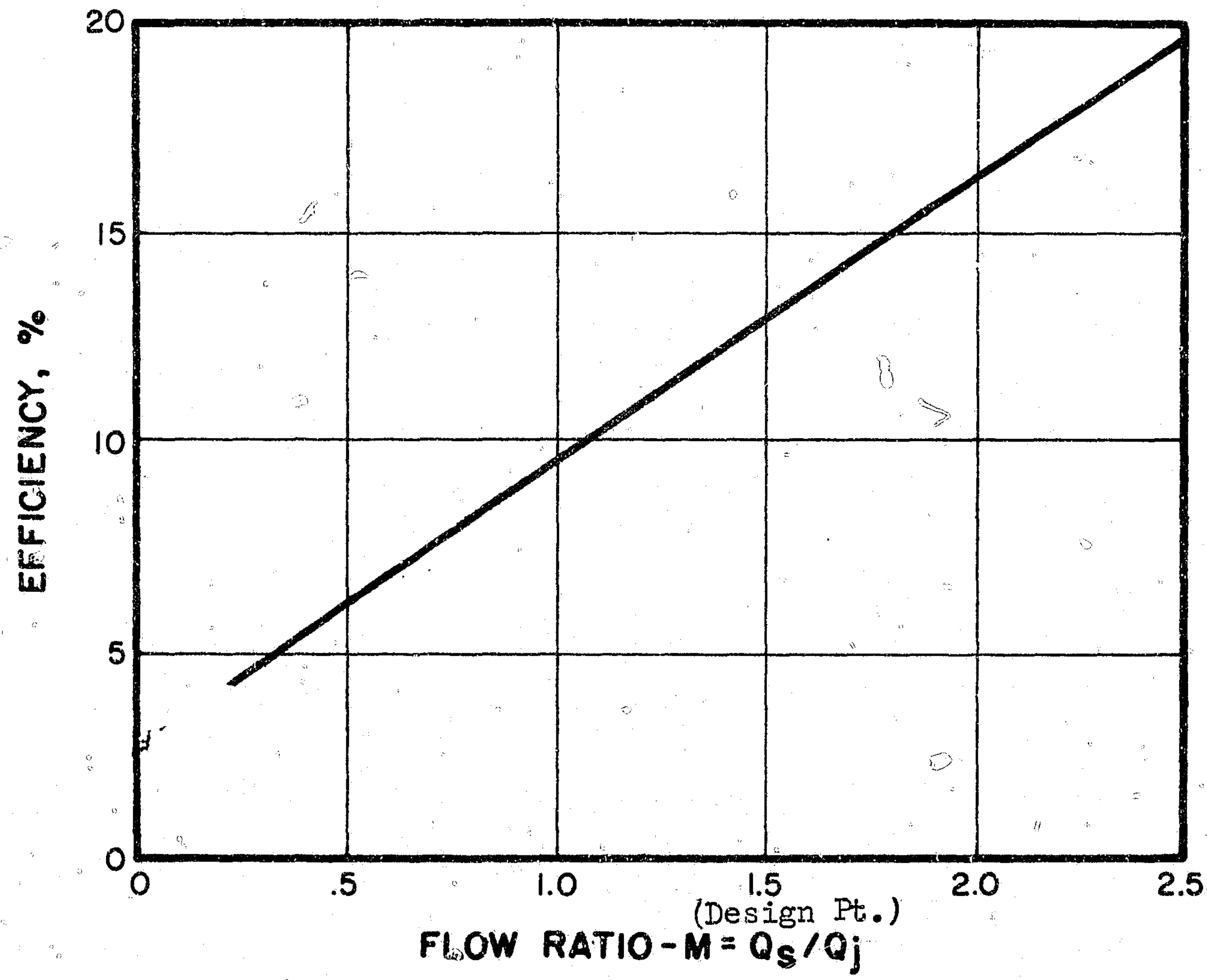


Figure 6. Jet Pump Performance Test

TABLE V. - MERCURY PUMP-MOTOR ASSEMBLY PERFORMANCE DATA

Components and Parameters	Test Results
Pump Jet/Centrifugal	
Flow, gpm	1.8
Speed, rpm	7830
Inlet temperature, ^{o}F	505
Inlet pressure, psia	10.5
Pressure rise (ΔP), psid	487
Head, ft	87
Hydraulic power, kw	0.38
Efficiency, %	29.0
Total Hydraulic Losses, watts	1150
Motor Scavenge Slinger Loss, watts	150
Motor-Induction Type	
Total electrical losses, watts	428
Motor electrical efficiency, %	87.8
Motor efficiency, % (electrical & hydraulic)	36.8
Input power, kw	3.0
Power factor	0.77
Overall PMA efficiency, %	12.7

Figure 7 shows 12.7% efficiency for the (entire) pump-motor assembly which includes all bearing, seal, and motor losses.

b. Temperature Distribution

Analytical results of the temperature distribution in the pump-motor were obtained for a pump operating in space environment. To simulate the space conditions, insulation was wrapped around the entire pump-motor for testing. The results indicated that all the temperatures were less than expected with the exception of the motor stator iron temperature and the

