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SUMMARY REPORT

HIGH-TEMPERATURE HYDRAULIC SYSTEM ACTUATOR SEALS FOR USE IN ADVANCED SUPERSONIC AIRCRAFT

by

J. Lee

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

September 15, 1967

CONTRACT NAS 3-7264

FAIRCHILD HILLER
REPUBLIC AVIATION DIVISION
FARMINGDALE, LONG ISLAND, NEW YORK

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Work on the program at Republic Aviation Division of Fairchild Hiller Corporation was under the direction of Mr. R. Schroeder, Program Manager, with Mr. J. Lee acting as Project Manager. Participation by Messrs. J. Fazio, T. Kondakjian and V. Petruzzeli of Republic, is acknowledged.

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ABSTRACT

This report summarizes the results and findings of a program to design, develop, and evaluate dynamic rod seals for high-temperature hydraulic systems applications. Ten potential seal materials selected from the general categories of plastics, soft metals, and hard metals were investigated. Fluid compatibility and sliding wear tests were conducted on these materials to determine the five most suitable candidates. Seventeen rod seal configurations were investigated, and five designs were developed and evaluated in dynamic screening tests with silicone fluid. Three seal/materials combinations, each incorporating a different basic sealing concept, were evaluated in long-term testing in a simulated actuator test system.

The specified leakage requirement of less than one drop per minute was achieved with the polyimide V-seals during 1359 hours of operation at temperatures up to 500°F. Leakage of less than one drop per minute was also achieved with a Vascojet 1000/silver alloy reed seal and a cobalt molybdenum alloy lip seal during 574 and 279 hours of operation, respectively. The major shortcoming of the metallic seals was their limited ability to operate under boundary lubrication conditions without damaging the chrome-plated piston rod. The polyimide material appears capable of withstanding temperatures of 500°F. However, cracking of the material due possibly to fluid-temperature effects or deficiencies in the present seal design usually occurs after prolonged operation.

HIGH-TEMPERATURE HYDRAULIC SYSTEM ACTUATOR SEALS FOR USE IN ADVANCED SUPERSONIC AIRCRAFT

by J. Lee
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SUMMARY

The object of this program, sponsored under NASA contract NAS 3-7264, was to investigate materials and reciprocating seal designs having the potential to operate reliably for 3000 hours in the temperature range -40°F to +600°F. Candidate seal materials consisting of plastics, soft metals, and hard metals were investigated. Fluid compatibility tests of these materials were conducted with F-50 silicone, modified polyphenyl ether, halogenated polyaryl, deep-dewaxed paraffinio mineral oil, and a perfluorinated polymer fluid. These tests were conducted at 400, 500, and 600°F, for 150 hours at each temperature. Sliding wear tests were also conducted at room temperature with circular segments of the material in contact with a reciprocating chrome-plated rod immersed in silicone fluid and blanketed with purified nitrogen. Tensile tests were performed at 600°F to obtain essential seal design data. From the foregoing, five materials were selected for further investigation in the seal design effort.

Two-stage rod seal concepts were evolved and refined to utilize to best advantage the properties of each of the chosen materials. Low-pressure cycling tests of five candidate second-stage seals were performed at 400°F, 500°F, and 600°F for 50 hours at each temperature. Further seal refinements were investigated, and three configurations were selected for the endurance tests. These were the polyimide V-seal, cobalt molybdenum lip seal, and Vascojet 1000/silver alloy reed seal. First-stage seals consisting of polyimide split sealing rings were selected and combined with the second-stage seals for the endurance tests.

The feasibility of the design concepts were well demonstrated in the endurance testing. The Polyimide V-seals met the leakage requirements of less than one drop per minute during 1359 hours of operation at temperatures up to 500°F. The Vascojet 1000/silver alloy reed seals and a cobalt molybdenum alloy lip seal also met the specified leakage requirements during 574 hours and 279 hours of operation, respectively. The shorter operating life experienced by the hard metal seals was due to scoring of the chrome-plated piston rod. The polyimide material appears capable of withstanding temperatures of 500°F. Factors attributable to the premature failure of the polyimide V-seals are long-term fluid effects or deficiencies in the seal design.

INTRODUCTION

The concept of sustained supersonic flight has become a reality within the past several years, and the trend for the future is toward the development of even higher speed aircraft. As operating conditions become more severe with succeeding families of vehicles, design margins will decrease substantially. Not the least to be affected by these considerations are hydraulic system components, particularly the dynamic seals. Elastomeric seals are now meeting the requirements of current aircraft operating in the temperature range of -65°F to +275°F. However, the temperature extremes anticipated for future air vehicles will impose significantly greater demands on these materials and will limit their use.

The objective of this present program was to investigate advanced materials and seal concepts for possible use in fluid power systems of future supersonic aircraft. This investigation was therefore directed to dynamic rod seals intended to function efficiently for 3000 hours in the temperature range of -40°F to +600°F and operating pressures to 4000 psi.

Empahsis was placed on integrating material properties and seal design to obtain the optimum seal-material combination. The specific tasks accomplished were:

Task I	-	Preparation of existing test facilities and design and fabrication of seal test actuators and fixtures
Task II	-	Selection, procurement, and evaluation of candidate seal materials
Task III	-	Design of seals for the one- and three-inch rod sizes
Task IV	-	Low pressure testing of one- and three-inch rod seals at temperatures of 400°F, 500°F, and 600°F
Task V	-	Long-term testing of the most promising seal-material combinations in the one- and three-inch rod sizes
Task VI	-	Development and evaluation of a single-stage high-pressure rod seal in the one-inch rod size.

This report presents the results and findings of this research undertaking. Description of test equipment and conclusions are also included.

SECTION I

TEST EQUIPMENT

Initial effort in this program was devoted to modification of the existing seal test rig. Design and fabrication of the following additional test apparatuses and cycling rigs were also accomplished:

1) High-temperature fluid test apparatus for determining compatibility of candidate seal materials with various fluids

2) Sliding wear tester for evaluating wear characteristics of the candidate materials sliding on chrome plate

3) Seal friction measuring apparatus

4) Seal test actuators for evaluating one and three-inch seals

5) One- and three-inch seal test cycling rigs

For the sake of clarity and continuity, detail descriptions of the above equipment are presented in the appropriate sections of this report.

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SECTION II

MATERIAL SELECTION AND EVALUATION

A. GENERAL

Ten (10) potential seal materials were selected jointly by the contractor and NASA. These materials were evaluated to determine (1) compatibility with several fluids at 400°F, 500°F, and 600°F; (2) mechanical properties at 600°F; and (3) sliding wear characteristics (room temperature) against hard chromium plating.

B. SELECTION OF MATERIALS

Representative plastics, soft metals, and hard metals were selected for evaluation. Experimental elastomers such as perfluoroalkylene-triazine rubber were considered, but they were not sufficiently developed at the time to be included in this program.

Conformability, low friction and wear, and retention of desirable mechanical properties at high temperatures were major considerations in the selection of seal materials. The plastics primarily offer flexible and anti-galling behavior. Hard metals are insensitive to high temperatures and are convenient for developing adequate contact loads and self-compensating characteristics with simple configurations. The soft metals exhibit some elements of both these extremes. Brief descriptions of materials selected from these three categories are presented below.

1. Polyimide Plastics

Chemically, the polyimides are organic condensation polymers derived from the reaction between pyrometallic dianhydrides and aromatic diamines. In the polyimide group, the unfilled (Polymer SP-1*) material appears to be the most suitable. Rod seals designed from this material have

* DuPont Trademark

been evaluated (Reference 1) for 300 hours at temperatures up to 600°F with no failure. The copper-filled and bronze-filled polyimides were included in the preliminary evaluation phase of this program as possible alternates. The main drawback to the filled materials is a further decrease in their already limited flexibility.

2. Polymet*

This metal/polymeric alloy is composed of silver, which is alloyed during sintering with a fluorocarbon polymer; the resultant material possesses properties of the metal. The polymer content contributes a low coefficient of friction. The alloy also exhibits good wear resistance without lubrication at high temperatures. Promising results were obtained from previous work at 600°F with rod seals fabricated from this material (Reference 2).

3. Silver-Impregnated Metallic Composites

These composites consist of metal fibers that are sintered, brazed, or mechanically interlocked to form a porous metallic skeleton that is then filled with silver. This process produces a material that combines the conformability of the soft metal with the strength, creep resistance, and other desirable properties of the stronger material.

Past (Reference 3) and current work on silver composites was reviewed to determine the most suitable composite system. Parameters such as fiber density, diameter, spacing, and their influence on the strength of the composite structure were analyzed. In essence, the findings reveal that fine fibers (0.0025 inch in diameter) produce a more immediate strengthening of the silver matrix than is produced by the coarse fibers. Closer fiber spacing also gives an increase in strength of the composite. On the basis of these findings, work on silver composites emphasized the use of fine fiber structures.

Fiber bodies consisting of stainless steel (Type 430) and nickel in 30% and 40% densities were procured from the Huyck Corp. Impregnation of these materials with a silver-lithium alloy was accomplished at Republic facilities.

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* Polymer Corporation of America

4. Westinghouse Self-Lubricating Composite

This self-lubricating composite (70% silver and 30% tungsten diselenide) offers exceptionally low friction and good wearing qualities. Its thermal coeffi ient of expansion (6.45×10^{-6} in./in./°F) closely matches that of the Type 440C material proposed for the piston rod.

5. Silver Base Alloys

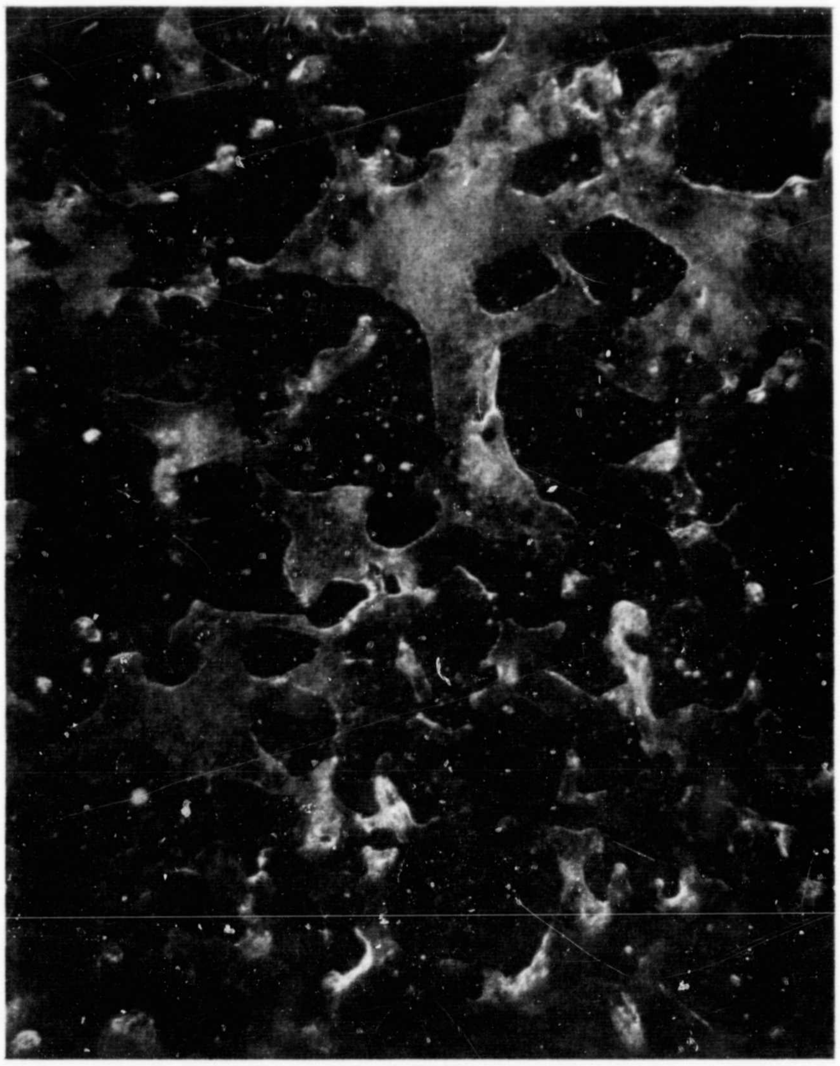
The candidate alloy selected was a silver-copper eutectic containing 72% silver and 28% copper. Past experience with silver alloys in seal applications has indicated that this eutectic exhibits better wear characteristics than commercially pure silver or coin silver, particularly from the standpoint of metal transfer in sliding applications. An alternate silver-copper alloy with up to 40% copper, which has the effect of raising the tensile strength in the annealed condition from 25,000 psi to 55,000 psi, was also considered.

6. Nickel Foametal

This material is prepared by a metal foaming process developed by General Electric Company's Metallurgical Products Department for abradable seal applications in gas turbine engines. It can be made in a wide range of hardnesses and with densities of between 2% and 75% of the unfoamed metal. Its performance may be enhanced by impregnating with plastics, metals, and ceramics. A 60% dense material was selected. This material was impregnated with a eutectic mixture of calcium fluoride and barium fluoride (see Figure 2-1). Based on preliminary work conducted by NASA Lewis Research Center (reference 4), this impregnant has exhibited good lubricating qualities and wear resistance.

7. Cobalt Alloy (75% Cobalt, 25% Molybdenum)

Cobalt in its hexagonal form (Reference 5) gives better friction characteristics than the cubic form of the metal in sliding application in vacuum. However, one difficulty encountered in the use of this material is that it undergoes a crystal transformation from hexagonal to the face-centered cubic form at temperatures of approximately 800°F. Initial transformation starts at about



Mag 250 x

Black - Metal
White - Fluoride

Figure 2-1. Nickel Foametal Impregnated with CaF_2 + BaF_2 Eutectic

600°F. Alloying cobalt with molybdenum in certain concentrations stabilizes the hexagonal crystalline form of the material up to at least 1300°F.

8. Titanium Alloy (84% Titanium, 16% Tin)

Titanium is an attractive metal for high-temperature applications because of its high strength-to-weight ratio. Little consideration has been given to this metal for sliding applications because of its relatively poor friction and wear characteristics. However, recent research (Reference 6) indicates that friction and wear characteristics may be improved by alloying with tin. Friction and wear data (Reference 6) obtained for tin-titanium alloys sliding against 440C in vacuum indicates that the addition of tin results in a marked decrease in friction coefficient.

9. Metco Flame-Plated Molybdenum Coating Burnished With MoS_2

This refractory material has been included in this investigation because of its good wear resistance and its good bond strength to most metal substrates. Titanium was selected as the base material because of its relatively low coefficient of thermal expansion (approximately 5×10^{-6} in./in.°F). The low coefficient of thermal expansion of titanium (as compared to steel) will minimize cracking of the coating due to differential expansion.

10. Vascojet 1000 (H-11 Type Tool Steel)

This alloy has a good combination of toughness and strength at elevated temperatures to 1000°F. An ultimate tensile strength of 268,000 psi can be obtained after air cooling from 1850°F and double tempering at 1050°F. Its nominal composition is 0.40% carbon, 5.0% chromium, 1.30% molybdenum, 0.5% vanadium, and approximately 0.013% sulfur. The presence of sulfur has proven to be beneficial in reducing friction, wear, and metal transfer of tool steels (Reference7).

11. Alternate Materials

In addition to the above materials, the following alternate materials were also considered:

1) Polymer SP plastics (a - 15% graphite filled, b - 30% bronze filled, and c - 20% copper filled)

2) Westinghouse composite (75% silver, 20% polyimide, and 5% tungsten diselenide)

3) Silver impregnated nickel fiber composite

4) Silver alloy (60% silver, 40% copper)

5) NM-100 high temperature bearing steel

6) Elastomers - newly developed elastomers were surveyed for high-temperature seal application. The most promising material uncovered during this survey was the perfluoroalkylene-triazine polymer, which was still in the laboratory phase of development and was not available. Another experimental elastomer was brought to the attention of the contractor during the survey. This material was developed by David Clark Co., Inc., and was designated as Omni Rubber No. X-FVF-19-15. However, high-temperature (400°F) aging tests conducted by the contractor indicated that this material did not exhibit any improvement in compression resistance as compared to a commercially available Viton compound. Consequently, the material was dropped from further consideration.

C. FLUID-MATERIAL COMPATIBILITY EVALUATION

Compatibility of the potential seal materials with five fluids was determined at temperatures of 400°F, 500°F, and 600°F. These five fluids were selected on the basis of their high temperature potentials and advanced stage of development:

- Perfluorimated polymer - (PR143AB)
- Modified polyphenyl ether - (MCS-293)
- Super refined paraffinic mineral oil - (MLO 60-294)
- Chlorophenyl methyl silicone - (F-50)
- Polyaryl - (MCS-3101)

Compatibility of the seal materials with the above fluids was determined over a 150-hour period at each temperature level. To reduce the time and material required to accomplish this task, initial tests were conducted at 600°F; those materials that did not perform satisfactorily at this temperature were re-evaluated at 400°F. The materials that were unsatisfactory at 400°F were eliminated from further consideration. Only those materials that exhibited good compatibility at 400°F were retested at 500°F.

Two additional fluids were later included in the program:

- Trifluoropropyl halophenyl substituted silicone copolymer - XF-1-0291
- Trifluoropropyl methyl polysiloxane - XF-1-0294

Compatibility testing with these two fluids was conducted at 600°F only (see Paragraph D).

1. Test Equipment and Procedures

The fluid-material compatibility test apparatus is shown in Figures 2-2 and 2-3. The setup consists of a bank of five manifolds, each manifold holding eleven capsules. The capsules in one manifold were all filled with the same hydraulic test fluid; ten of the capsules contain materials for compatibility tests and the eleventh contains fluid for use only as a control sample. Approximately 30 ml of fluid was used in each capsule. All the capsules were immersed in a container filled with molten salt. This entire assembly was placed in an oven and brought up to operating temperature. The molten salt bath ensured a uniform and constant temperature for all the capsules.

Leading from each manifold was a line joined to two manifolds located outside the oven. One of these manifolds was connected to a vacuum pump and was isolated from the other manifold by means of five valves. The second manifold was connected to a shut-off valve and thence to a nitrogen gas supply. The gas used to provide an inert atmosphere was 99.99 percent nitrogen by volume, with an oxygen content of not more than 50 ppm, a hydrocarbon content (methane) of not more than 5 ppm, and a dew point of -90°F or lower. The gaseous nitrogen was admitted into the system through check valves. The purpose of these check valves was to prevent intermixing of vapors from the five test fluids. However, it was discovered during initial testing that the check valves did not provide an effective seal against fluid vapors. To prevent resulting intermixing between manifolds, these valves were replaced with positive shut-off manual valves.

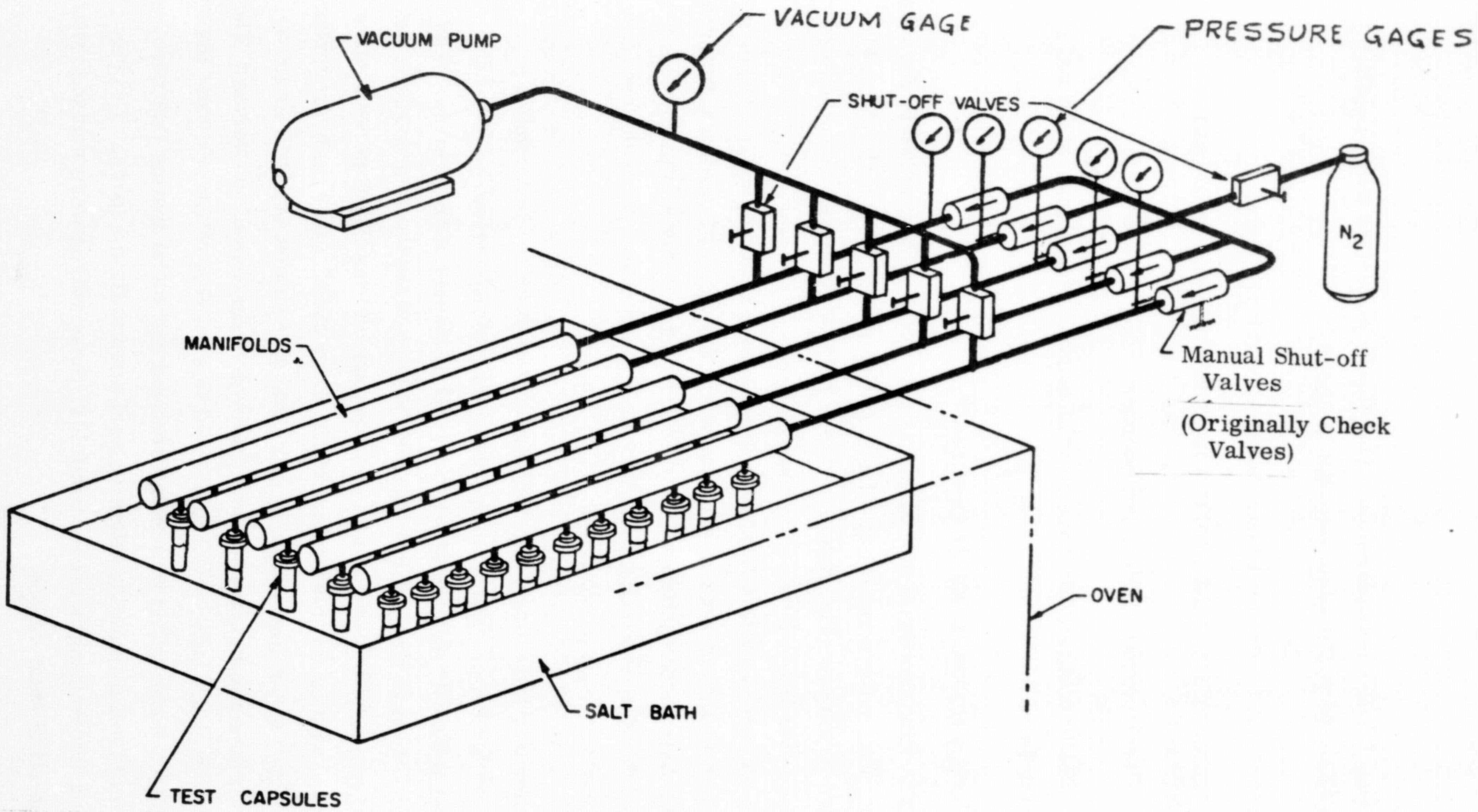


Figure 2-2. Schematic - Test Setup for Compatibility Tests

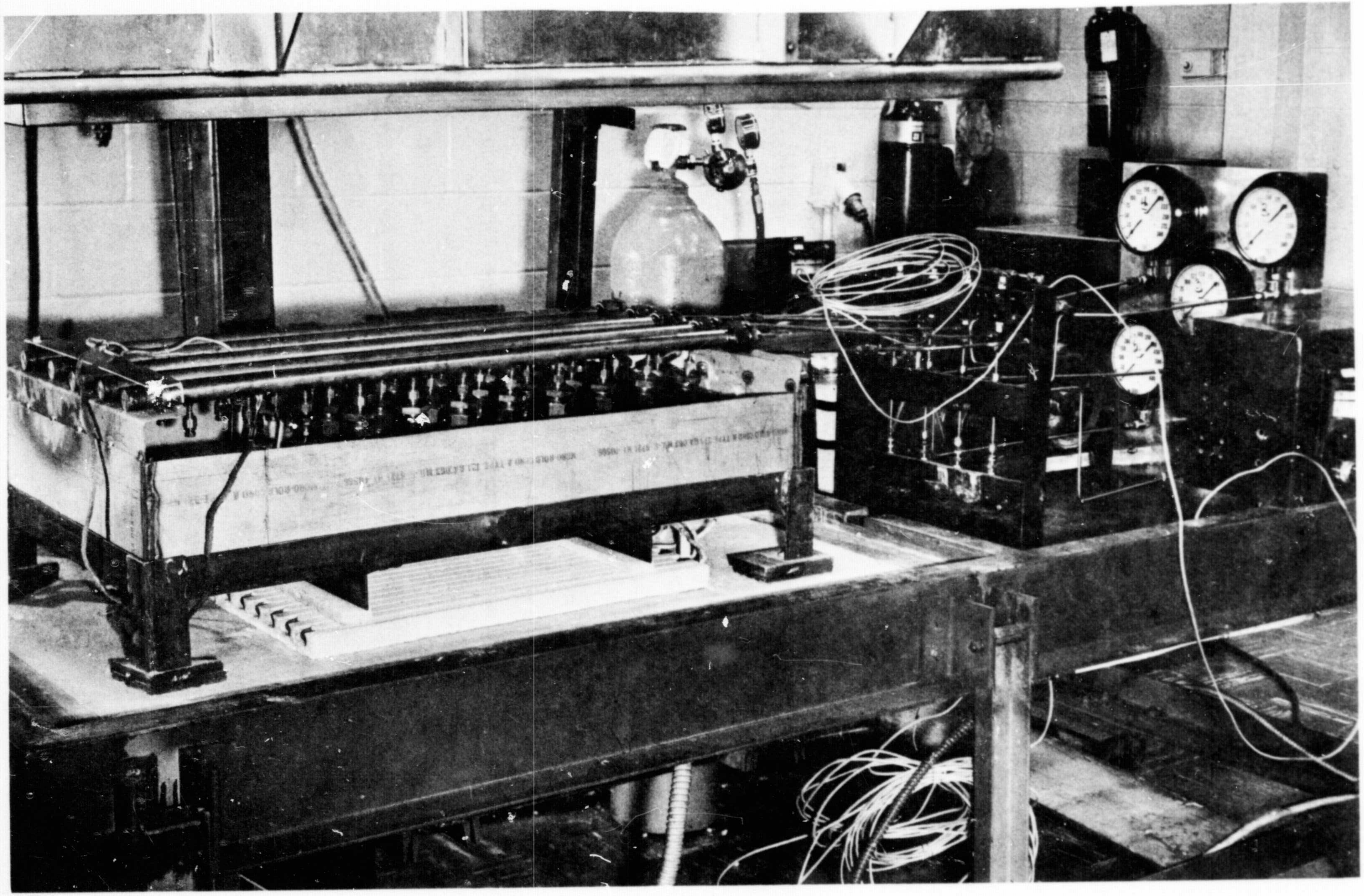


Figure 2-3. Test Setup for Compatibility Tests

Hardness readings were taken on the bulk material prior to machining the specimens. Each specimen was weighed prior to testing. As shown in Figure 2-4, the test specimen, which is approximately 7/8 inch in diameter, was kept in face contact with a polished, hard chromium-plated stainless steel button by means of a screw and spring assembly. This arrangement provided a constant contact pressure between the test specimen and button at elevated temperatures.

Each specimen was inserted in a capsule as shown in Figure 2-5. The capsules were fabricated from one-inch diameter stainless steel tubing with flared ends for attaching AN919-21 reducers (one-inch to 1/4-inch tube size). Stainless steel was selected as the capsule material because it is most representative of the seal environment during operation in flight actuators.

As the first step in the compatibility tests, the fluids were degassified for 72 hours at a vacuum of 29 inches of Hg. This was accomplished by drawing a vacuum in the system after the fluids have been warmed slightly. After degassification, the five vacuum isolating valves were closed, approximately one atmosphere of nitrogen was admitted into the system through check valves, and the entire apparatus was brought up to the test temperature.

2. <u>Fluid-Material Compatibility Testing at 600°F</u>

Two tests were conducted at 600°F. Test No. 1 was run with only those materials that were available during the early phase of the program. Test No. 2 included essentially all of the candidate materials.

Candidate materials included in Test No. 1 were as follows:

1) Silver - polymer composite (Polymet)
2) Polyimide - (unfilled)
3) Polyimide - (15% graphite filled)
4) Polyimide - (30% bronze filled)
5) Polyimide - (20% copper filled)
6) NM-100 (high-temperature bearing steel)
7) Silver-impregnated metal composite (430 SS, 35% dense)

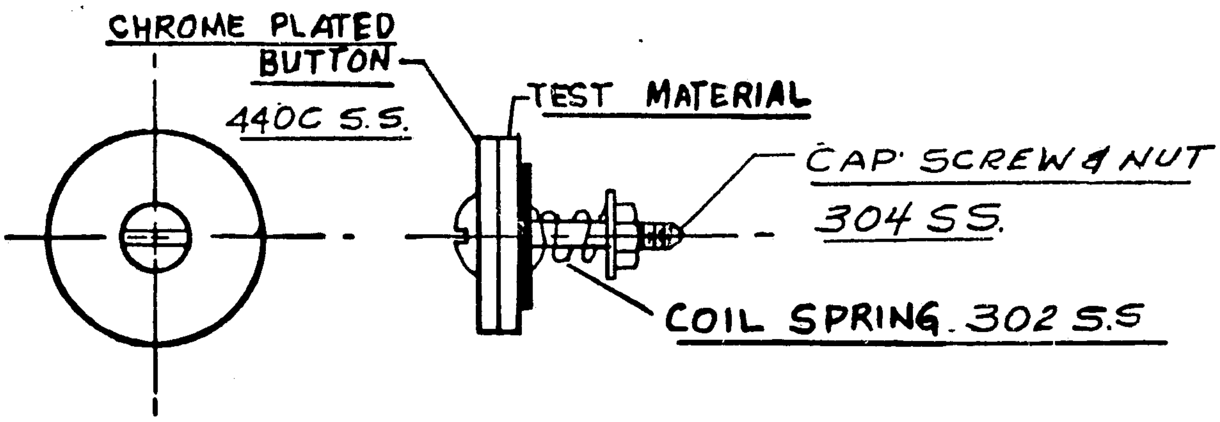


Figure 2-4. Test Specimen Assembly

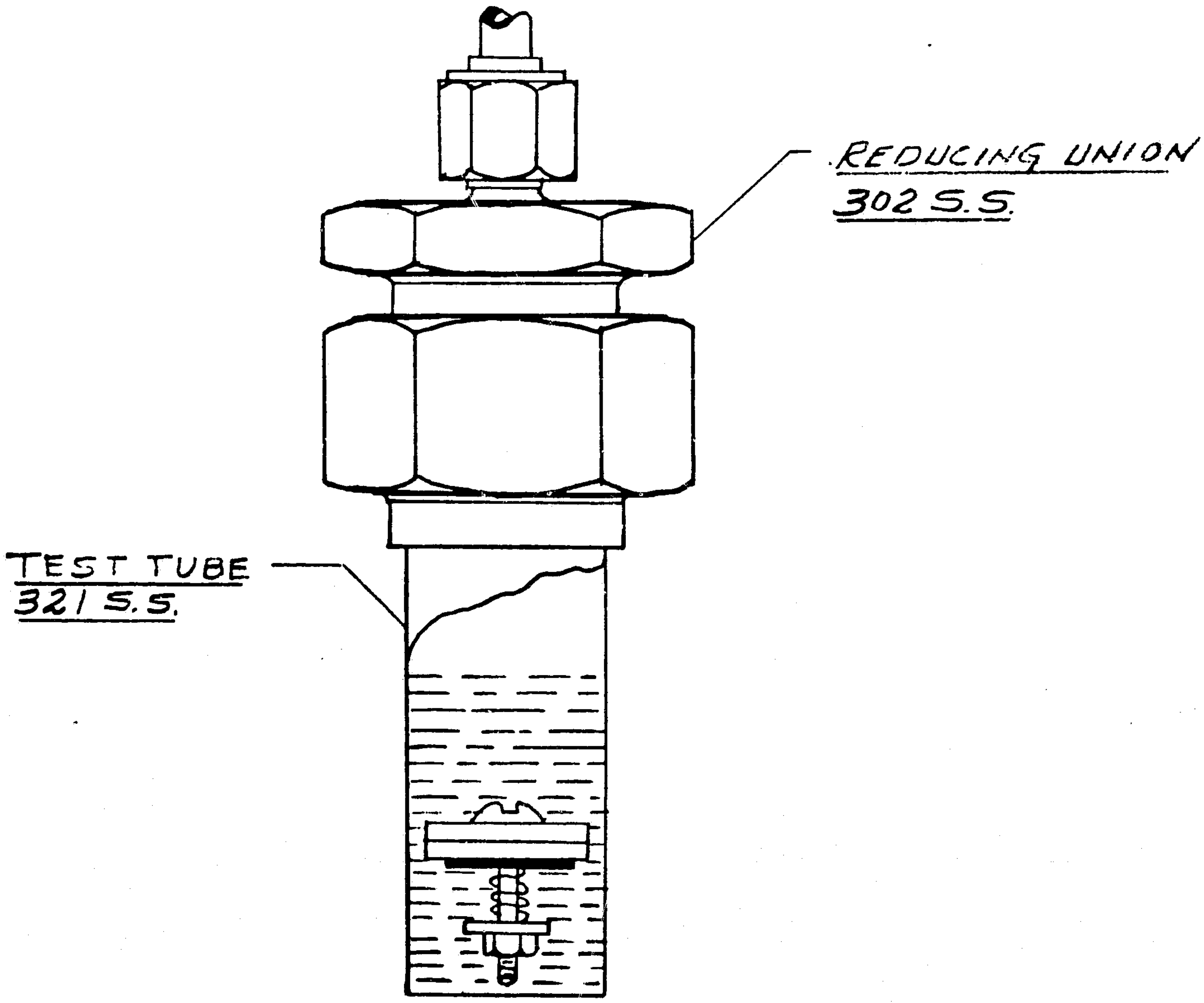


Figure 2-5. Test Capsule

With the exception of the MCS-3101, all of the fluids completed the 150-hour test. Evidence of degradation of the MCS-3101 was noticed after 95.5 hours at 600°F when pressure in the manifold built up from 15 psi to 210 psi. Consequently, this fluid was removed from the test. Visual inspection of this fluid showed considerable sludge formation accompanied by a strong acidic odor.

The data resulting from the compatibility test is summarized in Tables 2-1 through 2-5. The appearance of the material specimens are shown in Figure 2-6. In general, the most compatible fluid with this group of materials is the MLO-60-294, followed by PR-143-AB and MCS-293. The F-50 was fairly compatible with the polyimide base materials and the silver-polymer (Polymet) composite. However, the NM-100 steel alloy and silver-stainless steel composite exhibited some corrosion. The MCS-3101 was not compatible with any of the materials tested. The specimens showed evidence of oxidation and also exhibited a coating of baked oil. Considerable corrosion was exhibited by the NM-100 steel alloy and the silver-stainless steel composite material on the side opposite the chrome-plated disc.

Inspection of the chrome-plated mating buttons indicated that some form of protective coating is necessary on Type 440C stainless steel to prevent its corrosion in the presence of F-50, MCS-3101, MCS-293, and PR-143AB fluids when subjected to elevated temperatures. Corrosion in various degrees was observed on the unplated side of the test disc with these fluids. In the case of the MLO-60-294 mineral oil, only slight discoloration occurred.

Viscosity values at 100°F and 310°F, as well as the acid numbers at room temperature, were obtained for all the fluids except the MCS-3101. This fluid, which degraded after 95.5 hours at 600°F, was sent to Monsanto for analysis. The MLO-60-294 decreased in viscosity at 100°F and 210°F (29% and 18%, respectively) but increased slightly in acidity. The viscosity change of the PR143AB fluid was negligible. The acid numbers for this fluid could not be determined because the standard method (ASTM D664-58) was not applicable. Slight variations in the viscosity and acid number were exhibited by the MCS-293 fluid. The appearance of the fluid samples is shown in Figures 2-7 and 2-8.

TABLE 2-1

FLUID - MATERIAL COMPATIBILITY TEST NO. 1 - 150 HRS. @ 600°F
MCS-3101 HALOGENATED POLYARYL FLUID
30 ML PER TEST SPECIMEN

SPECIMEN	Polymet Ag	Polyimide (Unfilled)	Polyimide (Graphite)	Polyimide (Bronze)	Polyimide (Copper)	NM-100	Silver-SS Composite	Control
Weight Change of Specimen (Grams)	+.1677	+.0136	+.0367	+.5426	+.2281	-.0141	-.0679	
** Hardness Change	From H-48 to H-24	From H-90 to H-97 spec. cracked	From H-86 to H-85	From H-97 to H-70 spec. cracked	Specimen cracked	From C-38.5 to C-41.5	From H-97.5 to H-60.0	
Appearance of Specimen	Deposit of burnt oil	Darkened	Darkened deposit of burnt oil	Corroded, pitted, oxidized	Corroded, pitted, oxidized	Corroded	Corroded and pitted	
Appearance of Mating Surface	Deposit of burnt fluid. Heavy corrosion on unplated side.	Heavy deposit. Unplated side heavily corroded.	Discolored, uniform black deposit. Unplated side heavily corroded.	Heavy black deposit. Unplated side heavily corroded.	Heavy black deposit. Unplated side heavily corroded.	Heavy black deposit. Unplated side heavily corroded.	Uniform black deposit. Unplated side heavily corroded.	
Appearance of Fluid	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black	Extremely viscous and black
Viscosity @ 100°F, CS (4.34)	-	-	-	-	-	-	-	-
Viscosity @ 210°F, CS (1.32)	-	-	-	-	-	-	-	-
Acid No. mg KOH/g (0.11)	-	-	-	-	-	-	-	*

* Fluid severely degraded and shipped to Monsanto for analysis.

** Rockwell-C and Rockwell-H scale

TABLE 2-2

FLUID - MATERIAL COMPATIBILITY TEST NO. 1 - 150 HRS. @ 600°F
MLO-60-294 SUPER REFINED MINERAL OIL FLUID
30 ML PER TEST SPECIMEN

SPECIMEN	Polymet Ag	Polyimide (Unfilled)	Polyimide (Graphite)	Polyimide (Bronze)	Polyimide (Copper)	NM-100	Silver-SS Composite	Control
Weight Change of Specimen (Grams)	+.0069	-.0038	-.0077	-.0166	-.0072	-.0024	+.0015	
* Hardness Change	From H-48 to H-53	From H-90 to H-83	From H-86 to H-77	From H-97 to H-90	From H-89 to H-83	From C-38.5 to C-43	From H-97.5 to H-90.0	
Appearance of Specimen	No change on side mating with chrome disc, discolored on opposite side.	Slightly darkened	Slightly darkened	Slightly darkened	Slightly darkened	Discolored	No change	
Appearance of Mating Surface	No change. Unplated side discolored.	No change. Unplated side slightly discolored.	No change. Unplated side slightly discolored.	No change. Unplated side slightly discolored.	Slight discoloration on both sides.	No change. Unplated side discolored.	No change. Unplated side discolored.	
Appearance of Fluid	Dark amber	Dark amber	Dark amber	Dark amber	Dark amber	Dark amber	Dark amber	Dark amber
Viscosity @ 100°F, CS (15.02)	10.32	10.76	10.85	10.76	10.63	10.88	9.75	10.96
Viscosity @ 210°F, CS (3.26)	2.61	2.76	2.66	2.61	2.67	2.62	2.59	2.73
Acid No. mg KOH/g (0.02)	0.06	0.05	0.05	0.05	0.04	0.08	0.05	0.12

* Rockwell-C and Rockwell-H scale

TABLE 2-3

FLUID - MATERIAL COMPATIBILITY TEST NO. 1 - 150 HRS. @ 600°F
PR-143-AB FLUOROCARBON FLUID
30 ML PER TEST SPECIMEN

SPECIMEN	Polymet Ag	Polyimide (Unfilled)	Polyimide (Graphite)	Polyimide (Bronze)	Polyimide (Copper)	NM-100	Silver-SS Composite	Control
Weight Change of Specimen (grams)	+.0091	-.0062	-.0124	-.0259	-.0138	+.0004	+.1326	
* Hardness Change	From H-48 to H-41	From H-90 to H-85	From H-86 to H-75	From H-97 to H-93	From H-89 to H-80	From C-38.5 to C-43.5	From H-97.5 to H-61.0	
Appearance of Specimen	No change	Slight darkening	Slight darkening	Slight darkening	Slight darkening	Slight discoloration	Discolored on side mating w/chrome disc. corroded on opposite side.	
Appearance of Mating Surface	No change. Heavy corrosion on unplated side.	No change. Unplated side heavily corroded.	Green deposit. Unplated side heavily corroded.	Discolored. Unplated side heavily corroded.	Discolored. Unplated side heavily corroded.	Discolored. Unplated side heavily corroded	Discolored, heavy deposit. Unplated side heavily corroded.	
Appearance of Fluid	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.	Cloudy with a top layer of dark amber fluid.
Viscosity @ 100°F., CS (60.13)	58.07	59.13	62.81	59.72	56.34	56.94	59.27	59.88
Viscosity @ 210°F., CS (7.94)	8.16	7.89	8.53	8.30	7.79	8.24	7.87	8.21
Acid No. mg KOH/g	Standard method of test for neutralization number by potentiometric titration (ASTM D664-58) is not applicable with this fluid.							

* Rockwell-C and Rockwell-H scale

TABLE 2-4

FLUID - MATERIAL COMPATIBILITY TEST NO. 1 - 150 HRS. @ 600°F
MCS-293 - MODIFIED POLYPHENYL ETHER FLUID
30 ML PER TEST SPECIMEN

SPECIMEN	Polymet Ag	Polimide (Unfilled)	Polyimide (Graphite)	Polyimide (Bronze)	Polyimide (Copper)	NM-100	Silver-SS Composite	Control
Weight Change of Specimen (Grams)	+.0118	-.0062	-.0025	-.0266	-.0153	+.0868	-.0005	
* Hardness Change	From H-48 to H-49	From H-90 to H-87	From H-86 to H-74	From H-97 to H-92	From H-89 to H-80	From C-38.5 to C-46	From H-97.5 to H-83.0	
Appearance of Specimen	No change	Darkened	No change	Slight pitting	No change	No change on side mating with chrome disc. Corrosion on opposite side.	Tarnished on side mating with chrome disc. Corrosion on opposite side.	
Appearance of Mating Surface	Discolored Slight deposit. Corroded on unplated side.	Greenish deposit. Unplated side corroded.	Discolored. Slight uniform deposit. Unplated side corroded.	Heavy dark deposit. Unplated side corroded.	Slight deposit. Unplated side corroded.	Greenish deposit. Unplated side corroded	Greenish deposit. Unplated side corroded.	
Appearance of Fluid	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown
Viscosity @ 100°F, CS (48.18)	66.54	284.84	395.13	128.39	82.30	145.99	45.14	246.79
Viscosity @ 210°F, CS (15.99)	31.13	62.63	78.87	35.95	24.18	54.58	17.36	56.16
Acid No. mg KOH/g (0.03)	0.30	0.30	0.20	0.30	2.68	3.38	6.42	6.70

* Rockwell-C and Rockwell-H scale

TABLE 2-5

FLUID - MATERIAL COMPATIBILITY TEST NO. 1 - 150 HRS. @ 600°F
F-50 CHLOROPHENYL METHYL SILICONE FLUID
30 ML PER TEST SPECIMEN

SPECIMEN	Polymet Ag	Polyimide (Unfilled)	Polyimide (Graphite)	Polyimide (Bronze)	Polyimide (Copper)	NM-100	Silver-SS Composite	Control
Weight Change of Specimen (Grams)	+.0040	-.0028	-.0041	+.0070	+.0011	+.0007	+.0071	
* Hardness Change	From H-48 to H-43	From H-90 to H-89	From H-86 to H-78	From H-97 to H-84	From H-89 to H-79	From C-38.5 to C-43.0	From H-97.5 to H-70.5	
Appearance of Specimen	Slightly discolored	No change	No change	Discolored, surface roughened	Discolored, dark deposit	Slightly darkened	Darkened and roughened	
Appearance of Mating Surface	No change. Slight discoloration on unplated side.	No change. Slight discoloration on unplated side.	No change. Slight discoloration on unplated side.	No change. Slight discoloration on unplated side.	Slight discoloration. Unplated side slightly discolored.	Slight deposit. Unplated side slightly discolored.	Discolored. Slight deposit. Unplated side slight deposit.	
Appearance of Fluid	Light brown	Light brown	Light brown	Light brown	Light brown	Light brown	Light brown	Light brown
Viscosity @ 100°F, CS (25.30)	28.26	28.62	29.13	24.48	27.91	28.92	28.11	28.50
Viscosity @ 210°F, CS (4.18)	4.50	4.54	4.52	4.13	4.59	4.50	4.47	4.59
Acid No. mg KOH/g (0.01)	0.03	0.03	0.04	0.03	0.04	0.05	0.05	0.05

* Rockwell-C and Rockwell-H scale

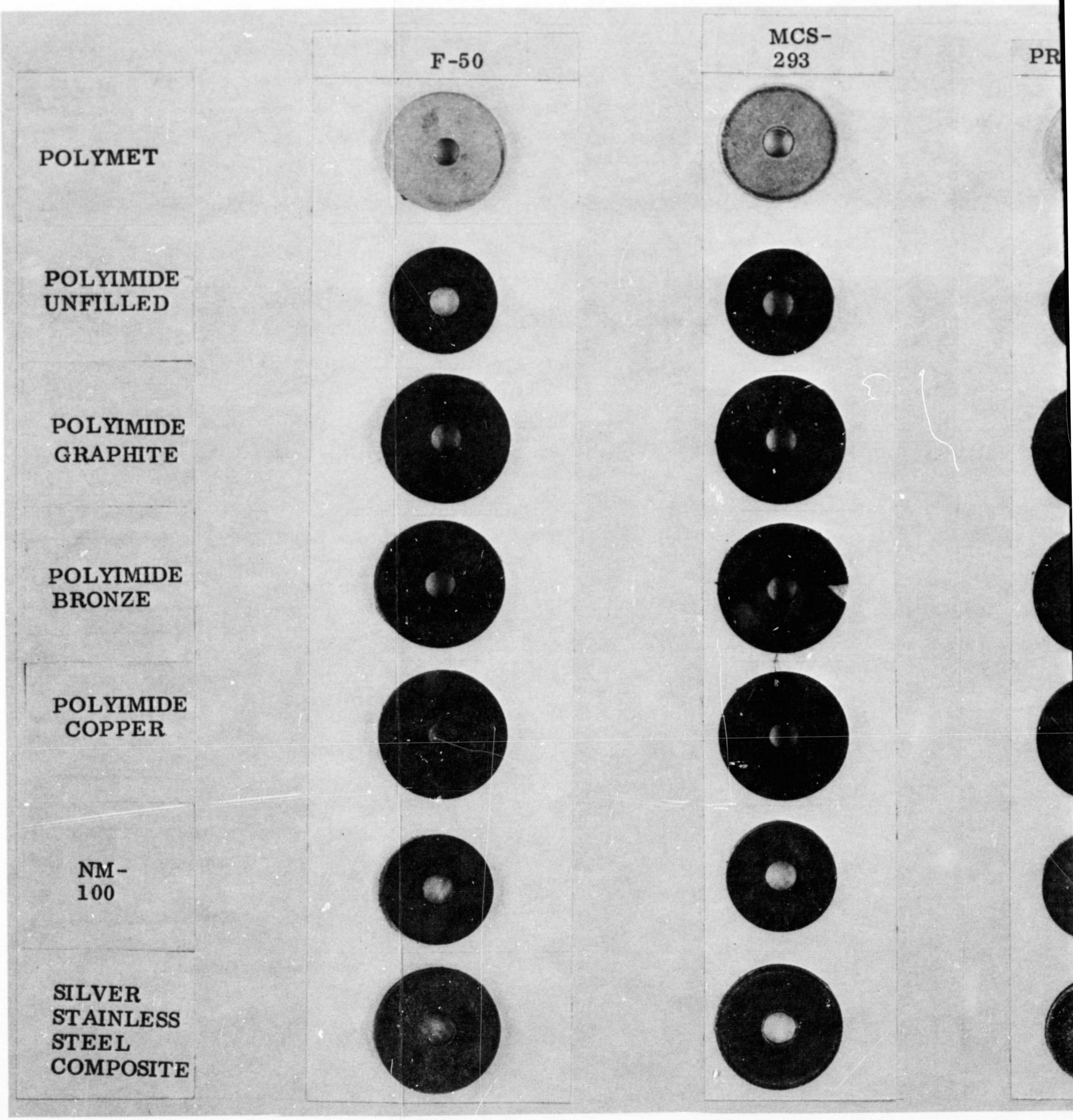


Figure 2-6. Test Specimens -- 150 Hours at 600°F

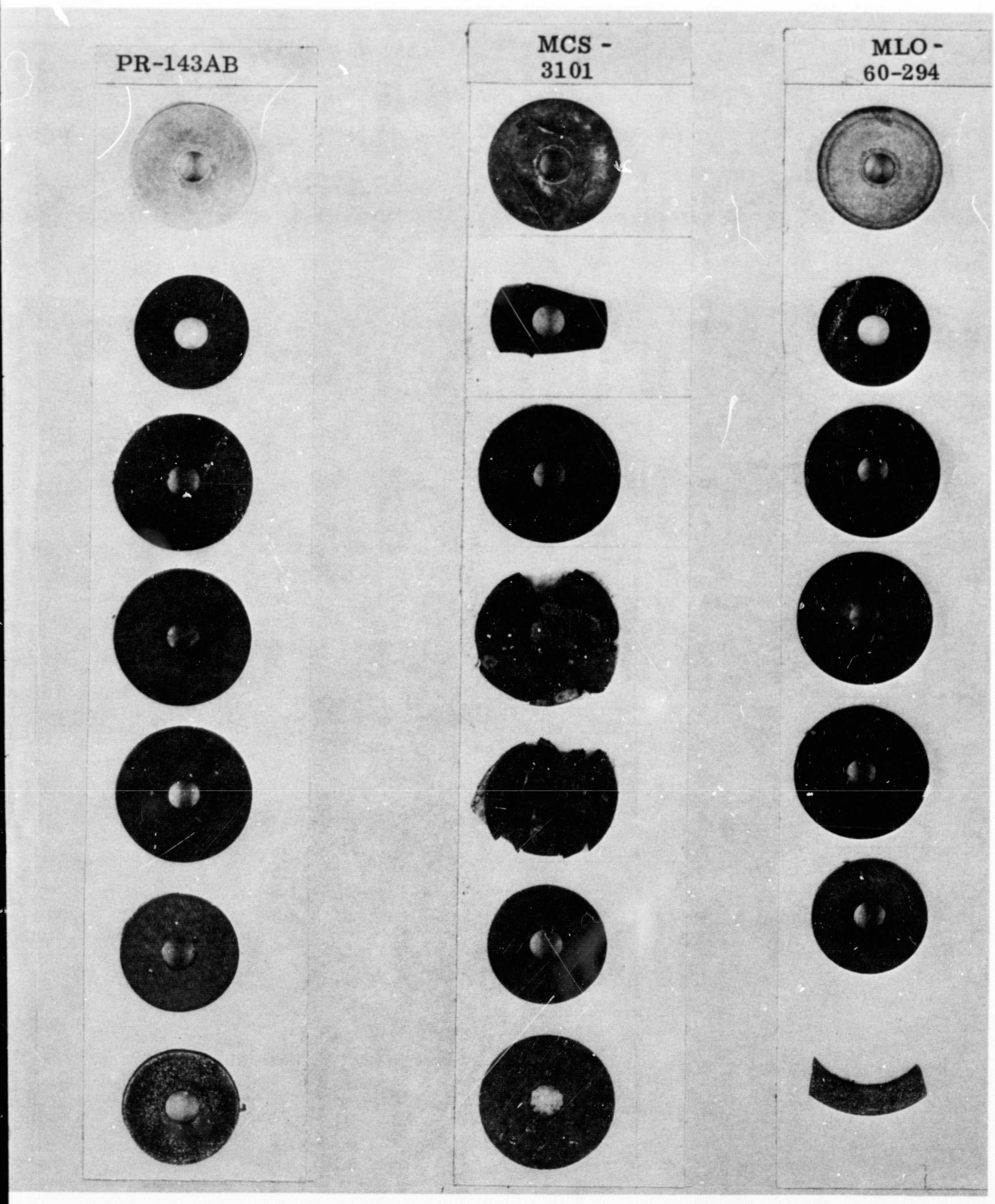

PR-143AB
MCS - 3101
MLO - 60-294

Figure 2-7. MCS-293 and MLO-60-294 Fluids After Testing at 600°F

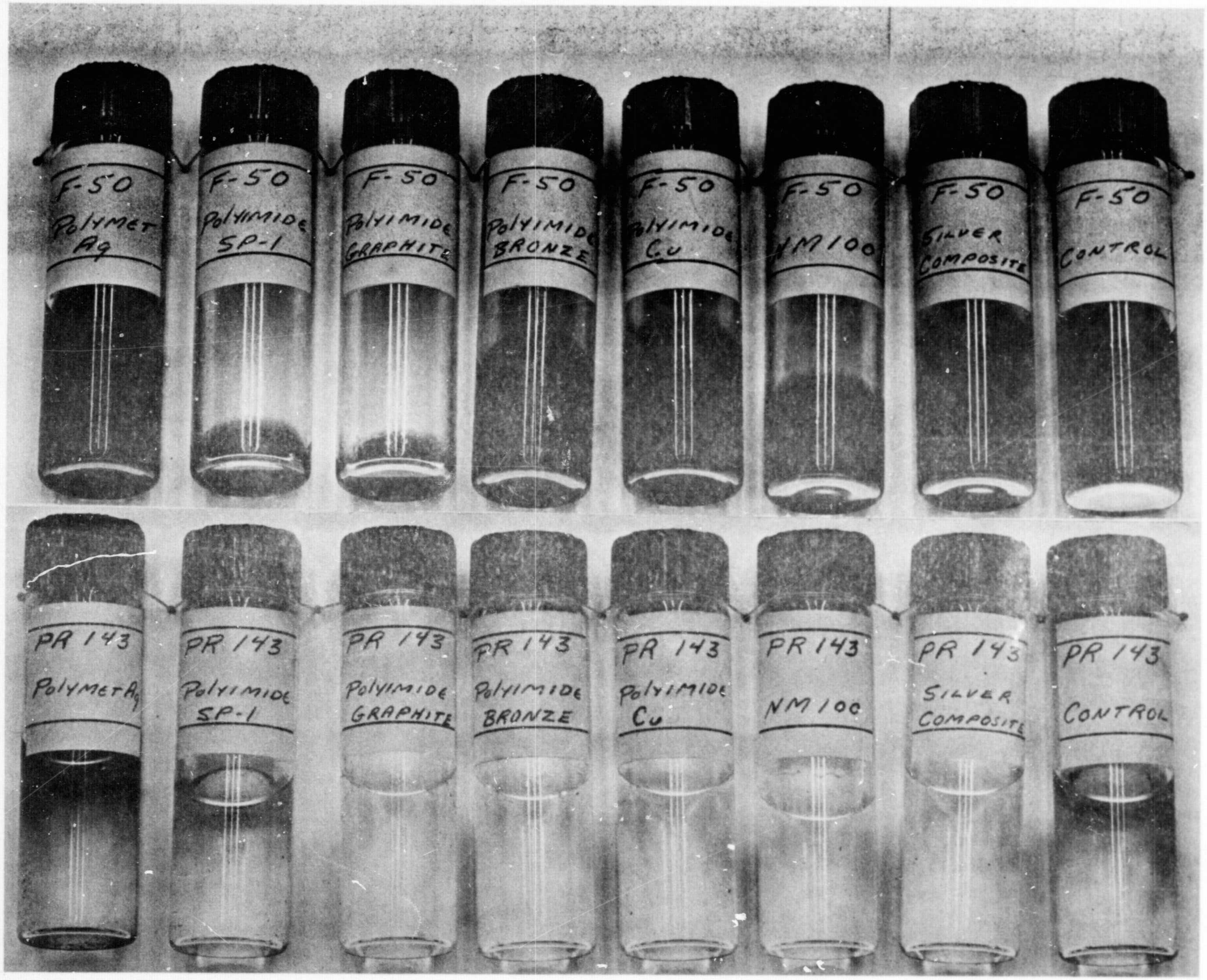


Figure 2-8. F-50 Silicone and PR-143-AB Fluids After Testing at 600°F

A considerable increase in viscosity was experienced by the F-50 silicone fluid. The acid numbers obtained were unusually high for the control fluid sample and the fluid samples containing the copper-filled polyimide, the NM-100, and the silver-stainless steel composite.