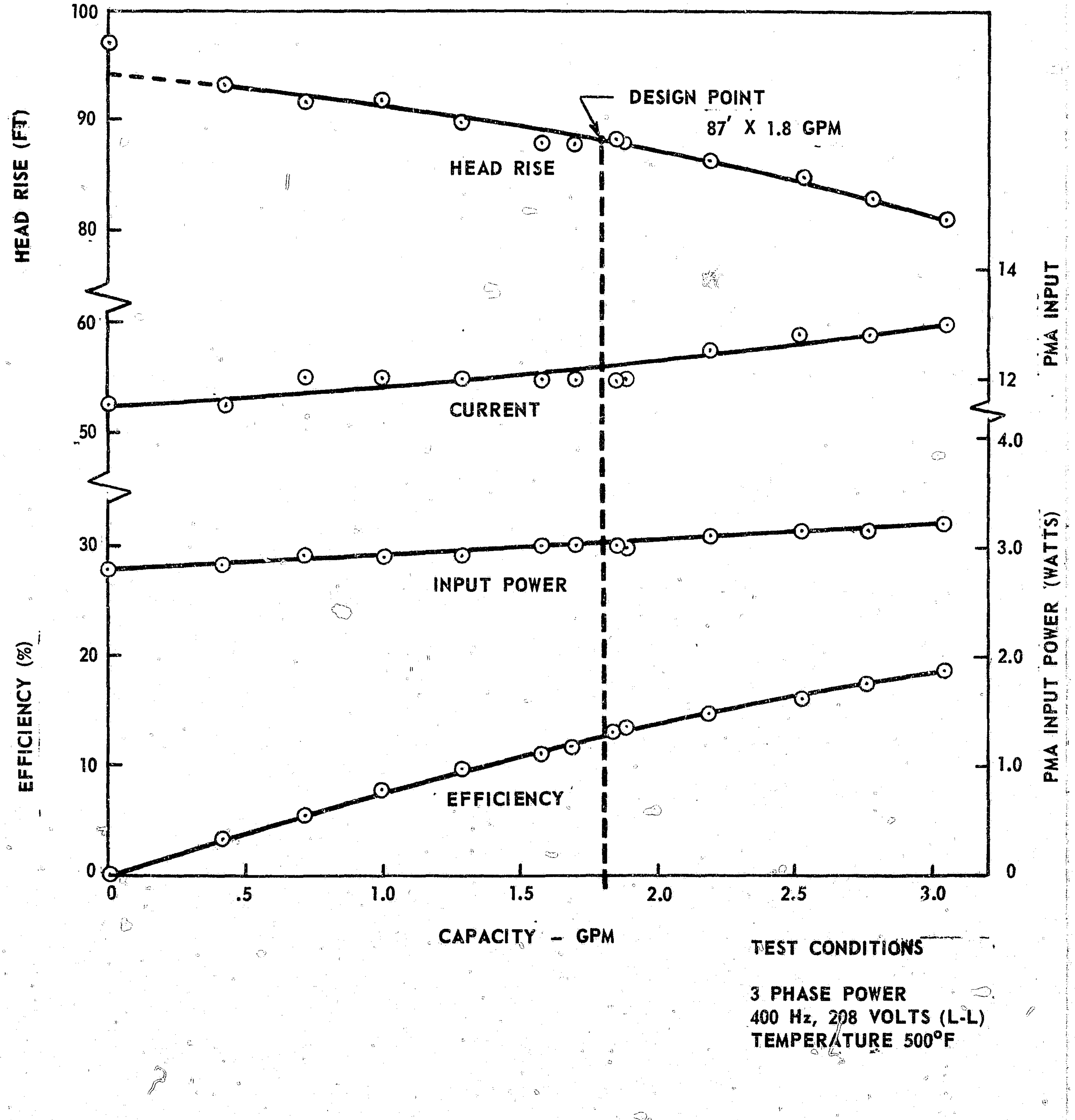


Figure 7. Test Performance Curves - SNAP-8 Mercury Pump Motor Assembly Operating at 500°F

motor housing temperature (Figure 8). The higher stator iron temperature was believed to have resulted from an unexpectedly high rate of heat transfer from the winding coil to the stator iron. The slightly higher temperatures of the stator iron and the motor housing were not considered significant.

c. Pressure Distribution

The pressure distribution around the centrifugal pump's impeller was used to determine the probable radial and axial thrusts acting on the impeller. Using the measured pressure distribution as a basis, the maximum axial thrust towards the motor was 63 pounds, compared to a calculated 60 pounds; and the maximum radial load was 17 pounds, compared to a calculated 15 pounds.

d. Hydraulic Performance

Figure 7 shows the head capacity characteristics of the mercury pump. The head generated at design flow was close to the predicted value. The head/flow curve corresponds to a typical pump with a specific speed of 522.

It was demonstrated for steady-state operation that the mercury pump was able to operate with the minimum system net positive suction head required of 1.53 foot at design flow without any performance degradation.

During start-up the power conversion system requires a 0.14 foot net positive suction head for a pump flow of 40% of design (Figure 9). Therefore, it is demonstrated that NPSH for the mercury pump is adequate to meet the design flow condition.

2. Mercury Pump Development

Three of the four major problems encountered during mercury pump development were solved by minor modifications. The problems of motor cavity flooding, pump impeller cavitation induced by the original impeller bolt and locking device, and jet pump nozzle blockage were solved completely. However, the problem of the actuating bellow's bursting has not been totally solved.

a. Motor-Cavity Scavenge Slinger

Testing showed that the motor cavity scavenging slinger was effective in purging the motor cavity of bearing lubricant and this required approximately 0.5 horsepower at steady-state conditions.

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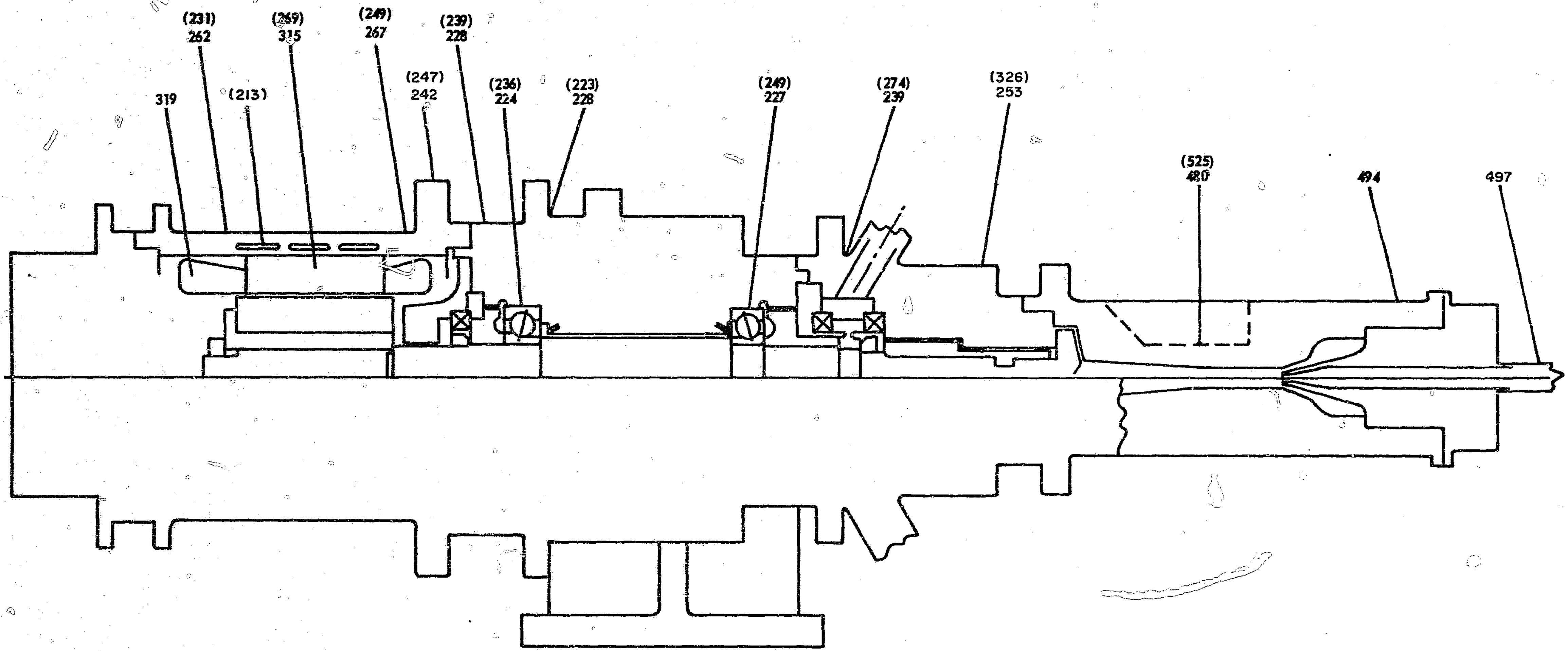

SPACE SEAL COOLANT FLOW 2000/2[illegible]00 LB/HR AT 210°F INLET TEMP.
MOTOR AND BEARING LUBRICANT-COOLANT FLOW 290/300 LB/HR AT 210°F INLET TEMP.
MERCURY FLOW 13,500 LB/HR AT 500°F SUCTION TEMP.
REF TEST 06 P32
ESTIMATED TEMPS IN PARENTHESIS °F eg (274)

Figure 8. Mercury Pump-Motor Assembly Test Data - Temperature Distribution

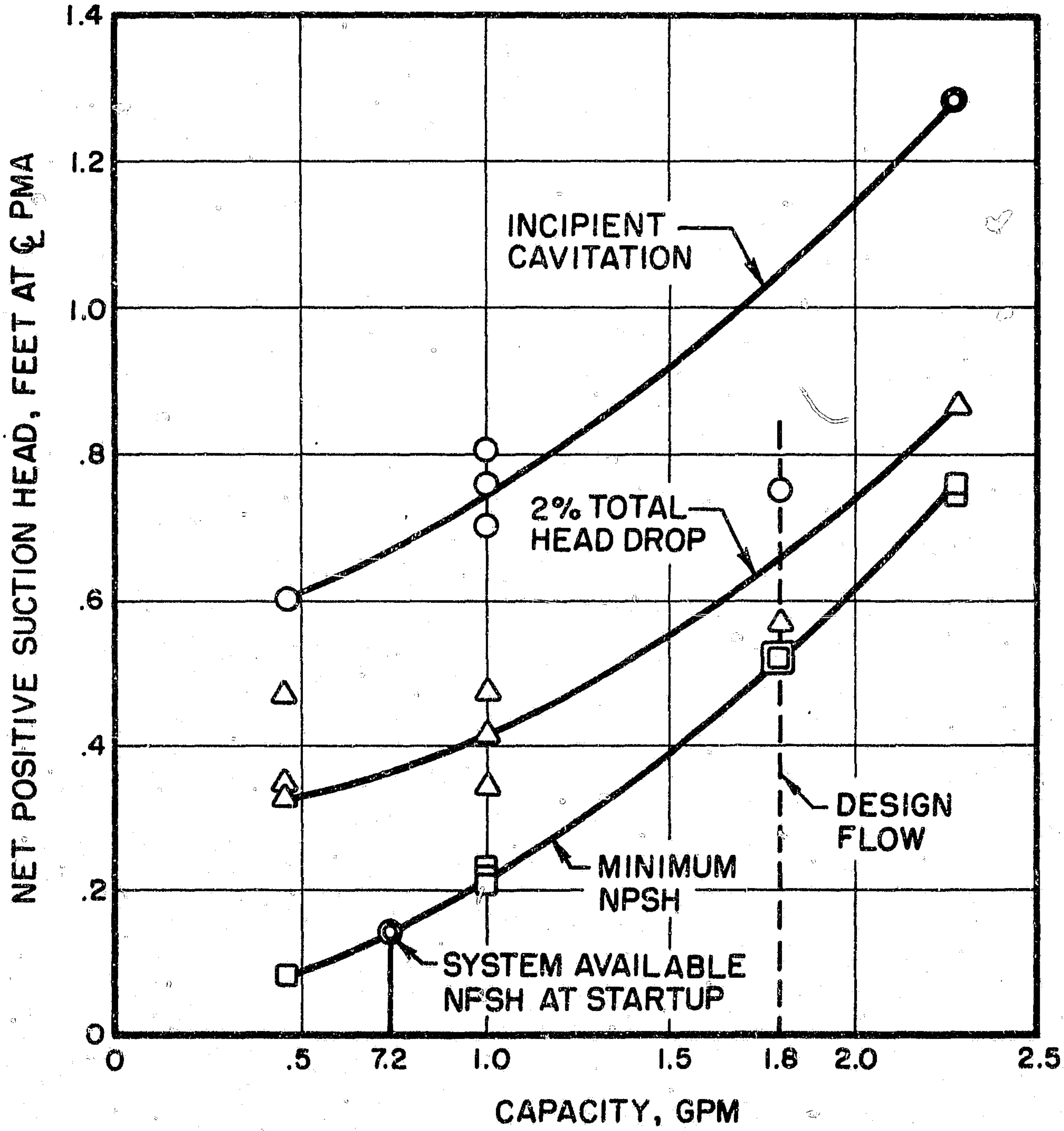

Figure 9. Summary of Test Results from Mercury Pump Motor Assembly Performance

b. Streamlined Bolthead and Locking Device

In the original impeller bolt design, the head was of the conventional Allen screw type. The bolt and its associated locking device caused minor interruptions in the mercury flow which resulted in cavitation damage. When the mercury pump was operated at extremely low suction pressures for cavitation tests, the depressions in the bolt head aggravated the cavitation damage to the impeller. A change to a more streamlined bolt head and locking device eliminated the problem.