A slight intermixing of the fluids had occurred during the test. The presence of a second fluid was very apparent in the PR-143AB because of the difference in density and the water-like appearance of the fluid. The small quantity of fluid (approximately 1 cc) floating on top of the PR-143AB was assumed to be MLO-60-294, because of its appearance and proximity to the PR-143AB test manifold. It was conceivable that fluid intermixing occurred in the other manifolds. However, this was difficult to ascertain visually because of the opaqueness of these latter fluids. Intermixing was caused by leakage past the check valves (see Figure 2-2). The test setup was reworked to replace the check valves with positive-closing manual shutoff valves prior to further use.

The candidate seal materials evaluated in Test No. 2 are listed below.

1) Westinghouse composite (70% silver, 30% tungsten diselenide)
2) Silver alloy (72% silver, 28% copper)
3) Westinghouse composite (70% silver, 25% polyimide, and 5% tungsten diselenide) - (alternate material)
4) Silver-stainless steel (Type 430) composite
5) Silver-nickel composite - (alternate material)
6) Nickel Foametal impregnated with calcium fluoride and barium fluoride
7) Vascojet 1000 (H-11 tool steel)
8) Titanium tin alloy
9) Cobalt molybdenum alloy
10) Metco flame-plated molybdenum (titanium base material)

The above materials were tested with F-50, MCS-293, PR-143AB, and MLO 60-294 fluids. Polymer SP and Polymet, which are prime candidate materials, were not included in the test with the foregoing fluids as Test No. 1

indicated that they were compatible. This provided an opportunity to obtain data on two alternate materials. The MCS-3101, fluid which exhibited poor thermal stability in Test No. 1 was virtually eliminated from further consideration as a 600°F fluid. However, a limited number of MCS-3101 fluid samples were included in this test to verify results obtained in Test No. 1. These samples were evaluated in four separate test tubes apart from the main test apparatus. One tube contained the MCS-3101 fluid as the control; the other three tubes contained Polymer SP, Polymet, and the silver-stainless steel (Type 430) composite, respectively.

All the fluids completed the 150 hour test at 600°F. General condition of the fluids and material specimens are shown in Figures 2-9 through 2-12. Changes in the viscosity and acidic condition of the fluid samples, and hardness changes of the materials specimens are summarized in Tables 2-6 to 2-9. With the exception of the material with the F-50 fluid, the results indicate that fairly good compatibility exist between the candidate seal materials and PR-143AB, MCS-293, and MLO-60-294 fluids.

The F-50 silicone fluid appears to be unstable at 600°F. A considerable increase in viscosity and acidity was exhibited by this fluid. The control fluid, in particular, showed a viscosity of 997 centistokes at 100°F, as compared to the original viscosity of 48.18 centistokes. High acid numbers (mg KOH/g), were exhibited by the fluid samples containing the silver alloy (72% Ag, 28% Cu) and the nickel Foametal impregnated with CaF_2 and BaF_2. Acid numbers for these fluid samples were 20.2 and 19.0 (mg KOH/g), respectively. The fluid crystalized in the tube containing the Westinghouse composite (70% silver + 30% tungsten diselenide). Discussions held with the producer (Westinghouse) of this material revealed that the incompatibility may have been due to an alloying element in the silver. Sight corrosion was exhibited by the silver-stainless steel (Type 430) composite. The Vascojet 1000 exhibited slight corrosion when exposed to air after being in contact with F-50 silicone. It was also discovered that the F-50 fluid containing the Metco flame-plated molybdenum specimen solidified in

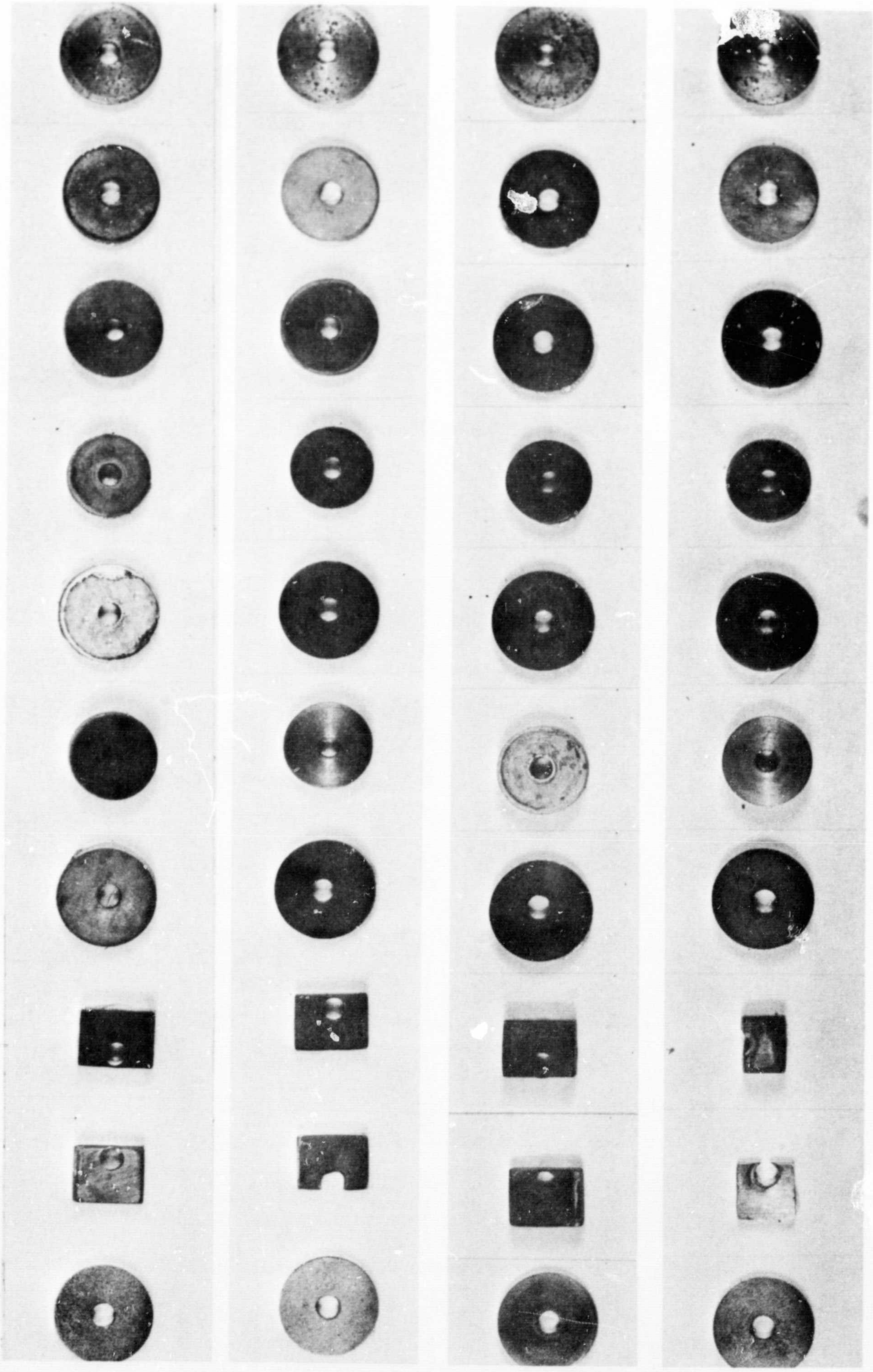


Figure 2-9. Test No. 2 - Material Specimens After Test at 600°F

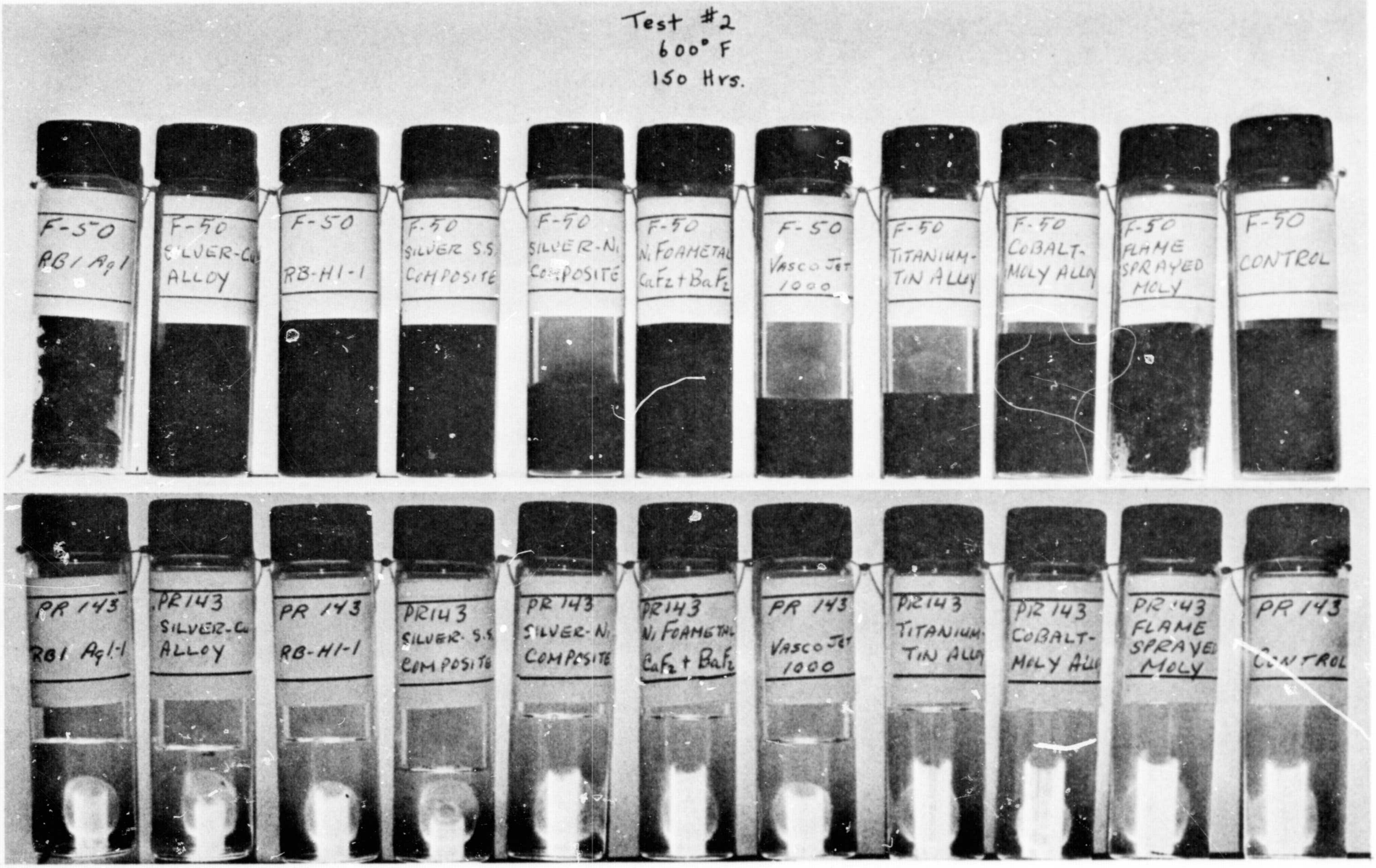


Figure 2-10. Test No. 2 - F-50 Silicone and PR-143AB After Test at 600°F

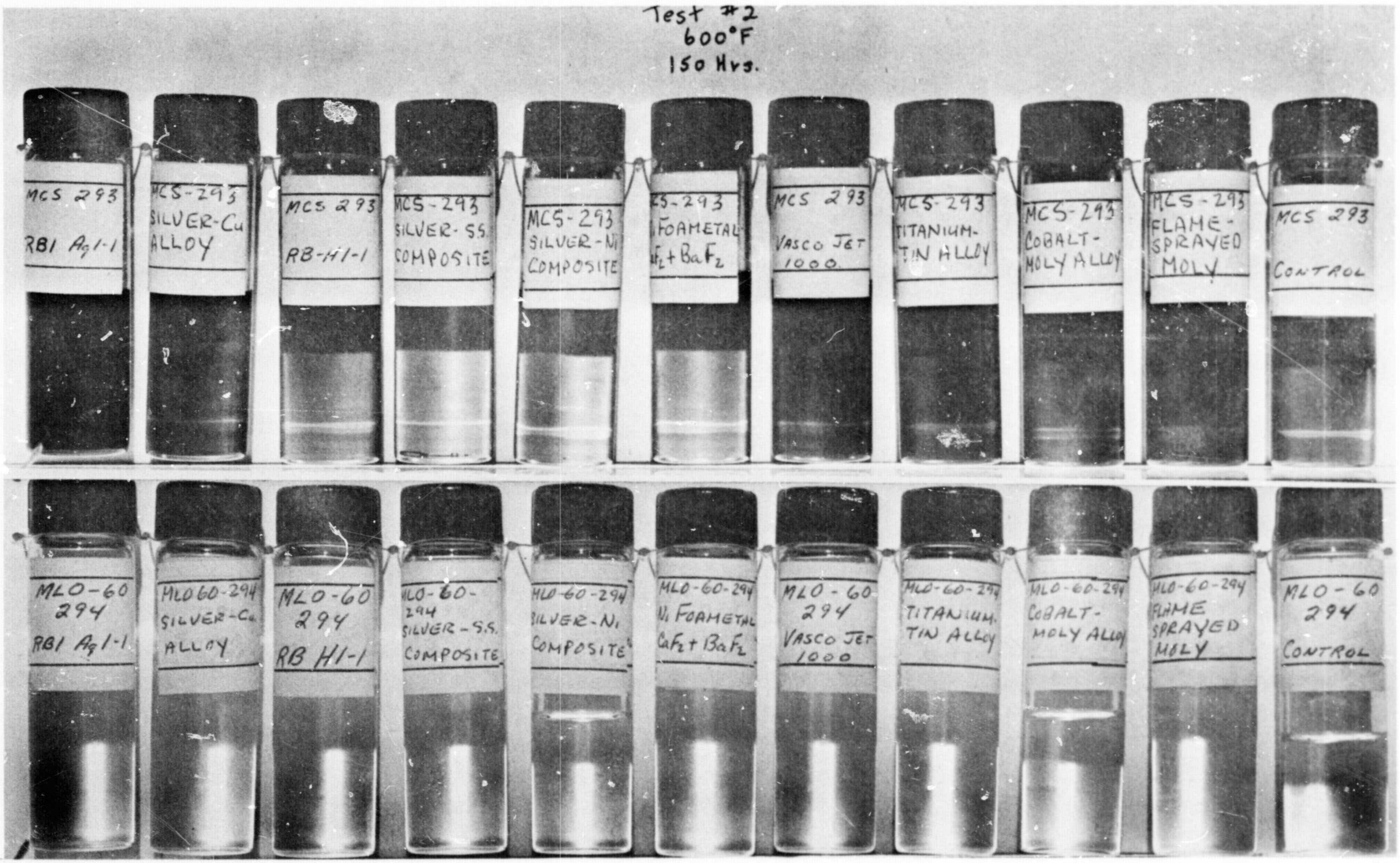


Figure 2-11. Test No. 2 - MCS-293 and MLO-60-294 After Test at 600°F

Figure 2-12. Test No. 2 - MCS-3101 After Test at 600°F

TABLE 2-6

FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
F-50 CHLOROPHENYL METHYL SILICONE FLUID
25 ML PER TEST SPECIMEN

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium-Tin Alloy	Cobalt-Molybdenum Alloy	Plasma Flame-Plated Molybdenum	Control
Weight Change (Grams)	-.0287	+.0020	+.0166	+.0185	+.1566	+.0149	-.0004	+.0010	+.0046	+.0016	
Hardness Change	H85 to H66	F89 to F91	H94 to H93.5	H90 to H96.5	H83 to H98	F87 to F9[illegible]	RC19 to RC16	RC36 to RC345	RC5 to RC2	RC39 to RC37	
Appearance of Specimen	Dull grey finish on both sides	Copper color on both sides	Dulled grey finish on both sides. Crystallized fluid deposit on side opposite of mating surface.	Brown deposit on mating side, discolored on opposite side.	Brown deposit on mating side, white coating on opposite side.	Thin dark deposit on both sides.	No change on mating side, thin coating on opposite side.	Slightly discolored on both sides	No change on mating side, thin dark coating on opposite side.	No change on mating side, slightly discolored on opposite side.	
Appearance of Mating Surface	Thin green deposit on chrome side, dark deposit on unplated side.		Green deposit on chrome side, deposit of crystallized fluid on unplated side.	Thin green deposit on plated side. Thin dark coating on opposite side.	No change on chrome side, thin dark coating on opposite side	Slight discoloration on chrome side, thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	No change on mating side. Thin dark coating on opposite side.	Plated side discolored, thin dark coating on opposite side.	
Appearance of Fluid	Dark brown	Dark brown	Fluid crystallized	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown
Viscosity @ 100°F (48.18)	102.00	133.50	*	98.00	277.50	113.80	113.50	102.70	92.60	107.00	997.0
Viscosity @ 210°F (15.99)	33.77	45.58	*	32.40	102.16	35.27	39.29	29.03	29.33	33.61	249.70
Acid No. mg KOH/g (0.03)	4.23	19.0	-	8.66	1.16	20.2	8.88	8.1	7.31	11.8	2.04

* Fluid crystallized

TABLE 2-7

FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
PR-143AB
25 ML PER TEST SPECIMEN

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium-Tin Alloy	Cobalt-Molybdenum Alloy	Plasma Flame-plated Molybdenum	Control
Weight Change	-.0184	+.0026	+.0503	+.0090	+.0070	+.0067	+.0014	+.0039	+.0030	+.0004	
Hardness Change	H85 to H71	F89 to F89.5	H94 to H99	H90 to H92.5	H83 to H82	F87 to F91.5	RC19 to RC18.5	RC36 to RC37	RC5 to RC6.5	RC39 to RC36.5	
Appearance of Specimen	Dulled finish on both sides	Copper color on both sides.	Thin dark coating on mating side,	Thin dark coating on mating side.	Thin dark coating on mating side,	Dulled finish on both sides.	Both sides slightly discolored.	Mating side discolored, slight deposit on opposite side.	No change on both sides.	No change on mating side, thin dark deposit on opposite side.	
Appearance of Mating Surface	No change on plated side, unplated side discolored.	Both sides discolored.	Both sides discolored.	Plated side discolored, unplated side darkened.	Plated side slightly pitted, unplated side discolored.	No change on plated side, unplated side discolored.	Both sides discolored.	Both sides slightly discolored.	Both sides discolored.	Both sides discolored.	
Appearance of Fluid	No change	No change	No change	No change	No change	No change	No change	No change	No change	No change	
Viscosity @ 100°F C.S. (60.13)	68.70	76.50	68.40	67.70	67.45	68.50	69.60	68.25	68.60	67.80	68.3
Viscosity @ 210°F C.S. (7.94)	8.63	9.44	8.67	8.48	8.48	8.84	8.79	8.71	8.72	8.88	8.80
Acid No. mg KOH/g (.01)	.12	.02	.03	.06	.06	.03	.06	.09	.03	.06	.09

TABLE 2-8

FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
MCS-293
25 ML PER TEST SPECIMEN

Specimen	70% Ag. 25% Polyimide +5 Tungsten Diselenide	72% Ag. + 28% Copper	70% Ag. 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium-Tin Alloy	Cobalt-Molybdenum Alloy	Plasma Flame-plated Molybdenum	Control
Weight Change	+.0355	+.0140	+.0206	+.0220	+.0103	+.0311	+.0004	+.0010	+.0010	+.0034	
Hardness Change	H85 to H36	F89 to F80	H94 to H98	H90 to H96.5	H83 to H82	F87 to F94	RC19 to RC11	RC36 to RC35.5	RC5 to RC5	RC39 to RC36	
Appearance of Specimen	Dulled finish on both sides.	Mating side discolored, dark deposit on opposite side.	Slightly discolored on both sides.	Discolored on both sides.	Brown deposit on mating side, slight darkening on opposite side,	Grey finish on both sides.	Thin dark coating on mating side, opposite side slightly discolored	Mating side discolored, no change on opposite side.	Thin dark coating on mating side, opposite side no change.	Thin dark coating on mating side, opposite side no change.	
Appearance of Mating Surface	No change on plated side, Unplated side darkened.	Plated side discolored, unplated side darkened.	Flated side discolored, unplated side darkened.	Plated side slightly discolored, unplated side darkened.	Plated side slightly discolored, unplated side darkened.	Plated side discolored, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	This dark deposit on plated side, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	Thin dark deposit on plated side, unplated side darkened.	
Appearance of Fluid	Light brown										
Viscosity @ 100°F CS. (25.30)	28.50	27.30	31.10	26.90	27.50	27.55	29.70	29.50	29.40	29.70	28.7
Viscosity @ 210°F CS. (4.18)	4.89	4.45	4.87	4.35	4.30	4.46	4.58	4.71	4.54	4.56	4.44
Acid No. mg KOH/g (0.01)	0.0	.07	.11	.02	0.0	0.0	.02	.07	.11	0.0	.02

TABLE 2-9

FLUID - MATERIAL COMPATIBILITY TEST NO. 2 - 150 HOURS AT 600°F
MLO-60-294
25 ML PER TEST SPECIMEN

Specimen	70% Ag. 25% Polyimide +5% Tungsten Diselenide	72% Ag. +28% Copper	70% Ag. + 30% Tungsten Diselenide	Silver - S.S. Composite (430 S.S.)	Silver-Nickel Composite	Nickel Foametal Impregnated w/Ca F2 + Ba F2	Vascojet 1000	Titanium-Tin Alloy	Cobalt-Molybdenum Alloy	Plasma Flame-Plated Molybdenum	Control
Weight Change	-.0737	+.0003	-.0606	+.0241	+.0066	+.0012	+.0004	+.0004	-.0107	+.0015	
Hardness Change	H85 to H47	F89 to F91.5	H94 to 90.5	H90 to H84	H83 to H82	F87 to F92	RC19 to RC125	RC36 to RC32	RC5 to RC2	RC39 to RC33.5	
Appearance of Specimen	Slightly discolored on both sides.	Copper color on mating side, opposite side tarnished.	Slightly discolored on both sides.	Thin brown coating on both sides.	Thin brown coating on mating side. Tacky film on opposite side.	Dull grey on both sides.	Slight discoloration on mating side, opposite side darkened.	Slightly discolored on both sides.	No change	No change	
Appearance of Mating Surface	No change on plated side, unplated side discolored.	No change on plated side, unplated side discolored.	No change on plated side, unplated side discolored.	Brown coating on plated side, unplated side darkened.	Brown coating on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	No change on plated side, unplated side darkened.	
Appearance of Fluid	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	Amber	
Viscosity @ 100°F CS.-(15.02)	12.75	12.90	12.55	12.40	13.00	12.80	12.90	12.75	13.10	12.75	13.1
Viscosity @ 210°F CS. (3.26)	3.04	3.02	3.08	3.04	3.06	3.05	3.07	3.06	3.09	3.00	3.05
Acid No. mg KOH/g (0.02)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

the bottle (Figure 2-10) approximately one month after testing. This condition was believed to be caused by the evaporation of the light ends of the fluid when exposed to air during the fluid analysis work.

As experienced previously, the MCS-3101 fluid exhibited considerable degradation. Fluid breakdown was evidenced after 50 hours of testing when the fluid pressure in the tubes started to build up from the initial 15 psi precharge. At the completion of the test, pressure in the tubes containing the control sample, Polymet, silver-stainless steel composite, and Polymer SP, was 85, 85, 110, and 150 psi, respectively. The control sample was extremely viscous and black. The fluid containing the material specimens was paste-like in appearance.

3. Fluid-Material Compatibility Testing at 400°F

This test was conducted with fluids that did not perform satisfactorily at 600°F. The fluids retested were F-50 silicone and MCS-3101. The MCS-3101 fluid was tested with all ten candidate seal materials. The F-50 silicone fluid exhibited a high acidic condition when in contact with certain candidate seal materials at 600°F and was tested with only those materials in question.

Both fluids completed the 150-hour test. However, results obtained with the MSC-3101 fluid were questionable because of the removal of the bulk of the fluid from the test capsules by vaporization during the degassing process. This was not discovered until the test was completed. Normally, fluid degassing is accomplished at approximately 250°F with a vacuum of 30 inches of Hg. However, during the last three hours of the 72-hour degassing period, the oven inadvertently heated to about 375°F to 400°F. At this temperature, the MCS-3101 vaporized and was drawn out of the tubes by the vacuum pump. However, degassing at this temperature did not appear to affect the F-50 silicone fluid, as the fluid was intact in the tubes after testing.

Results of the F-50 silicone fluid run are summarized in Table 2-10. As shown in Figure 2-13, the fluid was slightly discolored. The seal material specimens (Figure 2-14) were virtually unaffected at 400°F. The high acidic condition exhibited at 600°F by the silicone fluid when in contact with the nickel

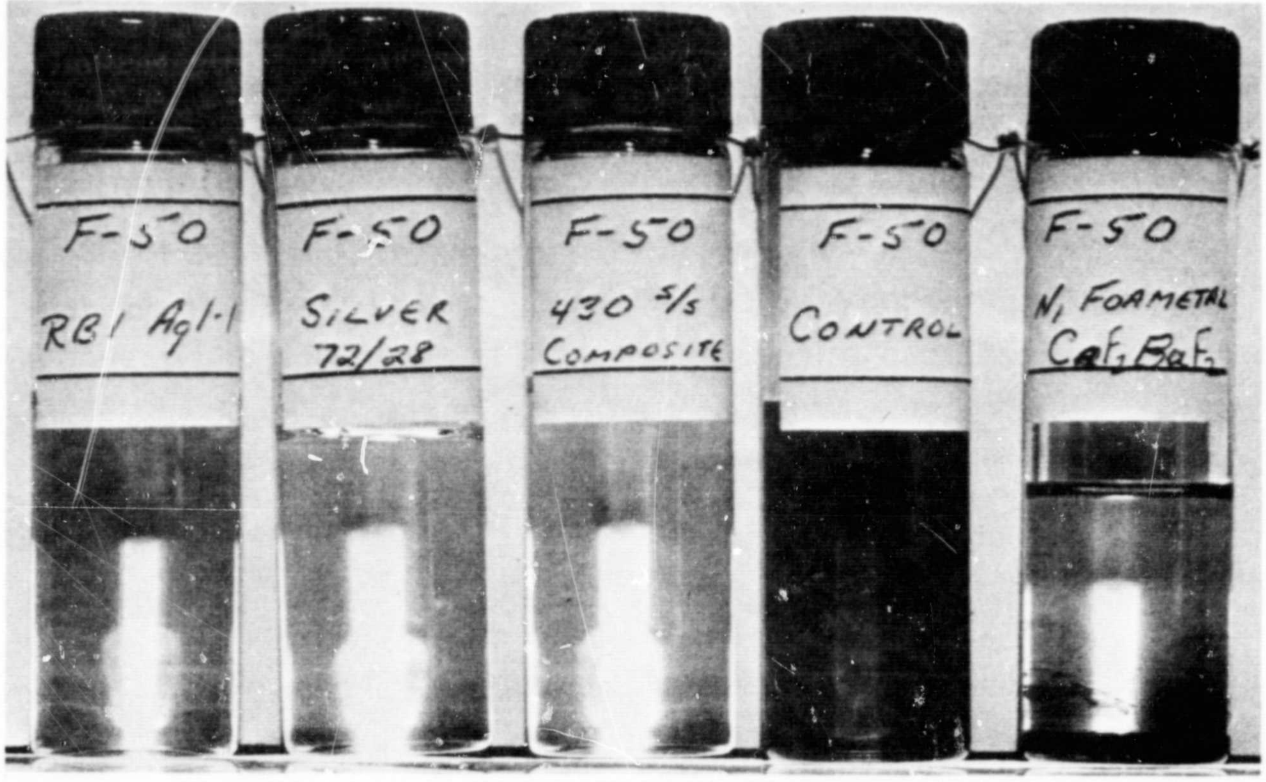


Figure 2-13. Test No. 3 - F-50 Silicone After Test at 400°F

Nickel Foametal Impregnated w/CaF_2 + BaF_2	70% Ag 30% Tungsten diselenide	Silver Alloy 72% Ag + 28% Cu	Silver - S.S. Composite

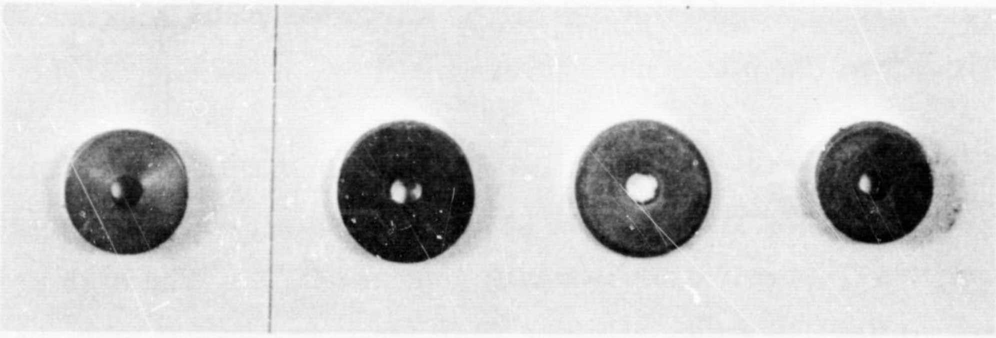


Figure 2-14. Test No. 3. - Material Specimens After Test at 400°F

TABLE 2-10

FLUID - MATERIAL COMPATIBILITY TEST NO. 3 - 150 HOURS AT 400°F
F-50 CHLOROPHENYL METHYL SILICONE FLUID
25 ML PER TEST SPECIMEN

Specimen	Nickel Foametal w/CaF_2 & BaF_2	Silver Alloy 72% Ag 28% Cu	70% Silver +30% Tungsten Diselenide	Stainless-Steel Silver Composite	Control
Weight Change (grams)	-.0019	-.0001	+.0137	+.0005	
Hardness Change (Rockwell)	From F-87 to F-91	From H-89 to H-89	From H-94 to H-99	From H-90 to H-96	
Appearance of Specimen	No change	No change	Slight discoloration	Contact side no change; opposite side corroded	
Appearance of Mating Surface	No change	No change	Slightly tarnished	No change	
Appearance of Fluid	Cloudy	Clear	Very light amber	Clear	Light amber
Viscosity @ 100°F CS (48.18)	30.59	46.83	41.78	45.18	34.98
Viscosity @ 210°F CS (15.99)	12.76	15.48	13.56	15.18	12.76
Acid No. mg KOH/g (0.03)	0.05	0.05	0	0.05	0

Foametal (impregnated with $CaF_2 + BaF_2$ eutectic), silver alloy, silver tungsten diselenide composite, and the silver-stainless steel (Type 430) composite was not present at 400°F. Variations in viscosity were also minor.

Due to the loss of the MCS-3101 fluid, there was not sufficient fluid remaining in the tubes to perform any fluid analysis. However, the material specimens appeared to be in good condition.

A repeat of the 400°F run with the MCS-3101 fluid is summarized in Table 2-11. The fluid (Figure 2-15) exhibited a slight discoloration (light amber) after the 150-hour run. Variations in viscosity at 100°F and 210°F were minor when compared to viscosity values obtained from the fresh fluid. Increase in acidity of the fluid was also negligible. In general, the material specimens (Figure 2-16) did not show any effects from the fluid. The exceptions were the silver-stainless steel composite (Type 430) and the Vascojet 1000, both of which exhibited slight corrosion. The chrome-plated mating buttons also exhibited no visual effects from the fluid.

4. Fluid-Material Compatibility Testing at 500°F

This test was conducted with MCS-3101 and F-50 silicone fluids. The MCS-3101 fluid was evaluated with the ten candidate seal materials. The F-50 silicone fluid was evaluated with only those seal materials that indicated some incompatibility at 600°F. These were: nickel Foametal, silver-copper alloy, silver-tungsten diselenide composite, and the silver-stainless steel composite.

Results of the run made with F-50 silicone fluid are summarized in Table 2-12. With the exception of the silver-stainless steel composite, which exhibited slight corrosion, the F-50 fluid produced minor effects (Figure 2-17) on the material specimens at 500°F. The chrome-plated (440C) test buttons were in good condition except for slight corrosion on the button which mated with the silver-stainless steel composite. The high acidic condition exhibited by the F-50 fluid at 600°F was not present at 500°F. Variations in fluid viscosity were also minor. General condition of the fluid samples is shown in Figure 2-18.

TABLE 2-11

FLUID - MATERIAL COMPATIBILITY TEST NO. 5 - 150 HOURS AT 400°F
MCS-3101 HALOGENATED POLYARYL FLUID
25 ML PER TEST SPECIMEN

Specimen	Polyimide (Polymer-SP)	Titanium-Tin Alloy	Cobalt Molybdenum Alloy	Silver Alloy 72% Ag + 28% Cu.	Nickel Foametal w/Ca F_2 + Ba F_2	Metco Flame-Plated Molybdenum	Polymet	Silver-Stainless Composite (430 S.S.)	Vascojet 1000	Silver-tungsten Diselenide Composite	Control
Weight Change (grams	+ .0108	- .0023	- .0008	+ .0079	- .0001	- .0001	+ .0044	+ .0040	- .0019	+ .0206	
Hardness Change	-3 H-90 to H-87	-4 R_c-36 to R_c-32	+2 R_c-5 to R_c-7	No change F-89	No change F-87	-2 R_c-39 to R_c-37	-2 H-48 to H-46	+6 H-90 to H-96	No change R_c 19	+3 H-94 to H-97	
Appearance of Specimen	No change	No change	No change	No change	No change	No change	No change	Slight corrosion	Slight corrosion	No change	
Appearance of Mating Surface	No change	No change	No change	No change	No change	No change	No change	Slight corrosion	No change	No change	
Appearance of Fluid	Light amber	Light amber	Light amber	Light amber	Dark amber	Dark amber	Light amber	Dark amber	Light amber	Light amber	Light amber
Viscosity @100°F, CS (4.34)	4.61	4.51	4.60	4.41	4.51	4.44	4.48	4.36	4.40	4.47	4.35
Viscosity @210°F, CS (1.32)	1.44	1.44	1.46	1.50	1.41	1.43	1.44	1.46	1.41	1.41	1.43
Acid No. (.01) mg KOH/g	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03	.03

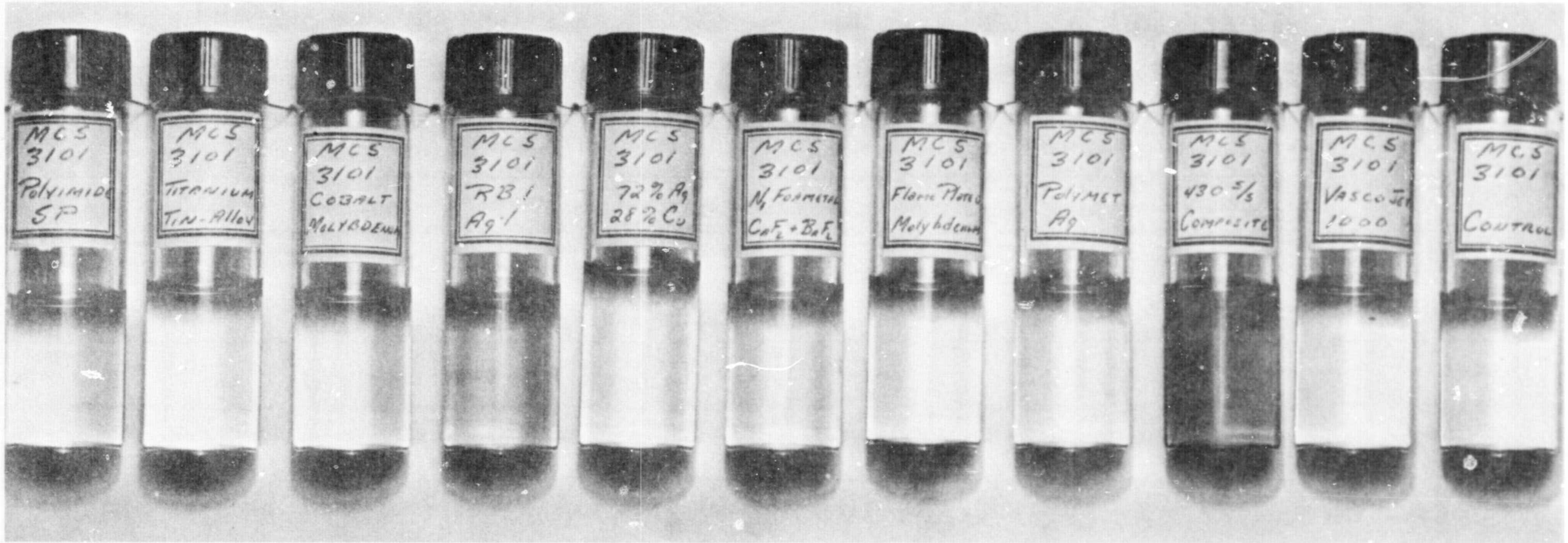


Figure 2-15. MCS-3101 Fluid After Test at 400°F

Polymer SP (Polyimide)

Titanium-Tin Alloy (10% tin)

Cobalt-Molybdenum Alloy

70% Silver-30% Tungsten Diselenide

Silver Alloy (72% Ag + 28% Cu)

Nickel Foametal with $CaF_2 + BaF_2$

Flame-Plate Molybdenum Coating

Polymet (silver-polymer composite)

Silver-Stainless Steel Composite

Vascojet 1000

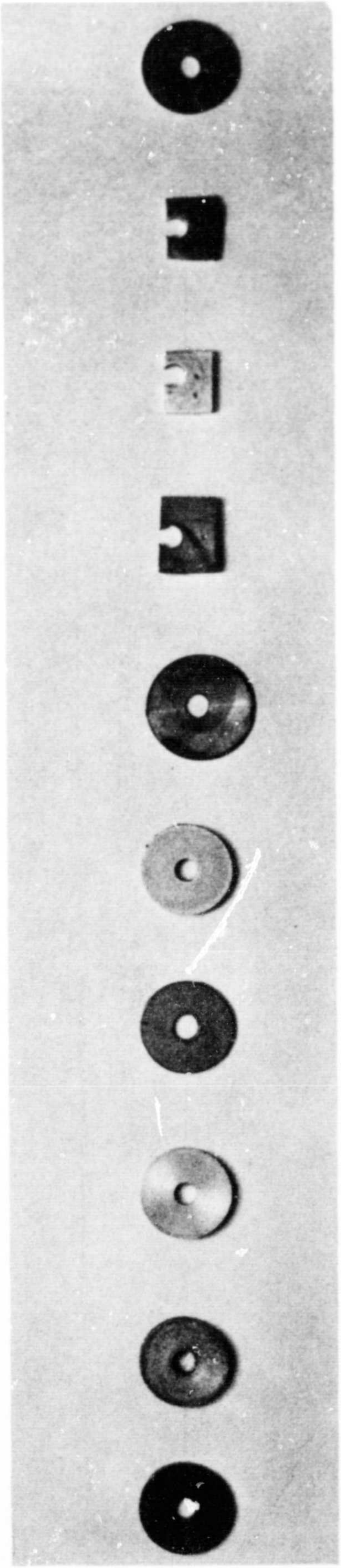


Figure 2-16. Material Specimens After Test at 400°F

TABLE 2-12

FLUID - MATERIAL COMPATIBILITY TEST NO. 4 - 150 HOURS AT 500°F
F-50 CHLOROPHENYL METHYL SILICONE FLUID
25 ML PER TEST SPECIMENT

Specimen	Nickel Foametal w/CaF_2+BaF_2	Silver Alloy 72% Ag+28% Cu.	70% Silver +30% Tungsten Diselenide	Stainless-Steel Silver Composite	Control
Weight Change (grams)	-.004	-.001	+.019	-.016	
Hardness Change (Rockwell)	H-87 to H-94	F-89 to F-76	H-94 to H-80	H-90 to H-82	
Appearance of Specimen	No change	Slightly tarnished	Slightly discolored	Slight corrosion	
Appearance of Mating Surface	No change	No change	No change	Slight corrosion	
Appearance of Fluid	No change	No change	Light amber	Light amber	No change
Viscosity @100°F CS (48.18)	45.0	47.1	56.9	50.6	45.5
Viscosity @210°F CS (15.99)	14.9	15.45	17.75	16.45	15.1
Acid No. mg KOH/g (0.03)	.001	.21	.22	.20	.001

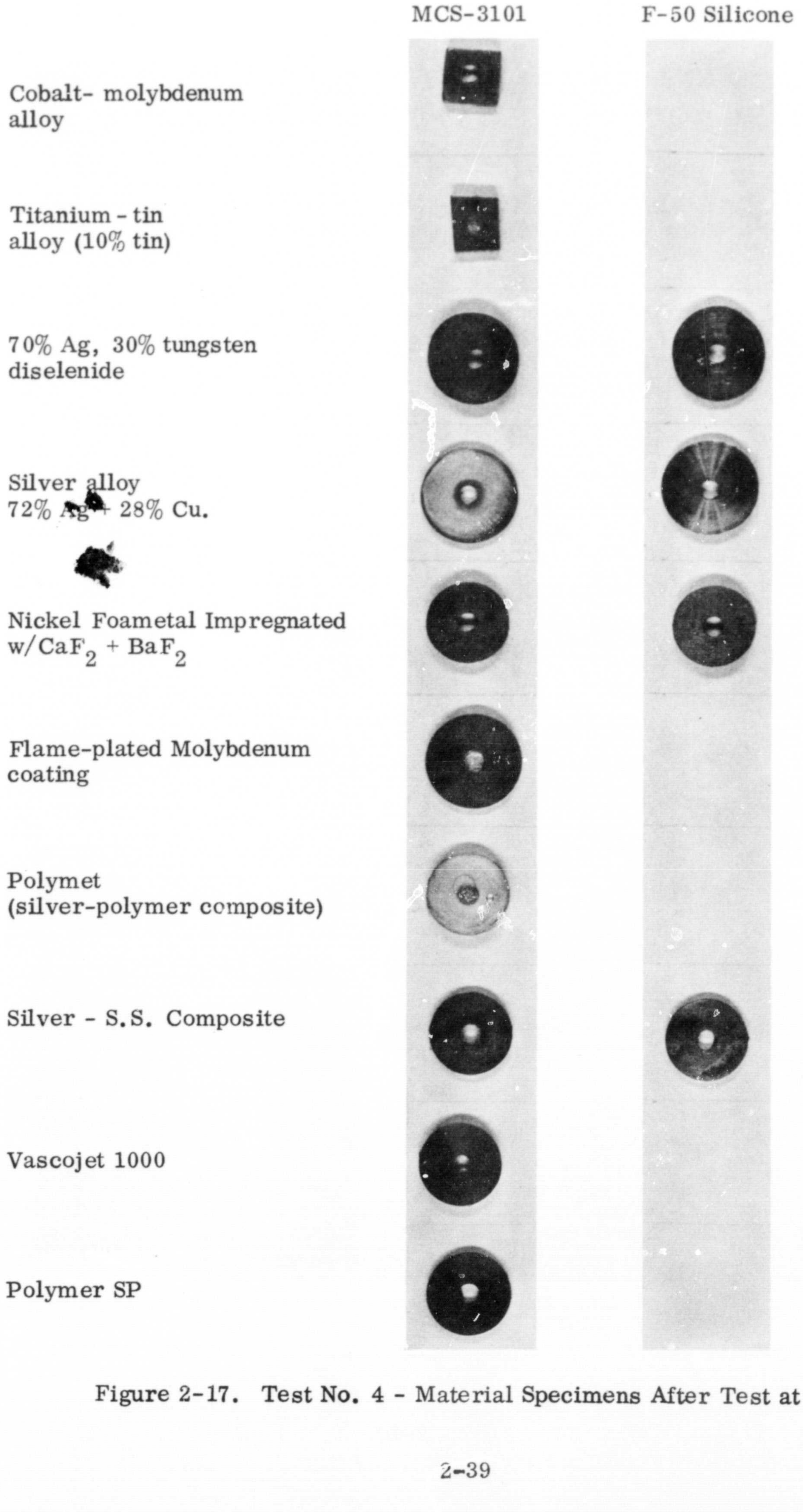


Figure 2-17. Test No. 4 - Material Specimens After Test at 500°F

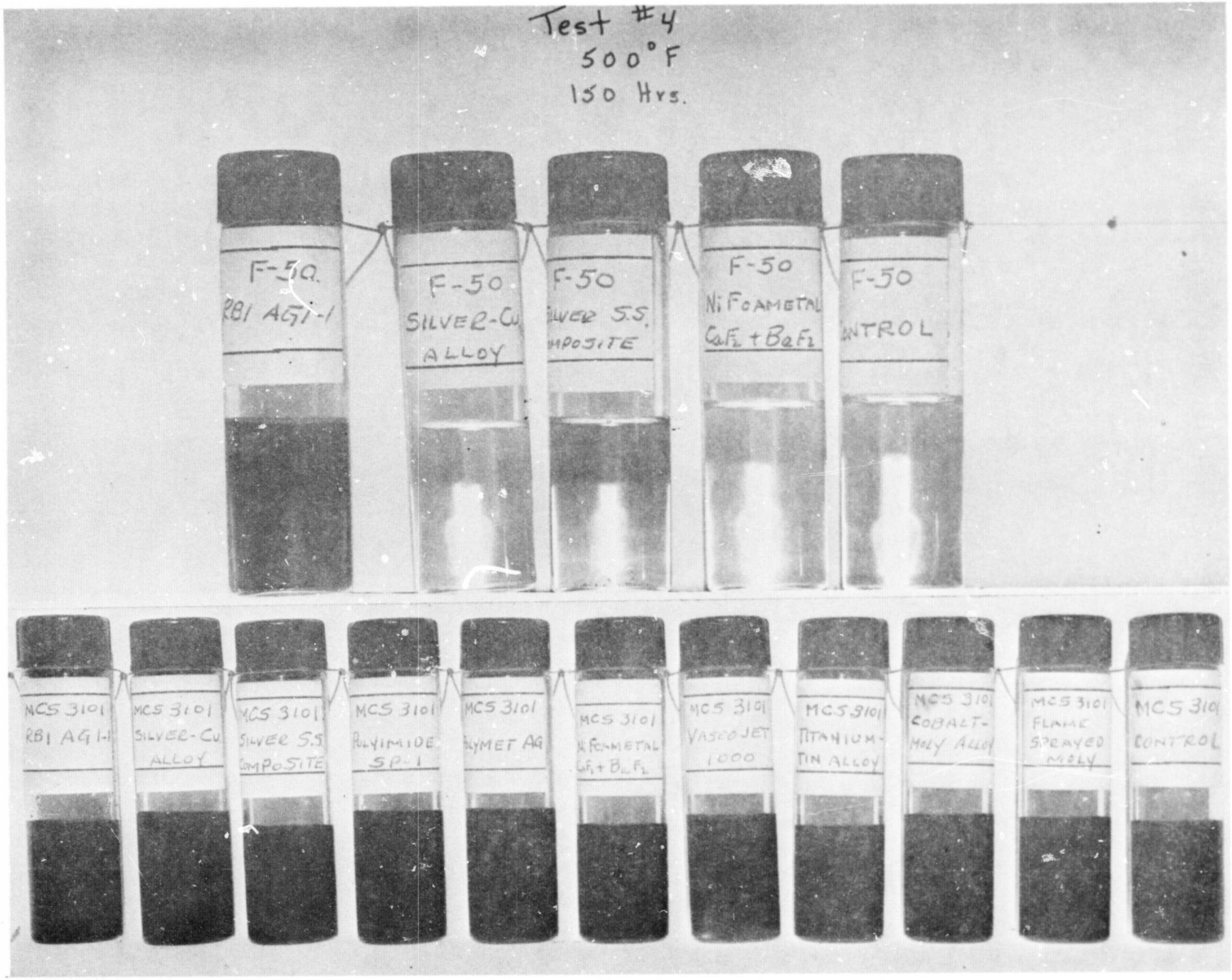


Figure 2-18. Test No. 4 - F-50 Silicone and MCS-3101 After Test at 500°F

TABLE 2-13

FLUID - MATERIAL COMPATIBILITY TEST NO. 4 - 150 HOURS AT 500°F
MCS-3101 FLUID
25 ML PER TEST SPECIMEN

Specimen	75% Cobalt 25% Molybdenum	Titanium-Tin Alloy	Silver-Tungsten Diselenide Composite	Silver Alloy 72%Ag+28%Cu	Nickel Foametal W/CaF_2+BaF_2	Metco-flame plated molybdenum	Silver-stainless steel composite	Polymer SP	Polymet	Vascojet 1000	Control
Weight change (grams)	+.0016	+.0022	-.1717	+.0607	-.003	+.0038	+.0067	+.0192	+.0138	+0008	
Hardness Change (Rockwell)	RC 5-No change	RC 36 to RC 39	H 94 to H 65	F 89 to F 69	H 87 to H 95	RC 39 to RC 36	H 90 to H 78	H 90 to H 71	H 48 to H 18	RC 19 - No change	
Appearance of Specimen	No change	No change	Heavy dark deposit on both sides	Dull finish on mating side. Dark coating on opposite side	No change	No change	Corrosion on both sides	No change	No change	Slight tarnish	
Appearance of Mating Surface	Greenish coating	Thin green coating on both sides	Plated side tarnished-dark deposit opposite side	Red coating on mating side. Dark coating on opposite side	Thin green coating on plated side. Opposite side tarnished	Slight etching on plated side. No change on opposite side	Thin dark coating on both sides	Greenish coating	Plated side no change. Dark coating on opposite side.	Slight etching on plated side	
Appearance of Fluid	Very dark	Very dark	Very dark	Very dark	Very dark	Very dark	Very dark	Dark Amber	Very dark	Dark amber	Dark Amber
Viscosity @ 100°F CS (4.34)	4.31	4.75	5.35	4.93	4.68	4.71	4.98	4.90	4.98	4.70	4.91
Viscosity @ 210°F CS (1.32)	1.45	1.49	1.45	1.50	1.50	1.48	1.47	1.52	1.49	1.43	1.45
Acid No. mg/KOH/g(0.01)	.02	.03	.15	.04	.02	.03	.01	.03	.02	.01	.02

Data on the run made with the MCS-3101 fluid are summarized in Table 2-13. Discoloration of the fluid samples (Figure 2-18) was quite noticeable; it ranged from dark amber to a very dark color. Variations in viscosity at 100°F and 210°F were minor when compared to the untested fluid. With the exception of the fluid sample that was in contact with the silver-tungsten diselenide composite, the fluids exhibited only nominal changes in acidity. Figure 2-17 depicts the condition of the material specimens. A heavy dark deposit of fluid was noticed on both sides of the silver-tungsten diselenide composite. The silver copper alloy exhibited a dull finish on the mating side and a dark coating on the opposite side. Evidence of corrosion was noticed on the silver-stainless steel composite. The remaining material specimens exhibited practically no change. The mating buttons all exhibited some coating or deposit on the unplated side.