c. Meshed Pump-Suction Screen

During one test, the jet pump nozzle was particularly blocked by what was believed to be a weld scale. Sustained mercury pump operation with a partly blocked nozzle resulted in some very slight cavitation damage to the pump impeller. This potential hazard was eliminated in later tests by the insertion of a fine mesh screen at the pump inlet upstream of the jet pump.

d. Static Shaft-Seal Lift-Off Device

The first lift-off device for the static seals consisted of bellows which were actuated by mercury from the mercury pump discharge. In the first two tests using this device, the weld seam ruptured. The "waterhammer" effect was produced when the mercury pump was started quickly. The combination of high weld seam stresses and the "waterhammer" effect produced the rupture. In a modified system the actuating medium was changed to 200 psia nitrogen gas. The substitution of nitrogen did not, however, solve the problem, and development has been continued.

3. Mercury Pump Endurance

The mercury pump has verified its capability to operate continually for 10,000 hours. The purpose of the endurance tests was to determine what factors might prevent a desired 40,000 hour operational life. Testing began in May of 1967 and continued through December 1968. A total of 12,227 hours testing time was accumulated including 109 starts.

Mercury pump performance was determined at the beginning of the endurance test and periodically thereafter. The results showed no significant changes in the mercury pump's head-capacity characteristics or efficiency. A post-test inspection, however, revealed damage to the impeller and the visco-pump. There were also minor cracks in the motor-rotor conductor bars. The other mercury pump components, however, were in good condition.

a. Centrifugal-Pump Impeller Back-Hub Damage

The pump impeller was found to be severely damaged at the hub area adjacent to the back vanes (see Figure 10), an area not generally damaged by cavitation. Cavitation was, nevertheless, considered the cause of damage, though the exact mechanism of cavitation was undetermined. The damage was judged to limit the mercury pump life to less than the desired 40,000 hours. Although the life already exceeded the 10,000 hour requirement, a solution to this problem is being sought to extend the mercury pump life to 40,000 hours.

b. Centrifugal-Pump Impeller-Vane Damage

Cavitation damage was observed on the centrifugal pump's impeller vane (see Figure 11). The damage was restricted primarily to the vane's high-pressure side near the impeller inlet. The damage consisted of surface roughening to an approximate 0.001 inch depth, which was attributed to the extremely low pressures encountered by the mercury pump during abnormal operating conditions of low suction pressure.