D. ADDITIONAL FLUID-MATERIAL COMPATIBILITY TESTING

As provided for by supplemental agreement to Contract NAS 3-7264, additional fluid-material compatibility testing was conducted with Dow Corning XF-1-0291 and XF-1-0294 silicone base fluids. These recently developed fluids were tested with materials having potential for hydraulic pump application or as seals. The candidate materials were as follows:

1) M-10 tool steel, hardened
2) K-82 carbide
3) Pheldor 10, iron-silicon-bronze
4) Ductile iron, D-2 hardened
5) Nitralloy G135 (modified)
6) Polyimide, unfilled
7) 72% Silver - 28% copper alloy
8) Nickel Foametal, 60% dense, impregnated with 38% CaF_2 - 62% BaF_2 eutectic
9) Vascojet 1000 (H-11 tool steel)
10) 75% Cobalt - 25% molybdenum alloy
11) Stellite Star J
12) S Monel

The test procedure was essentially the same as that used in previous compatibility tests. The exceptions were that the fluids be degassed at room temperature and that the fluid be vibrated during degassing.

The fluid compatibility test apparatus was modified to provide a means for vibrating the fluid during degassing. As shown in Figure 2-19, the fluid was vibrated by a linkage system attached to an eccentric plate, which was in turn driven by a variable speed motor. This setup produced a vibration of 100 to 120 cps at an amplitude of ± 1/8 inch.

Results obtained from the compatibility testing of the XF-1-0291 and XF-1-0294 silicone base fluids are summarized in Table 2-14 and 2-15. Degradation of the XF-1-0291 fluid (Figure 2-20 and 2-21) was quite evident. The fluid samples from the capsule containing the silver alloy and the capsule with the Pheldor 10 material were solid. The control fluid specimen and the fluid removed from the capsule containing the Nickel Foametal was very viscous. The remainder of the fluid specimens had a rubber-like consistency. Because of the degraded condition of the fluid, acid numbers and viscosity data were not obtainable. The fluid appeared to be quite corrosive when in contact with the unplated side of the 440C stainless steel mating discs. The condition of the material specimens are shown in Figures 2-22 and 2-23. Evidence of corrosion was also exhibited by the Nitralloy G, Ductile iron, and M-10 tool steel. The fluid produced slight discoloration of the silver alloy, cobalt moly alloy, and the nickel Foametal specimens. Compatibility of the fluid with S Monel, Polymer SP, and Vascojet 1000 appear to be good.

The XF-1-0294 fluid (Figures 2-24 and 2-25) appears to be in good condition except for the discoloration (dark brown) that took place. Viscosity values of the fluid samples at 100°F and 210°F were generally lower than the values obtained on the fresh fluid. Changes in acid numbers with respect to the original values were not detectable on any of the fluid specimens. As shown in Figures 2-26 and 2-27, compatibility of the fluid with the nickel Foametal, Vascojet 1000, and Polymer SP was good. Slight discoloration was exhibited by the silver alloy, cobalt moly alloy, and Stellite Star-J materials. Dark discoloration was exhibited by the Nitralloy-G, K-82, and Ductile iron specimens. The fluid showed no effect on the chrome plated side of the 440C stainless steel mating discs. However, slight discoloration was exhibited by the unplated side of the discs.

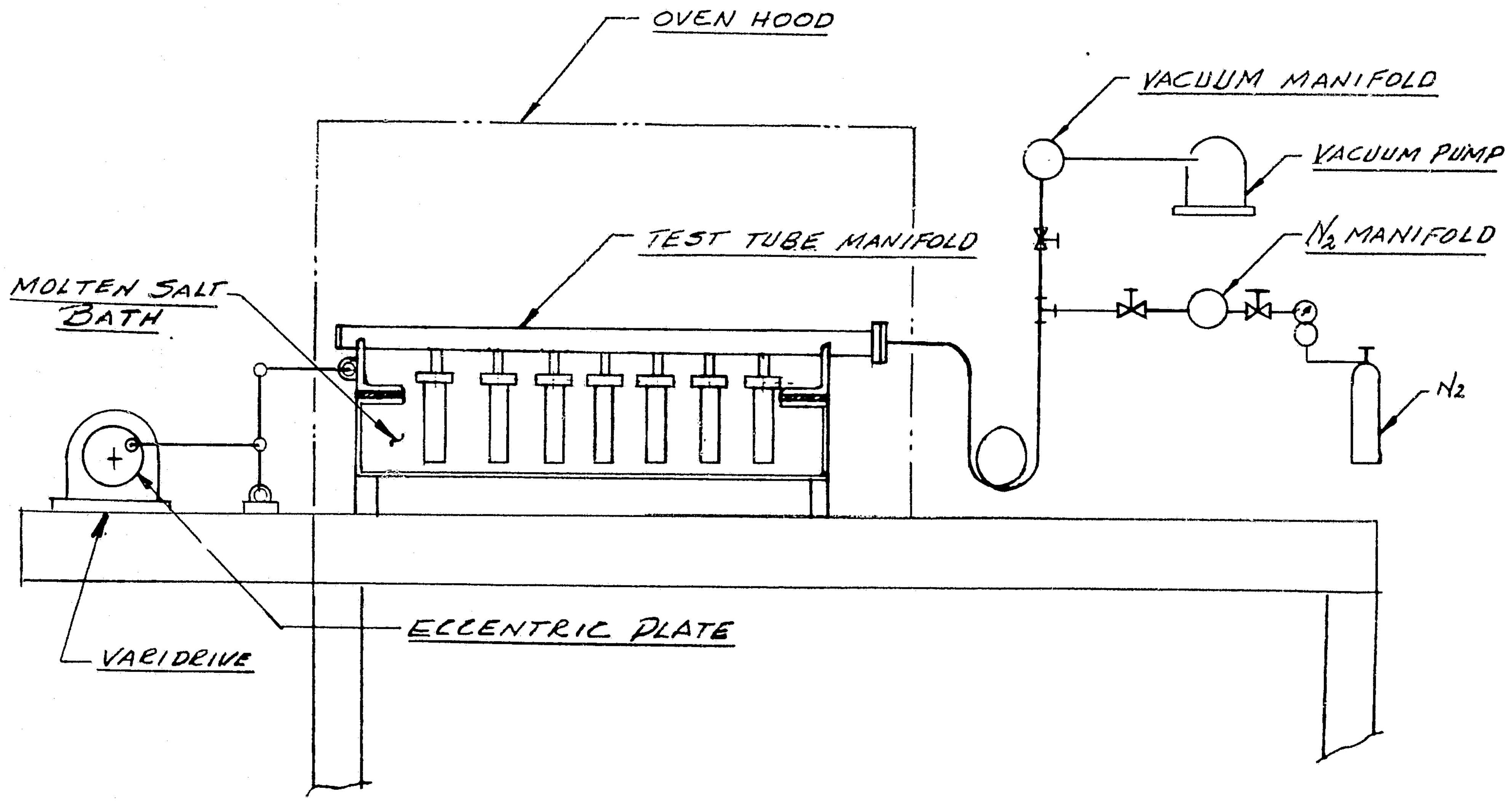
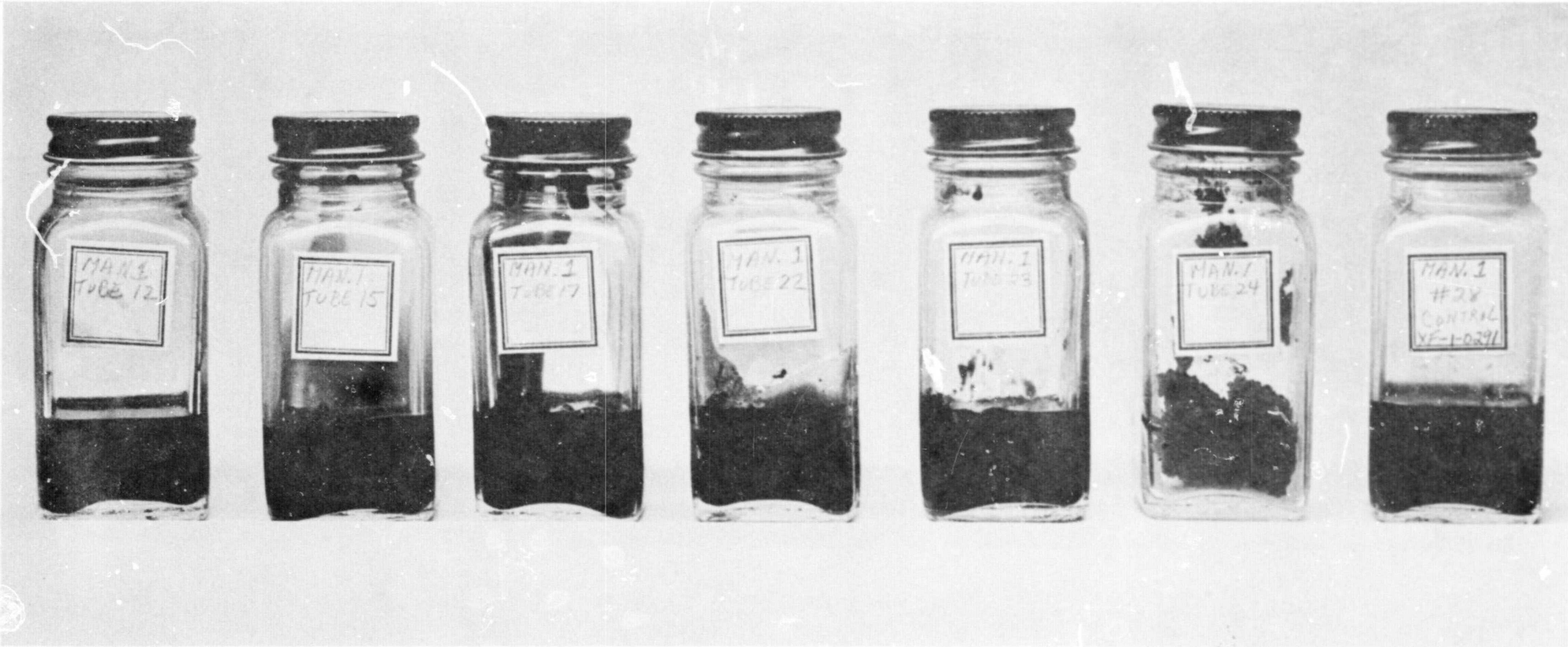


Figure 2-19. Fluid - Material Compatibility Test Setup

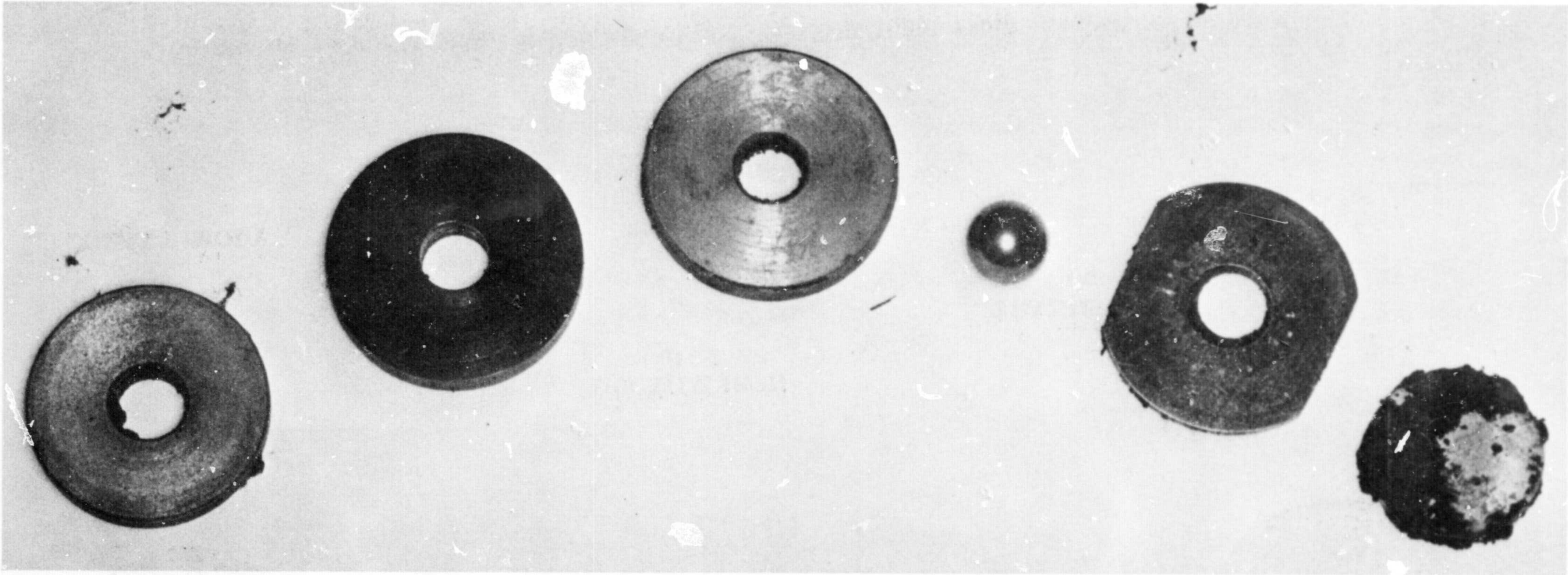


NICKEL FOAMETAL (CuF_2 + BaF_2)	POLYIMIDE	VASCOJET 1000	STAR - J	S - MONEL	PHELDOR - 10	CONTROL

Figure 2-20. XF-10291 Fluid After 600°F Test

CONTROL	COBALT MOLY	M - 10 TOOL STEEL	DUCTILE IRON - D-2	K - 82	NITRALLY - G	SILVER ALLOY 72% Ag + 28% Cu.

Figure 2-21. XF-10291 Fluid After 600°F Test



NICKEL FOAMETAL (CaF_2 + BaF_2)

POLYIMIDE

VASCOJET 1000

STAR-J

S-MONEL

PHELDOR 10

Figure 2-22. Material Specimens After Test in XF-10291 Fluid at 600°F

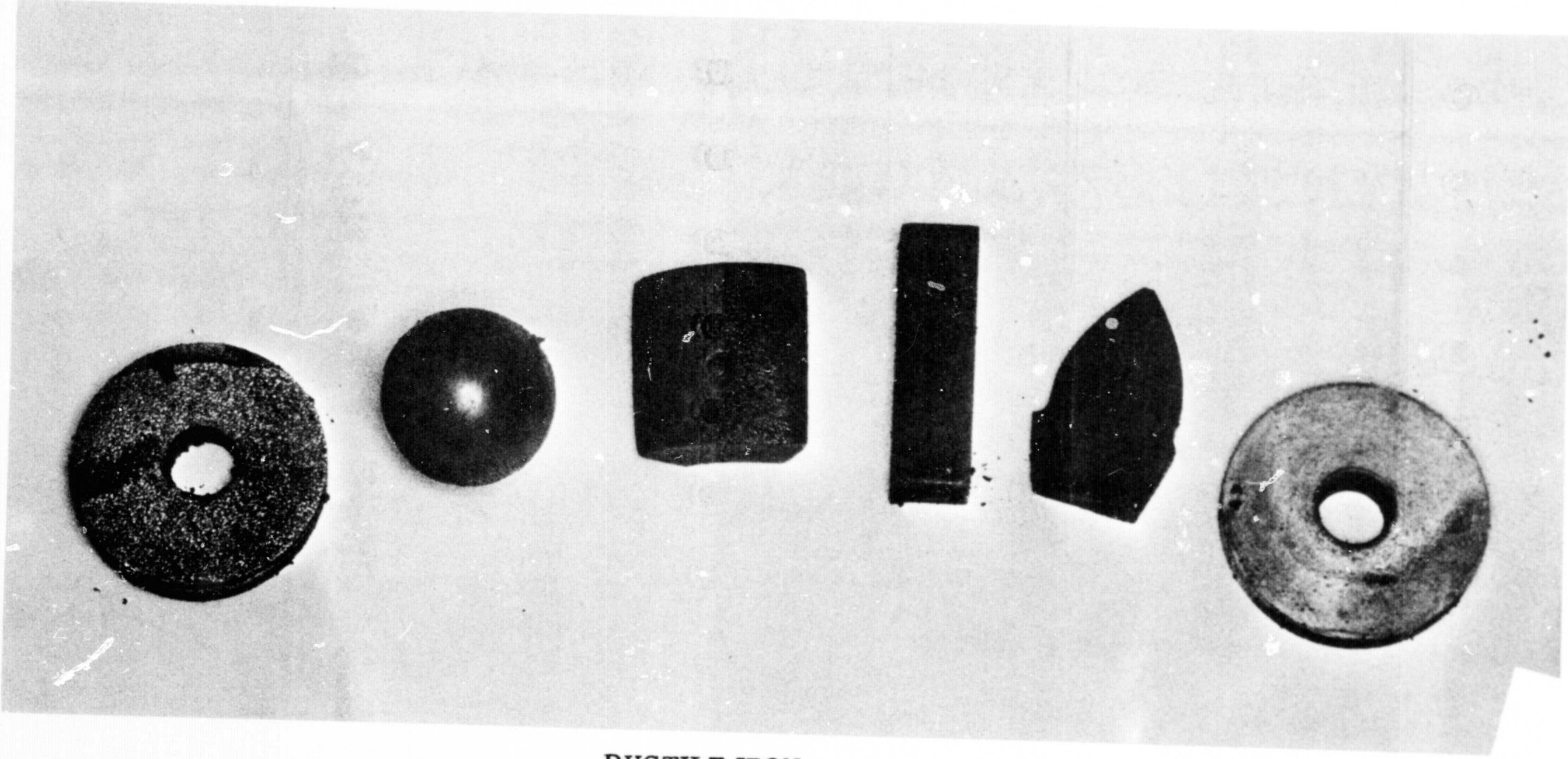
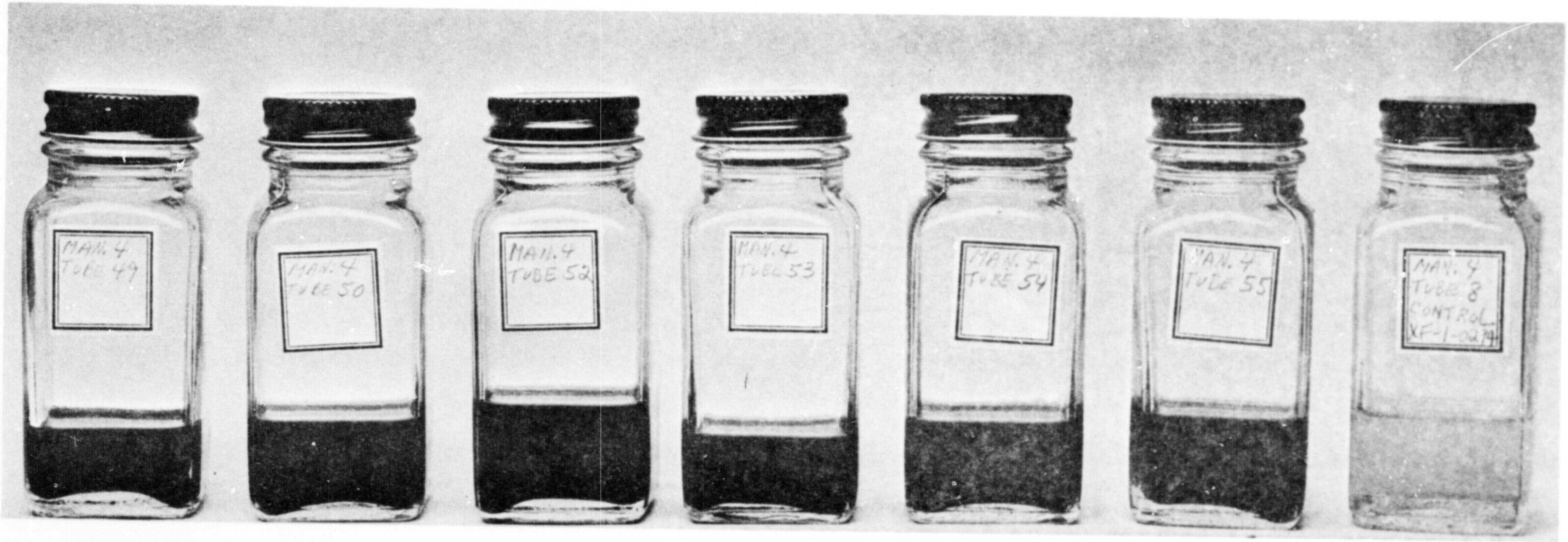


Figure 2-23. Material Spec iens After Test in XF-10291 Fluid at 600°F

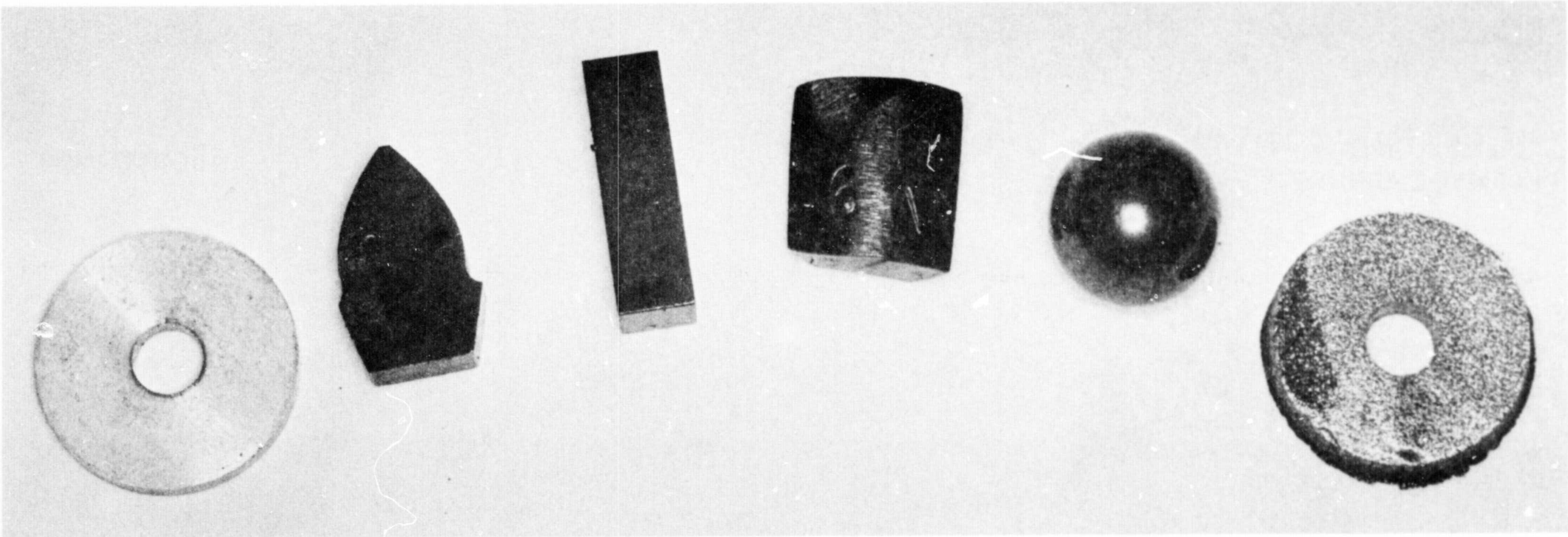
SILVER ALLOY 72% Ag + 28% Cu. | NITRALLOY - G | K - 82 | DUCTILE IRON D - 2 | M - 10 TOOL STEEL | COBALT MOLY | CONTROL

Figure 2-24. XF-10294 Fluid After 600° F Test



PHELDOR 10 | S-MONEL | STAR-J | VASCOJET 1000 | POLYIMIDE | NICKEL FOAMETAL (CaF_2 + BaF_2) | CONTROL

Figure 2-25. XF-10294 Fluid After 600°F Test



DUCTILE IRON

K - 82

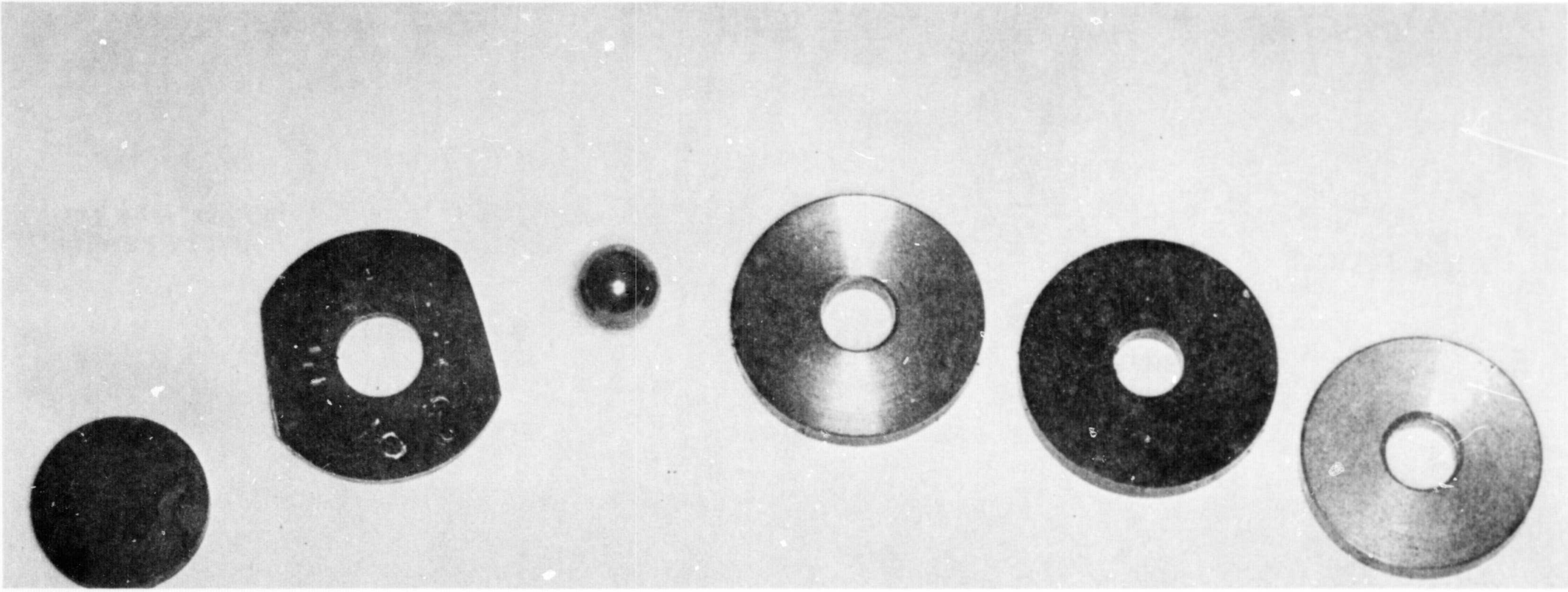
M - 10
TOOL STEEL

NITRALLOY - G

SILVER ALLOY
72% Ag + 28% Cu.

COBALT MOLY

Figure 2-26. Material Specimens After Test in XF-10294 Fluid at 600°F



STAR - J

S - MONEL

VASCOJET
1000

POLYIMIDE

PHELDOR 10

NICKEL FOAMETAL
(CaF_2 + BaF_2)

Figure 2-27. Material Specimens After Test in XF-10294 Fluid at 600°F

TABLE 2-14

FLUID - MATERIAL COMPATIBILITY TEST - 150 HOURS AT 600°F
XF-1-0291 FLUID
(Trifluoropropyl halophenyl substituted silicone copolymer)

Specimen	Silver Alloy 72% Ag + 28% Cu.	Nitralloy-G	K-82	Ductile Iron	M-10 Tool Steel	Cobalt Alloy (75% Co, 25% Mo)	Pheldor	S Monel	Stellite Star J	Vascojet 1000	Polymer SP (Polyimide)	Nickel Foametal (w/CaF_2 + BaF_2)	Control
Weight change (grams)	+.0001	-.0028	-.0002	-.0035	-.0049	+.0017	-.0058	+.0001	+.0004	-.0036	-.0072	+.0112	
Hardness change (Rockwell)	F-89 - no change	RC 35 to RC 33	RC 75 - no change	RC 25 to RC 23	RC 64 - no change	RC 30 to RC 35	RC 20 to RC 19	RC 35 to RC 33	RC 52 - no change	RC 28 to RC 22	H-90 - no change	F-87 to F-89	
Appearance of specimen	Discolored	Slight corrosion on surface	Slight discoloration	Rust coating on surface	Rust coating on surface	Slight discoloration	Heavy deposit of crystalized fluid	No change	No change	No change	No change	Slight discoloration	
Appearance of mating surface	Slight discoloration on chrome surface, corrosion on uplated side					Green deposit on chrome surface, corrosion on unplated surface		Chrome surface etched, corrosion on unplated surface		Slight pitting of chrome surface, corrosion on unplated side	Green deposit on chrome surface, dark coating on unplated side	Green deposit on chrome surface, dark coating on unplated side	
Appearance of fluid	Solid	Rubber-like	Rubber-like	Very viscous	Rubber-like	Rubber-like	Solid	Rubber-like	Rubber-like	Rubber-like	Rubber-like	Very viscous	Very viscous
Viscosity @ 100°F CS													
Viscosity @ 210°F CS													
Acid No. mg KOH/g													

TABLE 2-15

FLUID - MATERIAL COMPATIBILITY TEST - 150 HOURS AT 600°F
XF-1-0294 FLUID
(Trifluoropropyl methyl polysiloxane)

Specimen	Silver Alloy (72% Ag+28% Cu.)	Nitralloy-G	K-82	Ductile Iron	M-10 Tool Steel	Cobalt Alloy (75% Co. 25% Mo)	Pheldor	S Monel	Stellite Star J	Vascojet 1000	Polymer SP (Polyimide)	Nickel Foametal (w/CaF_2+BaF_2)	Control
Weight change (grams)	+.0012	+.0004	+.0009	+.0011	+.0011	+.0002	+.0057	+.0019	+.0006	+.0007	-.0093	+.0028	
Hardness change (Rockwell)	F-89 - no change	RC 35 to RC 33	RC 75 - no change	RC 25 to RC 23	RC 64 - no change	RC 30 to RC 39	RC 20 to RC 17	RC 35 to RC 33	RC 52 - no change	RC 28 to no change	H-90 - no change	F-87 to F-89	
Appearance of specimen	Slight discoloration	Darken	Darken	Darken		Slight discoloration	Slight discoloration	No change	Slight discoloration	No change	No change	No change	
Appearance of mating surface	No change in plated surface, unplated surface slightly discolored	-	-	-	-	No change on plated surface, unplated surface slightly discolored	-	No change	-	No change on plated surface, unplated surface slightly discolored	No change on plated surface, unplated surface darkened	No change	
Appearance of fluid	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Dark brown	Slight discoloration
Viscosity @ 100°F CS (62.0)	30.4	35.6	37.0	37.7	33.4	32.0	33.6	34.3	40.0	30.9	30.9	33.1	38.4
Viscosity @ 210°F CS (12.5)	6.7	7.84	7.93	8.32	7.41	7.12	6.72	7.74	8.87	6.68	6.47	6.79	8.92
Acid No. mg KOH/g (0.1)	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1

E. SLIDING WEAR TEST

Sliding wear tests were conducted on the majority of the candidate seal materials to determine their apparent wear characteristics when subjected to reciprocating motion against a hard chrome-plated piston rod. Initially, it was intended to conduct tests only on the hard base materials because some doubt existed as to their compatibility in sliding contact with hard chrome plate. However, these tests were extended to include the soft base materials as well.

1. Test Equipment and Procedures

The reciprocating wear tester, shown in Figures 2-28 and 2-29, was used to simulate a reciprocating rod seal arrangement. It consists of a chrome-plated Type 440C stainless piston rod (Hardness Rc 53) and a stationary test specimen. The piston rod is supported on both ends by linear roller bearings. A variable speed motor is used to vary the reciprocating speed of the piston rod. The cross-section of the test specimen consists of a segment of a circle with a radius conforming to the radius of the piston rod. The specimen is mounted in a keyed slider that is attached to the loading arm. A guide block with a thumb screw adjustment permits vertical motion of the arm, but prevents any side motion. Contact pressure between the test specimen and piston rod was varied by adjusting the weight attached to the loading arm. Running surfaces were immersed in an F-50 silicone fluid bath. The assembly was closed off to provide a nitrogen atmosphere over the degassed silicone fluid.

All tests were conducted on the same piston rod. The rod was shifted to a new position for each test to provide an untested surface for each material. Testing was conducted with a piston rod stroke of 0.5 inch at surface speeds of 0.8, 1.6, and 2.5 inches per second. The material specimens were run at contact pressures of 300, 600, and 1200 psi. Failure of a specimen was defined as a two-fold increase over the initial running friction determined for a given contact load. When failure occurs, shut-down of the test is automatic. This was accomplished with a calibrated spring capsule attached to the piston rod and connected in series with the driving motor. When friction exceeds the preset load of the spring capsule, the electric contact was broken and the driving motor shuts down.

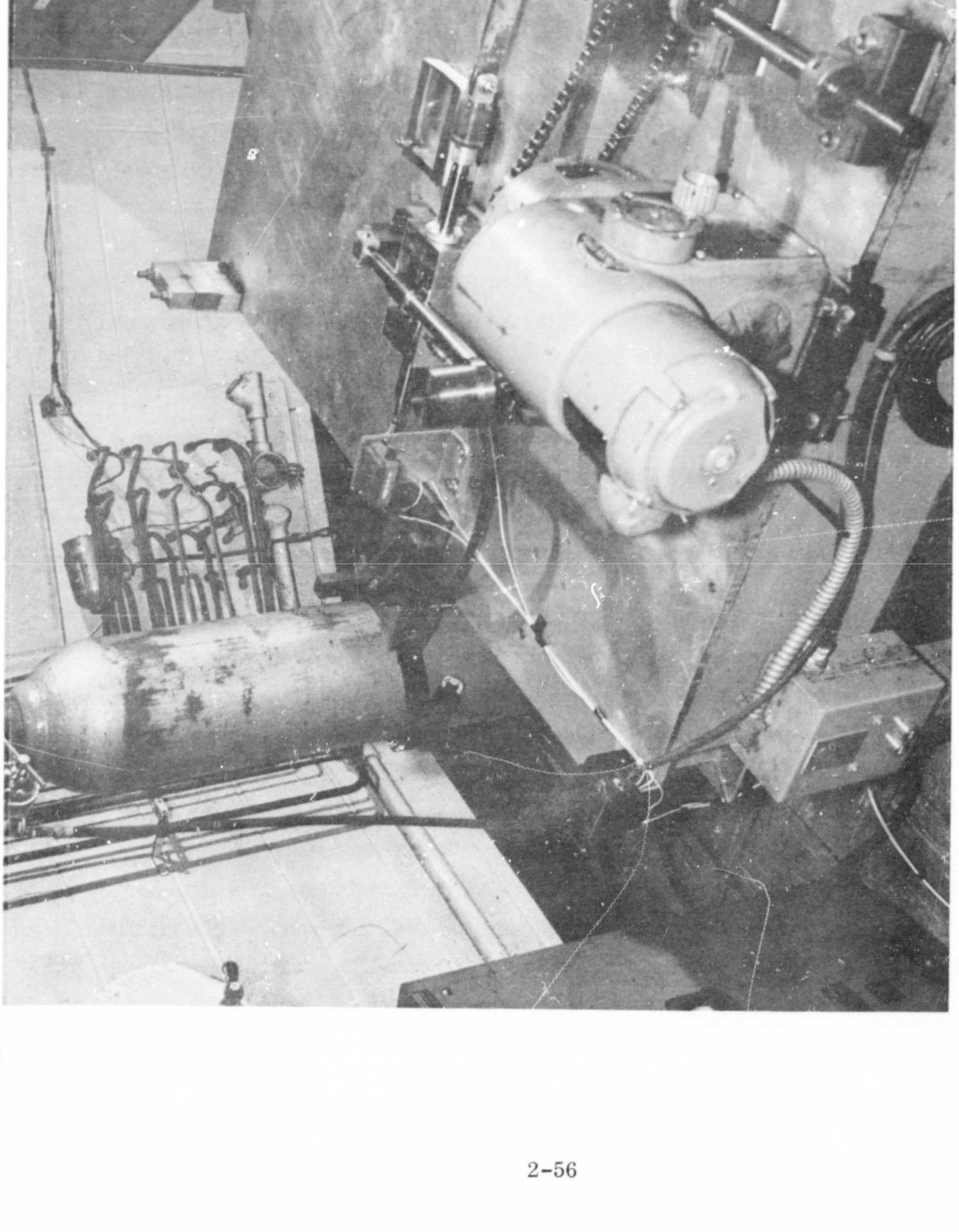


Figure 2-28. Sliding Wear Tester

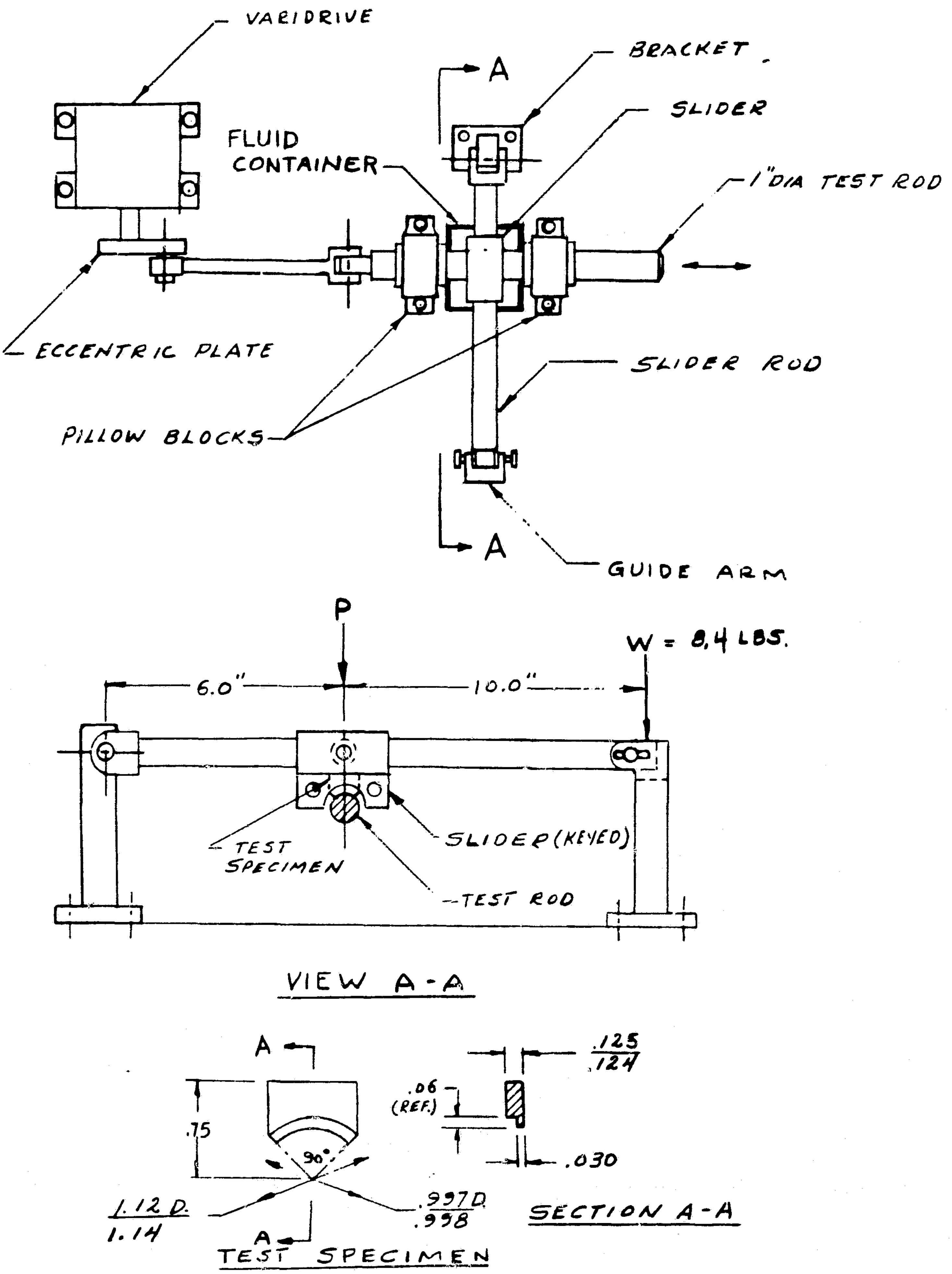


Figure 2-29. Details of Sliding Wear Tester

2. Results of Sliding Wear Tests

Test results are summarized in Table 2-16. The condition of the wear specimens and their respective mating surfaces is shown in Figures 2-30 and 2-31. In general, the results show that, of the hard base materials, the Vascojet 1000 and cobalt-molybdenum alloy did not produce any significant surface damage to the chrome plating. Although the titanium-tin alloy exhibited good compatibility with chrome plate, its wear rate was rather high. The flame-plated molybdenum coating produced scratching of the chrome plating at contact pressures above 300 psi. One difficulty encountered with this material is that of obtaining a good uniform surface finish. All of the soft materials tested exhibited low wear and good compatibility with chrome plating.

In examining the specimens after testing, it was noticed that they did not make full contact with the piston rod. Thus, the materials were tested under contact stresses that were approximately 25 to 50 percent higher than the calculated values.

Testing of the Vascojet 1000 material was terminated after completing 147,300 cycles at contact pressures up to 1200 psi. Inspection of the piston rod and specimen showed no evidence of scoring. However, light burnishing marks were visible on the chrome plated surface of the piston. The specimen indicated a weight loss of approximately 0.0011 gram.

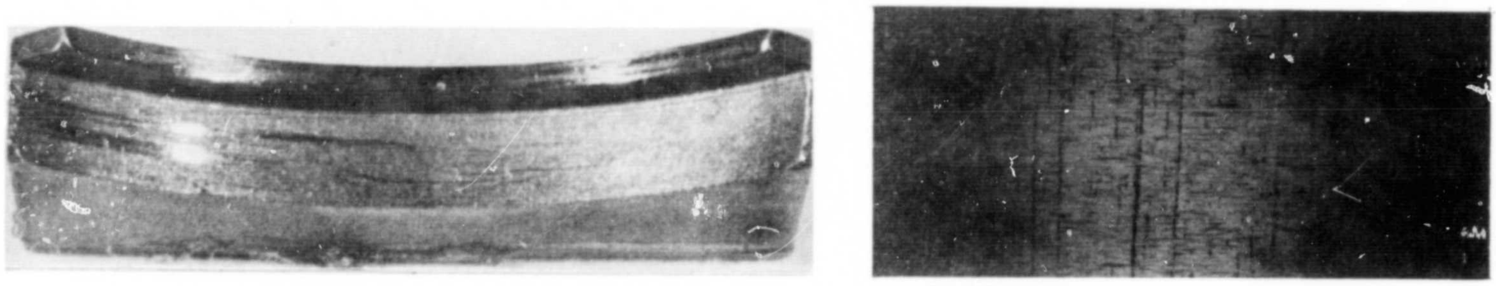
Testing of the titanium alloy, which contained 10% tin by weight, was terminated after 15.5 hours of cycling when excessive wear was observed at the 600 psi load level. Although this material exhibited a high wear rate (approximate weight loss of 0.131 gram), the material did not produce any galling or other damages to the chrome plating. Hardness of the test specimen was Rc-36. The wear particles from the test specimen were dispersed in the F-50 fluid, which turned black. However, after filtering the fluid through a five micron filter paper, the fluid regained its original color.

Testing of the cobalt alloy specimen (2% molybdenum by weight) was concluded after 38.5 hours of cycling. This material, which has a hardness of Rockwell B-86, showed very good compatibility with the chrome plated

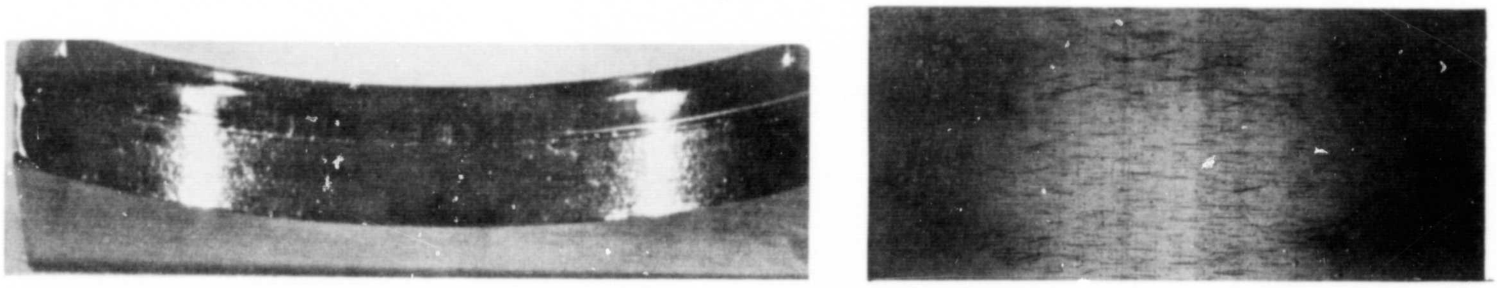
TABLE 2-16

SLIDING WEAR TEST - SUMMARY OF RESULTS

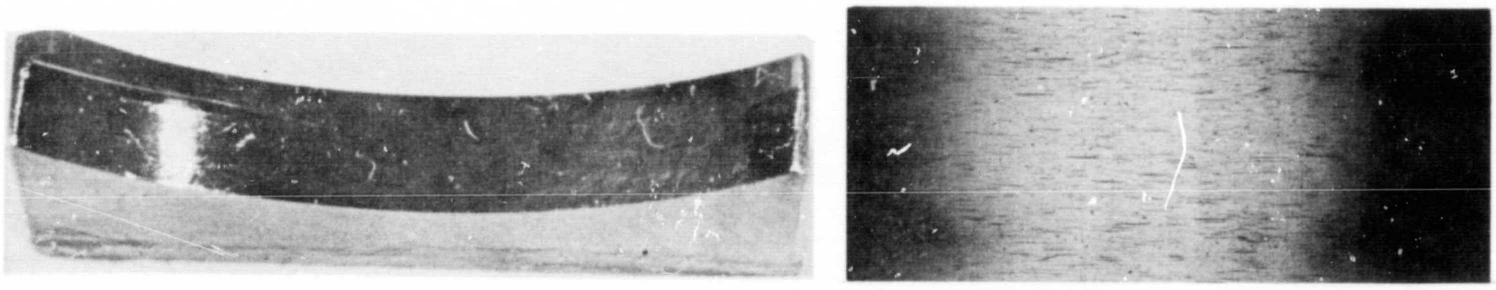
Material Specimen	Contact Pressure (psi)	Running Friction (lbs)	Surface Speed (in/sec.)	Cycles (1/2-in. Stroke)	Total Cycles	Time (hrs.)	Total Time (hrs)	Weight Loss of Specimen
Vascojet-1000	300	1.5	0.8	64,300		20		
	600	2.0	1.6	29,900		5		
	1200	2.0	2.5	53,100	147,300	6.5	31.5	.0011 gram
Titanium Tin Alloy	300	1.8	0.8	39,600		12.5		
	600	2.0	0.8	400		.5		
	600	2.0	1.6	14,300	54,300	2.5	15.5	0.131 gram
Cobalt-Molybdenum Alloy	300	1.2	0.8	31,800		12.0		
	600	2.2	1.6	64,800		11.0		
	1200	2.4	2.5	119,900	216,500	15.5	38.5	0.004 gram
Nickel Foametal w/ CaF_2 and BaF_2	300	2.0	0.8	39,400		12.5		
	600	2.0	1.6	47,600		10.0		
	1200	2.5	2.5	113,000	200,000	13.0	35.5	0.0017 gram
Polymet	300	1.2	0.8	38,800		12.0		
	600	1.3	1.6	53,200		9.5		
	1200	2.0	2.5	108,000	200,000	14.0	35.5	0.0004 gram
Silver Alloy (72% Ag + 28 Cu.)	300	.9	0.8	32,900		11.0		
	600	.9	1.6	57,100		10.5		
	1200	1.5	2.5	110,000	200,000	14.0	34.5	0.0004 gram
Pure Silver (Commercially Pure)	300	1.0	0.8	32,600		11.5		
	600	1.5	1.6	57,400		11.0		
	1200	1.7	2.5	110,000	200,000	12.0	34.5	0.0019 gram
Metco Flame-Plated Molybdenum (Vascojet-1000 base material)	300	2.5	0.8	39,000		13.0		
	600	3.0	1.6	52,000		8.0		
	1200	3.0	2.5	109,000	200,000	11.0	32.0	0.0008 gram



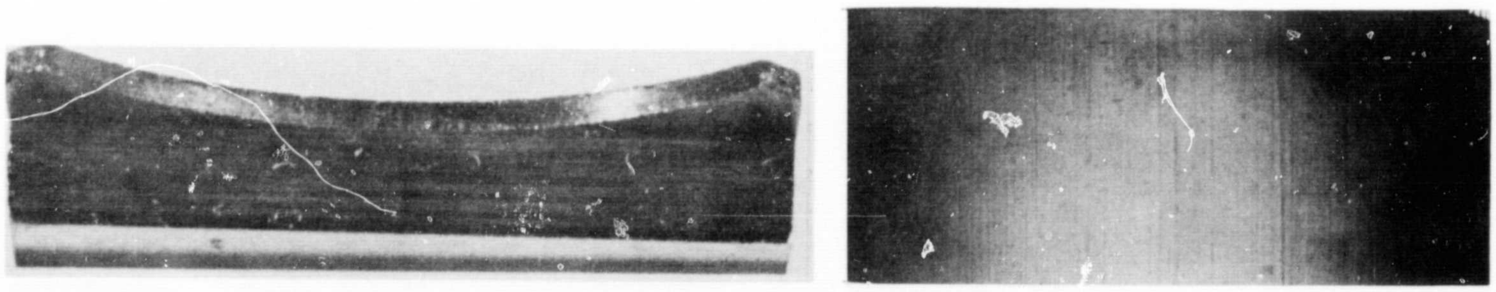
Vascojet 1000



Cobalt-Molybdenum Alloy (2% Molybdenum)



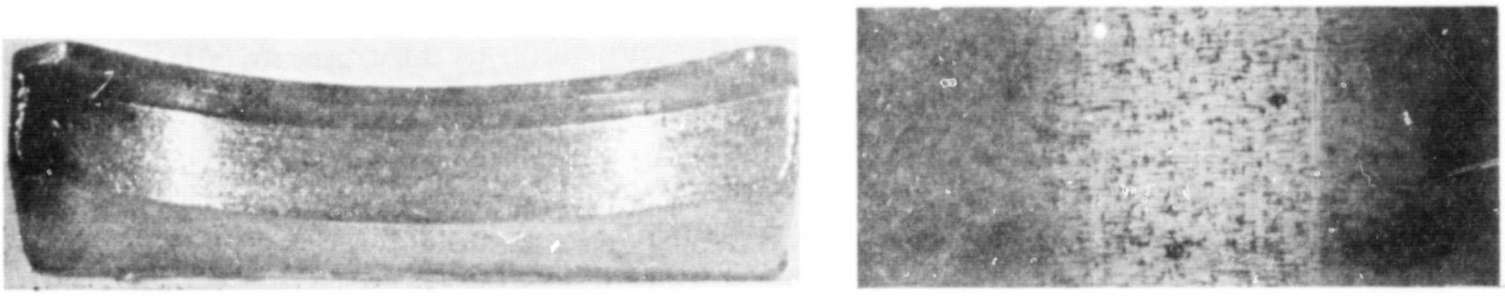
Titanium-Tin Alloy (10% Tin)



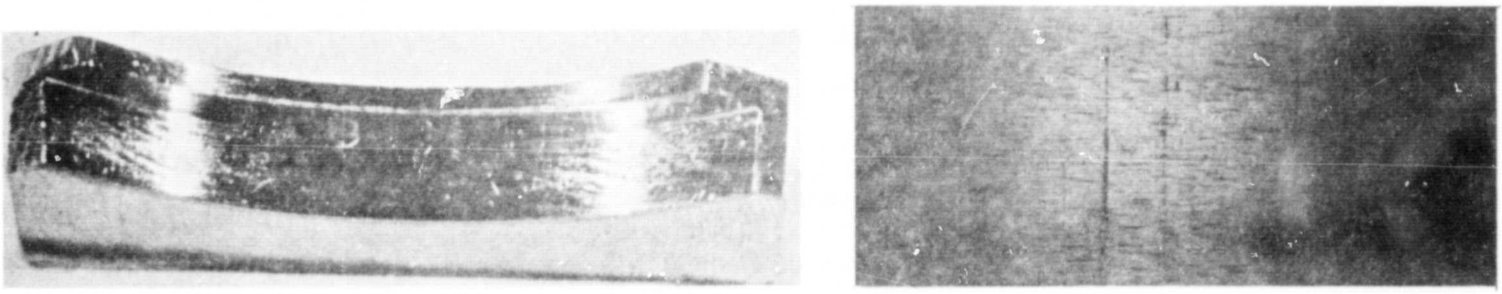
Flame-Plated Molybdenum on Vascojet 1000
(Burnished with Molybdenum Disulphide)

Wear Specimen 4x Magnification   Chrome Surface 25x Magnification

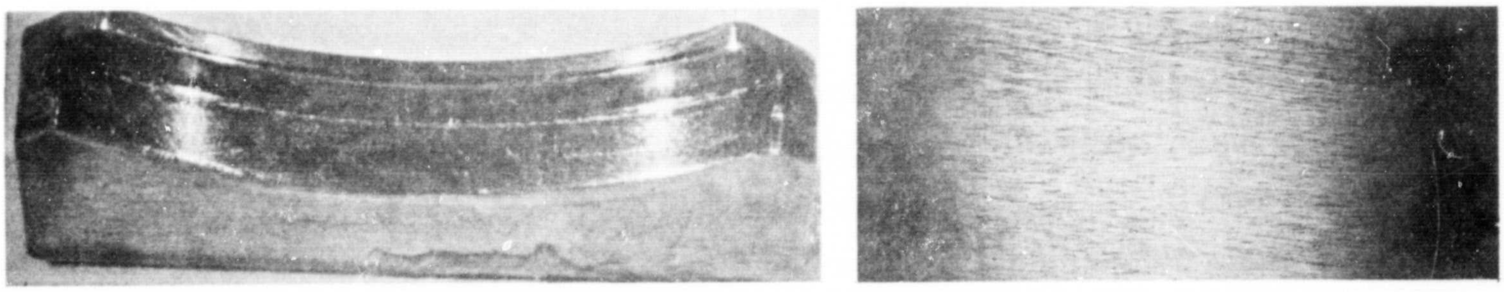
Figure 2-30. Sliding Wear Specimens and Chrome-Plated Mating Surfaces



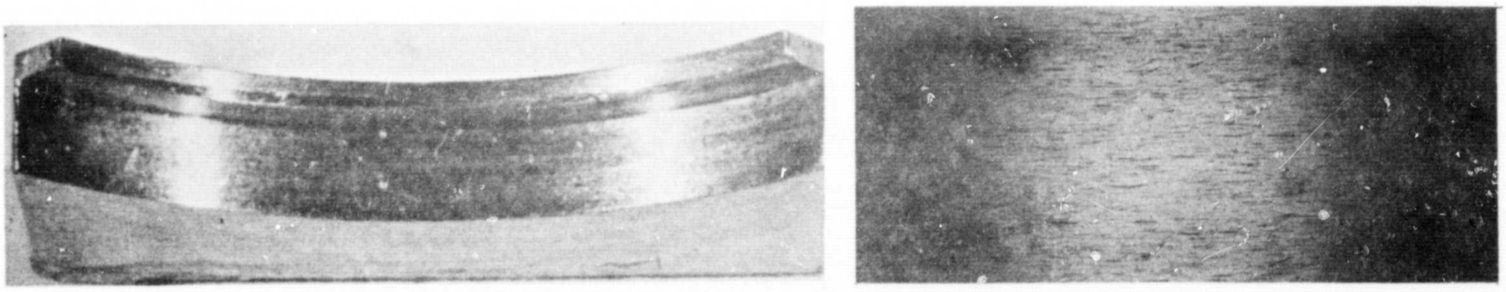
Nickel Foametal Impregnated w/BaF_2 + CaF_2



Silver Alloy (72% Ag + 28% Cu.)