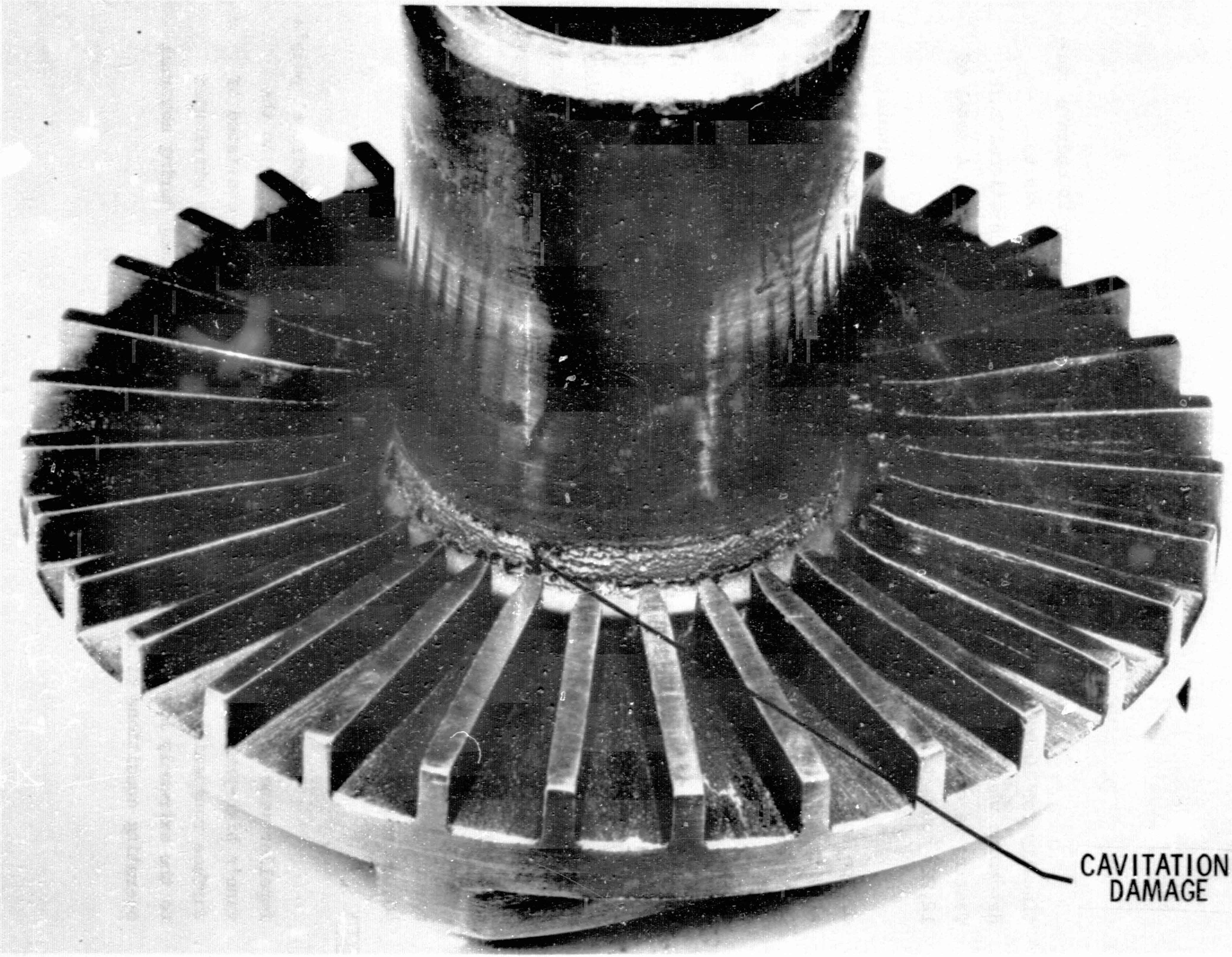

Figure 10. Appearance of Impeller Back Vane After 12,227 Hours

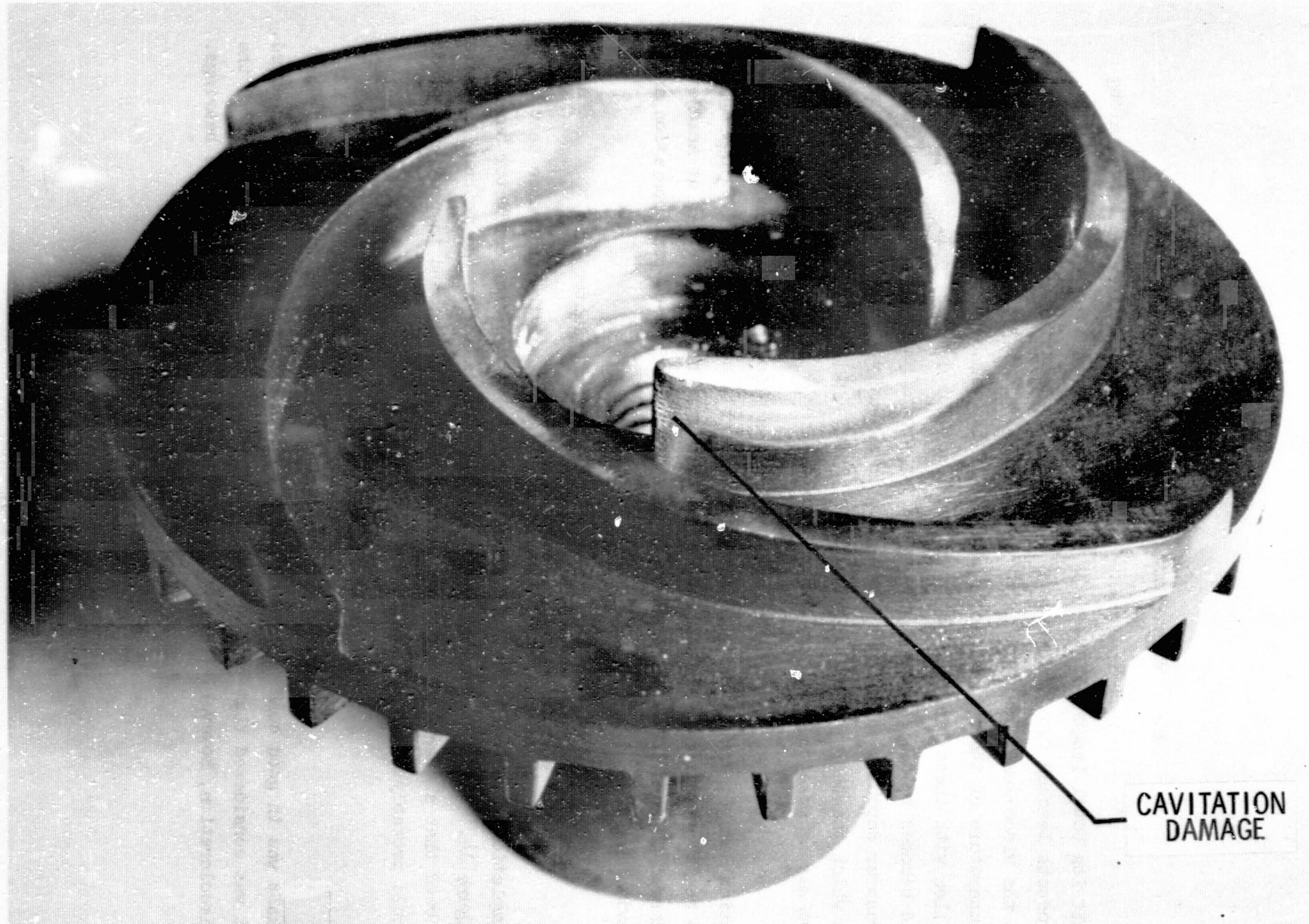

Figure 11. Mercury Pump Impeller After 12,227 Hours Operation and 109 Start Cycles

c. Visco-Pump Clogging and Cavitation Damage

The visco-pump was clogged with foreign matter along half its total length, Figure 12. This matter was mostly mass transfer products leached from the mercury system. Mass transfer products precipitate at the visco-seal, because the visco-seal is at the lowest pressure and temperature of the entire mercury loop. If the visco-pump becomes entirely filled with these transfer products, it loses its dynamic sealing capability and becomes totally ineffective for mercury containment. Moreover, the mass transfer deposits filled the visco-pump in a nearly isothermal loop (a loop in which the temperature is approximately constant), and therefore, this problem may be more severe in system operation.

The visco-pump grooves also showed evidence of cavitation damage, Figure 13. The damage was slight, however, and does not appear to be life-limiting. A test program is currently being conducted at Aerojet-General Corporation to determine the relationship of visco-pump operation and cavitation.

d. Cracks on the Motor-Rotor Conductor Bars

Examination of the motor after the endurance test completion revealed several minor cracks of the motor-rotor conductor bars, Figure 14. These cracks did not affect the performance of the motor. Analysis showed that a minor change in the fabrication technique, and inspection control would probably eliminate any future cracking.

e. Shaft Discoloration

Except for surface discoloration at the motor end, the shaft was in good condition. None of its dimensions were significantly changed. It was considered that heat soak-back along the shaft from the motor caused the discoloration, however, there appeared to be no adverse effect to the bearings.

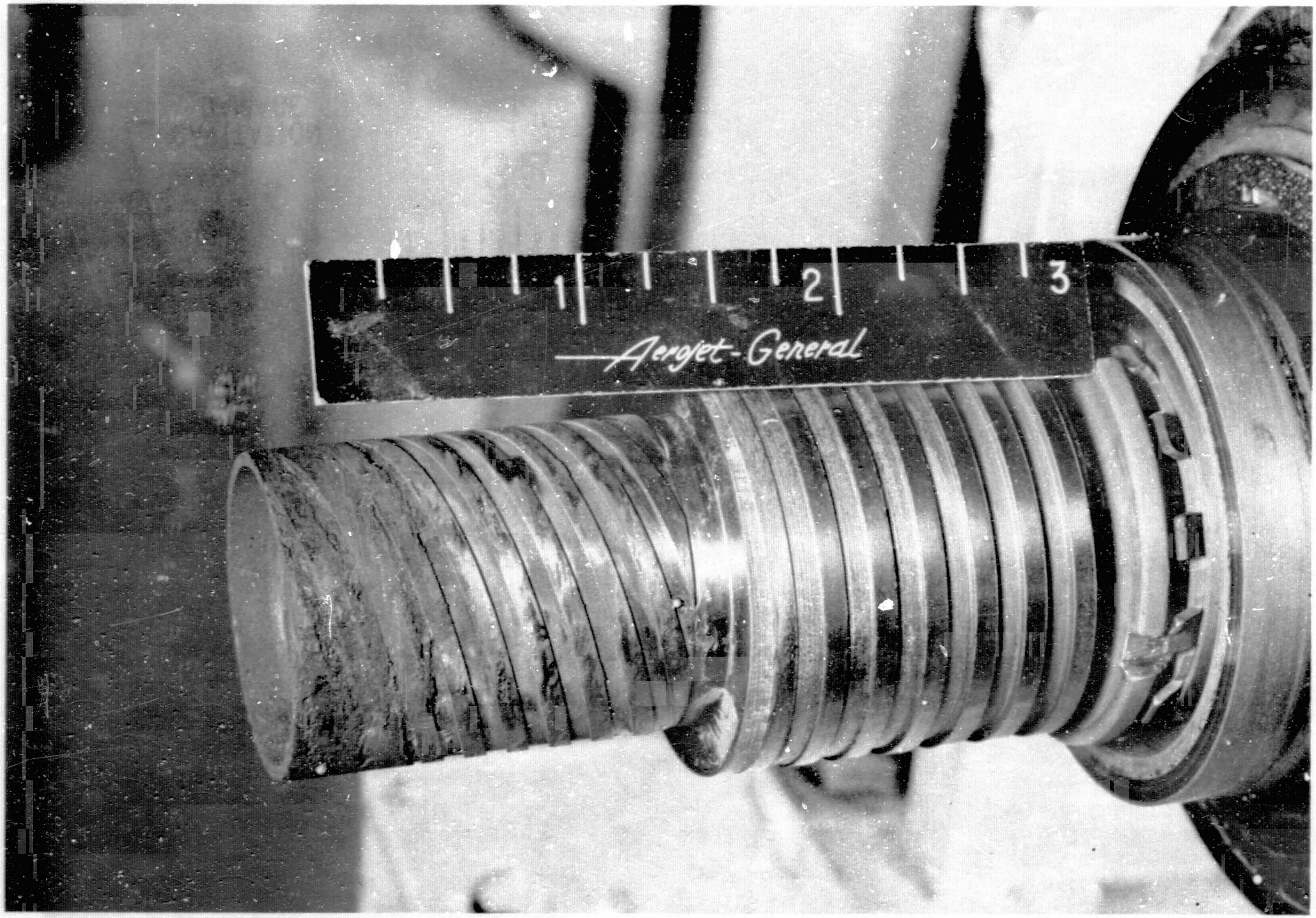

Figure 12. Visco Pump Sleeve Showing Filled Grooves at Disassembly After 12,227 Hours Operation and 109 Start Cycles