Silver Polymer Composite (Polymet)



Commercially Pure Silver

Wear Specimen 4x Magnification Chrome Surface 25x Magnification

Figure 2-31. Sliding Wear Specimens and Chrome-Plated Mating Surfaces piston rod. Wear rate was very low in comparison to the titanium alloy. Weight loss was approximately 0.004 grams after 216,500 cycles.

In the test on the flame-plated molybdenum coating, the coating was applied over a Vascojet 1000 base material and ground to an apparent finish of 4 rms. The coating was then burnished with molybdenum disulphide powder in an inerted atmosphere. Testing of this material was concluded after 32 hours of operation. The molybdenum coating showed good wear resistance. Weight loss of the specimen was approximately 0.0008 gram after 200,000 cycles. Although the weight of the specimen prior to burnishing with molybdenum disulphide was not taken, it is quite conceivable that part of the weight loss is attributed to the loss of the molybdenum disulphide powder. Inspection of the piston rod revealed a highly burnished surface with a series of light longitudinal scratches. Burnishing of the surface was first noticed during cycling at a contact pressure of 300 psi. Evidence of scratching was exhibited when contact pressure was increased to 600 psi.

The nickel Foametal impregnated with CaF_2 and BaF_2, which has a hardness of Rockwell F-87, exhibited very low wear during 200,000 cycles. Weight loss due to wear was approximately 0.0017 gram. Compatibility with the chrome plated piston rod was good.

The Polymet composite and the silver alloy (72% Ag + 28% Cu) exhibited very low wear after 200,000 cycles. Weight loss was 0.0004 gram for both materials. No wear marks were noticed on the chrome plated piston rod.

The pure silver showed a much higher wear in comparison to the silver alloy and Polymet. Weight loss due to wear was 0.0019 gram. The pure silver material was included in this series of wear tests for the purpose of obtaining an indication of the wear characteristic of the silver-impregnated stainless steel fiber composite. In the fabrication of seals with the composite material, a coating of silver will be retained on the contact surface of the seal to prevent the metal fibers from coming in contact with the piston rod. Therefore, the wear characteristic of the silver-metal composite is quite dependent on the wear resistance of the pure silver.

3. Mechanical Properties Testing

The tensile properties of the candidate materials are summarized in Table 2-17. Because of the high cost of most of these materials, a scaled-down tensile specimen (Figure 2-32) was used to conserve materials. All testing was done on an Instron test machine in air at 600°F, using quartz radiant lamps as the heat source. The time to temperature was approximately 10 minutes, and the time at temperature was 3 to 5 minutes. Temperature sensing was by means of a chromel-alumel thermocouple wrapped around the gage length of the test specimen. The specimens were tested at a strain rate of 0.02 in./in. to the yield and increased to 0.2 in./in. from the yield to fracture. The yield strength was obtained at 0.2% offset.

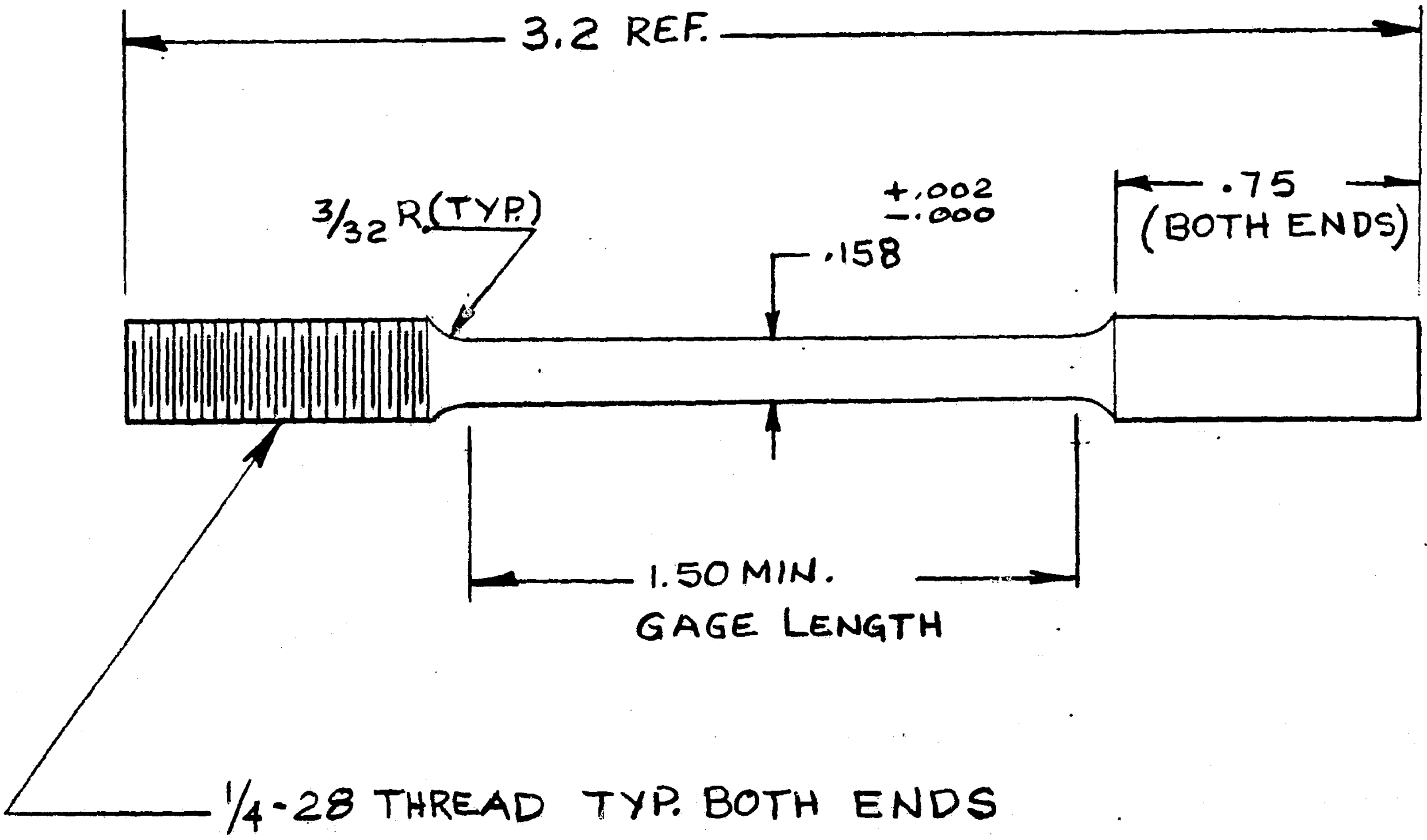


Figure 2-32. Tensile Specimen

TABLE 2-17. PROPERTIES OF CANDIDATE MATERIALS

Mechanical Properties at 600°F	Polymer SP-1	Polymet	Silver Composite 430 SS, 30% dense	Silver Composite Nickel,* 30% dense	Consil-720 Silver Alloy 72% - 28%	Consil-600 Silver Alloy,* 60% - 40%	Vascojet-1000 (H-11)	Nickel Foametal 60% dense Impregnated w/CaF_2 and BaF_2	Cobalt Alloy 75% Co + 25% Mo
Tensile Modulus of Elasticity (psi)	15.0×10^4	1.44×10^5	5.47×10^6	10.8×10^6	6.0×10^6	7.5×10^6	23.9×10^6	9.07×10^6	30×10^6
Tensile Yield Strength .2% offset (psi)	1285	3444	11,525	12,277	21,325	30,997	183×10^3	8297	117,100
Ultimate Tensile (psi)	1325	5805	18,762	19,333	32,550	41,361	225×10^3	8297	148,950
% Elongation	3.25%	7.6%	8.7%	11.5%	15.8%	24.7%	1.8%	2.45%	1.2%
Hardness**	Rockwell H-83-89	Rockwell H-53	Rockwell H-90	Rockwell H-83	Rockwell F-89	Rockwell H-97	Rockwell C-52	Rockwell F-87	Rockwell C-47-49
Coefficient of Friction	*** 0.08-0.15	*** 0.06-0.14							
Thermal Coefficient of Expansion (in./in./°F)	29.8×10^{-6}	8.9×10^{-6}					6.8×10^{-6}		

* Alternate material
** Hardness readings obtained at room temperature
*** Vendor's data

F. MATERIAL SELECTION

1. Selected Materials

Selection of the five candidate seal materials for the seal design phase was based on fluid compatibility, mechanical properties (Table 2-17), and sliding wear tests (Table 2-16). These materials are:

1) Polyimide plastic (unfilled)
2) Nickel Foametal 60% Dense (impregnated with CaF_2 and BaF_2)
3) Silver alloy (72% Ag + 28% Cu.)
4) Vascojet 1000 (H-11 tool steel)
5) Cobalt molybdenum alloy (75 Co, 25 Mo)

The unfilled polyimide plastic (Polymer SP), exhibited good fluid compatibility and temperature resistant up to 600°F. The mechanical properties of the material appear adequate for seal configurations such as V-seals and lip seals. (See Section III). For the latter design, the rolled sheets are preferred over the billet form of the material because of their improved flexibility and higher elongation.

The nickel Foametal impregnated with an eutectic mixture of CaF_2 and BaF_2 was selected from several soft base metals. The self-lubricating property and good wearing qualities are desirable features of this material. Although results obtained in the fluid-material compatibility tests at 600°F indicated that the silicone fluid in contact with the Foametal exhibited a high acidic condition, this condition may be due to breakdown of the silicone fluid itself. Fluid compatibility tests conducted at 500°F and 400°F showed no evidence of high acidity in the silicone fluid. Compatibility of this material with PR-143AB, MCS-293, and MLO-60-294 is good.

The silver alloy consisting of 72% Ag + 28% Cu was selected from among the silver base materials. The copper content of the silver alloy provides a harder material, which should result in better wear resistance than the pure silver materials. Sliding wear tests verify this assumption. The mechanical properties of this material appear to be well suited for soft metal seal applications. Results obtained in the 600°F fluid compatibility test indicated that the F-50 silicone fluid became quite acidic after being in contact with this material. However, partial breakdown of the silicone fluid may have been a contributing factor. The acidic condition of the fluid was considerably lower at 500°F and negligible at 400°F. Its compatibility with the PR-143AB, MCS-293, and MLO-60-294 fluids was good.

Vascoject 1000 (H-11 tool steel) is a tool steel type material that is suited for hard metal seal application. This material is capable of retaining high strength at elevated temperatures. It has also demonstrated low wear and fairly good compatibility in sliding contact with chromium plating.

The cobalt-molybdenum alloy selected consists of 75% cobalt and 25% molybdenum. A sliding wear test conducted on a cobalt-molybdenum alloy of similar composition indicated low wear and good compatibility with chromium plating. Compatibility with fluids is good. High-temperature mechanical properties of this alloy appear quite suitable for hard metal seal application where good tensile properties are required.

2. Eliminated Materials

Among the materials not selected was the silver-tungsten diselenide composite, which was grossly incompatible with the F-50 silicone fluid at 600°F. The silicone fluid in contact with this material crystallized. Another material not considered was the titanium tin alloy. This alloy exhibited excessive wear when in sliding contact with chromium plating. Although the Polymet material exhibited very low wear in the sliding wear test, it was not considered because of the organic filler used. The filler is a fluorocarbon type polymer which may tend to partially decompose during long exposure to elevated temperatures, resulting in a porous silver structure.

The silver-impregnated stainless steel (Type 430) fiber composite exhibited some corrosion effects when in contact with the F-50 silicone fluid. Therefore, it was dropped as a candidate material. The pure silver impregnant also exhibited higher wear than the silver-copper alloy.

The Metco flame-plated molybdenum coating exhibited good compatibility with chrome plating under low load (600 psi) conditions but tended to scratch the rod at higher loads.

[redacted].

SECTION III

SELECTION OF SEAL CONCEPTS

A. GENERAL

To meet the low external leakage (one drop per minute) and long life (3000 hours) requirements of this program, primary consideration was given to a two-stage design. In this approach, the first-stage (high-pressure) seal is subjected to the full operating pressure, and a controlled amount of internal leakage is returned to the system reservoir. The second-stage (low-pressure) seal is subjected only to the system return pressure and has practically zero external leakage. First-stage seal leakage is kept to a minimum because the large number of actuators used in a flight vehicle would otherwise present a substantial tare power drain on the system.

The approach to the first-stage seal was to use split-ring contracting rod seals. Experience has shown that this type of seal can provide effective sealing with acceptable leakage. Hydraulic balancing to reduce wear and friction was readily accomplished with these sealing rings.

For the second stage, consideration was given mainly to positive-contact-type seals. This approach had the greatest potential for meeting the leakage requirement of one drop per minute. A major portion of the effort was devoted to the second-stage seal because of its more stringent leakage requirements.

In accomplishing the objectives of this task, efforts were directed towards the investigation of potential seal concepts and selection of the five (5) most promising seal configurations for further development.

B. LOW-PRESSURE SECOND-STAGE SEAL CONCEPTS

1. <u>Seal Designs for Plastic Materials</u>

The seal designs shown in Figure 3-1 and 3-2 were intended for use with the polyimide plastic. These designs fully utilize the relatively

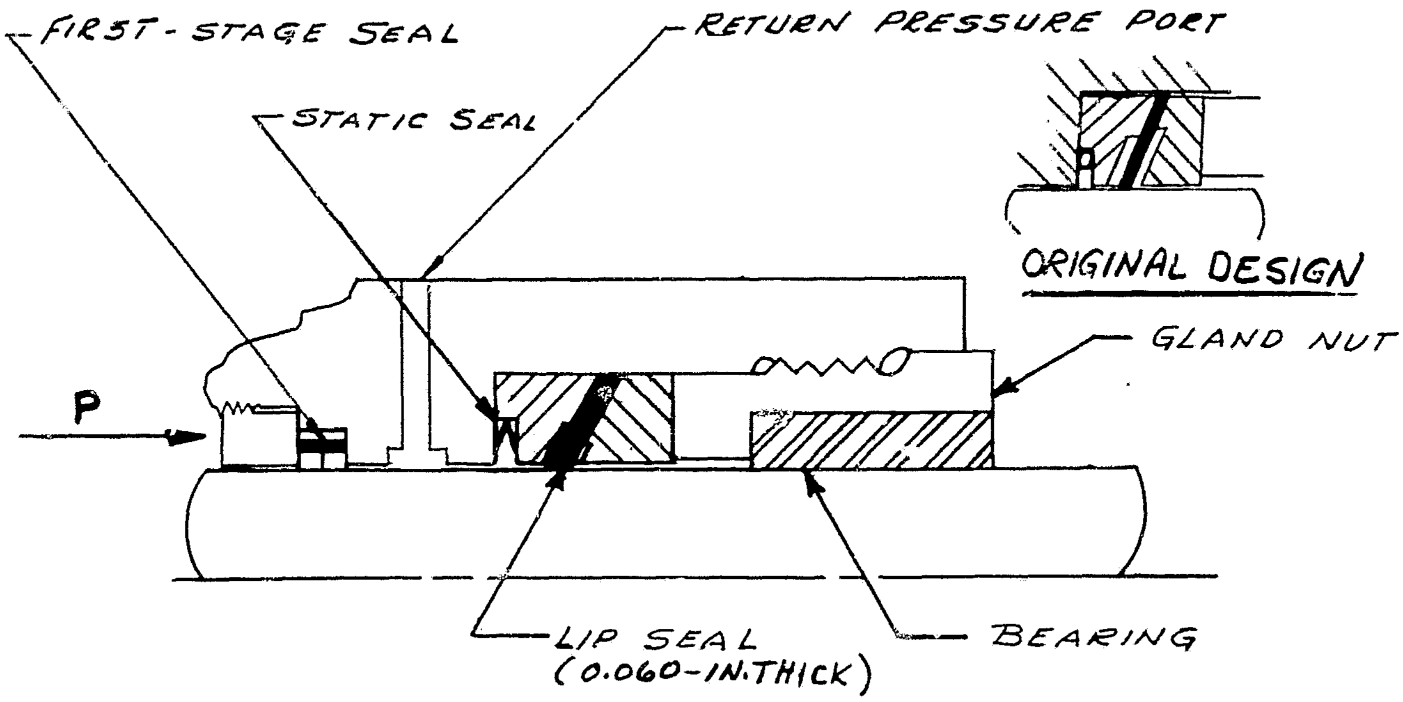


Figure 3-1. Design A - Polyimide Lip Seal-Low Pressure

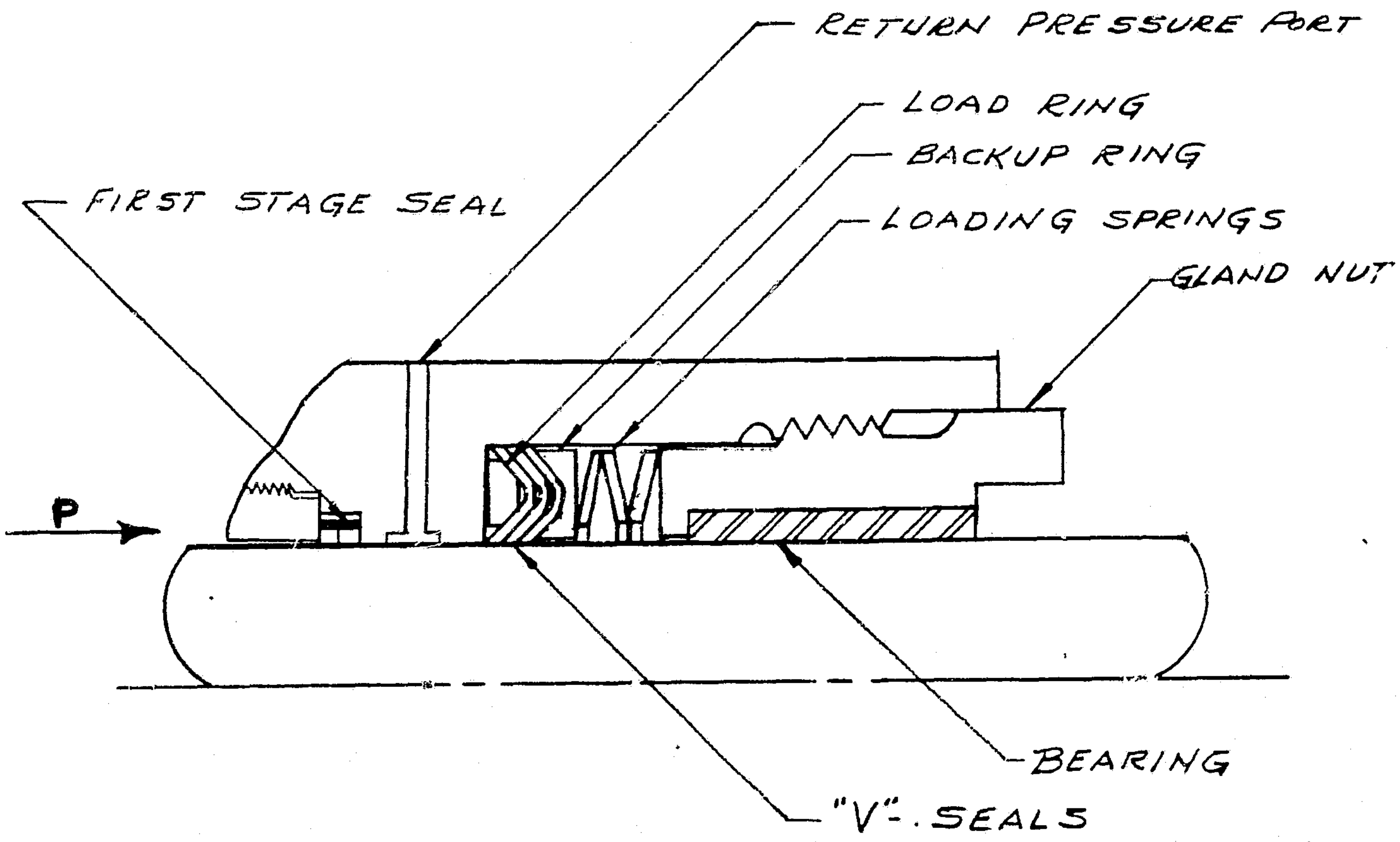


Figure 3-2. Design B - Polyimide V Seal-Low Pressure

high elastic properties and conformability of the material to effect a seal. The seal (Design A) shown in Figure 3-1 utilizes the relatively high elongation (approximately 3-to-4%) of the material to obtain a substantial interference fit over the shaft during initial assembly and thus to induce the initial seal contact pressure without use of an external loading device. This contact pressure is further augmented by fluid pressure. Preliminary tests were conducted with this seal-material combination at 400°F, 500°F, and 600°F. Results show that zero leakage was maintained at temperatures up to 500°F. (See Section IV for detailed discussion.)

The V-seal (Design B) design shown in Figure 3-2 has undergone substantial evaluation in previous seal programs conducted by the contractor (References 1 and 2). The seal consists of three sealing elements plus a load ring, a backup ring, and loading springs. The seal is assembled onto the shaft with essentially zero clearance. The outer leg provides the static seal and the inner leg effects a seal on the shaft. The springs are designed to give a constant load over a deflection of approximately 0.020-inch, which is considered adequate to compensate for wear.

2. <u>Seal Designs for Soft Metals</u>

The soft-metal seal approach relies on the plastic deformation of the material to conform to the surface geometry of the piston rod. One of the primary considerations in this approach was to minimize welding of the seal material to the mating member. This is best met by metals of relatively low shear strength, such as silver. Spring mounting of this type seal would also be required to provide uniform loading of the seal to compensate for wear. The feasibility of this approach was demonstrated (References 1 and 2) with the silver-base materials.

The seal design shown in Figure 3-3 (Design C) consists of conical-shaped sealing elements fabricated of a soft metal. The beveled edges of the inner and outer surfaces were used to effect sealing at the piston rod and seal gland, respectively. The loading rings were chamfered at an angle of 15 degrees so that, when they are tightened, they tend to induce axial and radial

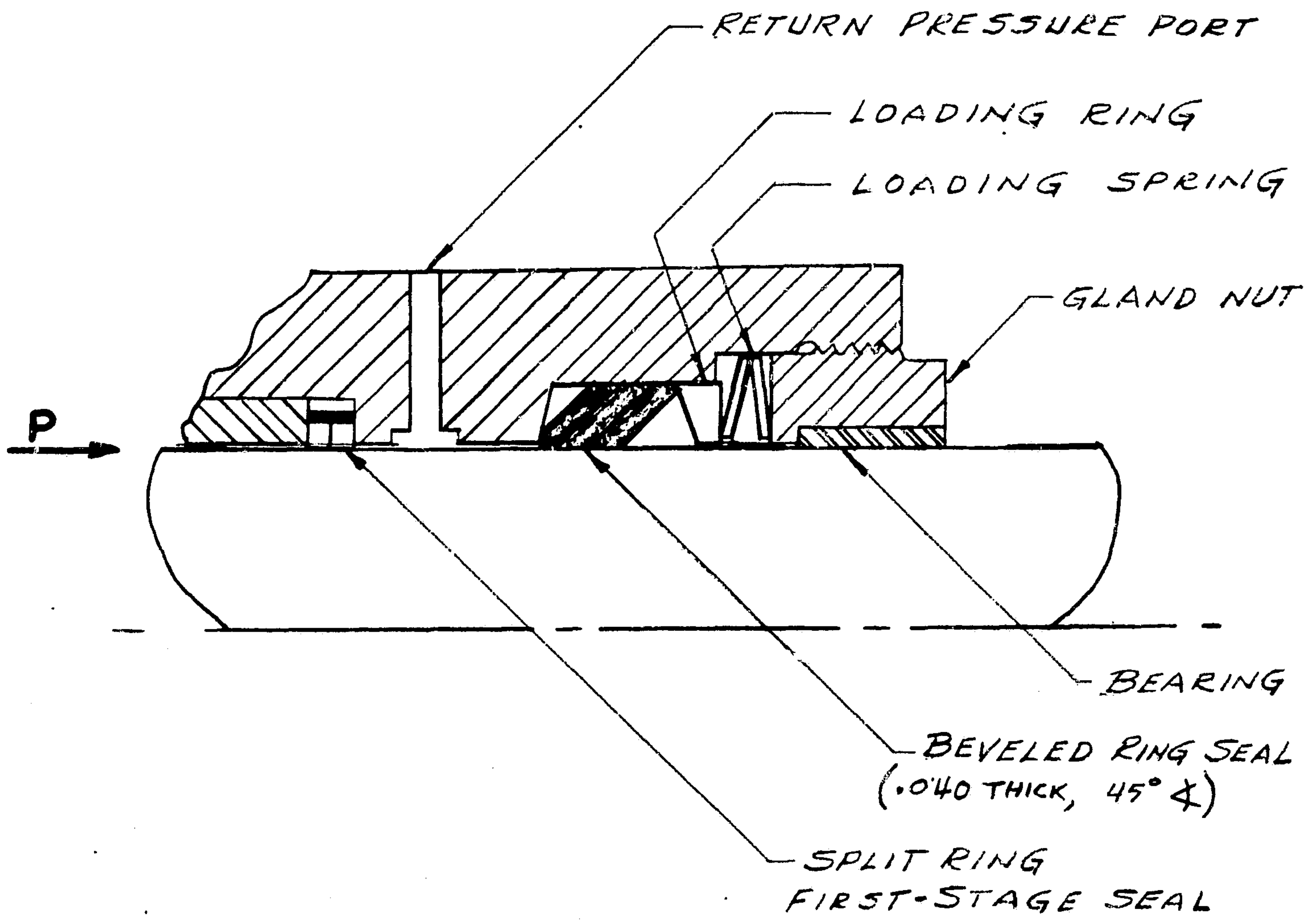


Figure 3-3. Design C - Bevelled Ring Seal (Soft Material)

motion of the seal. Static sealing at the gland face was accomplished by the beveled edge of the innermost ring. The small effective area of the seal that is exposed to the working pressure enables the use of a light loading spring. Therefore, contact stresses at the seal interface were kept to a minimum.

The seal (Design I) shown in Figure 3-4 consists of a wedge-shaped ring held in close contact with the piston rod by an axial force supplied by spring washers. The spring load also acts to compensate for wear. This configuration (Reference 10) gave satisfactory results up to 800°F and 4000 psi for short periods. Best results were obtained using graphite and the silver-stainless composites as the seal materials (Reference 9).

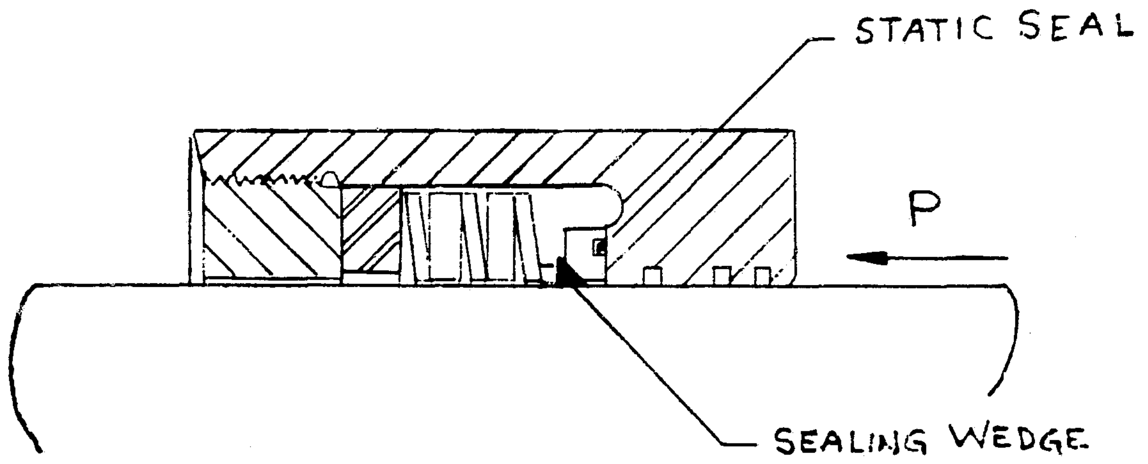


Figure 3-4. Design I - Spring Loaded Wedge Seal

The C-shaped seal (Design L) shown in Figure 3-5 utilizes a slight preload to provide initial contact at the static surface and between the seal and piston. The seal is also pressure-energized during operation. Seals fabricated from the silver-stainless composite have been tested up to 1000°F. Satisfactory performance was obtained at 100 psi (References 8 and 10).

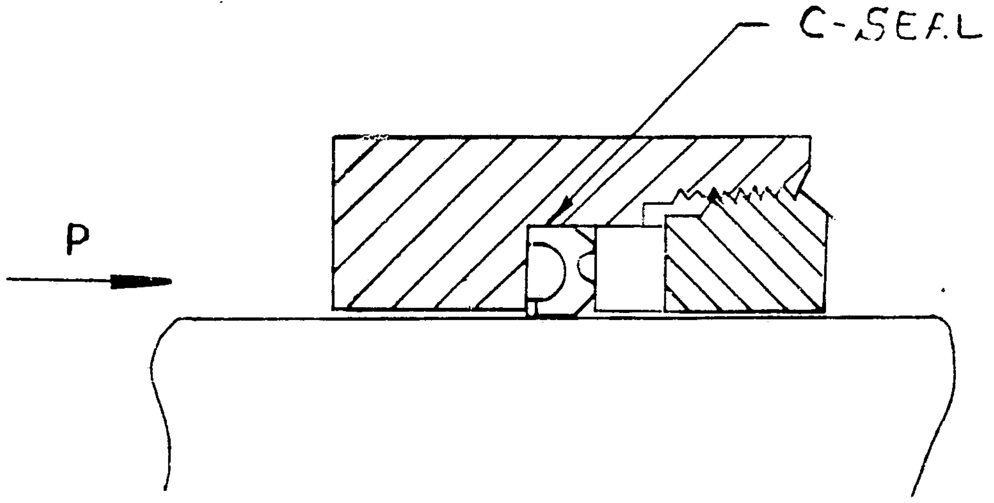


Figure 3-5. Design L - Metallic C Seal, Pressure Energized

3. Seal Designs for Hard Metal

The approach to a hard-metal seal was to design the seal to stretch elastically over the piston rod. The preload (hoop stress) induced will provide the contact pressure at the seal interface. However, the initial strain due to the preload must be within the elastic limits of the material. Because it is important that the material does not yield, the interference fit will be relatively small, and the material must possess good wear-resisting qualities in order to retain the induced load. Mechanical or pressure loading may be required to compensate for wear. Hard flame-plated coatings on the piston rod may be a requisite in obtaining good wear resistance.

Figure 3-6 (Design D) depicts a metallic lip seal developed by Republic. The design of the seal provides for an interference fit over the rod to effect a seal. In order to avoid a line contact condition, which would result in high contact stresses at the seal interface, the design provides a contact width of approximately 0.030 to 0.040-inch on assembly. This enables the sealing load to be distributed over a relatively large area, which results in lower unit contact stresses. By having the sealing lip facing away from the fluid, excessive build-up of contact stresses due to the pressure of the fluid is avoided.

This arrangement also permits relieving of the sealing load at the seal interface since the fluid pressure will tend to open the inner diameter of the seal. Such a feature was found to be advantageous in reducing friction and wear.

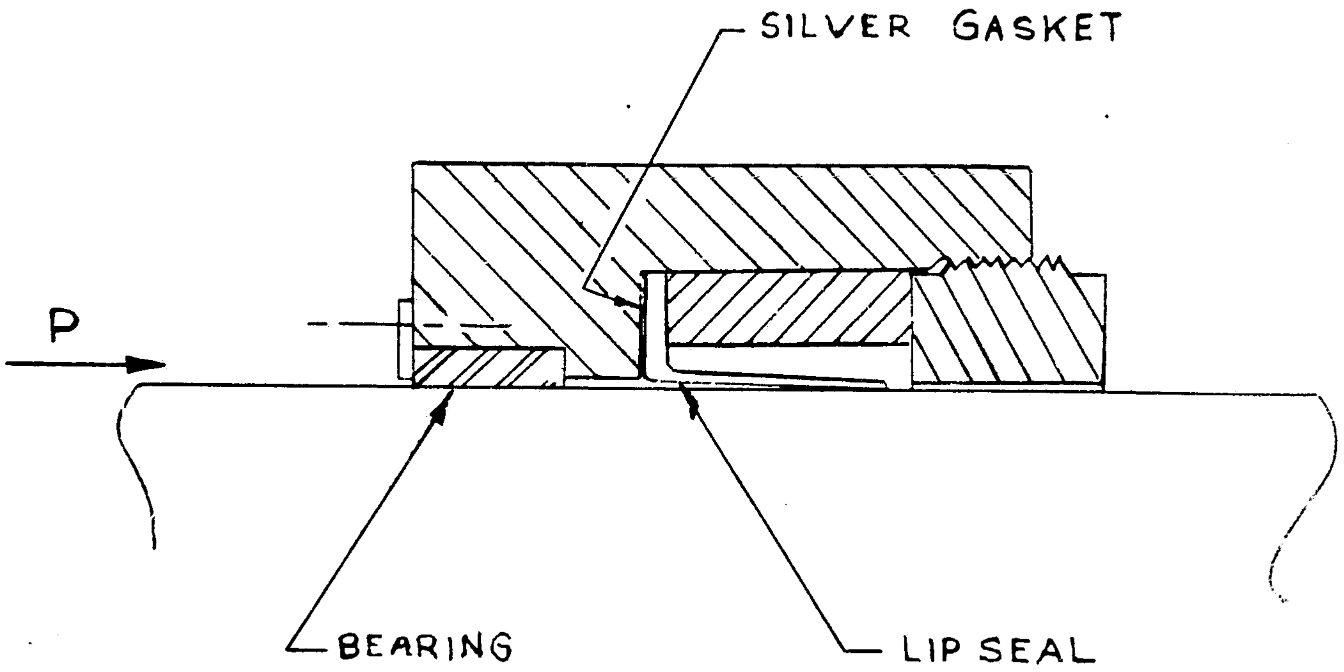


Figure 3-6. Design D - Metallic Lip Seal

The seal shown in Figure 3-7 (Design E) was designed for use with hard metals or as a bimetallic seal (hard and soft metal combination). It consists of a series of sealing reeds approximately 0.005 to 0.007 inch thick that are stretched over the piston rod to obtain radial loading. The design of the seal permits the working fluid to provide additional loading. A mechanical spring deflects the sealing lips against the shaft in the event of loss of preload (interference) due to wear. The bimetallic version of this design, which appears to be the most promising approach, consists of alternate reeds of a hard metal and a soft metal. The hard metal provides the strength and the soft metal provides good conformability.

Figure 3-8 (Design F) depicts a truncated-cone design, which permits independent loading of the static and dynamic portions of the seal by using separate adjusting nuts. Finer load adjustments are thus maintained. As shown in Figure 3-8, the sealing reed is preloaded against the rod by the initial inter-

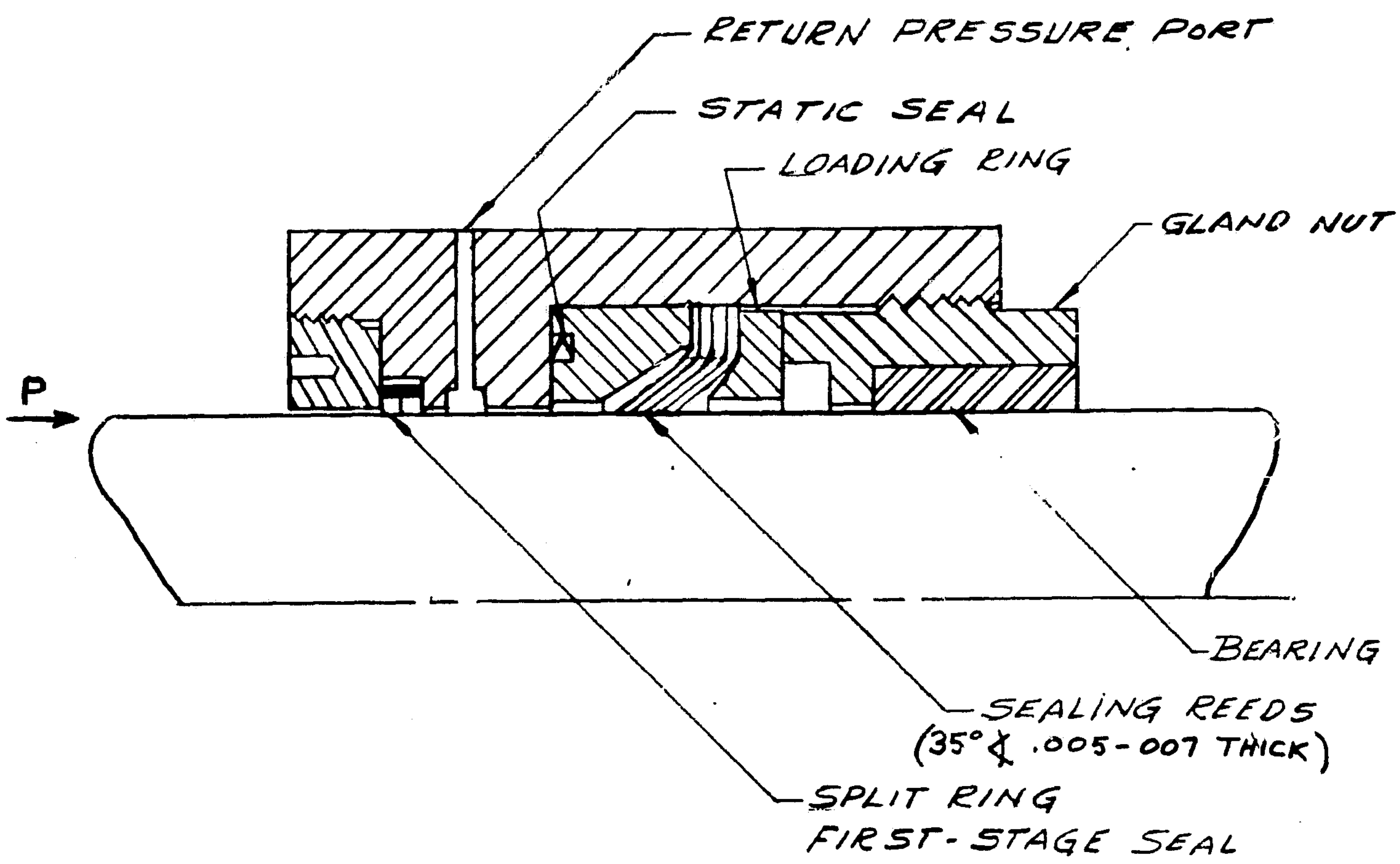


Figure 3-7. Design E - Metallic Reed Seal (Hard Metal)

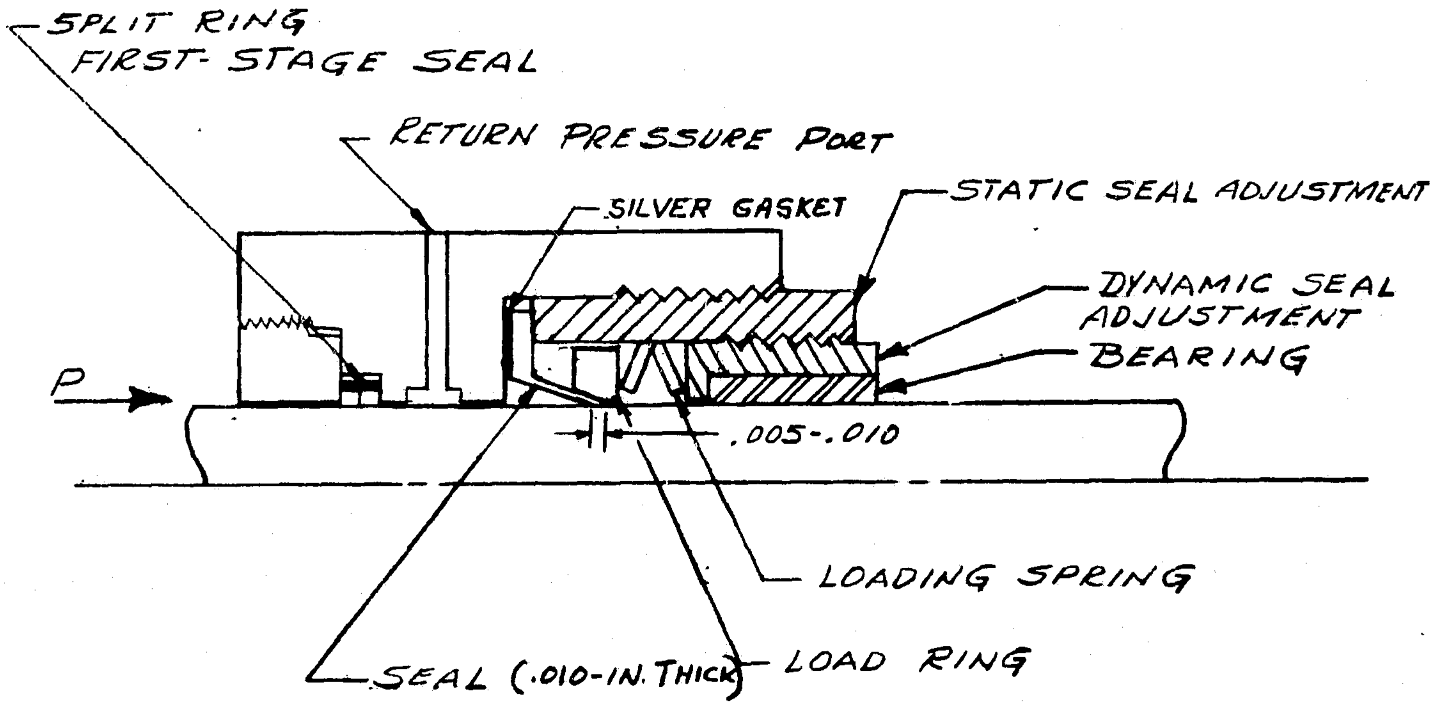


Figure 3-8. Design F - Metallic Lip Seal (Hard Metal)

ference fit. Based on stress calculations, using Vascojet-1000 (H-11 tool steel) as the seal material, an interference of approximately 0.004-inch can be achieved by working to a stress level of 135,000 psi (ultimate for Vascojet is 190,000 psi at 500°F). Analysis also shows that load relief at the seal interface by fluid pressure (100 psi) acting on the back of the reed is feasible. However, the radial deflection (0.00018-inch) of the seal is relatively small. The effects of this radial deflection can only be determined by actual test.

Figure 3-9 depicts a reed-type seal (Design H) which consists of a series of thin-walled elements preloaded on an assembly to provide positive contact with the piston rod. Limited testing at 600°F and 3000 psi indicated that effective sealing can be obtained. However, rapid wear of the seal and galling of the chrome plated piston rod were major problems.

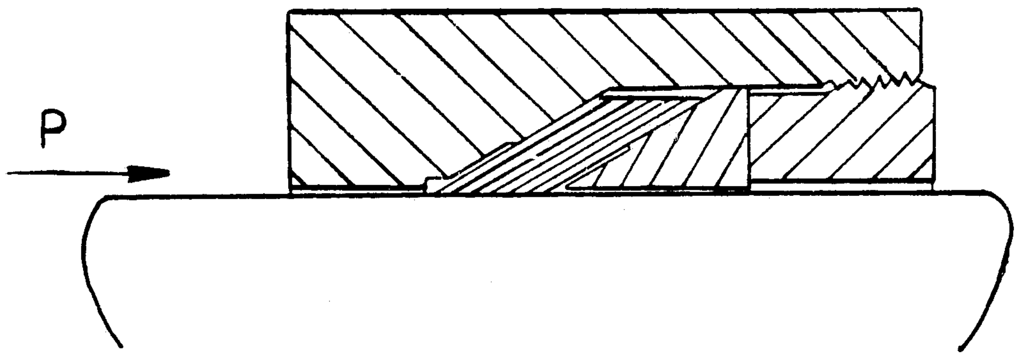


Figure 3-9. Design H - Metallic Reed Seal

The ring spring seal (Design J) shown in Figure 3-10 consists of two outer rings and one inner ring. Sliding of the two outer rings occurs along the tapered surfaces of the inner ring upon application of an axial load, resulting in compression of the inner ring against the piston rod. This configuration has been evaluated in the 600°F to 900°F range at 4000 psi (Reference 10). Seal life is relatively short due to low wear compensation.

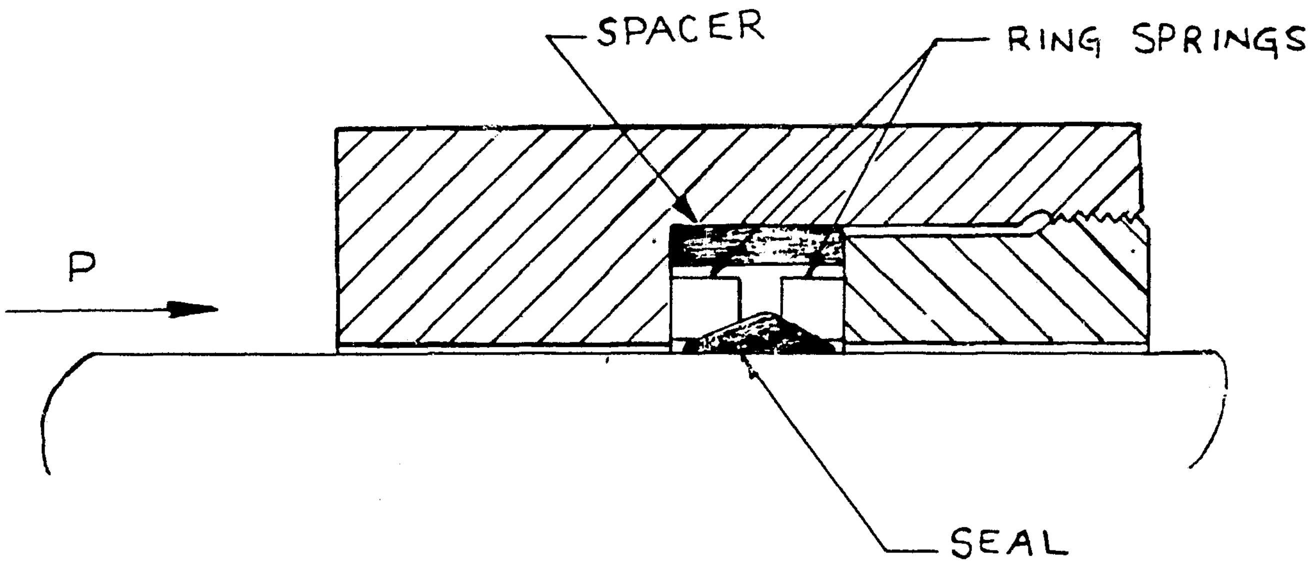


Figure 3-10. Design J - Metallic Ring Spring Seal

Figure 3-11 (Design K) depicts a design consisting of an X-shaped seal and two loading rings. Sealing is accomplished by an axial force applied to the load rings, which in turn deflect the legs of the X into intimate contact with the piston rod; in a similar manner, a seal is also effected on the outer diameter of the gland to seal statically. Lack of wear compensation was the major shortcoming of this design.

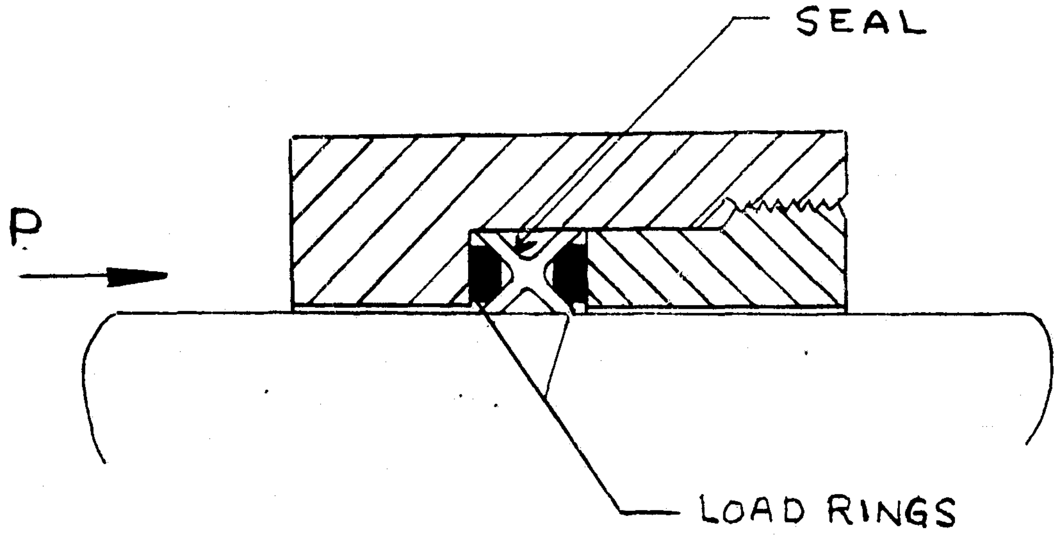


Figure 3-11. Design K - Metallic X Seal

C. HIGH-PRESSURE FIRST-STAGE SEAL CONCEPTS

The state-of-the-art of first-stage seals of the split contracting ring type is relatively well advanced and several proven designs were commercially available (Figures 3-12 to 3-15). Materials of construction include Type 440C stainless steel, polyimide and alloyed cast iron. All are preloaded to contact on the rod by means of a circumferential spring. A segmented graphite seal, consisting of two face-to-face rings with the joints staggered, is also available. This seal depends on pressure loading to attain low leakage.

Figure 3-15 illustrates a pressure-balanced sealing ring with axial loading rings that is also suitable for high-pressure applications. The split sealing ring is pressure balanced to reduce friction and wear. The friction backer ring, which consists of one split wedge ring and one solid wedge ring, provides the axial springing to the seal to maintain contact with the groove side. The backer ring assembly is also pressure balanced. Testing of this configuration with Oronite fluid by the Koppers Company indicated leakage of approximately 30 cc/hr. from 300 to 3000 psi at 450° (Reference 11).

D. COMMERCIAL SEAL DESIGNS

High-temperature metallic seals under development by commercial concerns were surveyed for their applicability to the subject program. An all-metal seal (Design M) submitted by B. F. Goodrich was considered for evaluation in the subject program. The seal design proposed by B. F. Goodrich is a two-stage configuration. Basic design of the first-stage (high-pressure) and second-stage (low-pressure) seal utilizes a knife edge or line contact method of sealing. However, only the low-pressure seal has provisions for external adjustment to compensate for wear. Initial sealing is accomplished by an interference fit of approximately 0.0012 inch over the piston rod. Although Republic has emphasized the use of the chrome plating for the piston rod, BFG recommends that a flame-plated tungsten carbide (LW-1) coating be used in order to provide a hard bearing surface for the seal.

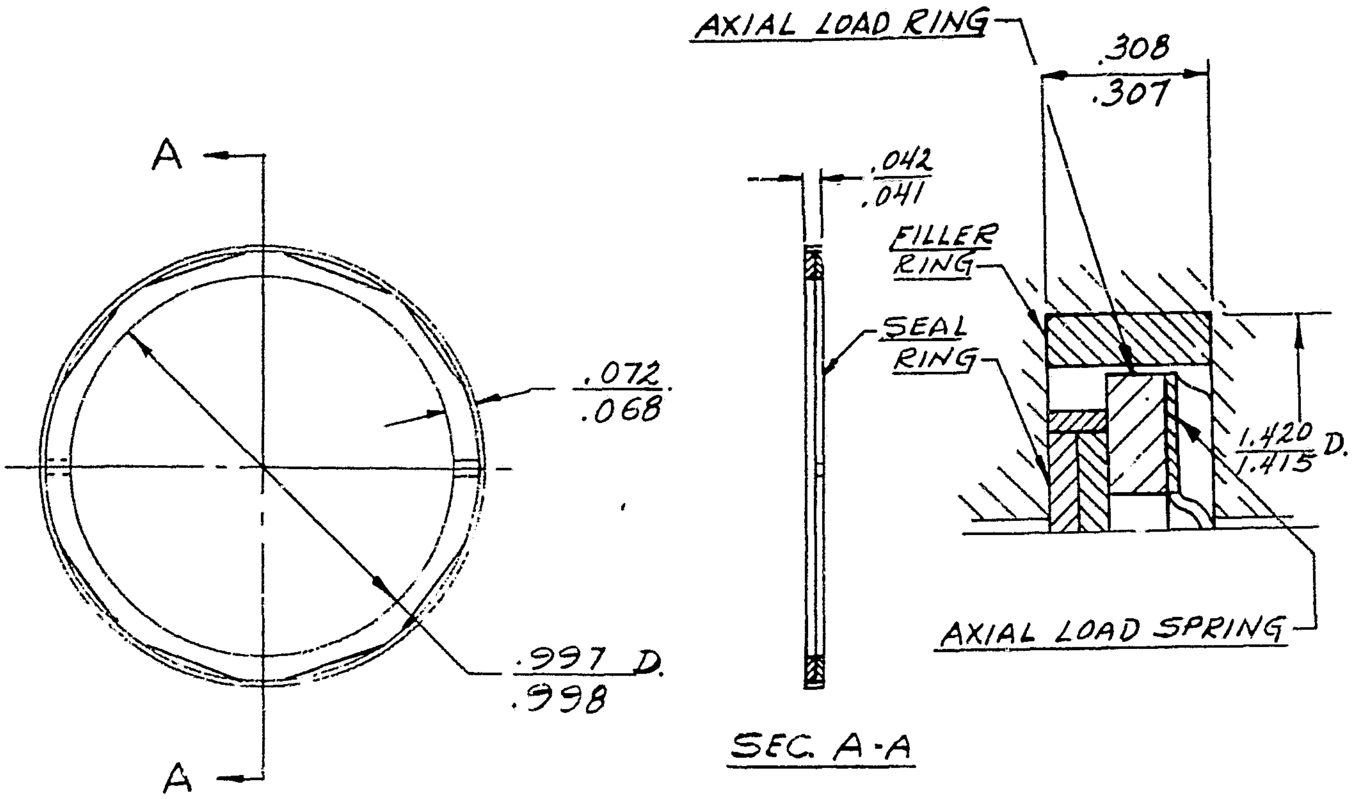


Figure 3-12. Cook Seal, Type MX-1 (Type 440C)

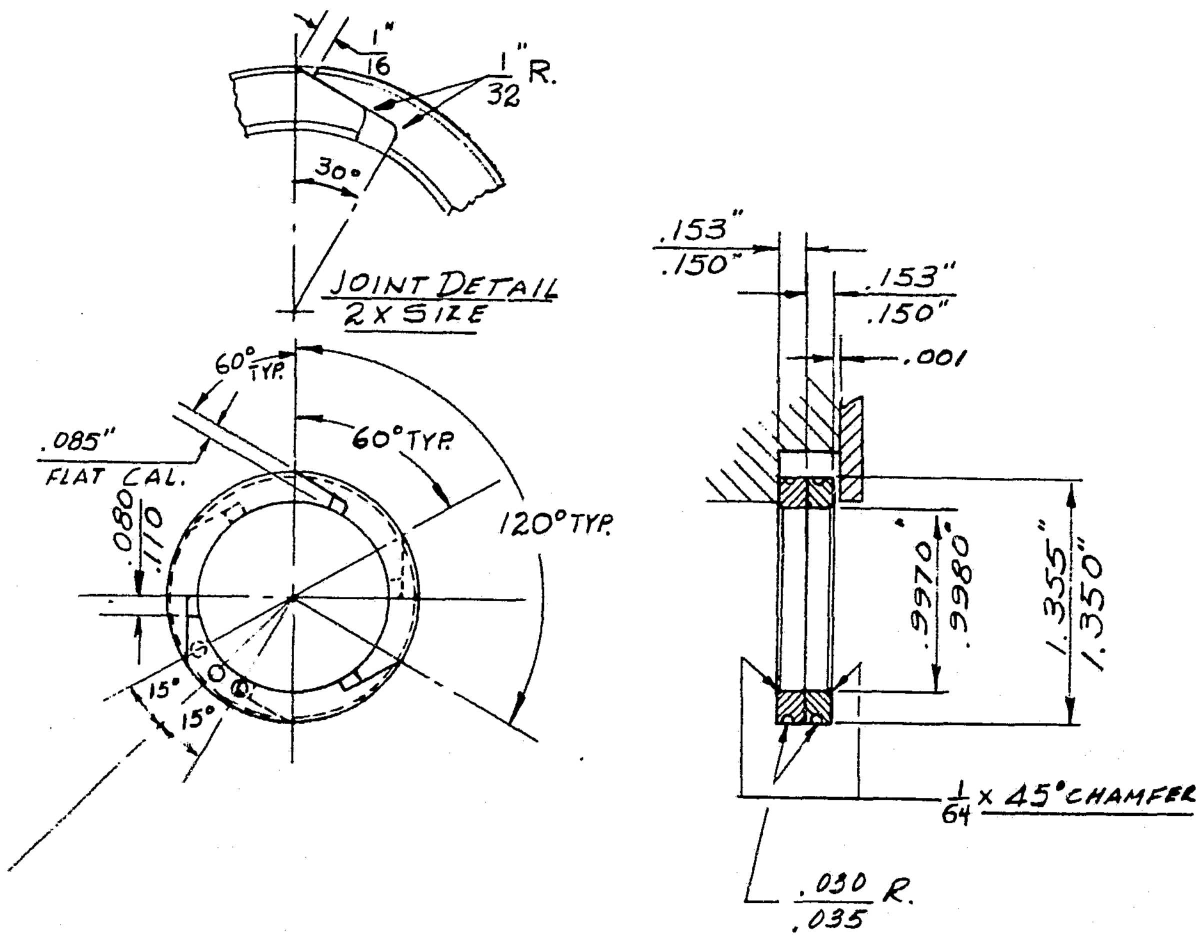


Figure 3-13. Segmented Graphitar* Seal Ring

* United States Graphite Co.

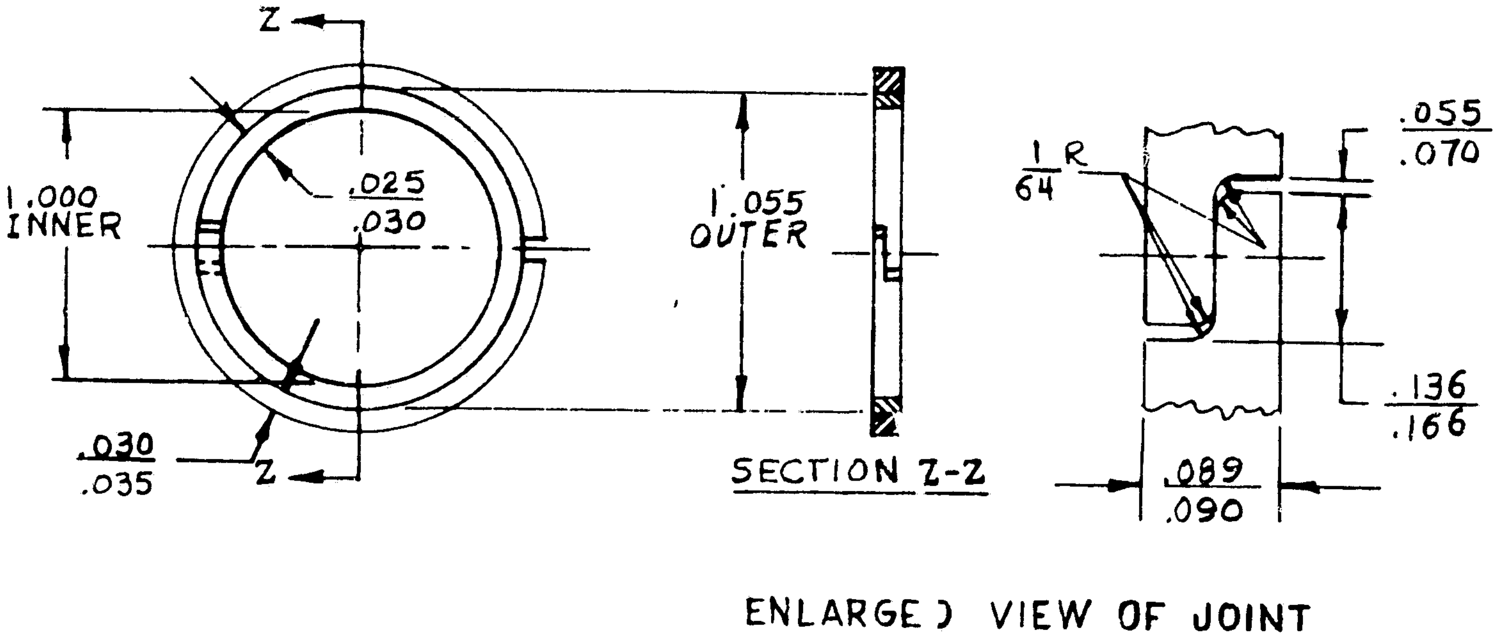


Figure 3-14. Contracting Two-Piece Split-Ring Seal

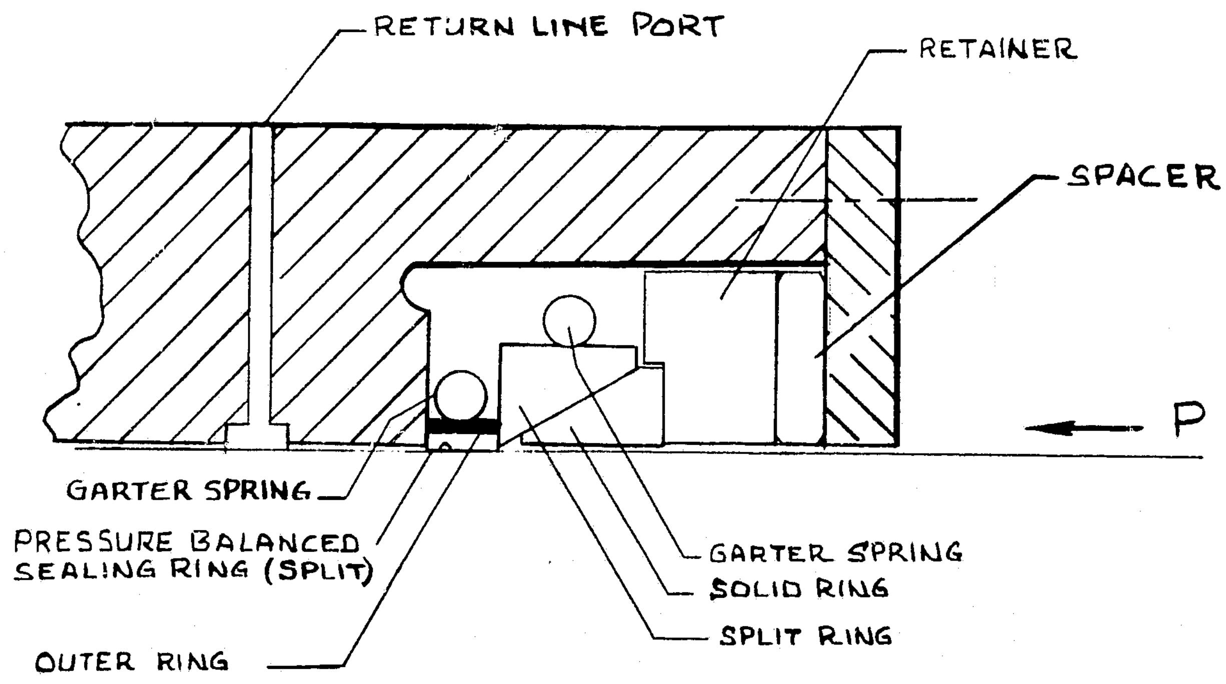


Figure 3-15. Pressure Balanced Sealing Ring with Axial Loading Rings

E. SELECTION OF CANDIDATE SEAL DESIGNS

In selecting the five most promising concepts for the second-stage seal, a rating system was established. The philosophy, basic approach, rating work sheets, and summary of the rating results are discussed in detail in Appendix A.

In brief, the rating system provided a means of selecting seal designs that warrant further development effort. Candidate designs were rated on the following factors deemed essential to attain program objectives.

1) Sealing Mechanism - Seal conforms to rod by means of a feasible loading technique. Appropriate stress levels and patterns are generated in the seal element throughout the temperature and pressure range. Geometry and behavior of elements can be predicted and controlled within workable limits. Effective static sealing is available. Pressure does not increase leakage.

2) Wear and Compensation - Seal exhibits either a low wear rate or the ability to compensate for wear, and wears rod surface at a low rate. Wear process is not detrimental to overall performance of seal installation (i.e., no adverse effect on static sealing). Pressure does not increase wear.

3) Reliability - Seal fails slowly, observable over a period of time. Rugged construction, inherently resistant to abuse in use. Redundant sealing elements. Precise definition of operating conditions and behavior is not critical; possesses inherent latitude to withstand pressure surges.

4) Short-term Potential - Knowledge of concept is available or can be acquired in the near future. Configuration can be optimized and evaluated within time period of program.

Seal designs that achieve a high score on the above factors are then rated on the basis of other factors, considered desirable but not absolutely essential, to aid in the final selection. These desirable factors are:

1) Design Features - Materials exhibit compatible coefficients of expansion. Rod does not require exotic plating. Friction is relatively low (for good servo performance). Seal has a relatively high degree of self-compensation for wear. Geometry lends to pressure balancing or pressure relief.

2) Cost - Relatively easy to manufacture (reasonable tolerance requirements, accessible geometry for machining). Materials are available and machineable.

3) Serviceability - Easy to install and remove. Does not require custom tailoring; no special installation tools required.

4) Past experience with similar approach.

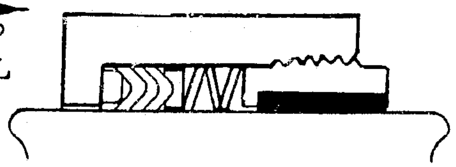
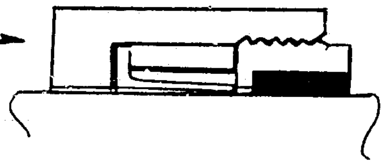
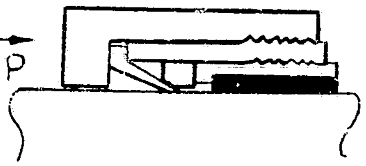
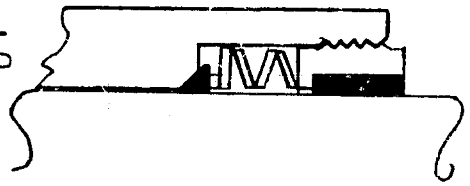
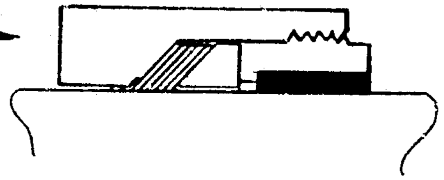
Approximately 17 designs (see Appendix A) were evaluated with the rating system. The five seal designs selected (plus one alternate design) and their applicable materials are shown in Table 3-1. Of the designs selected, the following were approved by NASA for further development.

1) Design B, V-seal with polyimide
2) Design I, wedge seal with nickel Foametals impregnated with a CaF_2BaF_2 eutectic.
3) Design AH, multiple reed seal with Vascojet 1000/silver - copper alloy
4) Design D, lip seal with cobalt molybdenum alloy
5) Design D, lip seal with Vascojet 1000

The designs selected represented a broad coverage of the various available seal concepts and material. It was recognized that other seal-material combinations may also have offered considerable potential. However, it was not feasible to evaluate all possible combinations within the scope of the program. The final selection of configurations was influenced by past and recent experience and was considered to be representative of the best features of the many designs reviewed.

TABLE 3-1

RECOMMENDED SEAL DESIGNS AND MATERIALS

SEAL DESIGN	SEAL TYPE	RECOMMENDED MATERIALS	ALTERNATE MATERIALS
B	P V-SEAL 	Polymer SP	Nickel Foametal with Ca F_2 + Ba F_2
D	P LIP-SEAL 	Vascojet-1000	
D	SAME	Cobalt-molybdenum alloy (75% Co + 25% Mo)	
F	P LIP-SEAL 	Silver alloy (72% Ag + 28% Cu)	Nickel Foametal with Ca F_2 + Ba F_2, Polymer SP
I	P WEDGE-SEAL 	Nickel Foametal with Ca F_2 + Ba F_2	Silver alloy (72% Ag + 28% Cu)
(ALTERNATE) H or AH	P REED-SEAL 	Vascojet - 1000 and silver alloy (72% Ag + 28% Cu) combination	Cobalt-molybdenum alloy and silver alloy (72% Ag + 28% Cu) combination, Polymer SP

SECTION IV

SEAL DESIGN AND DEVELOPMENT

A. GENERAL

Detail design of the candidate seal concepts (shown in Table 3-1) were accomplished in this phase. Preliminary testing was conducted where necessary to substantiate design assumptions. Tests were also performed to determine the effects of lateral motion (side load) due to bearing wear on the candidate seal configuration. The ability of the seals to operate at over-pressure conditions in the event of first-stage seal failure was also evaluated. In addition, limited testing was also performed on several first-stage high-pressure seal configurations to determine the most suitable design.

B. TEST EQUIPMENT AND PROCEDURES

1. Friction Measurement and Seal Loading Apparatus

The combined friction measurement and seal loading apparatus shown in Figure 4-1 was used to obtain experimental data to aid in the design of the seal and the seal-loading mechanisms. Such data included the normal force required at the interface to effect a seal at various pressures and the resultant friction. As shown in Figure 4-1, the test setup consists of a seal tester, a loading sleeve, and a loading cylinder. The seal tester was assembled with a rod seal on each end. Only one seal was subjected to test.

The testing procedure consists of pressurizing the seals to the desired pressure and then gradually loading the seal by applying pressure to the loading cylinder until zero leakage is achieved. The product of the pressure and the effective area of the cylinder piston determines the load applied axially to the seal. Seal friction is measured with a spring scale. The normal sealing force and the resultant contact pressure at the seal interface were then approximated on the basis of the friction force obtained, the friction coefficient, and the seal geometry.

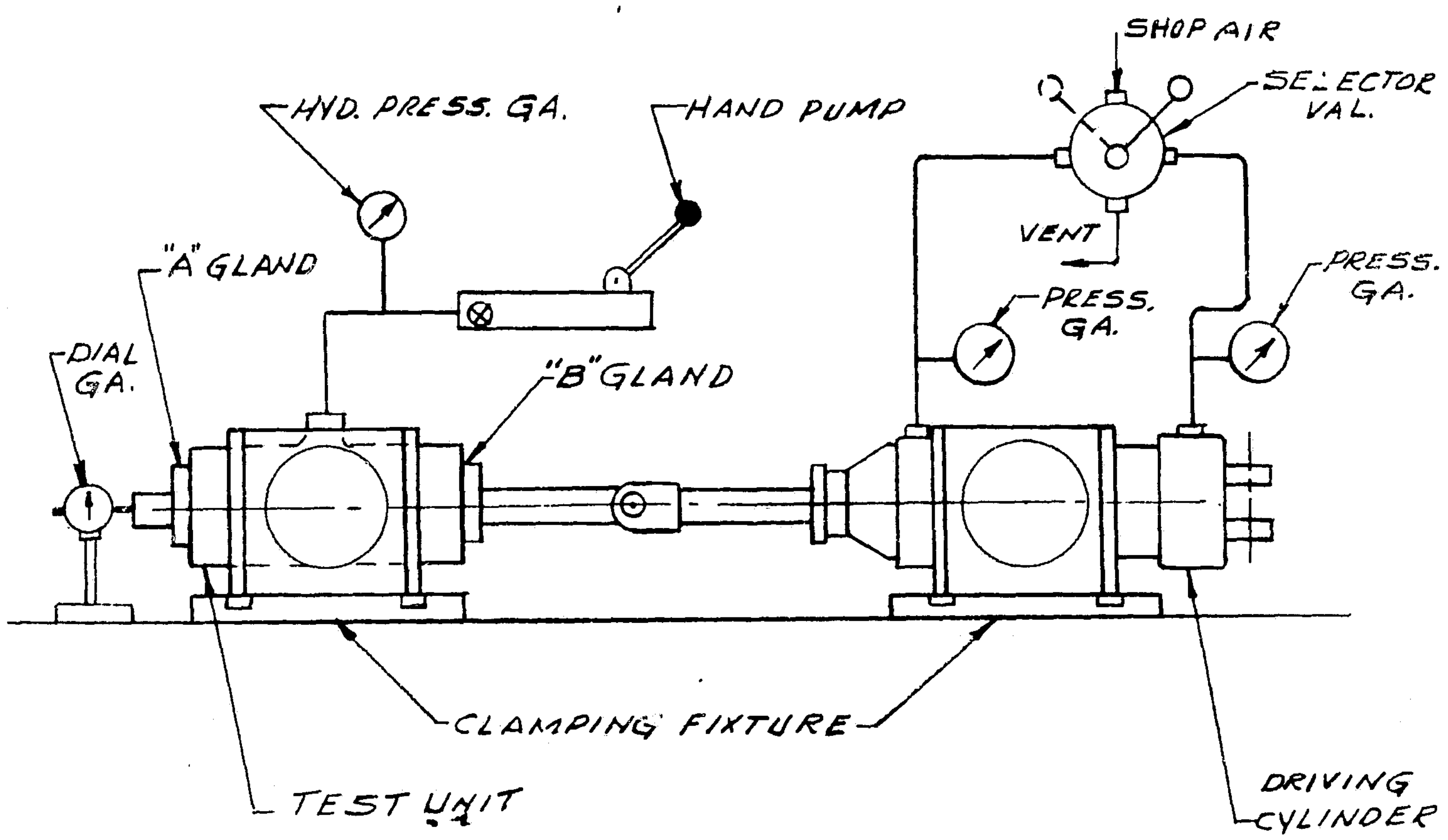


Figure 4-1. Seal Friction Setup

2. Preliminary Seal Test Rig

An existing test rig was modified to accomplish the preliminary evaluation of seal designs developed during the program. The test rig as shown in Figures 4-2 and 4-3 consists of a seal tester, a 3-horsepower variable-speed motor, and a hydraulic power source. The hydraulic power source comprises a power circuit and test circuit separated by a barrier cylinder, which also serves as a pressure booster. Although this arrangement was not a strict simulation of any aircraft system, it offers the following distinct advantages for laboratory testing:

- The power circuit can be operated at ambient temperatures, permitting the use of relatively inexpensive components and hydraulic fluid
- Packing changes in the power system are not required in the event of changing test fluids
- Total test fluid is limited to a small volume

The rod seal tester (Figure 4-4) consists of a cylinder that houses a seal gland on each end. The cylinder and seal glands are constructed of 17-4PH corrosion-resistant steel. The chrome-plated piston rod was fabricated of Type 440C stainless steel. The design of the seal glands permits their adaptation to various seal configurations with only minor modification. Graphite bearings are incorporated in each seal gland to prevent piston rod surface damage due to metal-to-metal contact with the gland shoulder. Premature failure of a seal due to piston rod scoring was thus minimized.

C. SECOND-STAGE LOW-PRESSURE SEALS

1. Design B - Polyimide V-Seal

Detail design of this configuration is shown in Figures 4-5, 4-6, and 4-7. Sample seals were fabricated and load-deflection tests were conducted to determine the ability of the seal to compensate for wear. These data was used in the design of the loading springs. The test setup, depicted in Figure 4-8 consists of a seal gland, dial indicator, and a loading cylinder. A single V-seal was assembled in the gland with a load ring and backup ring. The dial indicator, which measures the lip deflection, was placed at the outer edge of the lip. Axial

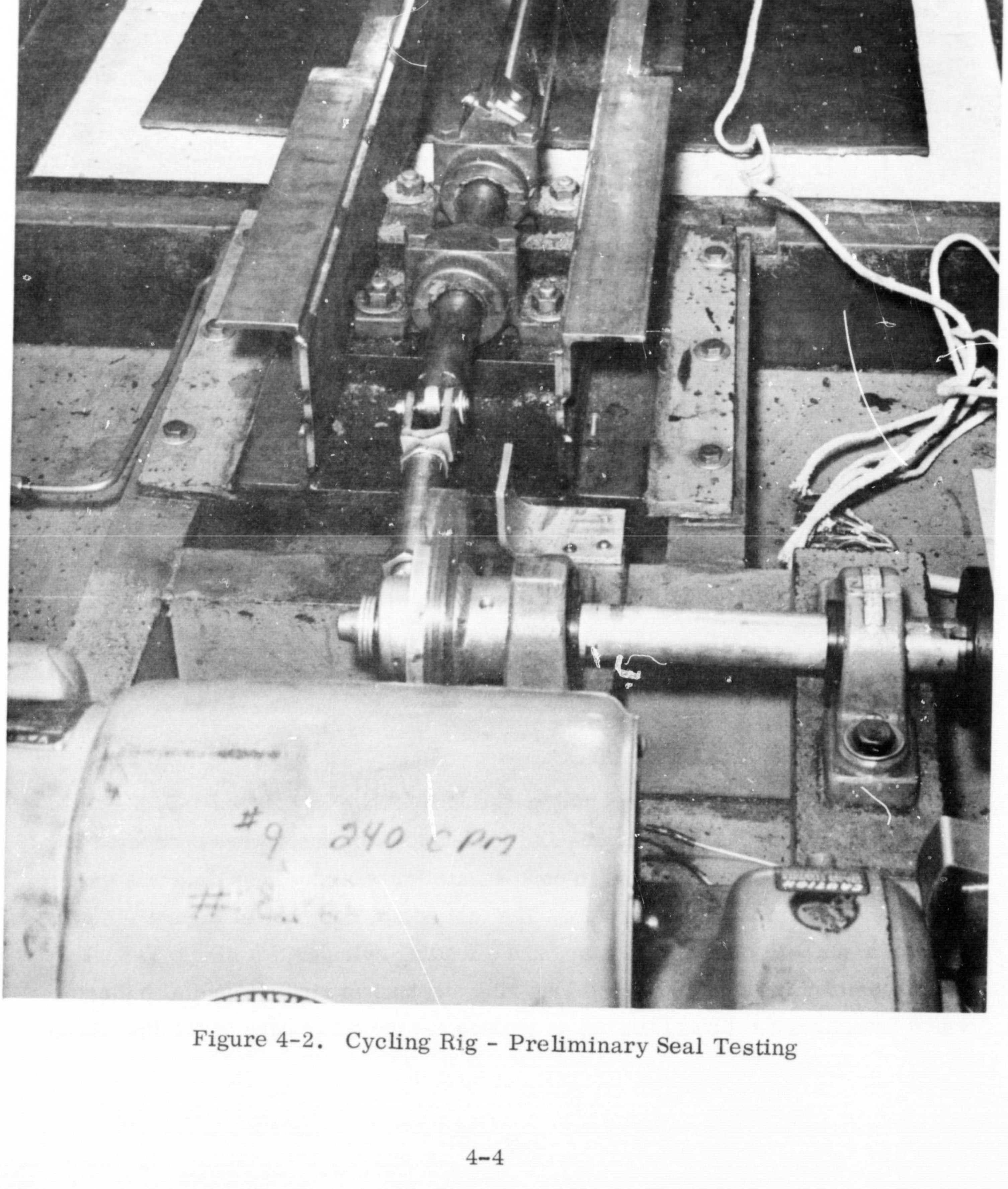


Figure 4-2. Cycling Rig - Preliminary Seal Testing

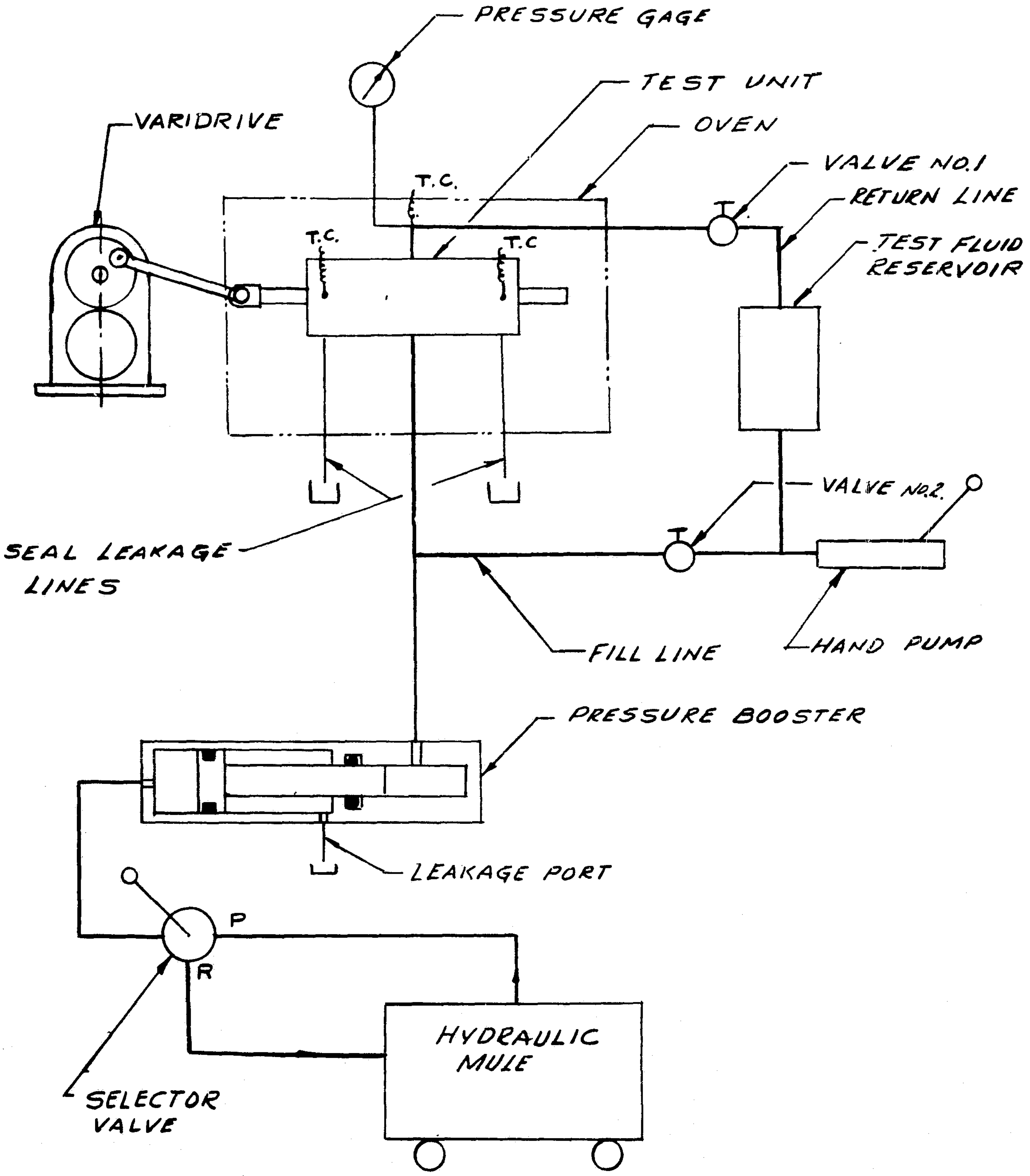


Figure 4-3. Schematic - Cycling Rig - Preliminary Seal Testing

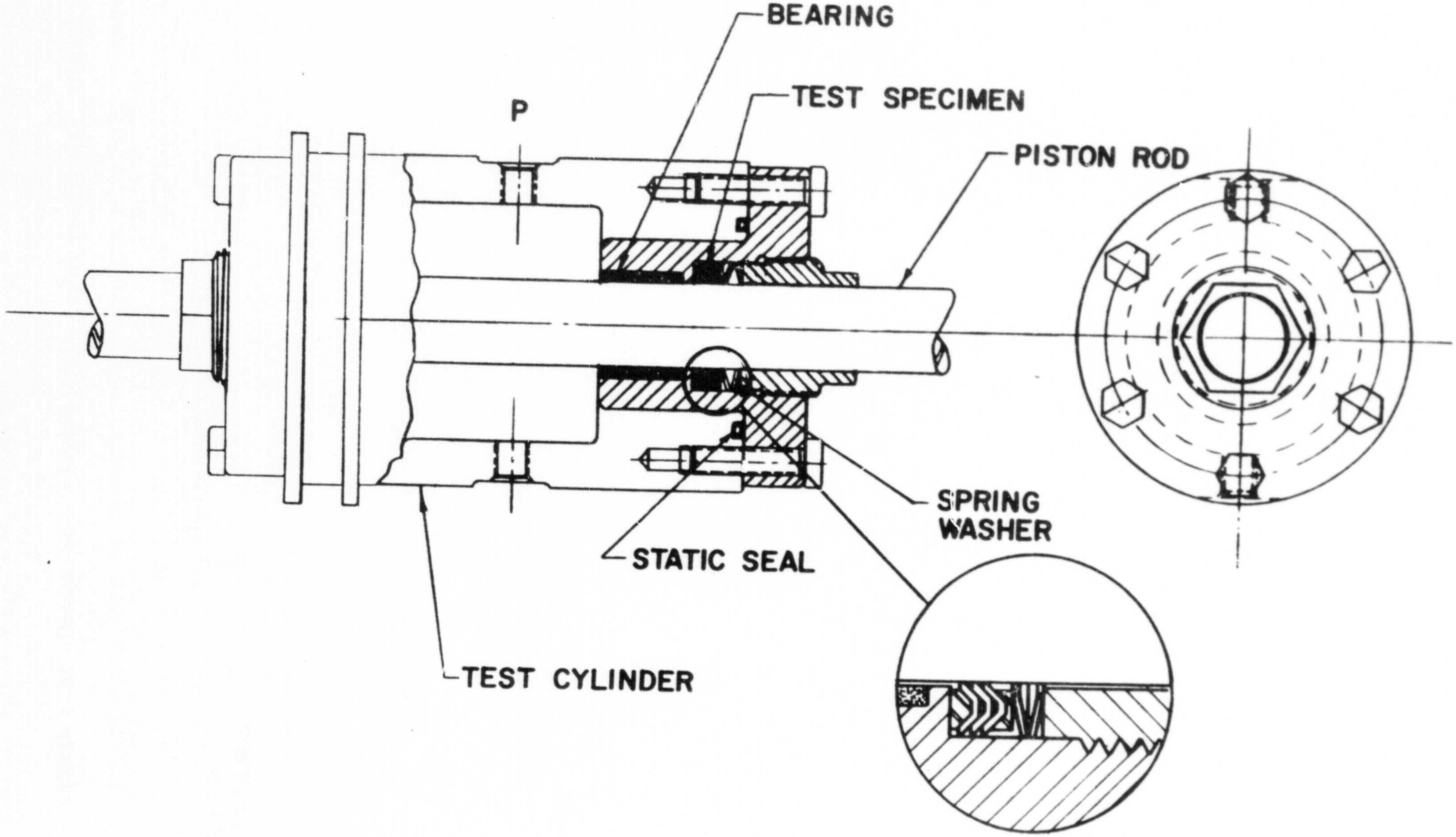


Figure 4-4. Rod Seal Test Unit

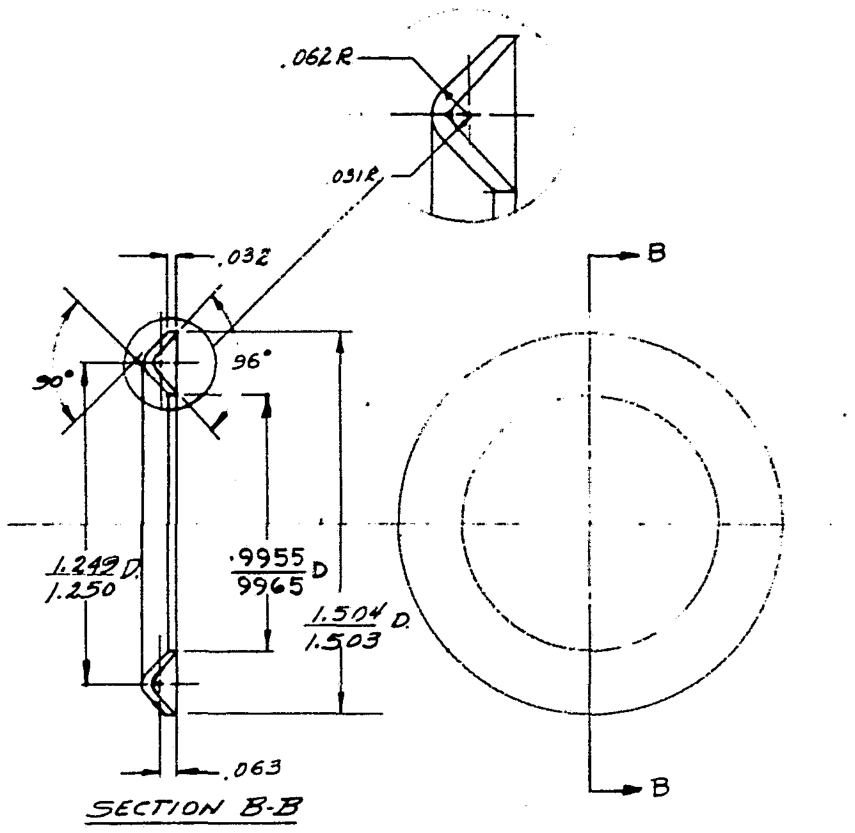


Figure 4-5. Polyimide V-Seal

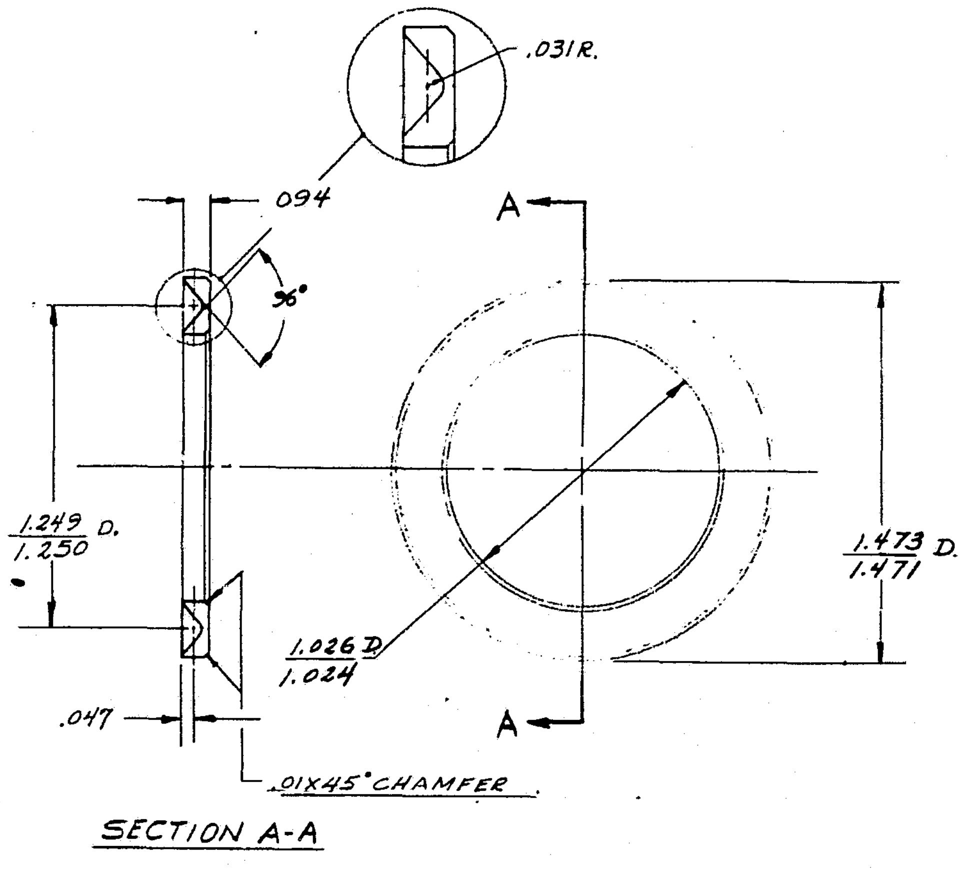


Figure 4-6. Backup Ring - 321 Stainless Steel

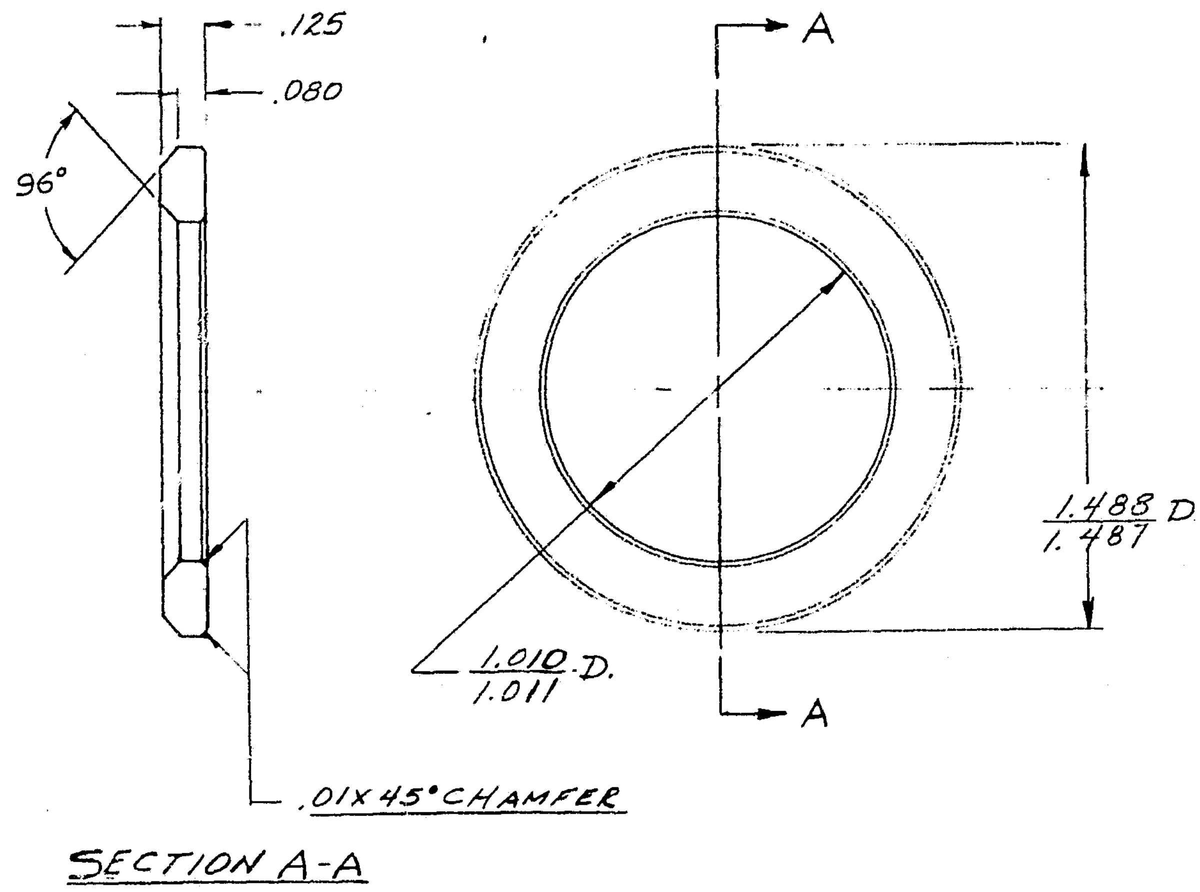


Figure 4-7. Load Ring - 321 Stainless Steel

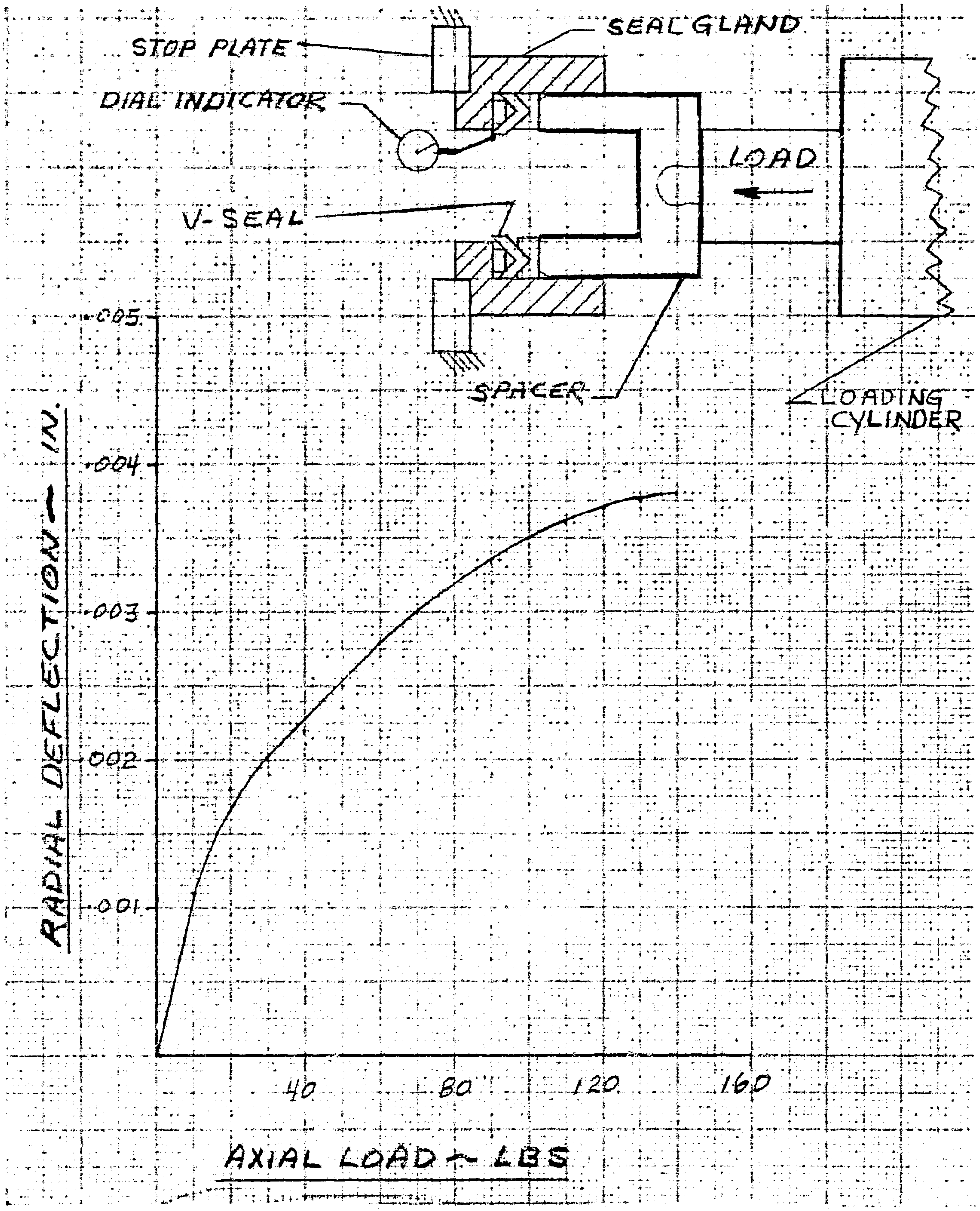


Figure 4-8. Polyimide V-Seal Test Radial Deflection versus Axial Loading

loading was applied to the seal by the loading cylinder. Data obtained (Figure 4-8) indicate that a radial deflection of 0.0038 inch was obtained with an axial load of 140 pounds. As multiple seal elements were to be used in the final configuration, additional tests were conducted using the loading springs to determine lip deflection with the seals stacked. Seal deflections obtained under these conditions (Figure 4-9) were 0.0075 to 0.0085 inch with a spring load of 200 pounds.