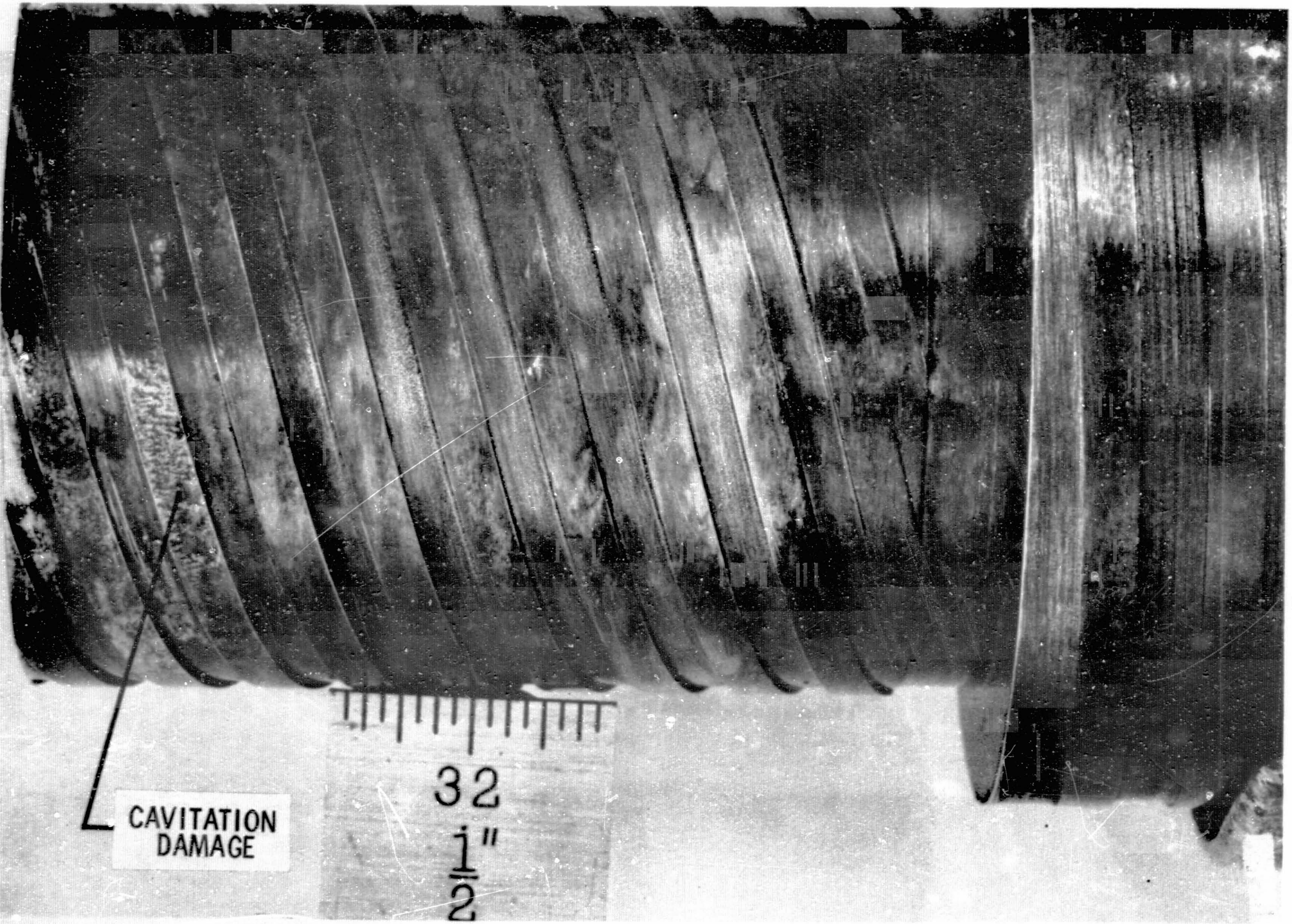

Figure 13. Visco Pump After 12,227 Hours Operation and 109 Start Cycles

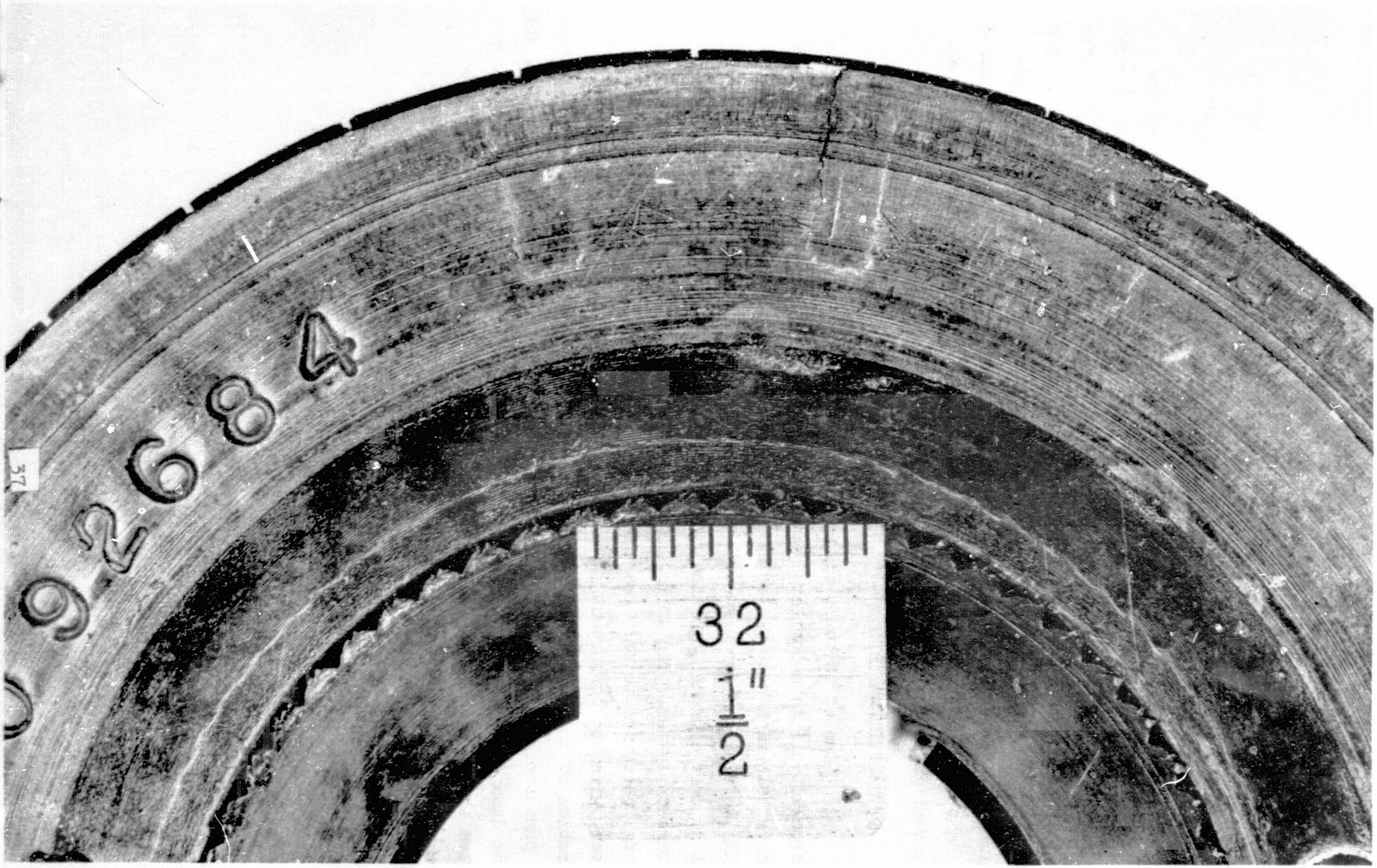

Figure 14. Motor End Ring Cracks - Inboard End After 12,227 Hours Operation and 109 Start Cycles

f. Bearing-Race Dimensional Change

The bearings were in excellent condition after the endurance test, Figure 15. There was no evidence to indicate possible future failure. The balls and the outer races showed excellent dimensional stability, but the inner ring revealed an apparent 0.0005 inch growth. The change in the inner race dimension and the tracks on the ball bearings indicated that the bearing clearance was less than optimum. If the dimensional change was caused by shaft over-heating, it is possible that the bearing life may be shortened. An investigation of this possibility is being conducted.