The effects of bearing wear and lateral motion on this seal were simulated with the test setup shown in Figure 4-10. Three V-seals were assembled on each end of the fixture. The rod bearings, which were machined with sufficient clearance to simulate wear, were placed inboard of the seal. Side motion on the rod was induced by use of nylon screws fitted on the end plugs. A dial indicator was placed on the piston rod (4.230 inches from the contact point of the seal) to measure the lateral displacement of the rod. The seal was pressurized to 100 psi with benzene used as the test fluid. Benzene was selected because of its low viscosity (0.7 centistrokes at room temperature), which is similar to the viscosity of a hydraulic fluid at 500°F to 600°F.

In the above side-load test, the piston rod was displaced laterally at 0.001-inch increments. No seal leakage was observed with lateral deflections up to 0.011 inch (which corresponds to a deflection of 0.0055 inch at the seal). At this point the rod bottomed on the gland bearing and testing was concluded. The results indicate that the V-seal can withstand lateral deflections of at least 0.0055 inch. This ability is attributable to the flexibility of the seal configuration and the Polyimide material. In addition, the ability of the seal to achieve a fairly high initial deflection (squeeze) against the piston rod enables it to remain in contact with the rod to provide an effective seal under the side-load condition.

At the completion of the lateral deflection test, the V-seals were tested to determine their ability to withstand high pressures in the event of first-stage seal failure. The seals were tested in the fixture shown in Figure 4-10, at pressures up to 3500 psi, with the F-50 silicone fluid. Although no leakage was observed at pressures up to 3500 psi, higher pressures had a considerable

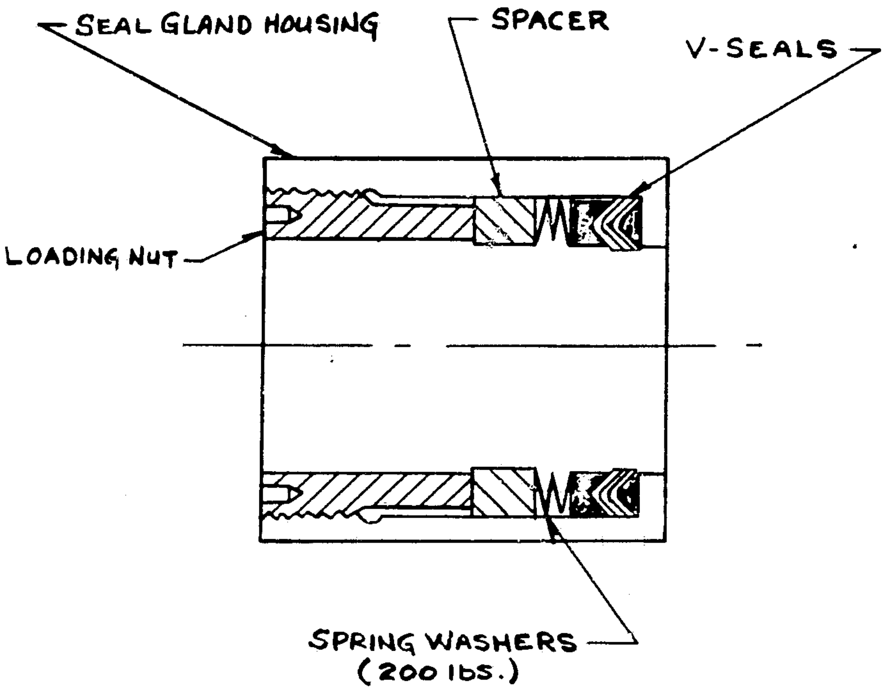


Figure 4-9. Test Gland - Seal Deflection Test

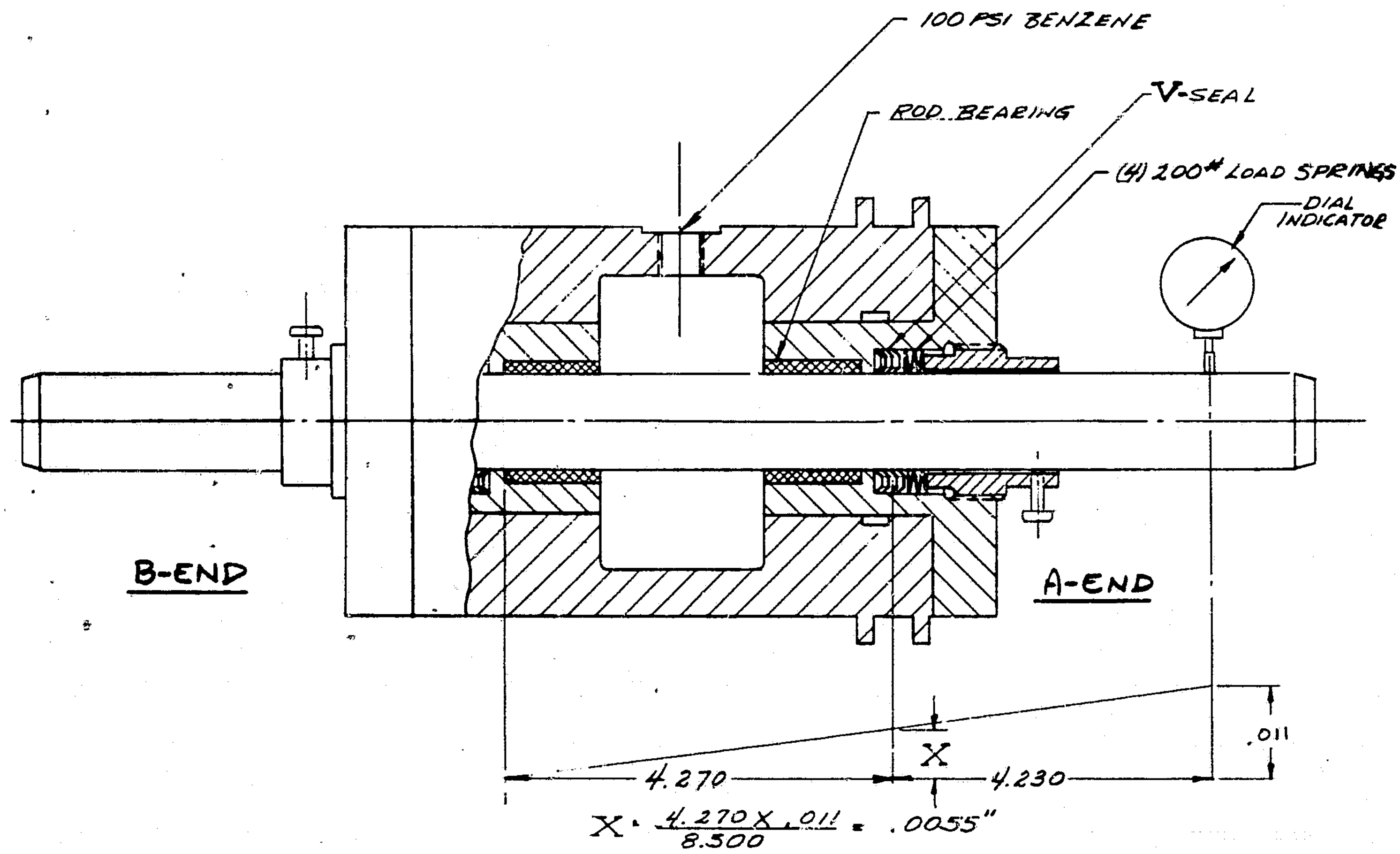


Figure 4-10. Polyimide V-Seal - Radial Deflection Versus Leakage Setup

effect on the seal breakout friction. As shown in Figure 4-11, total friction for two sets of seals increased from 160 pounds at 100 psi to 465 pounds at 3500 psi. However, in an actual application, this high friction condition resulting from failure of the first-stage seal would exist for a relatively short period. Presumably, the failure of the first-stage seal would be detected and the seal replaced. At the completion of the test, the seals were disassembled and inspected for damage. No evidence of cracking or permanent deformation of the seals was noted.

2. Design D-Vascojet 1000 Lip Seal

Several variations of this design (Figure 3-6) were evaluated to determine friction and leakage, and wear under dynamic cycling conditions. The ability of the seal to withstand side load (due to bearing wear) and over-pressure conditions were also investigated. These tests were conducted in the test unit shown in Figure 4-4.

The first set of seals was designed to provide a seal contact load of approximately 50 pounds per inch of circumference with lip deflection (radial) of 0.0019 inch when stretched over the piston. This was accomplished with a seal lip sectional thickness of 0.009 inch. However, the actual radial deflection of one of the seals (A-seal) was approximately 0.0015 inch due to dimensional changes resulting from the heat treating process. This resulted in a somewhat lower seal contact load for the A-seal. The B-seal was within design tolerance.

The second set of seals was configured to provide a seal contact load of 25 pounds per inch of circumference for the same (0.0019 inch) radial deflection. For this configuration, the lip sectional thickness was approximately 0.0055 inch. Actual seal contact loads for both seals were slightly less than the design values due to build up of machining tolerances.

Friction and leakage characteristics of the seals were determined in the test setup shown in Figure 4-1 using the seal test unit shown in Figure 4-4. Each set of seals was tested separately. Figure 4-12 depicts the leakage and friction characteristics of the two seal designs under fluid pressurization.

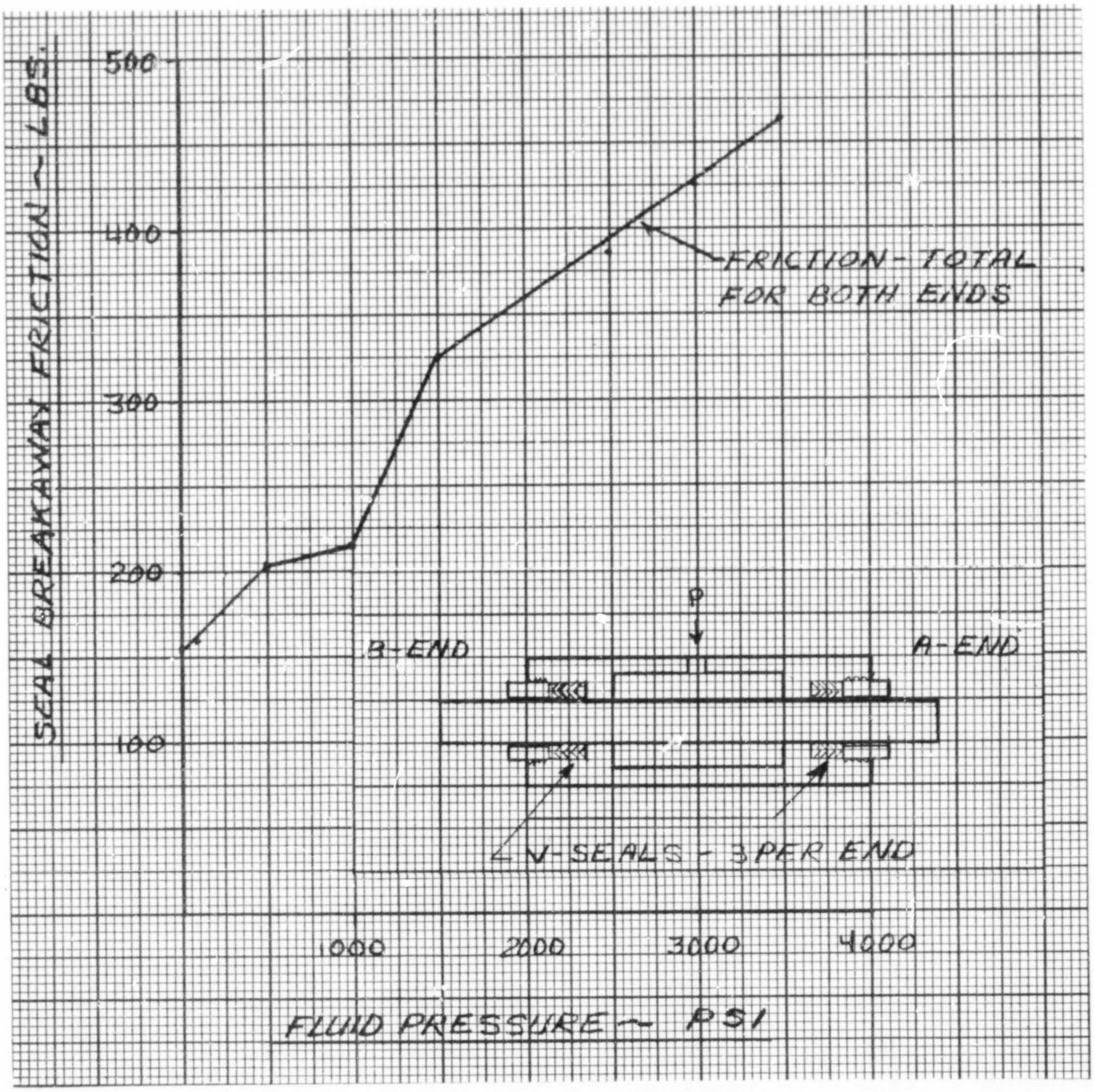


Figure 4-11. Polyimide V-Seal - Seal Breakaway Friction versus Pressure (F-50 Fluid at Room Temperature)

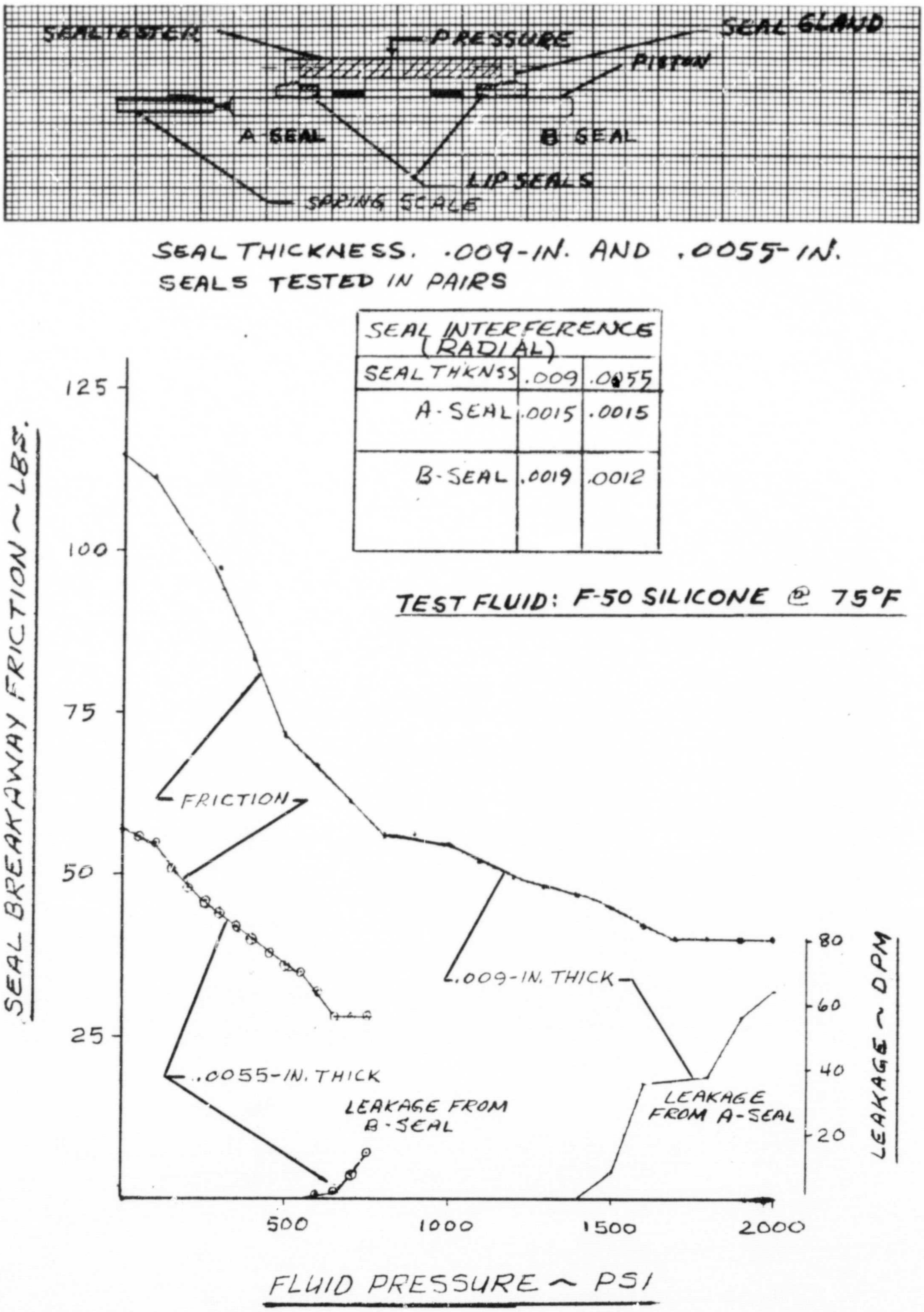


Figure 4-12. Seal Friction and Leakage versus Pressure, Vascojet 1000 Lip Seal - 0.0055 Inch Seal Thickness

Initial breakaway friction for the second set of seals (lower contact load) was approximately 50% lower than the first set of seals, which had a higher seal contact load. Both configurations exhibited a rapid decrease in seal friction when subjected to increasing fluid pressure. For the first set of seals, the friction leveled off to a constant value of approximately 40 pounds between 1500 and 2000 psi fluid pressure. Part of the friction indicated for this pressure range may be attributed to friction generated by the fluid film between the close fitting graphite bearings and the piston rod. It was also noticed that, at the point where friction leveled off, leakage was observed from the "A" seal (which had a lower interference fit). Leakage started at approximately 8 drops per minute at 1500 psi and increased to 75 drops per minute at 2000 psi.

Seal breakaway friction for the second set of seals leveled off to a constant value of approximately 30 pounds at a fluid pressure of 600 to 800 psi. This was expected because these seals were designed to a much lower contact load. Leakage of 1/2 drop per minute from the B-seal was first noticed at 600 psi and increased to 14 drops per minute at 800 psi.

The above results demonstrated the feasibility of this approach to the design of a hard metal seal. The leakage and friction data indicate that the seal contact load can be relieved by the pressure of the working fluid, thus minimizing wear. For this approach, the second set of seals appear to be the most suitable. However, it would not be able to withstand transient high pressures (surges) normally experienced by system return lines. Therefore, the first set of seals was considered for the program as a practical compromise.

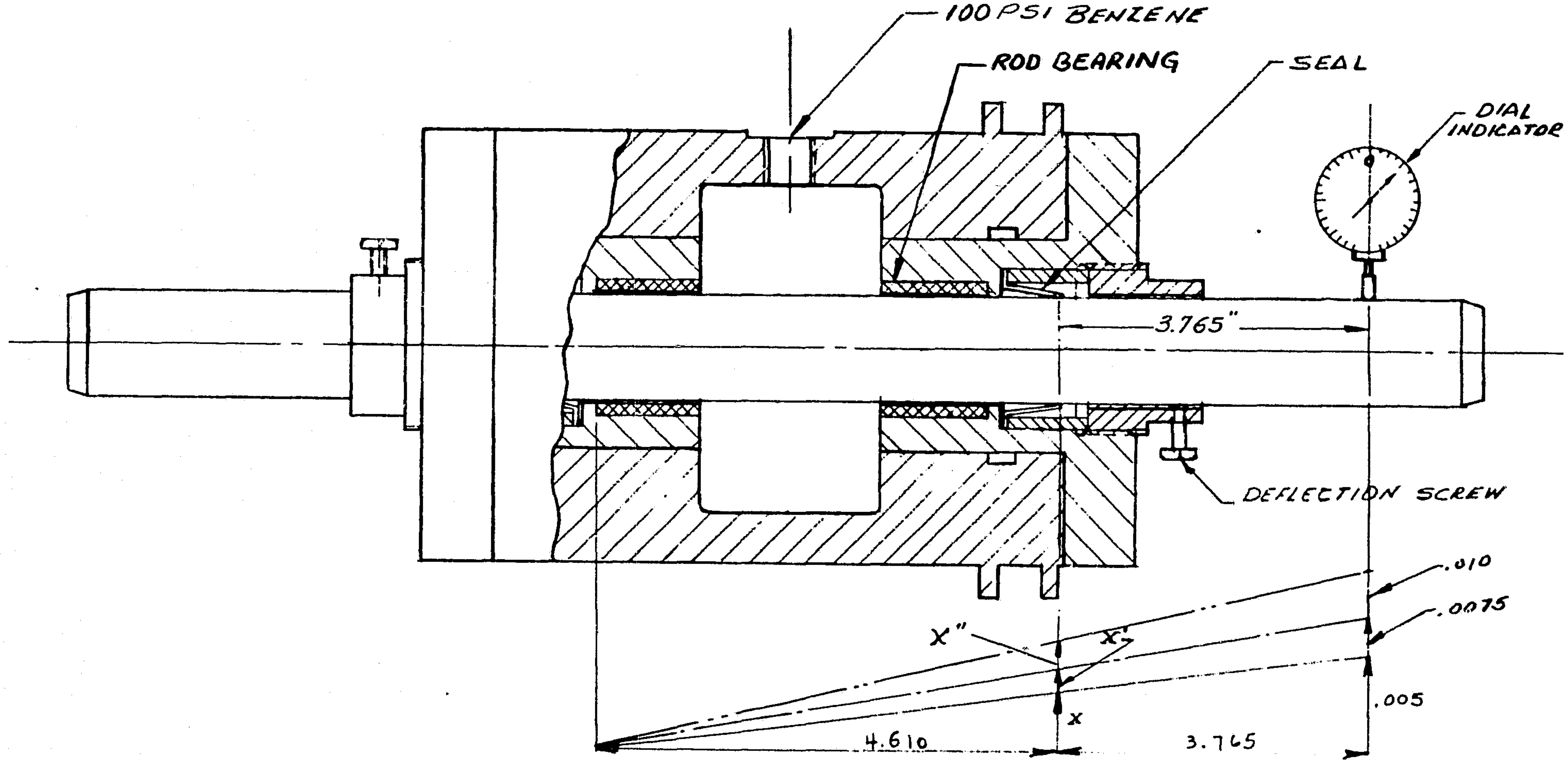
Upon completion of the above investigation, the "A" seal from the first set was subjected to mechanical cycling in the rig shown in Figure 4-2, at room temperature and at 400°F to determine the behavior of the Vascojet 1000 material sliding on chrome plate and the leakage characteristics of the seal under dynamic conditions. The "B" seal was removed from the seal test unit, replaced with an O-ring seal, and retained for future investigations. During the cycling, the seals were pressurized to 100 psi. The piston rod was cycled at a rate of 40 cpm with a stroke of 1.5 inches. Testing was discontinued after

15.5 hours (34,200 cycles) cycling. No leakage was observed during 7.5 hours of room temperature cycling. However, light burnishing marks were observed on the chrome plated surface of the piston rod. During 2.75 hours of cycling at 400°F, total leakage of approximately 0.25 cc was collected. This leakage represents the film of fluid that was carried out by the piston rod.

Investigations were conducted to determine the effects of bearing wear and side load on the lip seal configuration. These conditions were simulated with the test setup shown in Figure 4-13. The rod bearings were placed inboard of the seals. Side motion on the rod was induced by means of Nylon screws fitted on the end plugs. A dial indicator was placed on the piston rod (3.765 inches from the contact point of the seal) to measure the lateral displacement of the rod. The seal was pressurized to 100 psi using benzene as the test fluid.

The above tests were conducted on three different lip seal configurations. Seals Nos. 1 and 2 were designed with a lip thickness of 0.009 and 0.0055 inch, respectively. Seal No. 3 also had a lip thickness of 0.009 inch, but the thickness of the seal flange was decreased from an original thickness of 0.010 inch to simulate wear and the graphite rod bearings were machined to provide a diametral clearance (between the rod and the bearings) of 0.006 inch. To simulate a condition wherein wear has taken place on the seals as well as the bearings, Seals Nos. 1 and 3 were designed to provide an interference fit of 0.0012 inch. Seal No. 2 was designed to an interference fit of 0.0015 inch. (Design interference is 0.0038 to 0.004 inch.)

Of the three designs investigated, Seal No. 3 (modified flange) was the most acceptable. As shown in Figure 4-14, Seal No. 3 exhibited the greatest flexibility when subjected to side loading. This seal withstood a lateral deflection of 0.0055 inch with no leakage. This amount of deflection was equivalent to a rod deflection of 0.010 inch. The maximum amount of lateral deflection that can be tolerated by this seal was not determined because the rod had already bottomed on the gland bearing.



BEARING CLEARANCE FOR .009" TK SEAL = .0025"

BEARING CLEARANCE FOR .0055" TK. SEAL = .004"

$$\frac{4.610 \times .005}{8.375} = X = .00275 \text{ (FOR .009 TK. SEAL)}$$

$$\frac{4.610 \times .0075}{8.375} = X' = .00413 \text{ (FOR .0055 TK. SEAL)}$$

$$\frac{4.610 \times .0075}{8.375} = X'' = .0055 \text{ (FOR MODIFIED FLANGE)}$$

Figure 4-13. Metallic Seal - Radial Deflection versus Leakage Test Setup

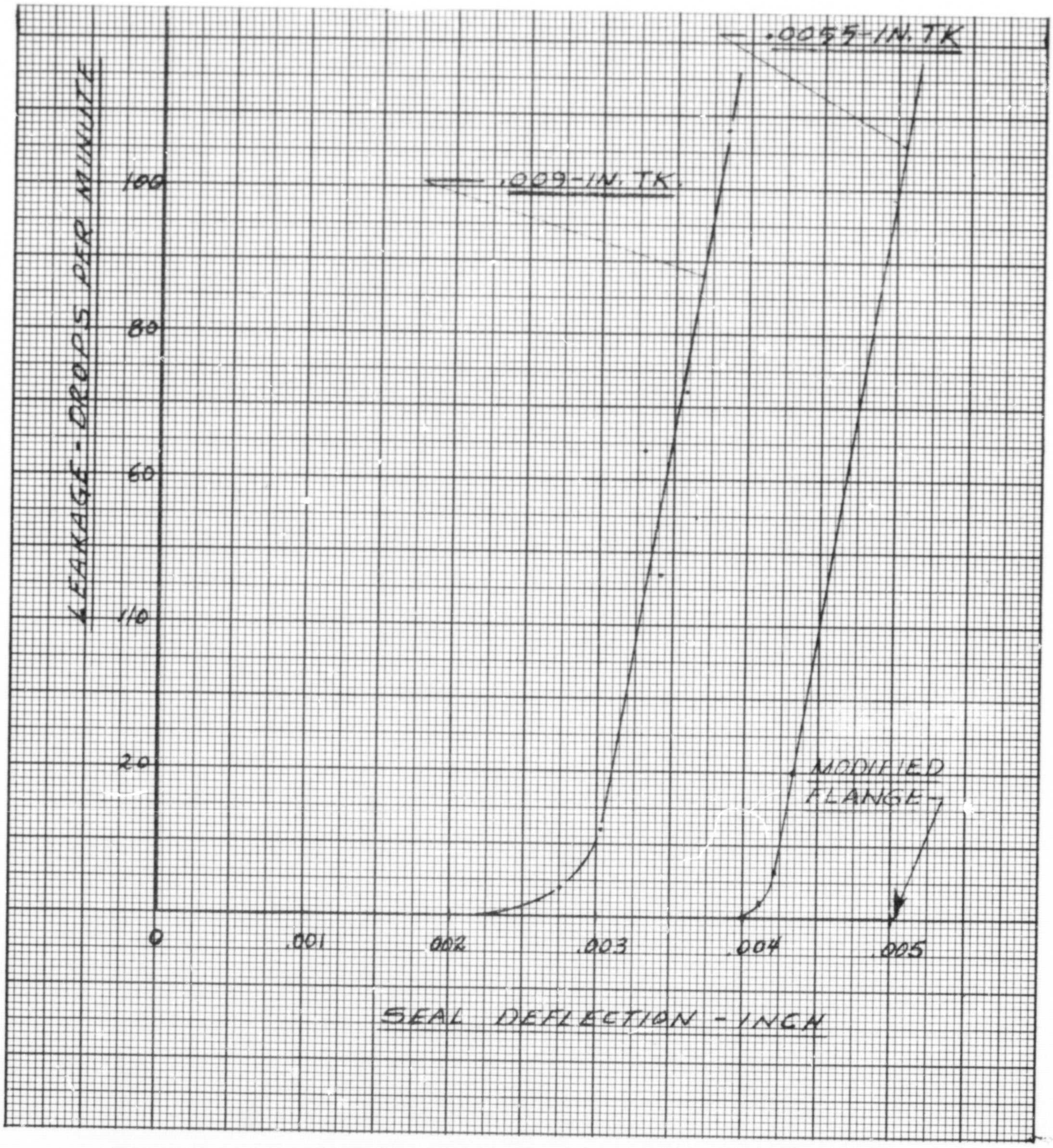


Figure 4-14. Design D Vascojet 1000 - Leakage versus Seal Deflection

For Seal No. 1 (0.009 inch) a leakage of 2 drops per minute was first observed when the seal deflected 0.00275 inch. Leakage on Seal No. 2 (0.0055 inch) did not occur until a deflection at the seal of 0.0041 inch was induced. Both seals exhibited a rapid increase in leakage when subjected to increasing deflection.

Additional modifications to the lip seal design to enable it to withstand greater side motion were also considered. Figure 4-15 depicts a design which is similar to a ball joint. The design incorporates a flange nestled in a mating seat. The seat also acts as a static seal. A spring washer provides the seating force at zero pressure conditions. Nickel Foametal impregnated with CAF_2 and BaF_2 was selected as the seat material because of the potential lubricating ability of the impregnant and good conformability of the relatively soft nickel matrix. Figure 4-16 depicts the position of the seal under exaggerated side motion. Another configuration based onthe same approach is shown in Figure 4-17. However, this design is somewhat complex because of the separate static and dynamic seals required. The foregoing concepts offer potential improvements to the lip seals. However, considerable effort and time would have been required to develop these approaches. Consequently the simpler approach (modified flange) was undertaken to improve the tracking ability of the seal.

As recommended by NASA, the lip seal design was further reviewed to determine what possible means can be incorporated into the seal configuration to provide a fail-safe backup for high-pressure conditions. As the seal was designed for optimum low pressure performance, its ability to withstand high pressures is limited. Under normal operating conditions, the seal would not be subjected to high pressures. Even with a catastrophic failure of the first-stage high-pressure seal, pressure would build up against the second-stage seal only under conditions of high inflow to the actuator that reached the flow limits of the downstream lines.

However, it is recognized that it may be desirable to build safety features into the design of certain high performance actuators so that a cata-

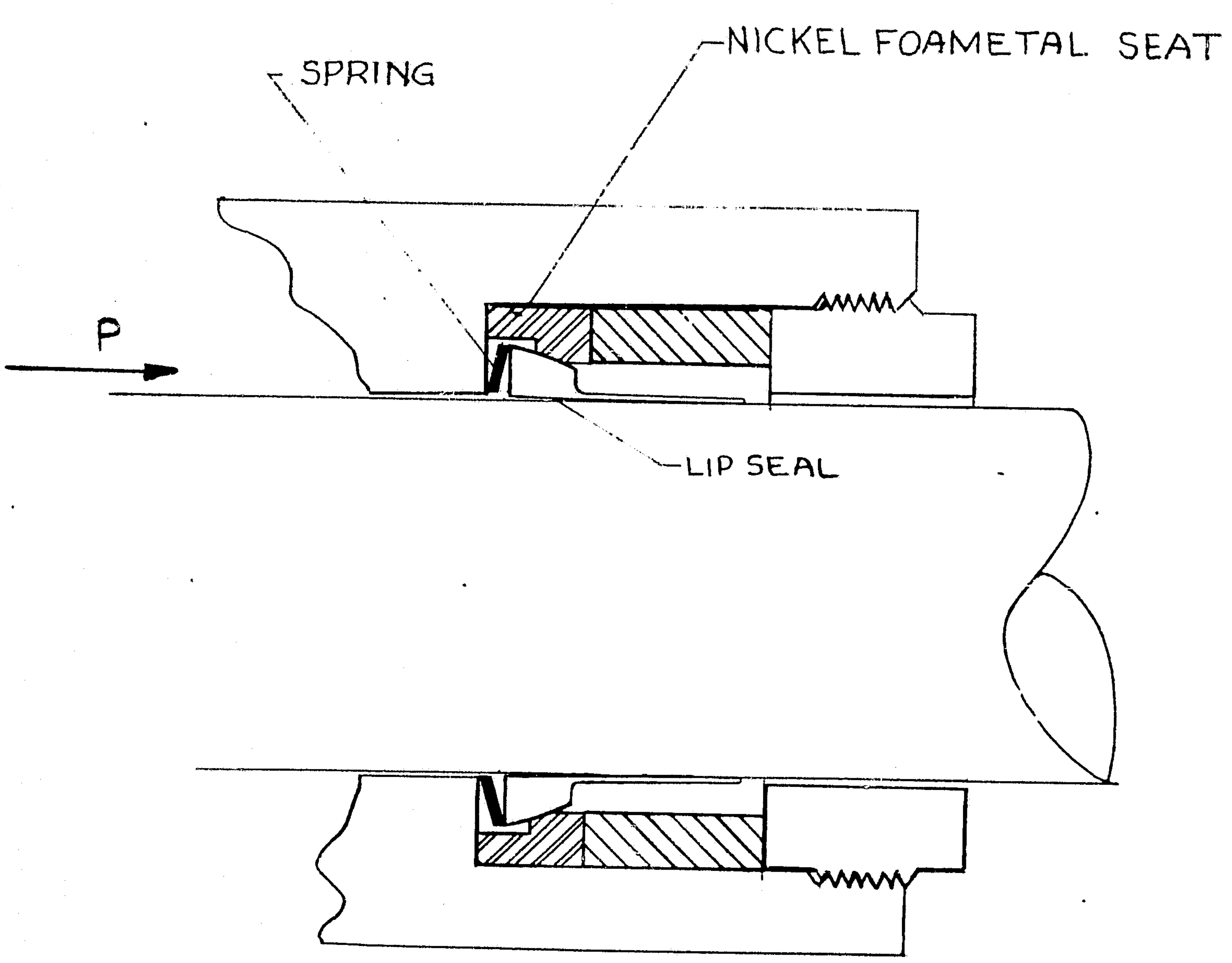


Figure 4-15. Modified Metal Lip Seal

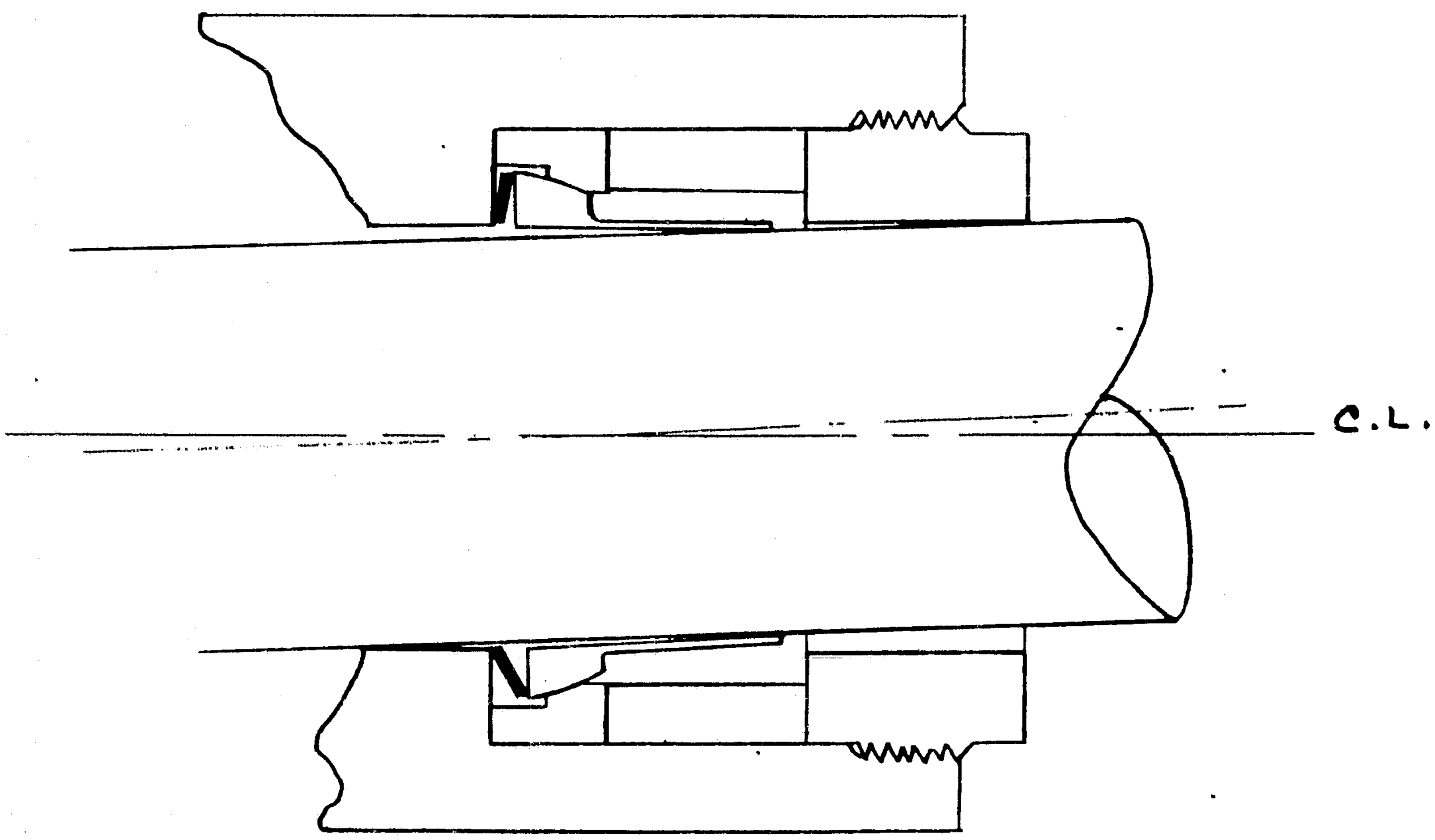


Figure 4-16. Modified Metal Lip Seal With Side Load

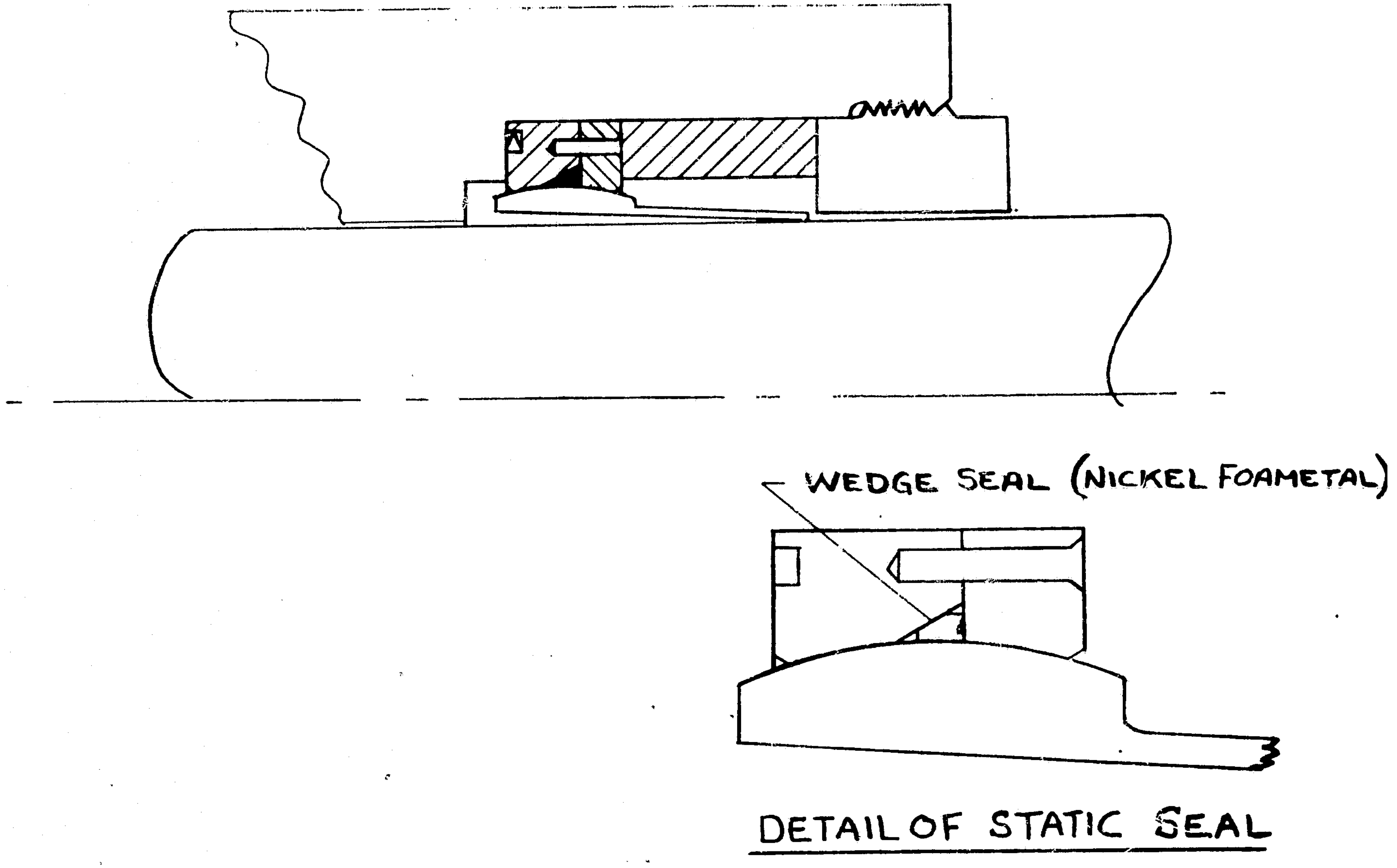


Figure 4-17. Modified Metal Lip Seal

strophic failure of the first-stage seal does not render the component inoperative. Such a feature generally consists of a restrictor-check valve in the return port between the first- and second-stage seal. (The check valve prevents reverse flow in the event of second-stage seal failure.) The restrictor limits the internal leakage flow resulting from first-stage seal failure so that pressure can be maintained in the actuator. Under these conditions, the second-stage seal would be required to withstand practically full system pressure.

The critical part of the present lip seal design is at the contact area. This portion of the seal is stressed to approximately 110,000 psi (hoop stress) due to a radial stretch of 0.0019 inch (interference fit) over the piston rod. An increase in fluid pressure would increase the induced hoop stress at this point. To prevent the seal from being over-stressed, a backup device would be required to limit the deflection due to pressure. However, in order not to limit the built-in flexibility of the seal, the clearance between the backup and the seal must be closely controlled.

Based on the assumption that the seal can withstand a maximum hoop stress of 160,000 psi (tensile yield strength of Vascojet 1000 at 600°F = 183,000 psi), then the allowable increase in hoop stress due to fluid pressure is:

$$160,000 - 110,000 = 50,000 \text{ psi}$$

The fluid pressure required to produce a hoop stress of 50,000 psi is:

$$P = \frac{St}{R} = \frac{50,000 \times .0055}{.5} = 550 \text{ psi}$$

Radial deflection of the sealing lip (from the assembled position) due to a fluid pressure of 550 psi is:

$$R.D. = \frac{R}{E}(S_2 - vS_1)^*$$

---

* "Formulas For Stress and Strain," First Edition, R.J. Roark, p 227

where

R = mean radius of seal

E = modulus elasticity (for Vascojet 1000, $E = 23.9 \times 10^6$ at 600°F)

S_2 = hoop stress ($S_2 = \frac{PR}{t}$)

S_1 = meridional membrane stress ($S_1 = \frac{PR}{2t}$)

v = Poissons ratio = .26

$$R.D. = \frac{.5}{23.9 \times 10^6} (50,000 - (.26 \times 25,000))$$

$$R.D. = 0.000908 \text{ inch}$$

Based on this analysis, the sealing lip can expand 0.0009 inch radially without being overstressed. Beyond this point, the seal will have to be backed up. In order not to inhibit the seal during low pressure operation or in its ability to follow radial motion, the equivalent clearance will have to be maintained between the seal and backup. Such a clearance would be extremely difficult to machine and maintain upon assembly.

An alternate approach to protecting the low-pressure second-stage seal was to provide a fail-safe backup to the first-stage seal. The backup arrangement for the first-stage seal is shown in Figure 4-18. This configuration incorporates two contracting sealing rings with a vent between them. During normal operation, the upstream seal performs the sealing function. Leakage past this seal is returned to the system return line through the vent. The downstream sealing ring provides only nominal sealing during normal operation. Wear on this ring will be minimal because it is not energized. The vent is a calibrated orifice which will accept normal seal leakage without pressure buildup. In the event that failure of the first-stage seal occurs, the increased leakage will saturate the orifice, causing pressure to build up and energize the second sealing ring. Thus, pressure is maintained in the cylinder for adequate operation. Although this arrangement was not incorporated in the endurance testing of the lip seal, this approach would provide a fail-safe backup in practice without placing tight performance restrictions on the low-pressure seal.

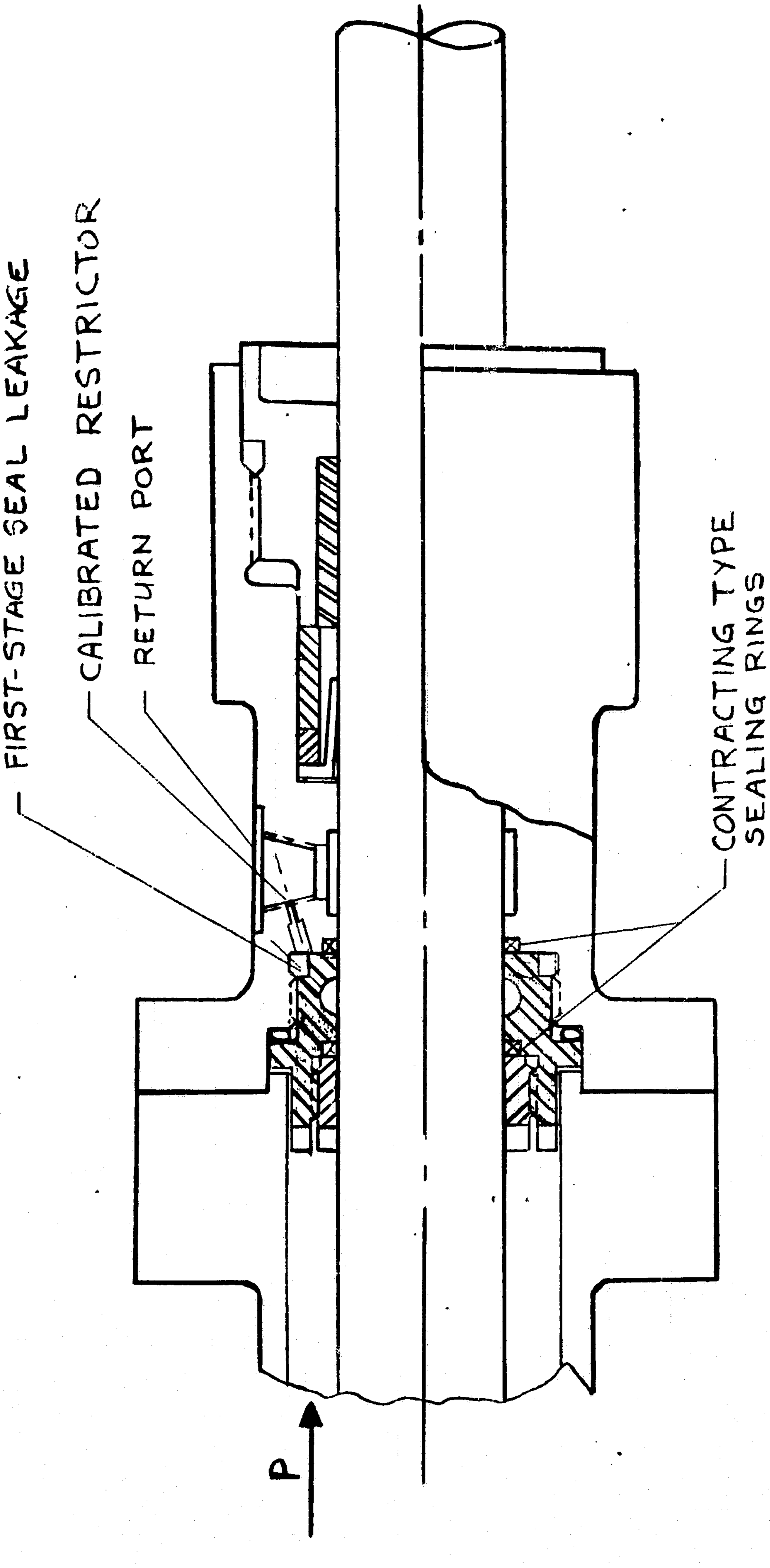


Figure 4-18. Fail-Safe First-Stage Seal

Based on all the preliminary test data, the final design evolved for the Vascojet lip seal is shown in Figure 4-19.

3. Design D - Cobalt-Molybdenum Alloy Lip Seal

As shown in Figure 4-20, this configuration is similar to the Vascojet 1000 lip seal. Because of the lower tensile yield strength (at 600°F) of the cobalt alloy (117,000 psi) as compared to the Vascojet 1000 (183,000 psi at 600°F), slight modifications were made to obtain the comparable deflection (seal interference) and flexibility of the Vascojet 1000 seal. For the seal modification, a seal contact load of 25 pounds per inch of circumference was selected. This load enabled a seal thickness of 0.007 inch to be used. The resultant seal diametral interference was 0.0027 inch as compared to 0.0038 inch for the Vascojet 1000 seal. The lower interference would appear to limit the ability of the seal to compensate for wear. However, this condition may be offset by the use of a lower seal contact load.

Because of the difficulties anticipated in machining the cobalt alloy, which has a nominal composition of 75% cobalt and 25% molybdenum, the seal was initially roughed out by using an Elox machine. The final seal dimensions were obtained by grinding. The finished seal was then heat-aged for 72 hours at 720°F to transform the crystalline structure of the cobalt to a hexagonal form in order to obtain better friction characteristics. Seal measurements taken at the conclusion of the heat aging process showed that the inner diameter of the sealing lip increased by approximately 0.001 inch. As this dimension is critical in controlling the amount of seal interference, it was decided to fabricate another seal for test. The material for the second seal was heat aged prior to machining. However, inclusions in the material were discovered during machining. The inclusions (see Figure 4-21) appeared to be solid pieces of molybdenum that pulled out during machining. Because a long lead time was required to obtain additional material, it was necessary to forego the preliminary testing on that seal.

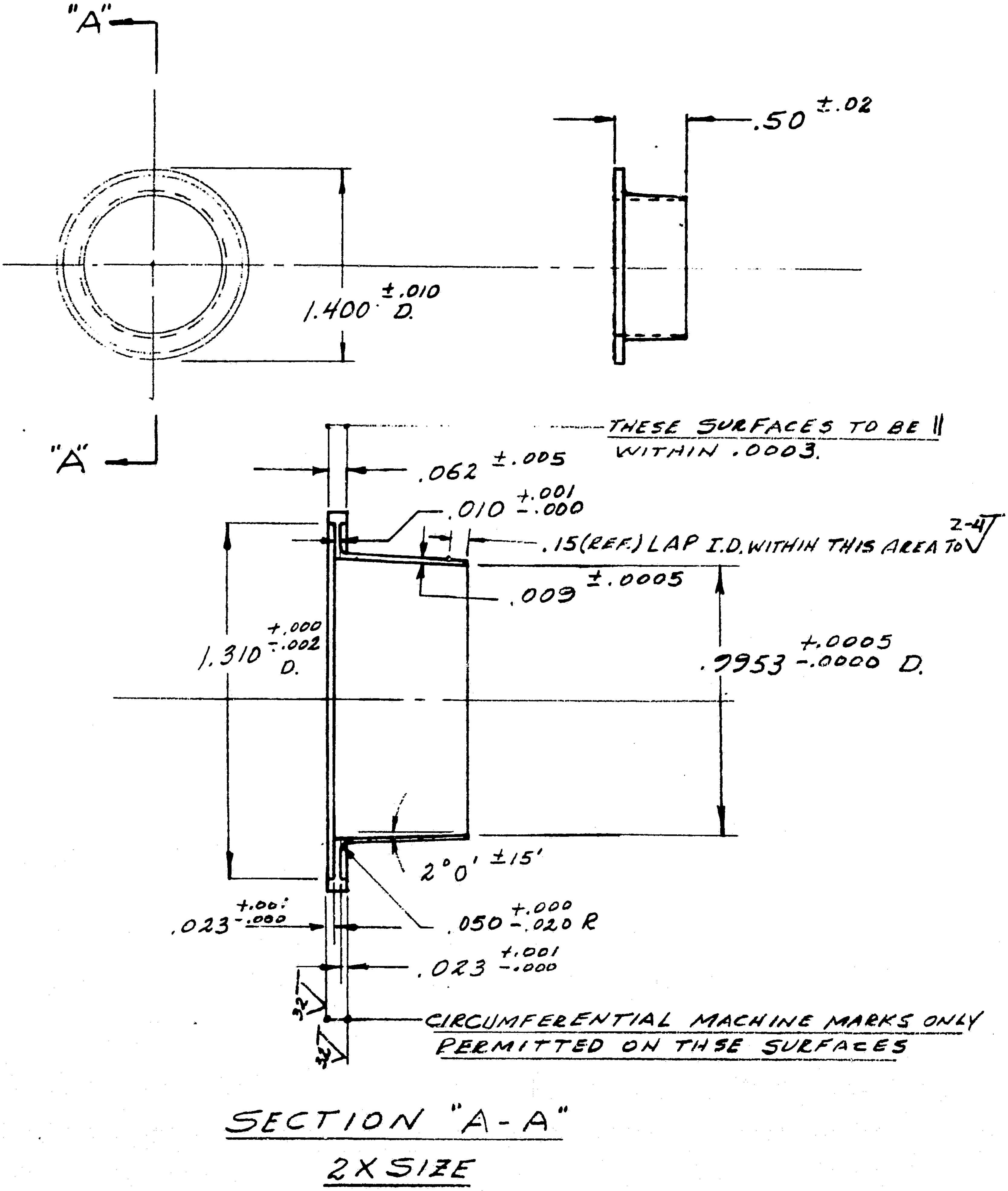


Figure 4-19. Vascojet 1000 Lip Seal (Modified Flange)

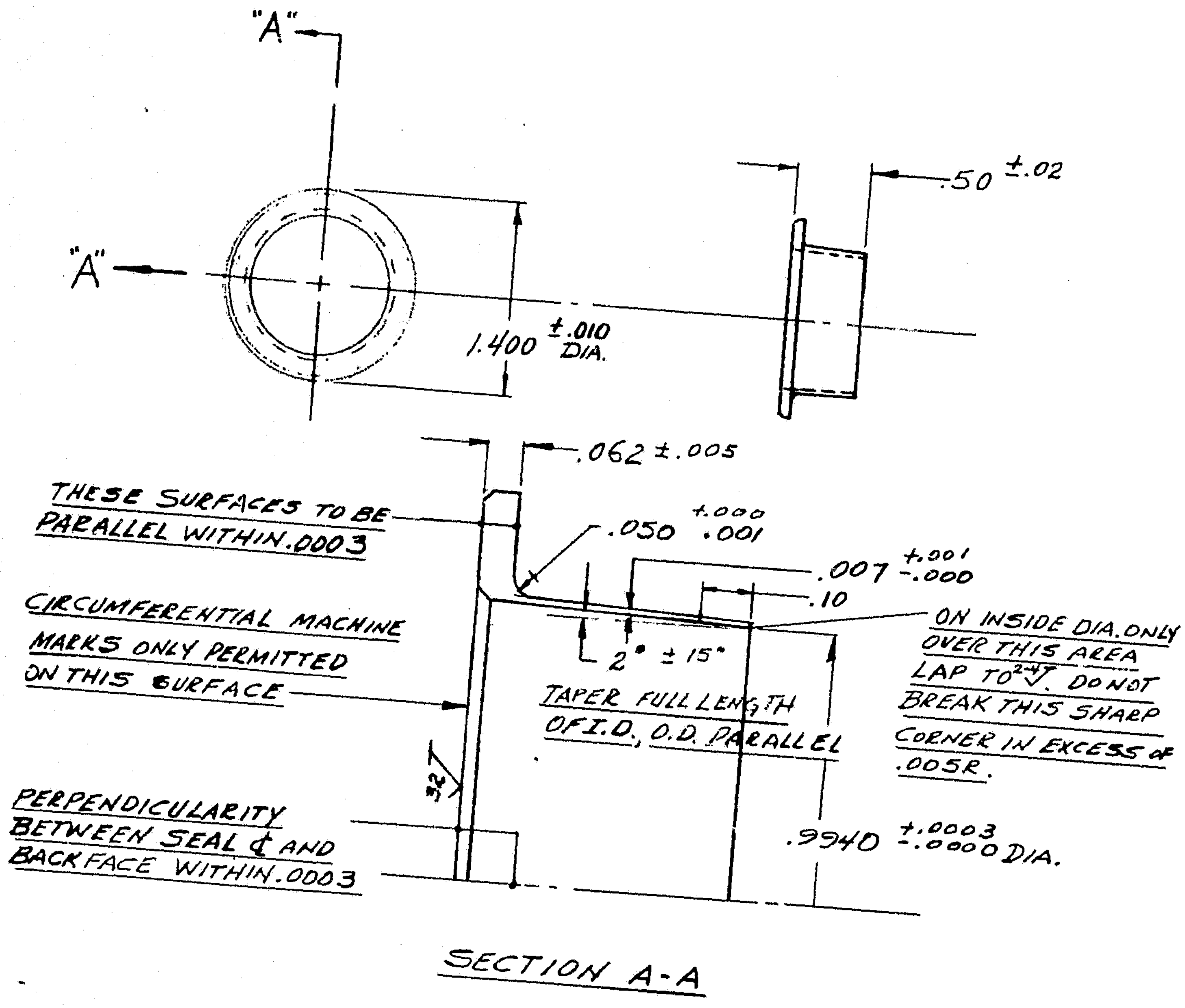


Figure 4-20. Cobalt-Molybdenum Alloy Lip Seal

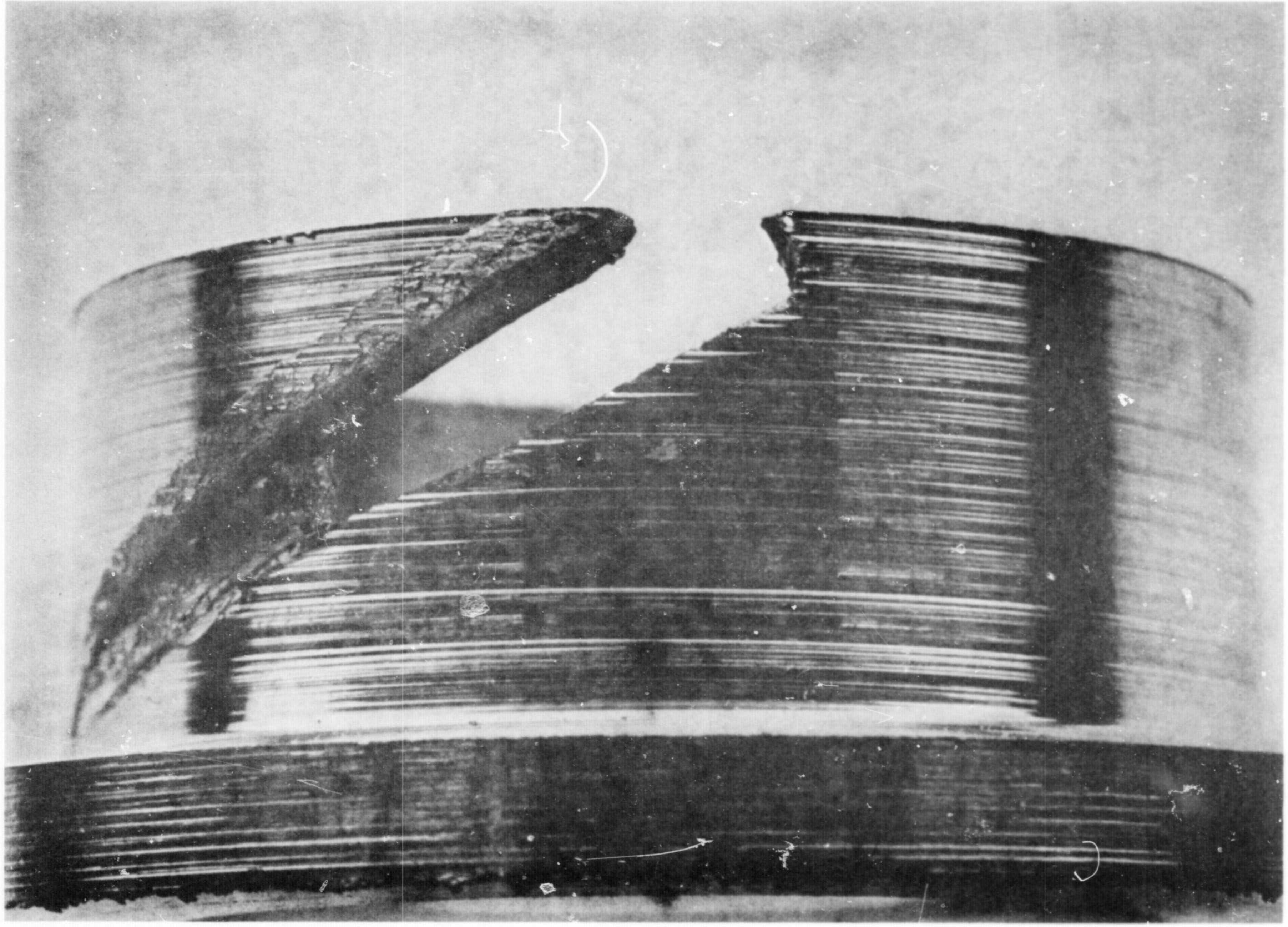
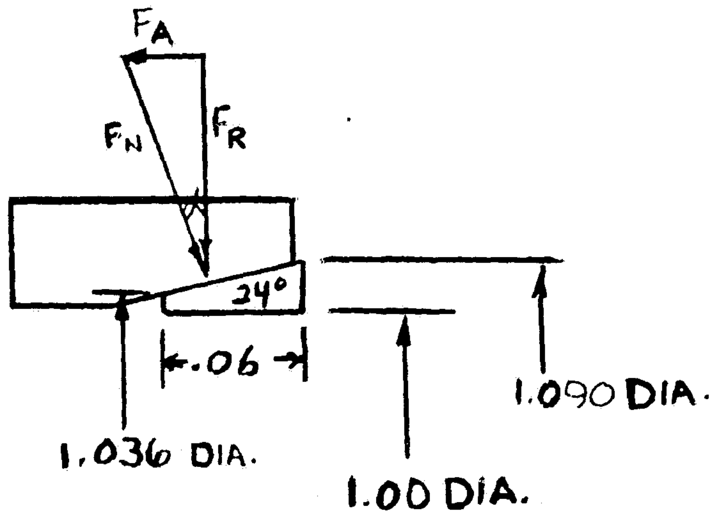


Figure 4-21. Inclusions in Cobalt-Molybdenum Alloy

4. Design I - Nickel Foametal Wedge Seal

Investigations were conducted on wedge seals with various taper angles to determine the most suitable configuration. Because wear compensation is one of the important factors of the seal, load-deflection characteristics of the various designs were first investigated. The theoretical load deflection of the wedge seal was approximated by the following method.



Seal wedge angle, α = 24 degrees

Coefficient of friction, μ = 0.2 for nickel Foametal

Mean seal radius, r = 0.521 inch

Seal cross-sectional area, A = 0.0019 in.^{2}

Seal radial load, F_R = 50 lb/in.

Modulus of elasticity, E = 9×10^6 for nickel Foametal (Reference Table 2-17)

By assuming a radial load (F_R) of 50 lb/in., the total circumferential load (P) is

$$P = F_R \times 2\pi r$$

$$P = 50 \times 2\pi \times 0.521$$

$$P = 163.7 \text{ lb}$$

Then the normal load (F_N) becomes

$$F_N = \frac{P}{\cos\alpha - 0.2 \sin\alpha}$$

$$= \frac{163.7}{0.9135 - 0.2(4067)} = 196 \text{ lb}$$

Using a radial seal load of 50 lb/in. of seal circumference, the compressive stress (S_c) induced is

$$S_c = \frac{F_R \times r}{A}$$

$$S_c = \frac{50 \times 0.521}{0.0019}$$

$$S_c = 13{,}700 \text{ psi}$$

The radial deflection of the seal becomes

$$R.D. = \frac{S_c \times r}{E}$$

$$R.D. = \frac{13{,}700 \times 0.521}{9 \times 10^6}$$

$$R.D. = 0.000795 \text{ in.}$$

The axial load (F_A) required to produce the above deflection is

$$F_A = N (\sin\alpha + \mu \cos\alpha)$$

$$F_A = 196 (0.4067 + (0.2 \times 0.9135))$$

$$F_A = 115 \text{ lb}$$

Similar calculations were performed on seals with wedge angles of 12, 30, and 45 degrees. The calculated values of loads and deflections for these configurations are summarized in Table 4-1.

TABLE 4-1

LOAD DEFLECTION TEST

Seal Wedge Angle (degrees)	Radial Seal Load (lb/in.)	Axial Seal Load	Radial Deflection		
			Calculated	(1) Experimental Data	(2) Experimental Data
12	50	72	0.0014	0.0008	0.001
24	50	115	0.0008	0.001	0.0016
30	50	146	0.00063	0.0005	0.0015
45	50	249	0.0005	(3) -	(3) -

(1) Seal wedge angle same as seat angle

(2) Seat angle 5 degrees greater than seal wedge angle

(3) 45-degree wedge seal not tested - configuration required excessive loading

Actual seal deflections under load were determined in the test setup shown in Figure 4-22. The test apparatus consists of a seal gland, holding fixture, force gauge, and hydraulic loading cylinder. The seal was deflected by applying an axial load to the seal with the hydraulic cylinder. The loading force was measured directly from the force gauge, and the radial deflection of the seal was measured with a dial indicator located at the sealing surface.

Results obtained with seals having wedge angles of 12, 24, 30, and 45 degrees are shown in Table 4-1. The actual radial deflections obtained correlate quite well with the calculated values. The exception was the 12-degree wedge seal, which exhibited approximately half the calculated deflection. This is believed to be due to the higher friction inherent in the low seal angle.

During inspection of the seals after test, it was noted that the seals were not loading near the tip of the taper. As the angle of the seat and the seal are the same, a slight variation in machining could result in improper seating of the seal. In view of this condition, the seal cavities were reworked to provide a seat angle that was 5 degrees greater than the seal angle. Load deflection tests were re-run using the modified seal cavities to determine the behavior of the seal. As shown in the last column of Table 4-1, additional deflection of the seals was obtained. However, this is partly attributable to higher local loading at the tapered end of the seal. Examination of the seals indicated that loading did take place at the tip of the taper.

Based on the results obtained, a seal angle of 24 degrees and a seat angle of 29 degrees appear to give the best load-deflection characteristics. This configuration (Figure 4-23) was further investigated to determine seal friction, effects of lateral motion, and over-pressure conditions.

Friction of the wedge seal was determined in the test apparatus shown in Figure 4-24, which consists of a seal test fixture, force gauges, and loading cylinder. The axial seal load was produced by the latter. Load values were read directly from the No. 1 force gauge. Seal friction was measured by the No. 2 force gauge. Figure 4-25 depicts seal friction at various load levels. For comparison purposes, seal deflection and seal contact pressure as functions of axial seal loading are also included in Figure 4-25. Values of seal contact pressures were calculated based on the axial load and apparent seal bearing area. The seal deflection data was obtained in the test apparatus shown in Figure 4-22. As shown in Figure 4-25, the increase in seal friction at the higher load conditions appear to be moderate for a one-inch seal. The data also indicates that additional seal deflections can be obtained by increasing axial loading. However, this also results in considerable increases in seal contact loads. The benefits of the added seal deflection could be offset by the increase in wear due to the higher seal loading.

Side-load tests were conducted to determine the ability of this seal to withstand lateral deflections due to bearing wear and misalignment. The test fixture (Figure 4-13) and procedures were the same as those described previously. Results showed that zero leakage was maintained with a deflection at the seal of 0.0028 inch. Because of its configuration, the seal is restrained from excessive deflections by the seal cavity. Therefore, further deflections

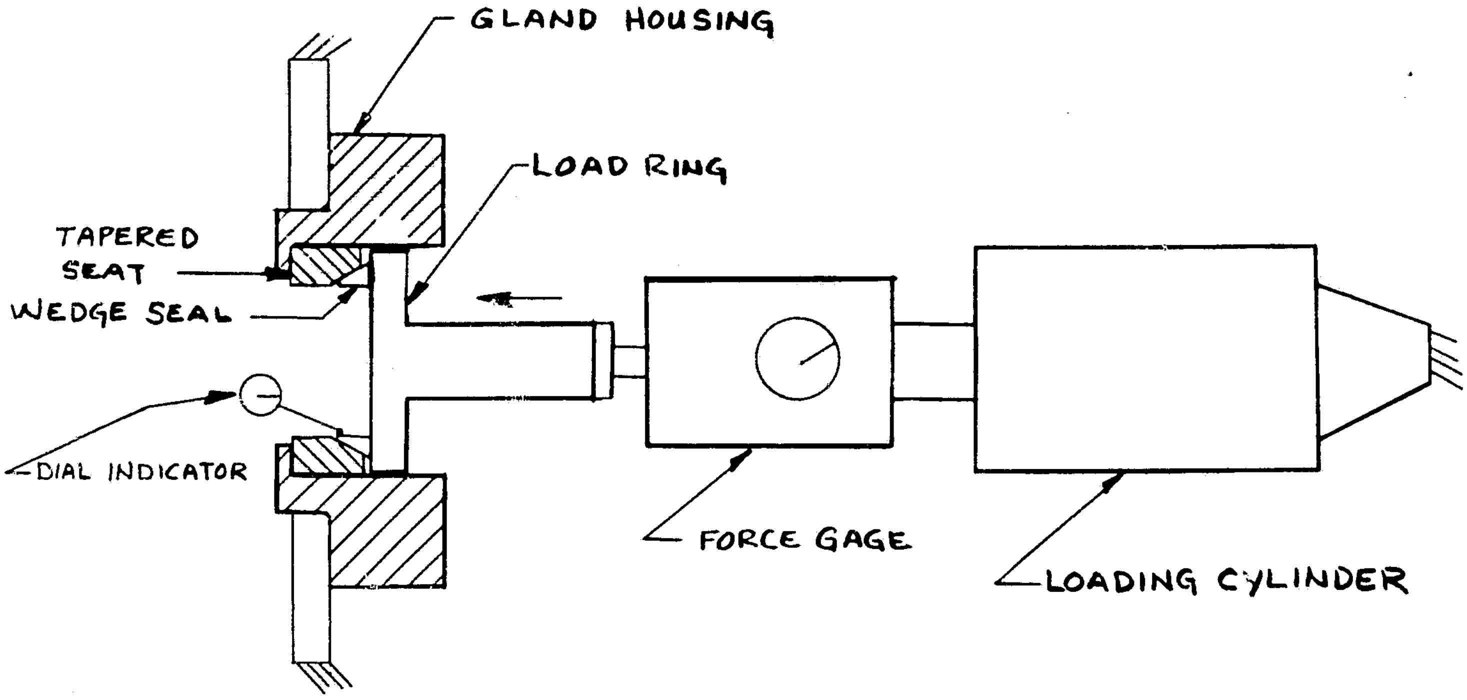


Figure 4-22. Wedge Seal Deflection Test Setup

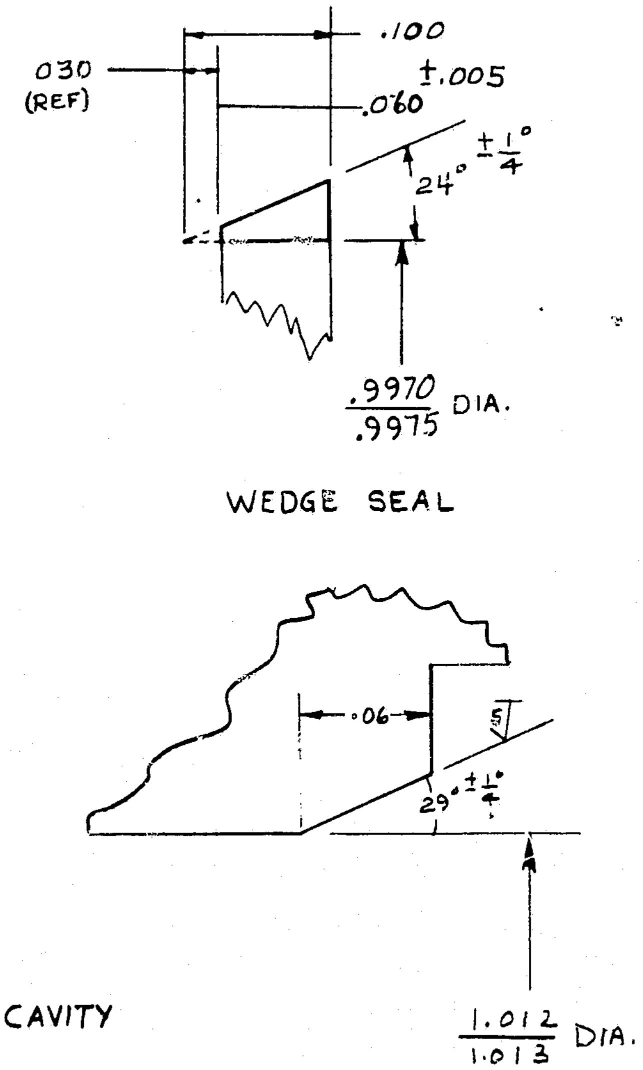


Figure 4-23. Geometry of Nickel Foametal Wedge Seal

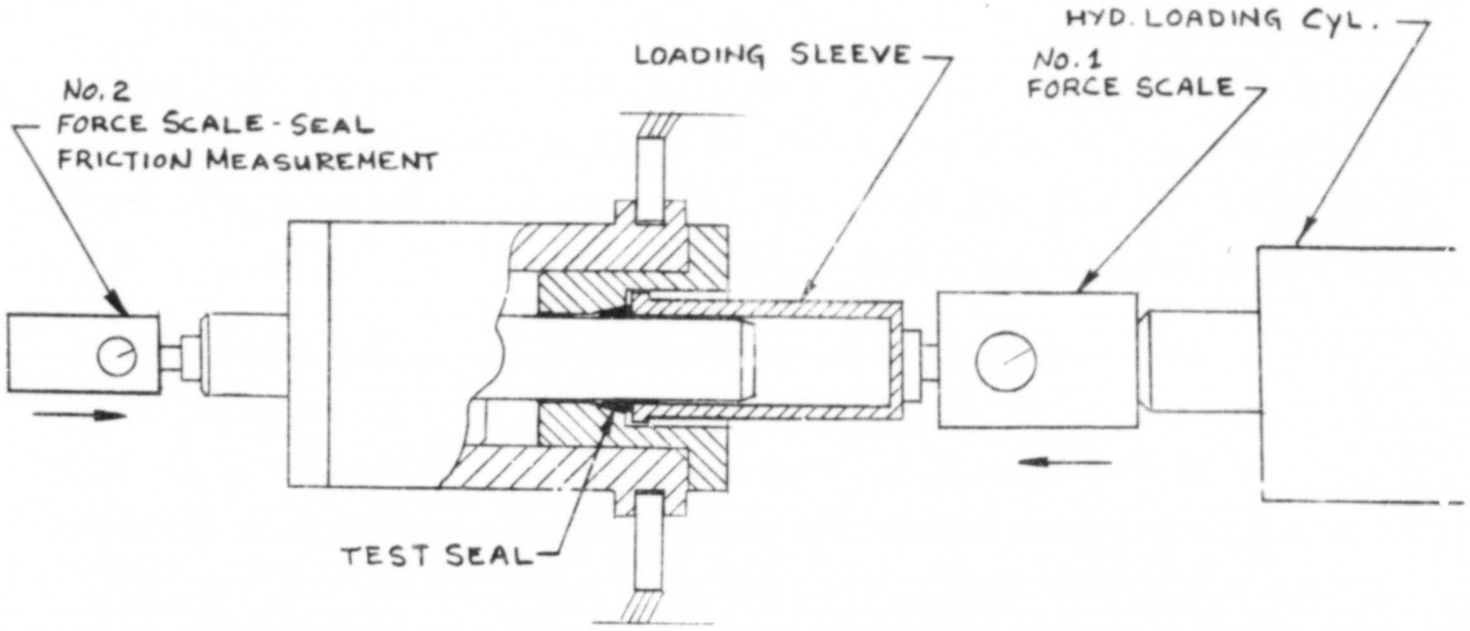


Figure 4-24. Test Setup for Wedge Seal Friction Measurement

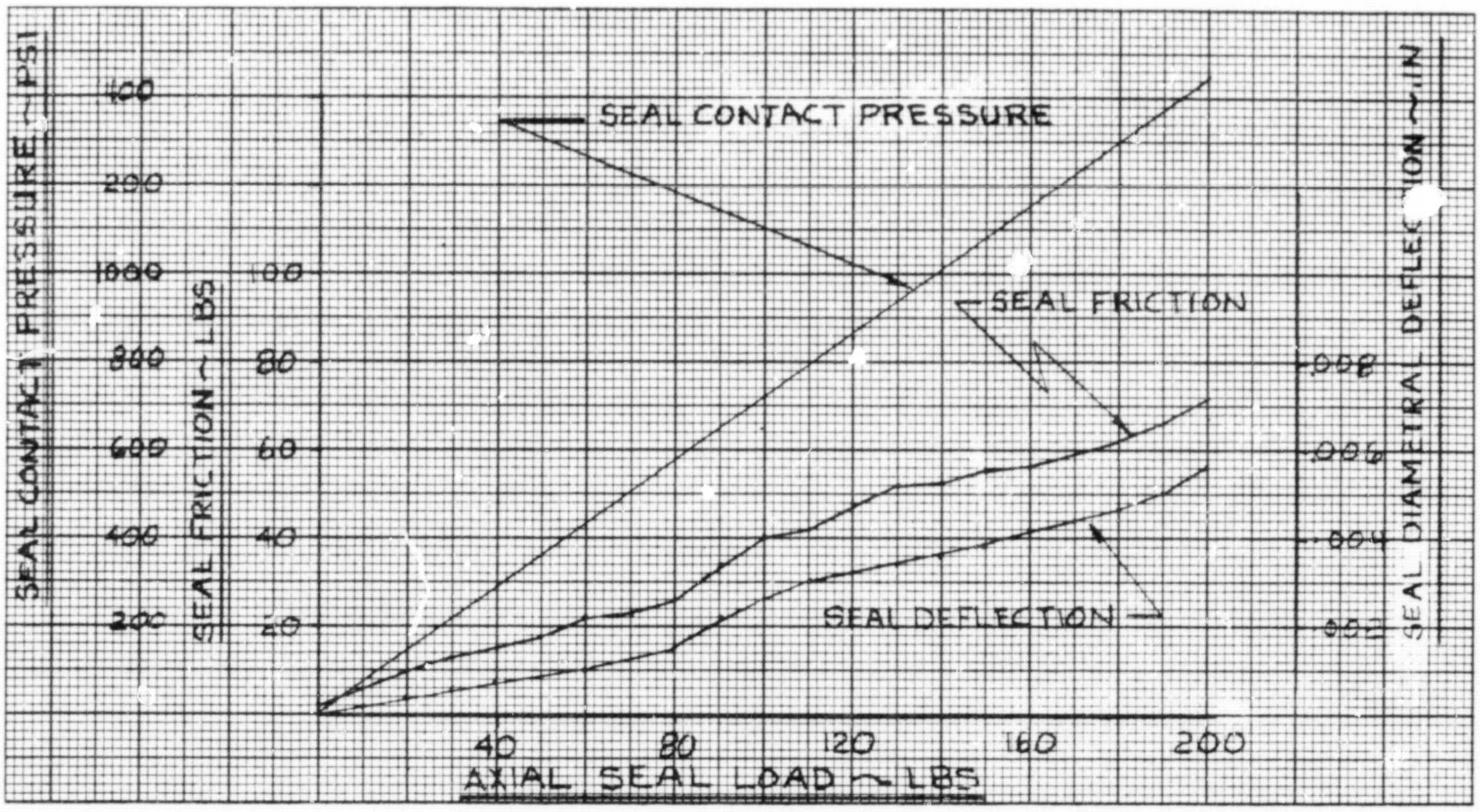


Figure 4-25. Friction Deflection and Contact Stress of Wedge Seal at Various Axial Seal Loads

would only induce compression of the wedge seal against the cavity and bending of the piston rod. Testing was repeated to determine the actual load on the seal at a deflection of 0.0028 inch. This was accomplished by using a force scale instead of the nylon set screw to deflect the rod. The load was applied at a distance of 5.50 inches from the seal to obtain a deflection of 0.0028 inch at the seal. The load required to obtain this amount of deflection was 80 pounds, which is equivalent to 180 pounds at the seal.

Following the above investigations, a series of tests was conducted to determine the seal's ability to withstand over-pressure conditions due to possible first-stage seal failure. The test setup used was essentially the same as that shown in Figure 4-24 for the friction test. The procedure for the pressure test consisted of applying a nominal axial load to the seal and then pressurizing the seal at pressure increments of 500 psi. Leakage was measured at the different pressure levels. When leakage was found to be excessive, the axial seal load was increased and the test repeated. The results (Figure 4-26) indicate that, at the design axial load of 115 to 120 pounds, seal leakage was 2 drops per minute at 1000 psi and 10 drops per minute at 1500 psi. However, above 1500 psi, leakage was in the form of a steady stream. Although reduced leakage can be accomplished at the higher pressures by increasing the axial seal, the higher seal load could result in excessive seal wear.

At the completion of the above investigations, a cycling test was conducted on the 24-degree wedge configuration. Testing was conducted in the low-pressure seal test actuator. In the test configuration shown in Figure 4-27, a seat angle of 29 degrees was used to ensure proper seating of the seal. The seal was loaded to obtain maximum deflection, which was approximately 0.0055 inch (diametrally). According to the data shown in Figure 4-25, an axial load of 200 pounds was required, which resulted in a seal contact pressure of approximately 1450 psi. Breakout friction for the seal under these conditions was approximately 70 pounds. Based on the data shown in Figure 4-25, the foregoing conditions would provide a seal that possesses high deflection capability for wear compensation and also minimum leakage at over-pressure conditions. As only the A-end of the actuator was reworked for the wedge seal, a set of Polyimide V-seals was installed on the B-end of the unit.

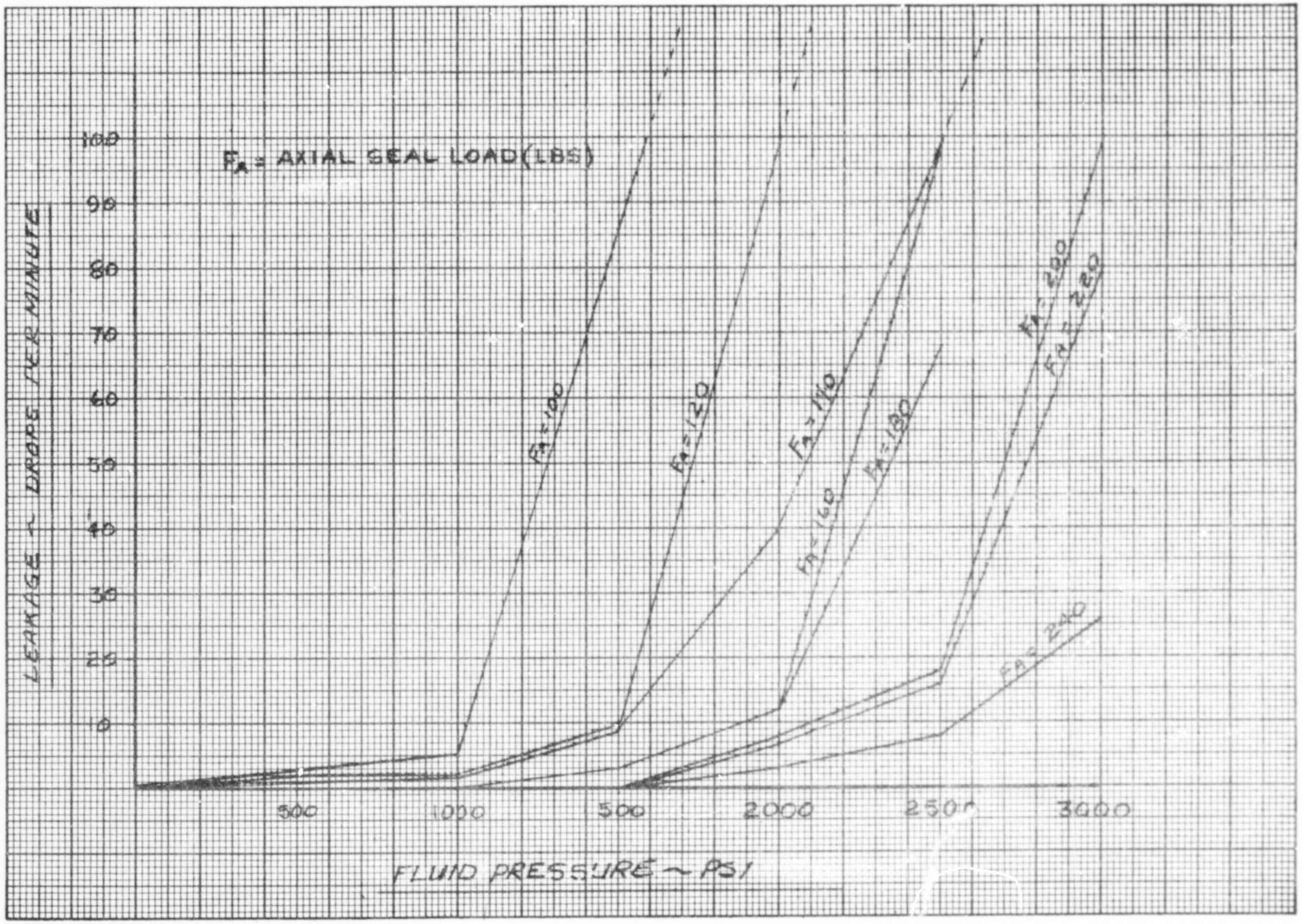


Figure 4-26. Leakage versus Pressure - Nickel Foametal Wedge Seal

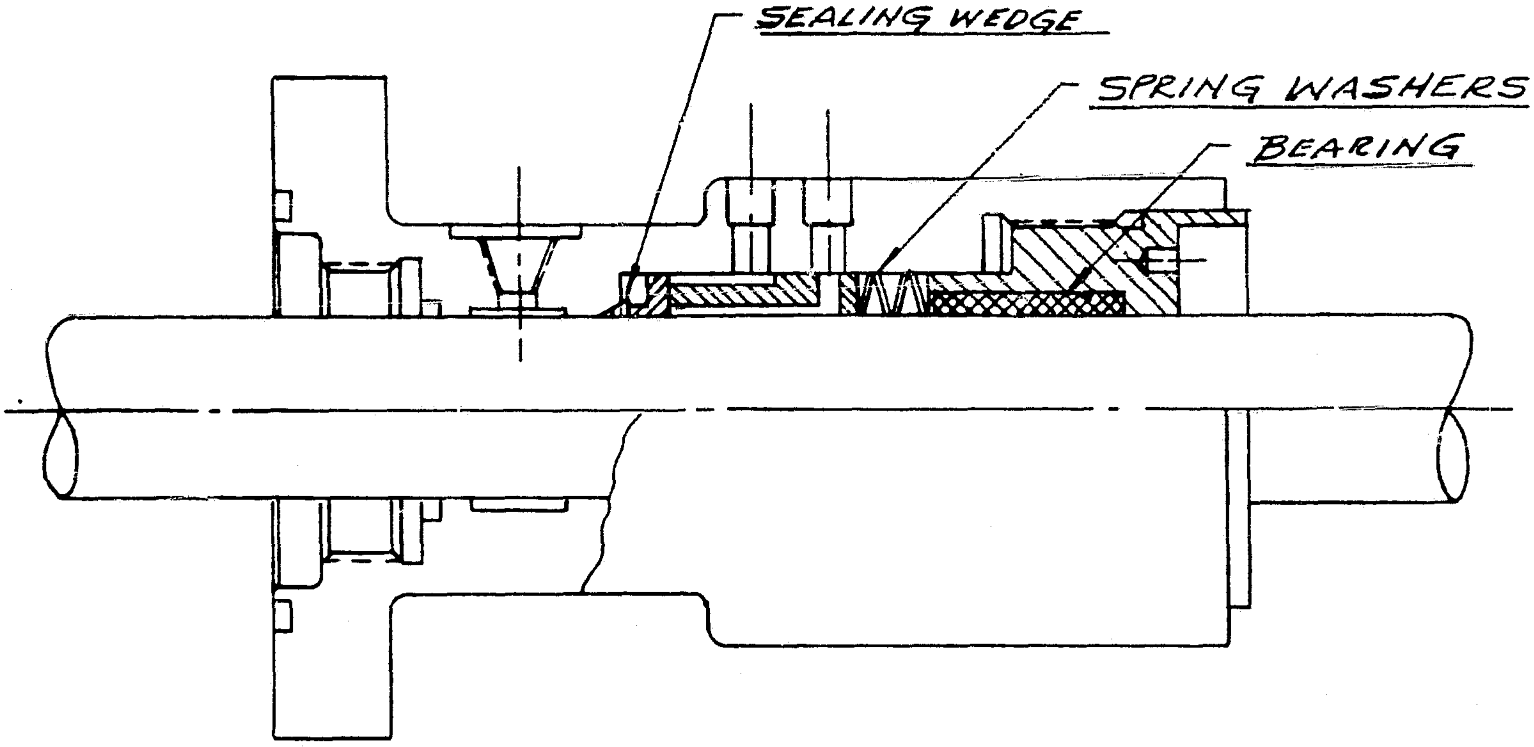


Figure 4-27. Gland Configuration - Nickel Foametal Wedge Seal

Results of the above cycling test showed that this loading condition was excessive. Wetting of the rod during the extension stroke was noticed during four hours of cycling at room temperature. Rod cycling was conducted at 20 cpm with a stroke of four inches. After 6700 cycles of room temperature operation, heat was applied to the test actuator. At 220°F fluid temperature, leakage of 60 drops per minute was experienced. At this point, the test was terminated and the actuator disassembled for inspection. Examination of the seal showed that the open pores of the sealing surface were burnished. As shown in Figure 4-28, a series of longitudinal scratches was noticed on the burnished surface. Inspection of the rod showed similar scratches. However, these scratches were not on the rod during initial assembly. A thin greyish film was also noticed on the piston rod, which appeared to have been transferred from the nickel Foametal. This film was easily rubbed off with fine emery paper. As the seal friction load showed only a slight decrease at the termination

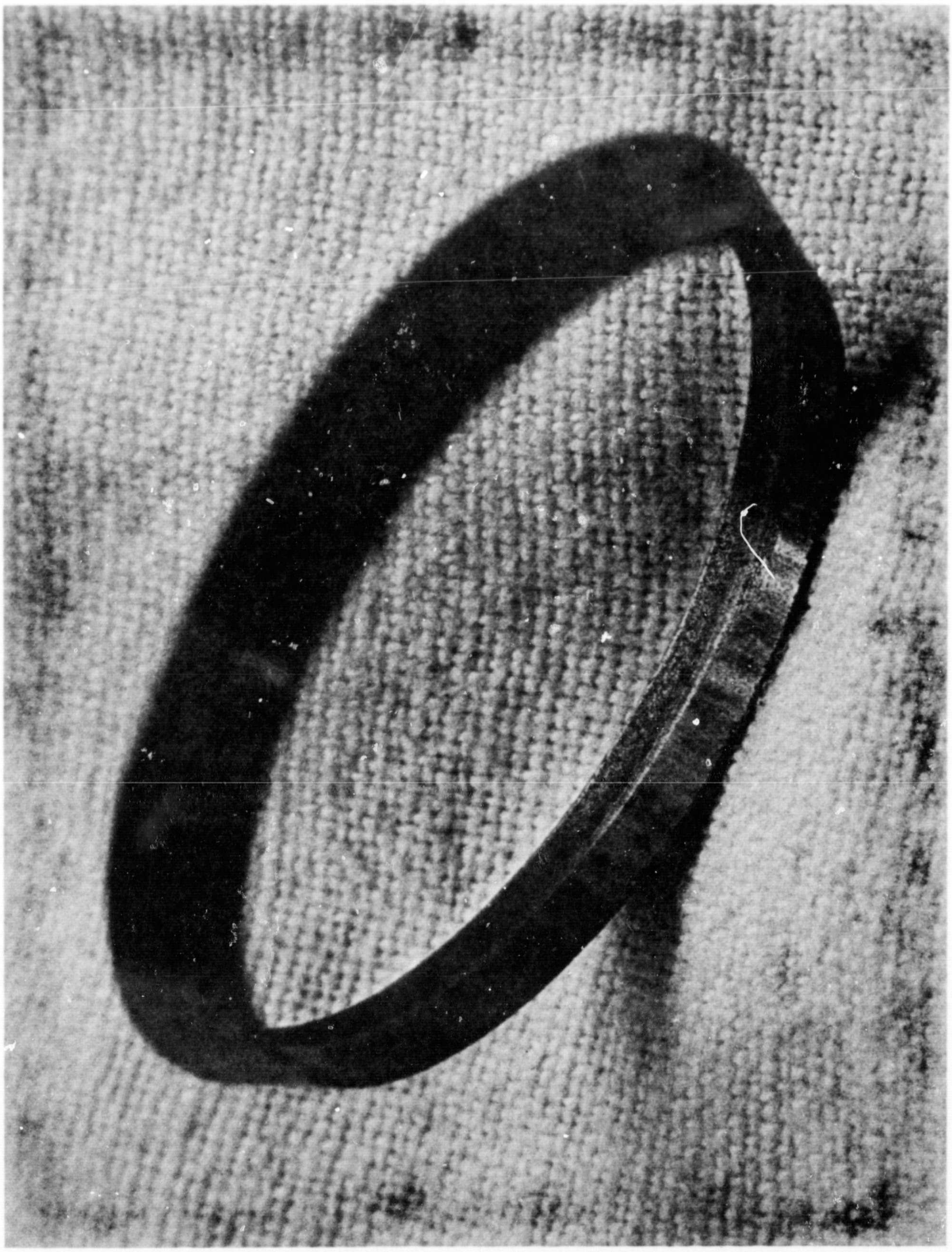


Figure 4-28. Nickel Foametal Wedge Seal After Test

of the test, it was concluded that the seal contact load was not lost. Therefore, the leakage experienced during the test was caused by the numerous leak paths formed by the scratches on the sealing surface of the wedge. Closer inspection revealed that the scratch marks resulted from the pull-out of material at the sealing surface. This condition was due to sliding of the nickel Foametal seal on a transfer film of the same material on the rod. This sliding resulted in welding and subsequent shearing of the material from the sealing surface of the wedge.

5. Design AH - Vascojet 1000 and Silver Alloy Reed Seal

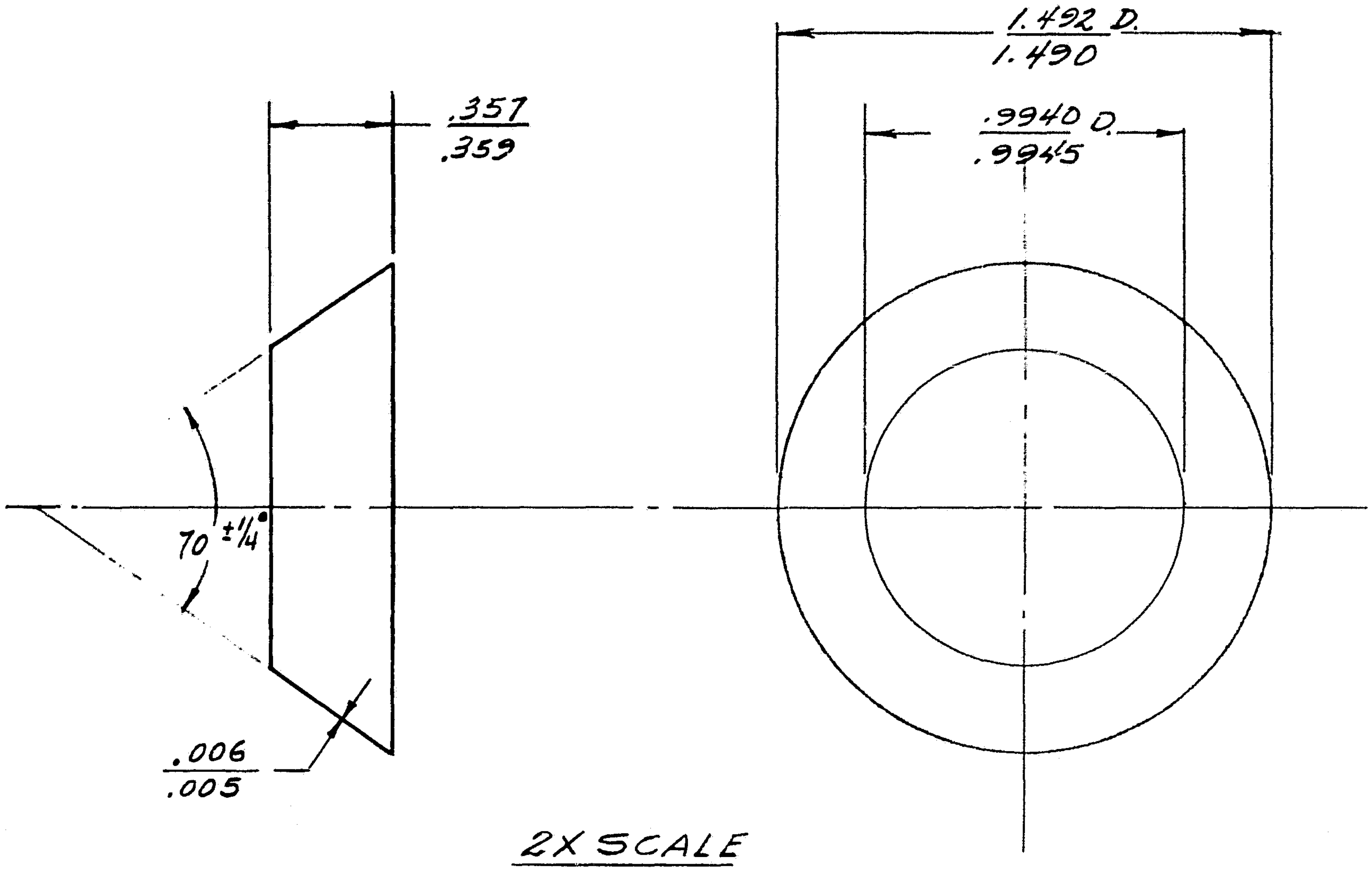
The detail design of this seal is shown in Figures 4-29, 4-30, 4-31 and 4-32. The silver reeds and Vascojet reeds were designed to provide an interference fit over the piston rod of 0.003 inch to 0.0035 inch. Thickness of the silver reeds was 0.010 inch as compared to 0.005 inch for the Vascojet reeds.

As shown in Table 3-1, the seal assembly consists of alternate reeds of silver and Vascojet 1000. Five each of silver and Vascojet 1000 are used for each assembly. Because of the thin cross-section of the reeds, this configuration offers good flexibility to withstand a reasonable amount of side. The design also provides sufficient back up to the reeds to withstand over-pressure conditions.

6. Seal Designs A and F

Although these two designs were not among the selected candidates, their relatively high ranking in the seal rating system warranted some exploratory development.

Seal design A shown in Figure 3-1, which consists of a truncated cone, was formed from a 0.030-inch thick Polyimide sheet. The seal was formed with an included angle of 150 degrees. The inside diameter of the seal was machined smaller than the piston rod to allow for an initial interference fit of 0.020 inch. The hoop stress induced by the interference fit provided the seal contact pressure for zero pressure sealing and to compensate for wear. During operation, the initial sealing force is augmented by fluid pressure acting



REED SEAL - VASCOJET 1000

NOTE:
HEAT TREAT PART AS FOLLOWS:
(1) PREHEAT AT 1500°F, (2) HOLD AT 1850°F FOR 20 TO 30 MIN. IN NEUTRAL ATMOSPHERE AND COOL TO RM. TEMP. RATE OF COOLING FR. 1850°F TO 1200°F IN 60 SEC. (3) TEMPER (IN NEUTRAL ATMOS.) IMMEDIATELY UPON REACHING RM. TEMP. FOR 3 HRS. AT 1050°F, AIR COOL AND RETEMPER FOR ADDITIONAL 3 HRS. AT 1050°F AND AIR COOL. (BRIGHT SURFACE REQ'D.) TEST BUTTON HARDNESS TO BE Rc 48-52.