C. SYSTEM TESTS

One of the most important aspects in mercury pump testing was its incorporation into a complete power conversion system. There have been four such systems with a total testing time of 8,755 hours as of December 1968. During testing the mercury pump was subjected to the unexpected behavior of test support equipment, and the other SNAP-8 components comprising the power conversion system. The following are several incidents which have occurred during in-service tests: leakage of sodium-potassium alloy into the mercury system, with the consequent formation of an amalgam in the mercury pump; operation of the mercury pump without mercury and running with a debris-clogged jet pump nozzle. Throughout all these tests the mercury pump has sustained only minor damage. It has never been the prime factor in causing power conversion system shutdowns. As of December 1968, two pumps are being tested and have accumulated test times in excess of 1,400 hours in actual systems without problems.

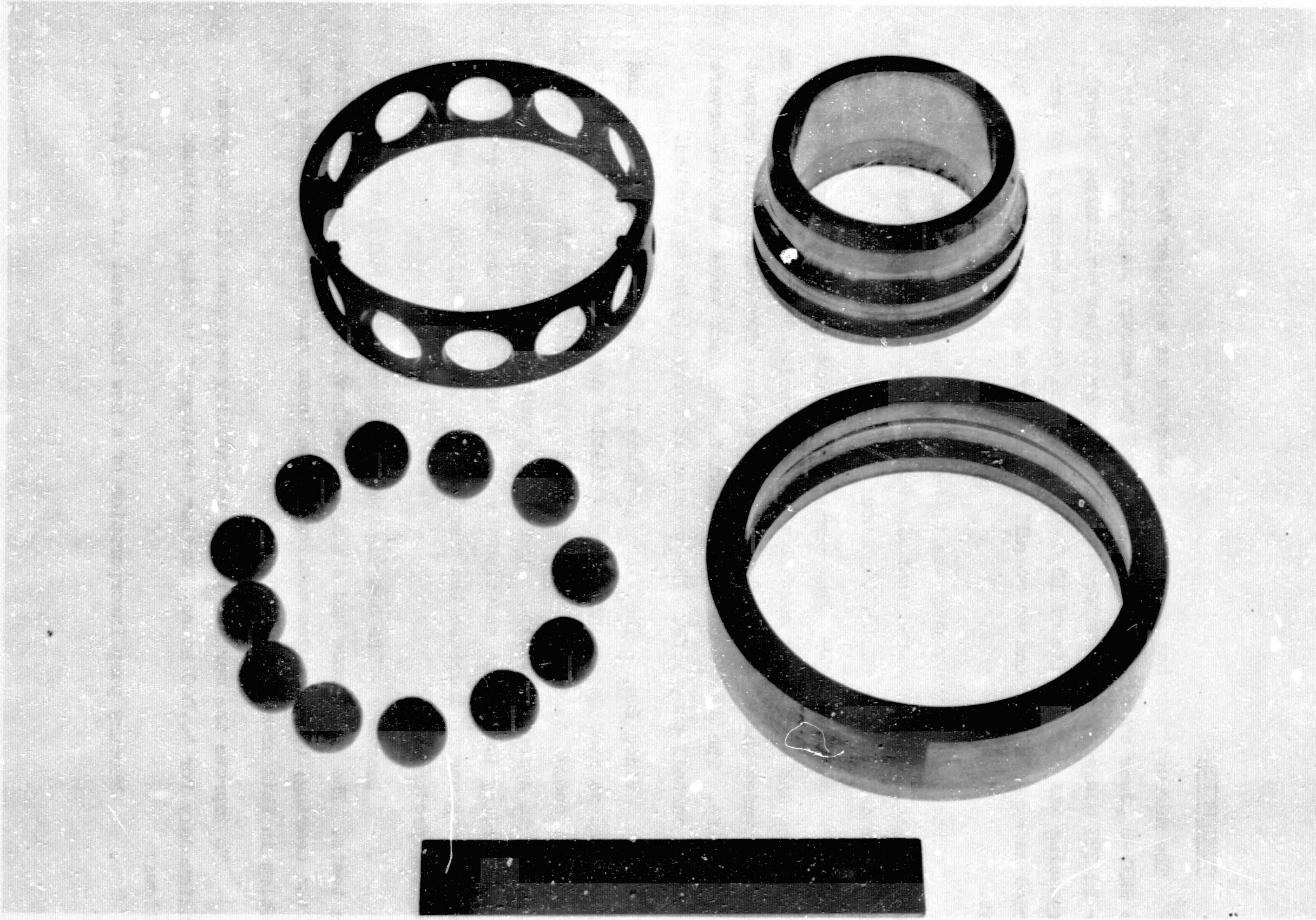

Figure 15. Motor End Bearing After 12,227 Hours Operation and 109 Start Cycles

IV. CONCLUSION

The mercury pump has proven to be adequate as a boiler feed pump for the SNAP-8 power conversion system. One of the mercury pumps has operated for 12,227 hours of satisfactory operation under system conditions, thereby surpassing its design life of 10,000 hours. The overall mercury pump performed as expected and had an overall efficiency of 12.7% as determined by test.

Although a mercury pump was tested for 12,227 hours without incident, post-test analysis indicated that the following factors might be life limiting beyond the design life of 10,000 hours:

- Pump Impeller - The pump impeller had cavitation damage on the front vanes and at the hub next to the back vanes. The cavitation damage on the front vanes was as expected and is not a cause for concern with respect to pump life. The damage on the hub next to the back vanes was more severe and might curtail pump life. The problem is currently being studied.

- Visco Pump - The visco pump was fouled with foreign matter and also showed cavitation damage. The cavitation damage was not considered to be serious enough to curtail pump life, but the fouling by the debris poses a problem in system operation. The fouling was attributed to mass transfer in the mercury loop. The solution is to minimize the mass transfer in the mercury loop, and to increase the pump's capacity in accepting the mass transfer products.

- Bearings - The bearings were in excellent condition following the tests. The inner race had a slight growth which could shorten the life of the bearing, but improved cooling of the inner race should eliminate any problem in this area.

To improve the mercury pump's reliability and capability to operate continuously for 40,000 hours, further development is being continued as follows:

- Mercury pump incorporation of a new face seal lift-off device.

- Analysis and resolution of the cavitation problem which exists at the impeller hub next to the back vanes.

- Study of techniques which might improve the cooling of the bearing inner race.

- Continuation of system and design refinement tests and analysis.

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