AD6095-36

Figure 4-29. One-inch Reed Seal - Vascojet 1000

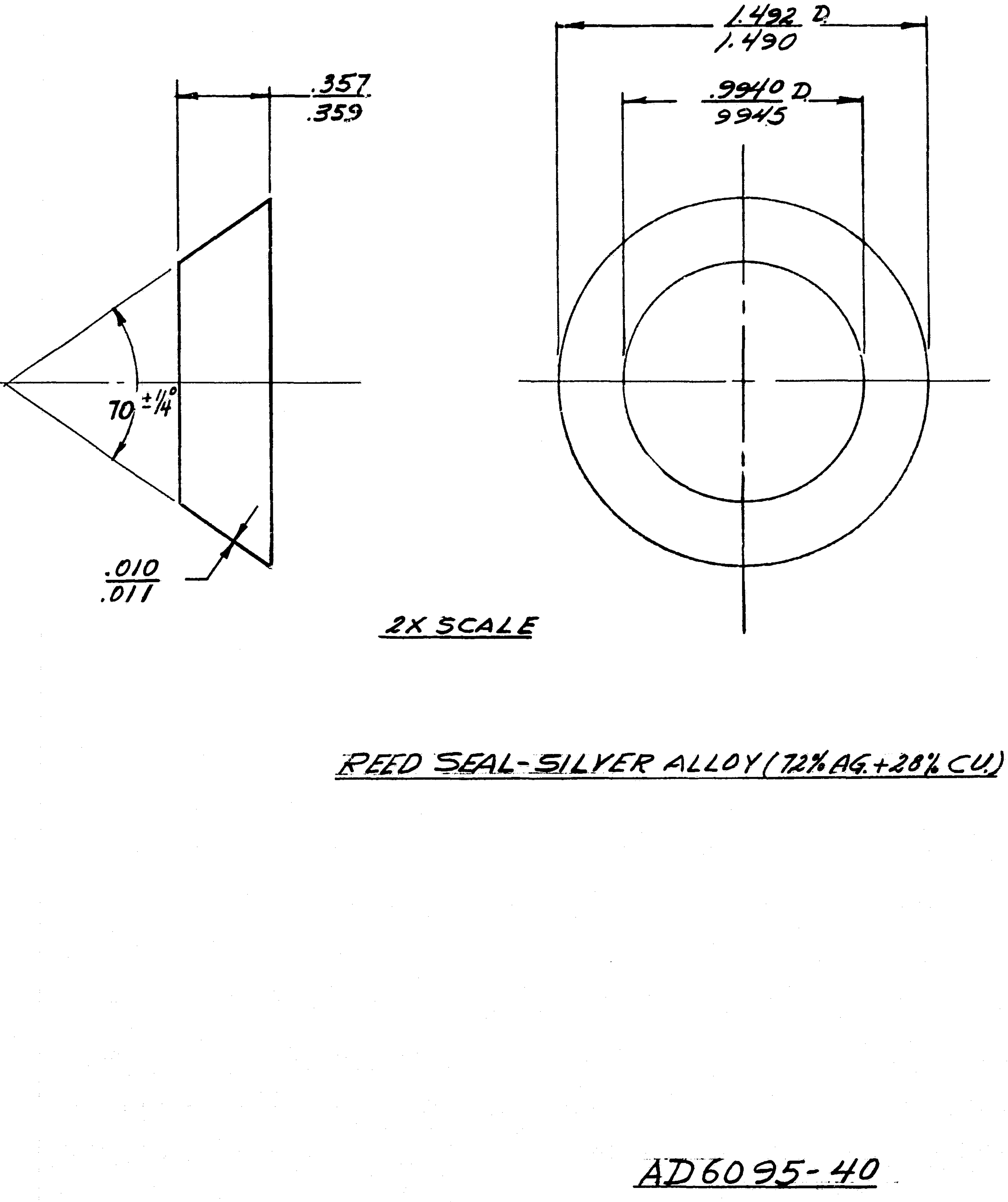
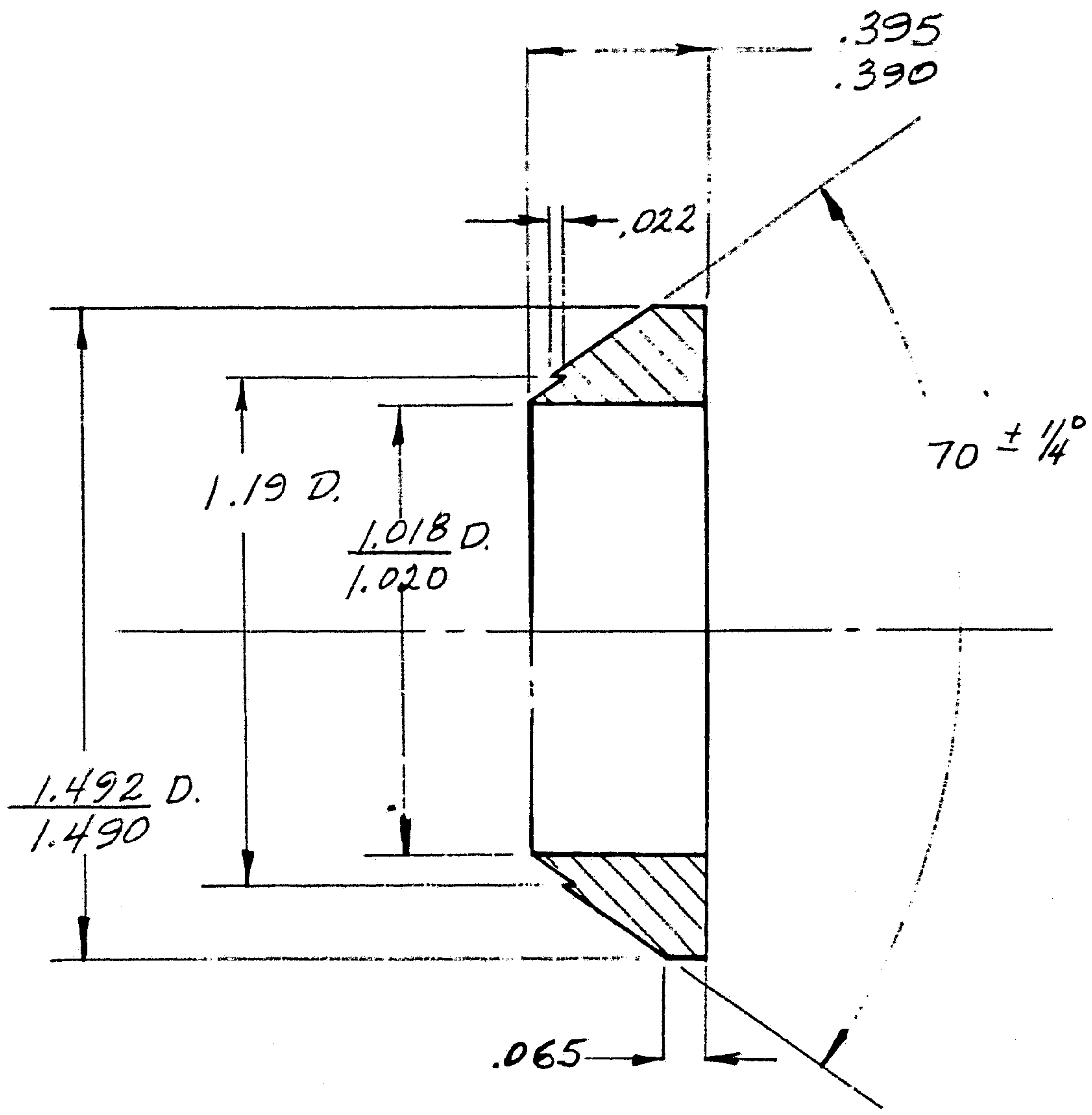


Figure 4-30. Reed Seal - Silver Alloy



MALE ADAPTER - REED SEAL

MAT'L - 17-4 PH CRES.

2X SCALE

AD6095-38

Figure 4-31. Male Adapter - Reed Seal

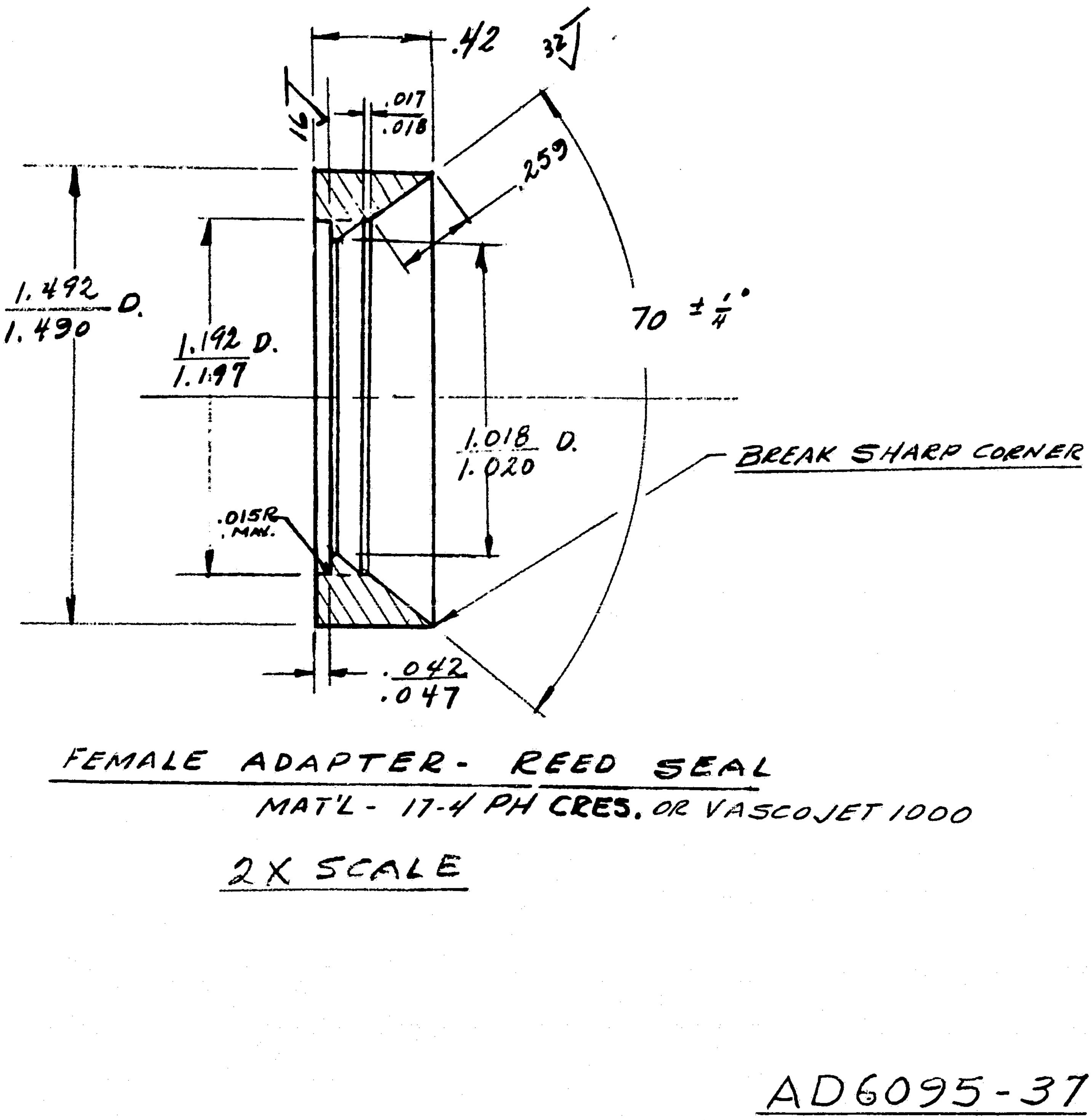


Figure 4-32. Female Adapter - Reed Seal

on the sealing element. Seal friction, thermal, and mechanical cycling effects were determined on this configuration.

Seal friction data at pressures up to 200 psi were obtained in the test setup shown in Figure 4-1. As shown below friction values for this design are relatively low for a one-inch seal.

Fluid Pressure (psi)	Avg. Seal Friction (lb) (total for both seals)
0	131
50	133
100	136
150	146
200	147

Following the friction test, a second set of seals of the same configuration was fabricated and subjected to mechanical cycling in the test rig shown in Figure 4-2. In this test, the seals were subjected to approximately 15 hours of cycling; 5 hours each at 400°F and 500°F and 6.3 hours at 600°F. The piston rod stroke was 2-inches and was reciprocated at a rate of 30 cpm. Friction was measured after completing operation at each temperature plateau.

The seals completed the 400°F and 500°F cycling with no leakage. However, at the completion of the 600°F operation, leakage of approximately 5 drops per minute occurred on the B-seal. This leakage was caused by the seal gland nut backing off, resulting in excessive axial clearance in the assembly. This enabled the seal to move axially during cycling and caused it to crack in several places. Evidence of permanent deformation of the seal was noticed, which indicated excessive deflection at the unsupported area.

Approximately 39% interference was retained on the A-seal. The B-seal showed a retention of about 11% to 20%, the lower values being attributable to the cracks in the seal which relieved much of the hoop tension. The decrease in interference was indicated by the reduction in seal friction (Figure 4-33) recorded after operation at each temperature level. However, it is also possible

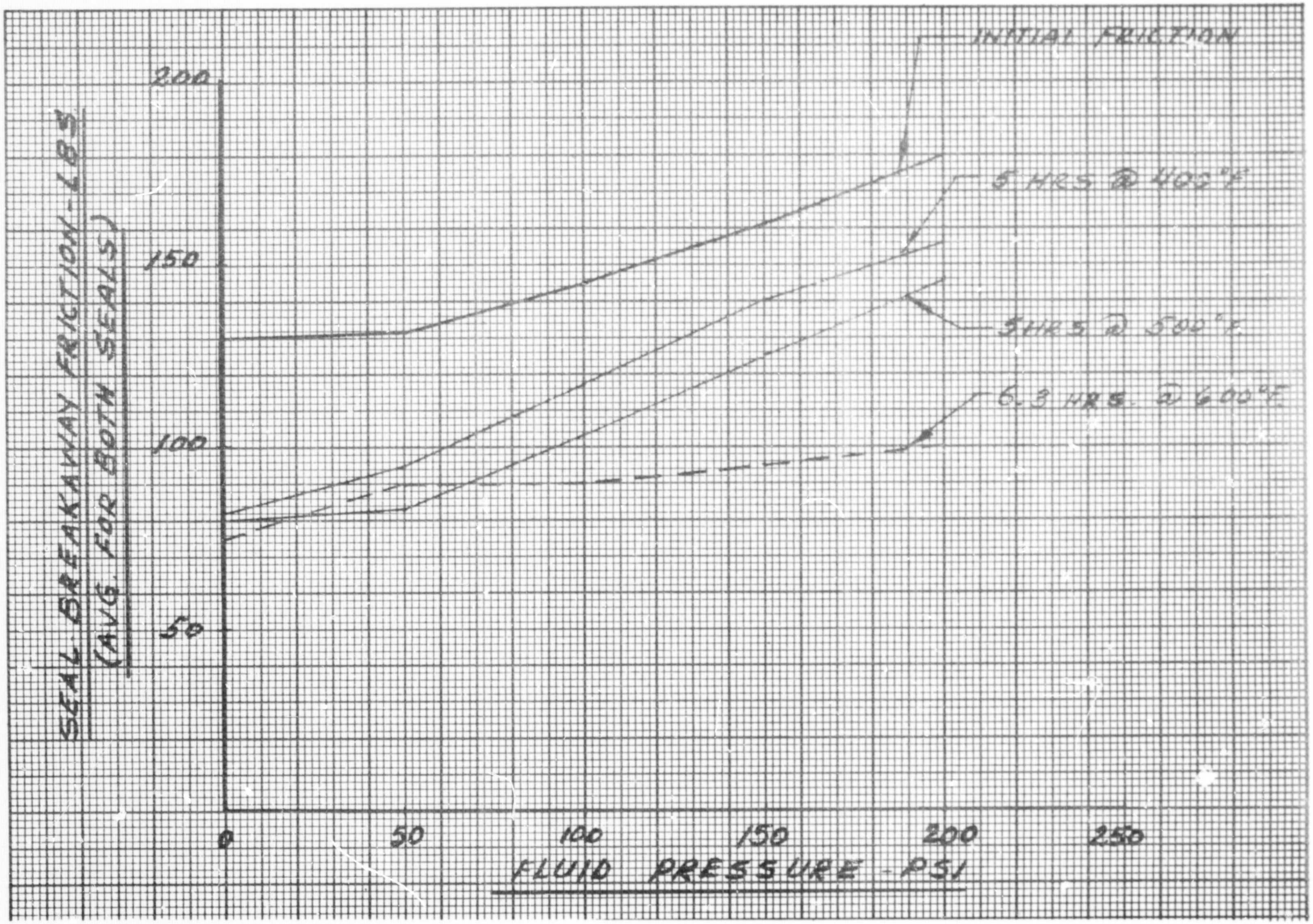


Figure 4-33. Seal Breakaway Friction vs Temperature Cycling

that the lower friction was due to the burnishing of the piston with the Polymer SP seals during cycling. In general, the results obtained were very promising.

Additional aging tests were conducted to determine stress relaxation characteristics during long-time exposure at 500°F and 600°F. The aging test at 500°F was conducted with Polymer SP seals 0.030 and 0.060 inch thick. Each aging cycle consisted of heating from room temperature to 500°F, maintaining this temperature for 50 hours, and then cooling to room temperature. After each aging cycle, the seals were disassembled and measured. The results obtained after 350 hours of aging are presented below and in Figure 4-34.

As indicated in Figure 4-34, the 0.030 inch thick seal cracked after 150 hours at 500°F, resulting in the loss of 80% of its initial interference. Failure was caused by high stresses being concentrated at the outer periphery because of the method used in holding the seal in the cavity. Consequently, the seal was replaced with another 0.030 inch thick seal and testing continued. However, testing was monitored on the 0.060 inch thick seal only. A total of 350 hours was obtained with the latter with only a 43% loss in the initial interference.

Results of the Polyimide aging test conducted at 600°F are shown below and in Figure 4-34. Relaxation of the seal material was quite pronounced at 600°F. As shown in Figure 4-34, a rapid drop occurred after 112 hours, representing a loss of approximately 88% and 92% for the A- and B-seal, respectively.

Aging Cycle	Hours at 600°F	Seal Friction (lb) at 0 psi	A-Seal I.D. (in.)	B-Seal I.D. (in.)	Seal Interference (in.)	
					"A"	"B"
Before Aging		105	.9841	.9789	.0144	.0196
1	50	56	.9920	.9935	.0065	.0050
2	112	37	.9973	.9961	.0012	.0024

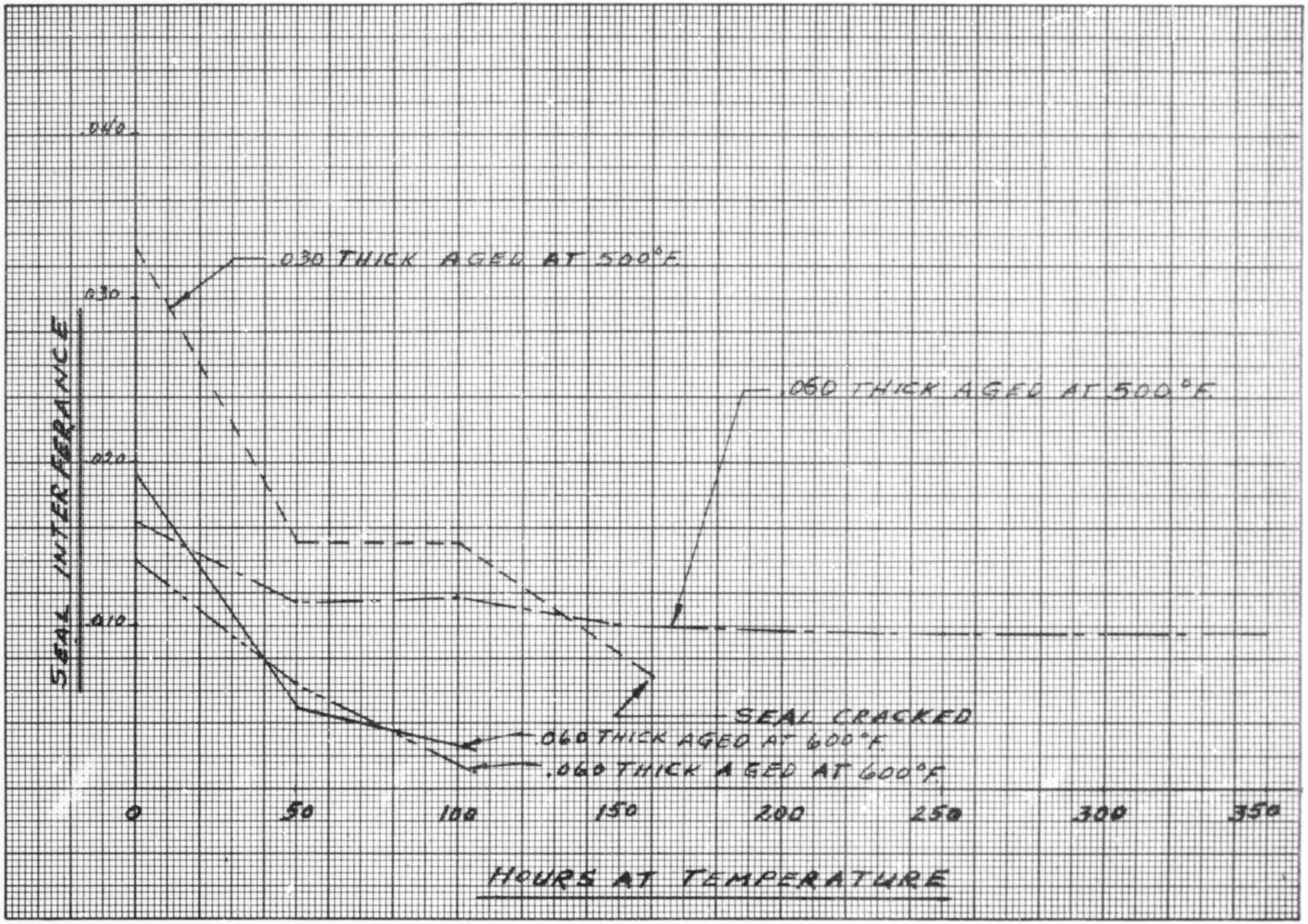


Figure 4-34. Polyimide Aging Test

As the lip seal design relies on the radial stresses induced at the inner diameter to effect a seal, the use of this seal at 600°F appears marginal with Polyimide. At 500°F, the material is able to retain a good portion of the initial stretch to maintain an effective seal; however, its long-term behavior at this temperature is uncertain.

Tests conducted on the externally loaded lip seal (Design F) shown in Figure 3-8, demonstrated the feasibility of this design approach. The lip seal was fabricated from a silver alloy (72% Ag + 28% Cu) with a seal section thickness of 0.025 inch. The diametral clearance between the seal and piston rod (chrome plated, Type 440C) was approximately 0.002 inch. The seal was assembled in the test unit shown in Figure 4-4. Loading of the sealing lip to achieve contact with the piston rod was readily accomplished. Seal contact with the rod was indicated by an increase in seal breakaway friction (approximately 40 pounds at 0 psi). During the initial loading of the seal, leakage was approximately 5 drops per minute at 100 psi using the F-50 silicone fluid. This leakage was due to the seal not making sufficient contact with the piston rod. Consequently, the seal was run-in for 8400 cycles (30 cpm at 2-inch rod strokes) and the gland nut was retightened. Following this procedure, a pressure check of the seal was made at 100 psi with no leakage.

After demonstrating sufficient sealing ability, the seal was subjected to cycling at temperatures to 400°F. Leakage collected during 6.5 hours of cycling was 12 cc. Time and cycles are summarized below.

Run-in cycles	8,400	-	4	hrs
Cycles - room temp. to 400°F	4,200	-	2	hrs
Cycles at 400°F	9,450	-	4.5	hrs
Total cycles and time	22,050	-	10.5	hrs

Disassembly of the seal showed that an even wear pattern was obtained around the sealing surface. Although the design was intended to be used with spring washers for wear compensation, no springs were used in this test because the main purpose was to demonstrate the loading approach.

D. FIRST-STAGE HIGH-PRESSURE SEALS

Several designs of contracting sealing rings were evaluated for possible use as first-stage rod seals. These include sealing rings fabricated of Graphitar grade 80, Type 440C stainless steel, Polyimide, and an alloyed cast iron. Typical designs are shown in Figures 3-12 through 3-15.

Static leakage checks were obtained on the sealing rings shown in Figures 3-12, 3-13, and 3-14. These were checked in the one-inch rod size, at pressures between 0 and 4000 psi. Leakage measurements were made with the rings assembled in a single-ended actuator. Leakage characteristics of there rings are shown in Figure 4-35. The three-piece graphite rings exhibited fairly high leakage at low pressures, but leakage decreased with increasing pressure. The C. Lee Cook three-piece metallic sealing exhibited fairly low leakage throughout the pressure range. The two-piece Polyimide sealing rings (Figure 3-14) were evaluated in two thicknesses. Results showed that a ring of thinner section provides better conformability and smaller gap area and, consequently, lower leakage. Seal friction was not obtainable on these rings due to the unbalanced condition of the single-end actuator used in these tests. Leakage rate and friction of new and used Polyimide sealing rings of the same configuration were compared. The used rings were tested in a previous high-temperature seal program at 500°F fluid (Ref. 1). Total test time on these rings was 166 hours, with approximately 120 hours at 500°F. As shown in Figure 4-36, leakage rates of the used rings were slightly higher at 2000 psi. At 3000 psi the used rings exhibited lower leakage than the new rings. Seal breakaway friction varied slightly for the new and used rings at pressures to 2000 psi.

Based on the above investigation, the split ring polyimide sealing ring was selected as the most suitable configuration for the first-stage seal. As shown in Figure 3-14, the seal consists of a polyimide sealing ring and a 17-4PH corrosion-resistant steel spring.

In view of the 3000-hour life requirements for these rings, it was decided that pressure balancing should be incorporated to minimize ring wear. Figure 4-37 depicts a pressure balanced version of the sealing ring proposed by Koppers.

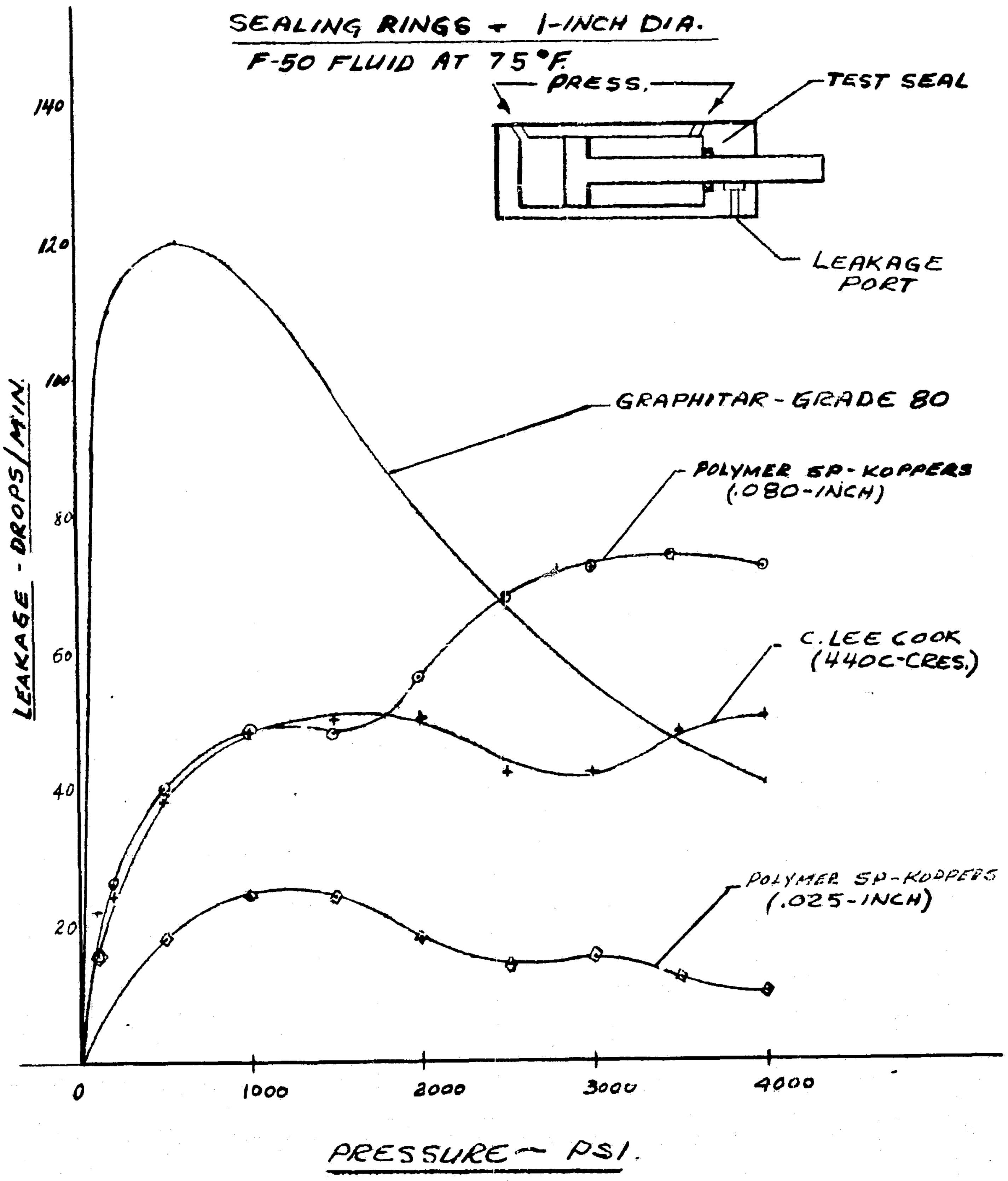


Figure 4-35. Seal Leakage Test - Contracting Sealing Rings

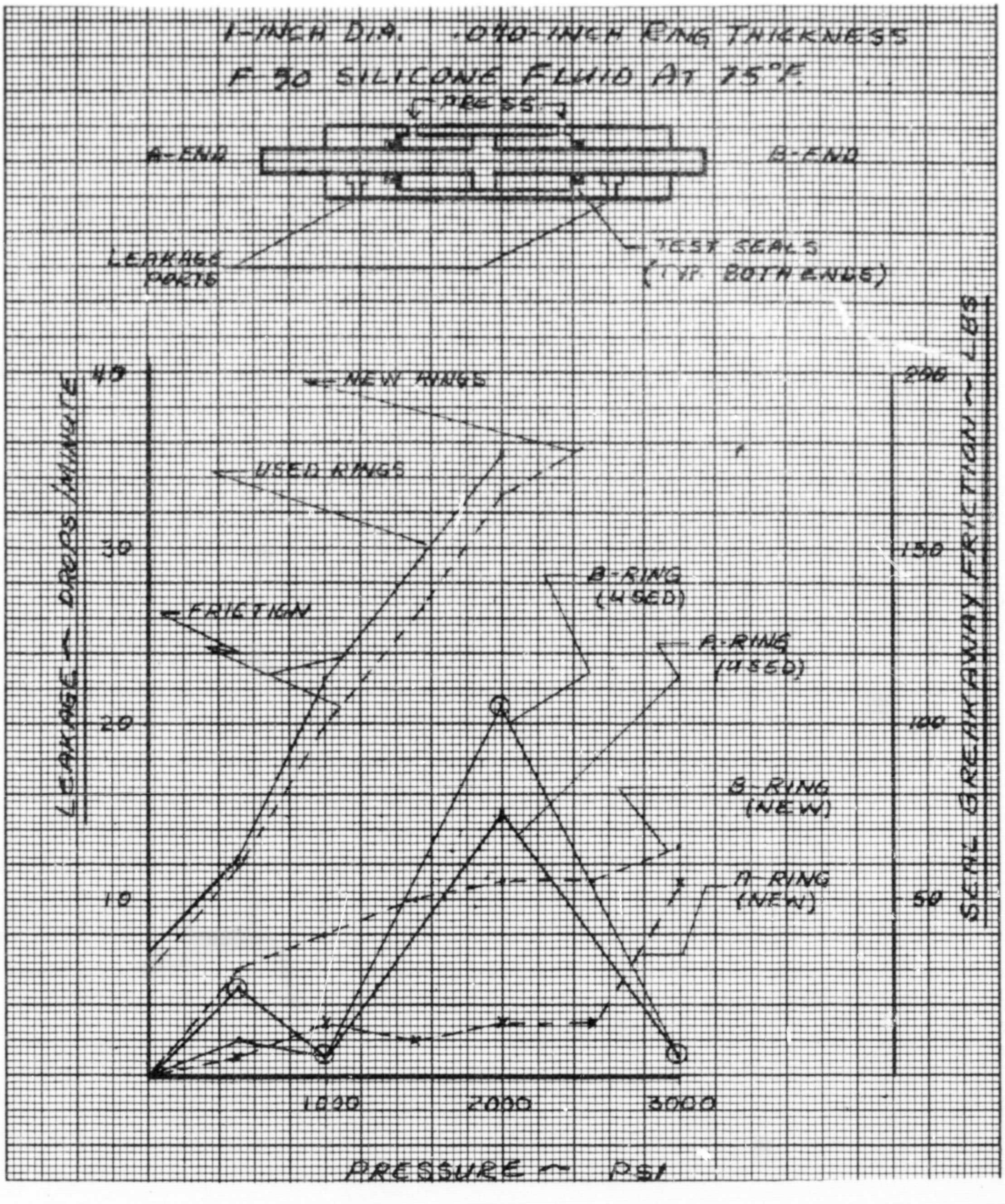


Figure 4-36. Leakage and Friction of New and Used Polyimide Contracting Sealing Rings

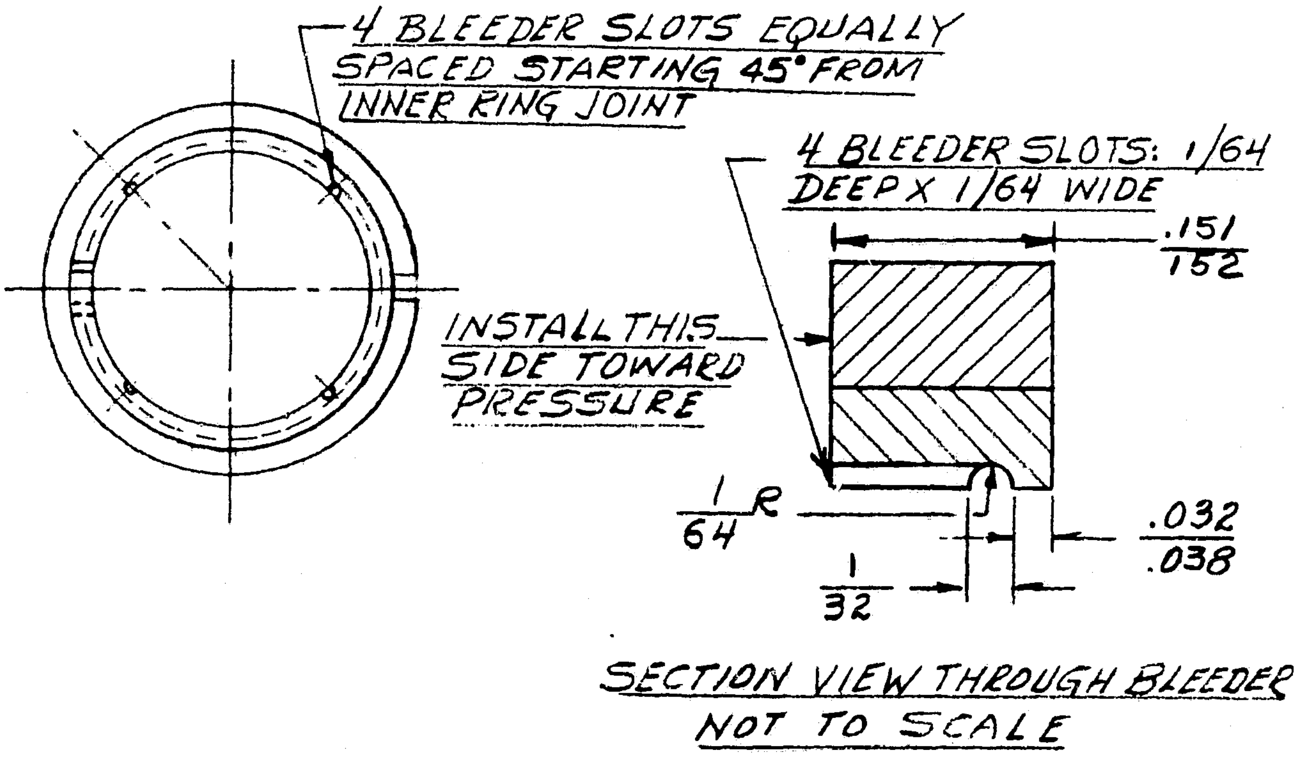
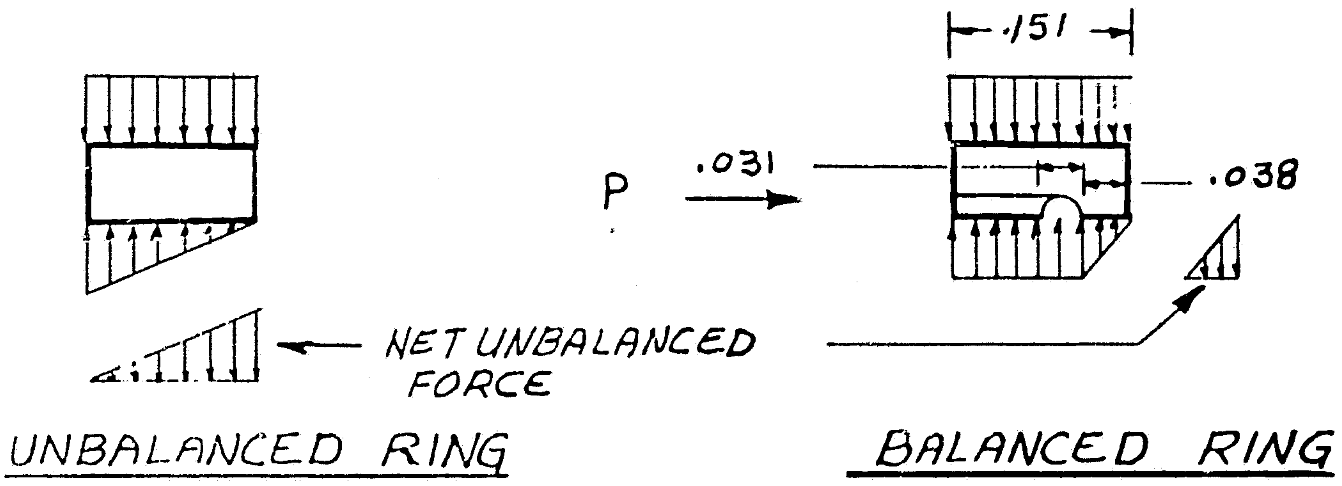


Figure 4-37. Pressure Balanced Sealing Ring (for 3-inch Rod)

Company, Inc. for the three-inch rod size. This design provides approximately 67 percent pressure balancing. The difference in contact pressures between a balanced and unbalanced ring of this configuration is depicted below.



Assuming the pressure gradient to be linear, the contact pressure for the unbalance ring is:

$$P_c = \frac{4000}{2} = 2000 \text{ psi}$$

For the balanced ring, the contact pressure is reduced to:

$$P_c = \frac{\frac{4000}{2} \times .038}{.151 - .038} = 673 \text{ psi}$$

In the above calculations the contact pressure generated by the outer ring (shown in Figure 4-39) was not included, since the spring load (approximately four pounds) was negligible. However, it can be seen that the bearing pressure is greatly reduced by pressure balancing. With a pressure-balanced ring, the thickness of the ring is adjusted to keep the axial forces in balance with the radial forces. This ensures adequate closing of the ring in the radial direction and, at the same time, maintains ring contact with the gland wall.

SECTION V

LOW-PRESSURE SEAL TESTING

A. GENERAL

Low-pressure tests were conducted on the five candidate seal configurations to determine the three most promising designs for endurance testing. Testing of all five candidate seals were performed in the one-inch rod size. The three best configurations were then evaluated in the three-inch rod size.

B. TEST PARAMETERS

The test profile is shown in Figure 5-1. Testing was conducted with F-50 silicone fluid at 100 psi and at temperatures of 400°F, 500°F, and 600°F. Test time for each of these temperature levels was 50 hours or until failure. Seal leakage in excess of one drop per minute or a two-fold increase in friction was considered failure.

The test profile consisted of the following operations:

1) Long-stroke cycling (± 2 inches) at 25 ± 5 cpm during the heat-up from room temperature to the designated temperature level
2) Continuous operation at the designated temperature level for 50 hours. Operation consists of alternate short-stroke cycling (±0.1 inch at 300 cpm for two hours) and long-stroke cycling (±2 inches at 30 cpm for one hour)
3) Long-stroke cycling during the cool-down from the designated temperature level to room temperature
4) Repeat of steps 1, 2 and 3 for the next temperature level

Seal leakage and friction was also monitored during the operation.

C. TEST EQUIPMENT

1. Cycling Rigs

Two cycling rigs were designed and fabricated for the low-pressure testing of one-inch and three-inch seals. Shown in Figure 5-2 is a typical rig which consists of a seal test actuator and a driving actuator. The test actuator

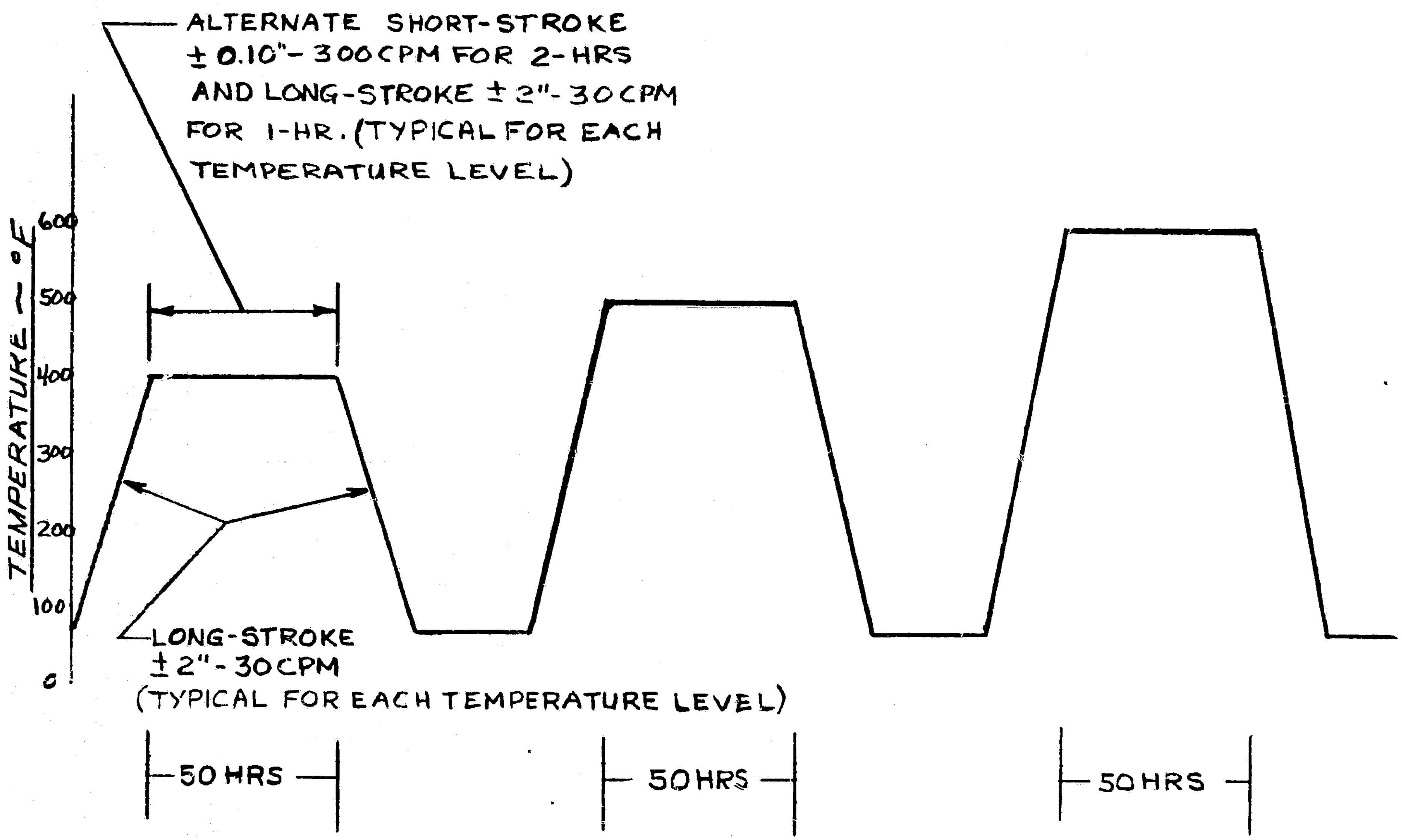


Figure 5-1. Test Profile - Low Pressure Seal Test

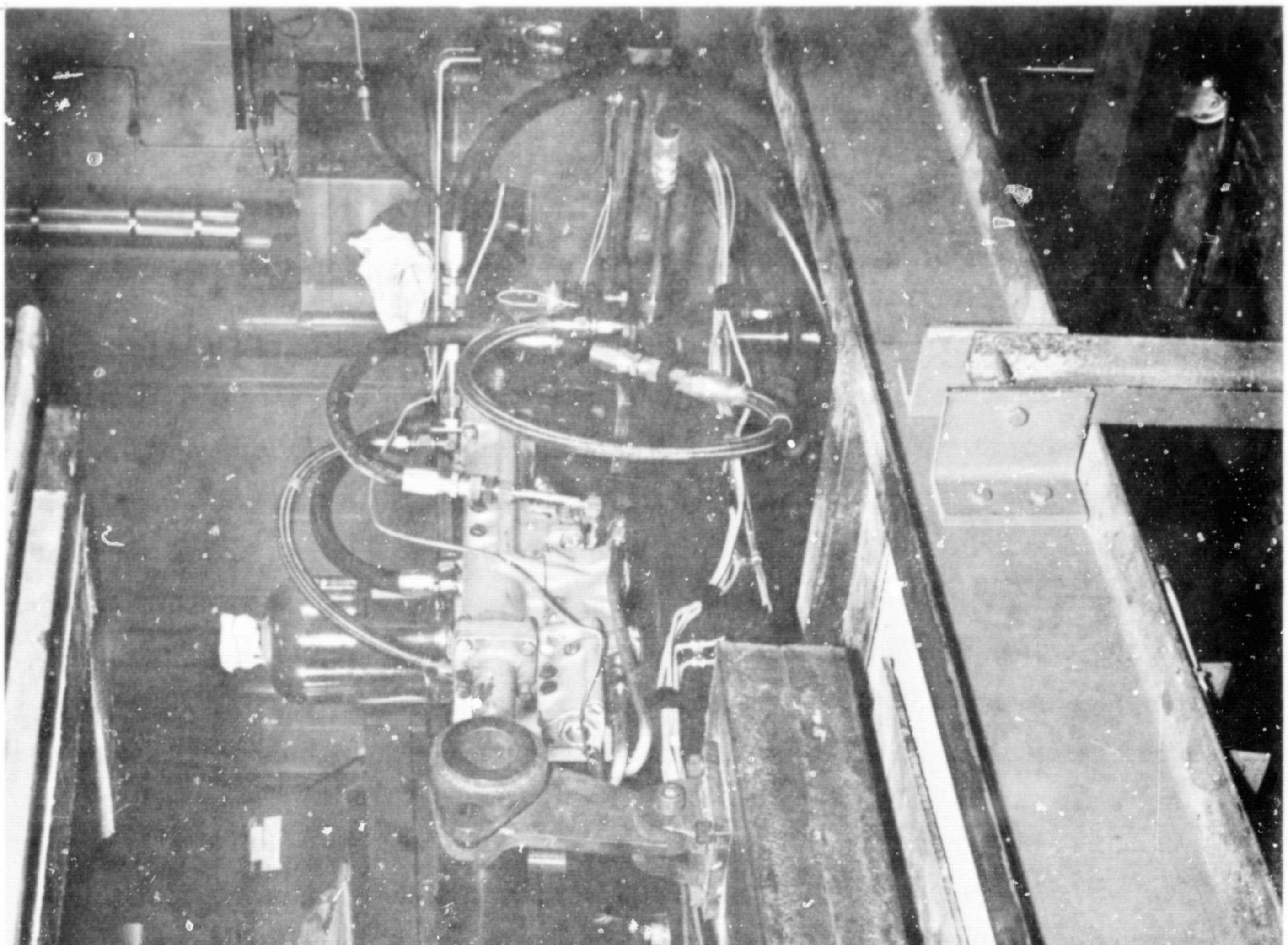


Figure 5-2. One-Inch Cycling Rig

is trunnion mounted in the oven and driven through a bell crank. The driving actuator located outside of the oven is a mechanical input servo controlled type. The servo valve input link is connected to an eccentric cam which is driven by a variable speed motor. Thus, the length of the piston rod stroke and cycling rate can be varied by adjusting the cam radius and motor speed, respectively.

A schematic of the hydraulic system for the test rig is shown in Figure 5-3. The system consists of a power circuit and a test circuit. The power circuit supplies pressure from a hydraulic mule to the driving actuator. The test circuit is comprised mainly of a booster pump, which is used for filling, and an accumulator for maintaining fluid pressure during testing. Leakage lines from the test actuator are brought outside of the oven to facilitate constant monitoring during testing. Thermocouples are strategically placed to measure ambient conditions, actuator skin temperature, and seal temperature. Thermocouples for monitoring seal temperatures are placed inside the actuator adjacent to the test seal.

Pressure gauges were installed in each cylinder chamber of the driving actuator to monitor the pressure required to drive the seal actuator. The pressures registered on the gauges indicated the resisting load on the driving actuator. This load consisted of the friction of the seals and the bearing friction in the crank arm. The pressure readings obtained at the start of the test served as base line values. An increase over this value would indicate friction build-up in the seals or bearings, which was cause for shut-down and subsequent determination of the source of the friction.

2. Seal Test Actuator

Three each of the one-inch and three-inch seal test actuators were designed and fabricated for the low-pressure testing.

A typical test actuator is shown in Figures 5-4 and 5-5. The actuator consists of a double-ended cylinder with removable seal glands on each end. The actuator housing is fabricated of 17-4PH corrosion-resistant steel. Piston rods are fabricated to Type 440C stainless steel and heat treated to a hardness of Rockwell C 52-53. The rod surface is plated with hard chromium

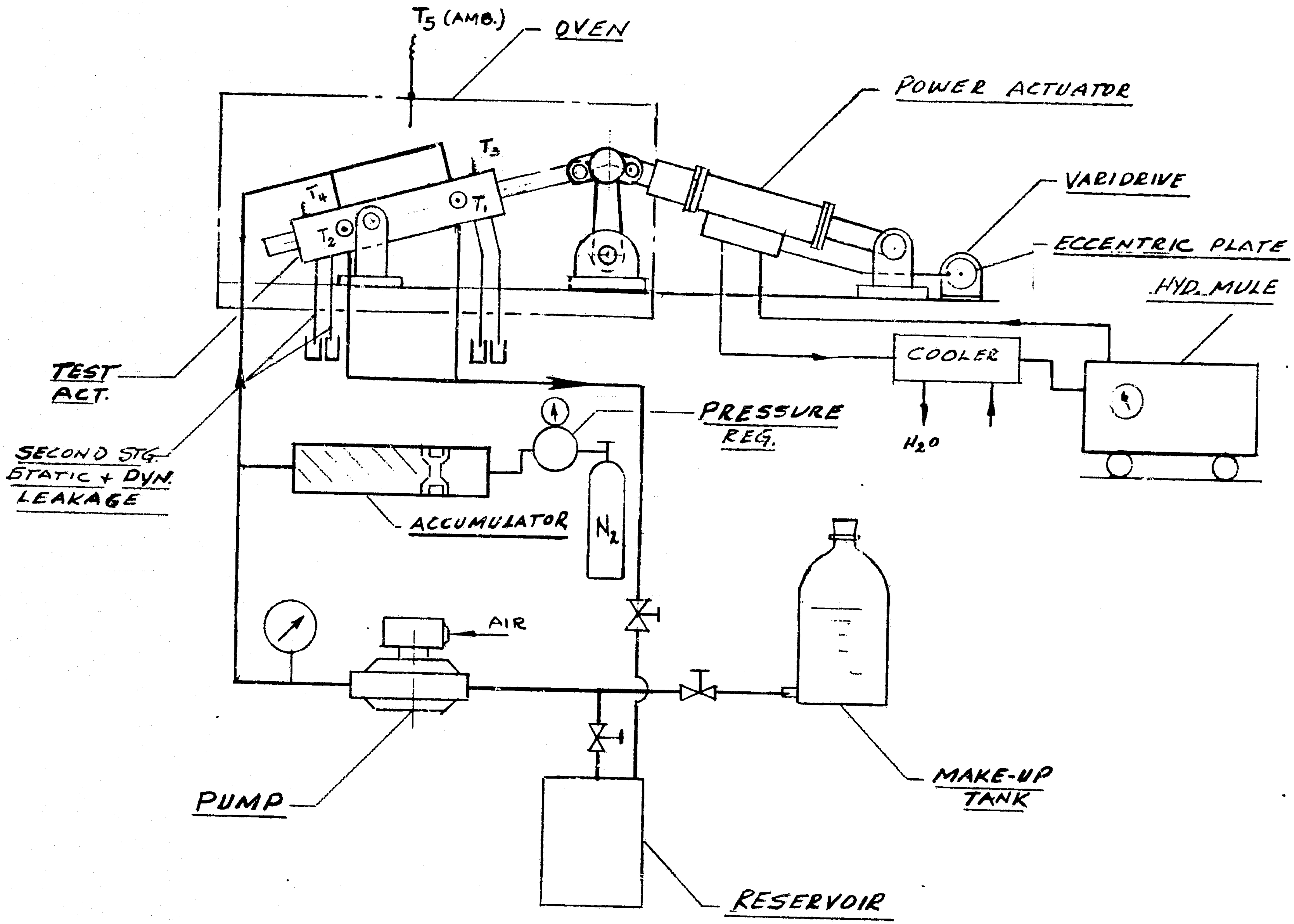


Figure 5-3. Hydraulic System Schematic

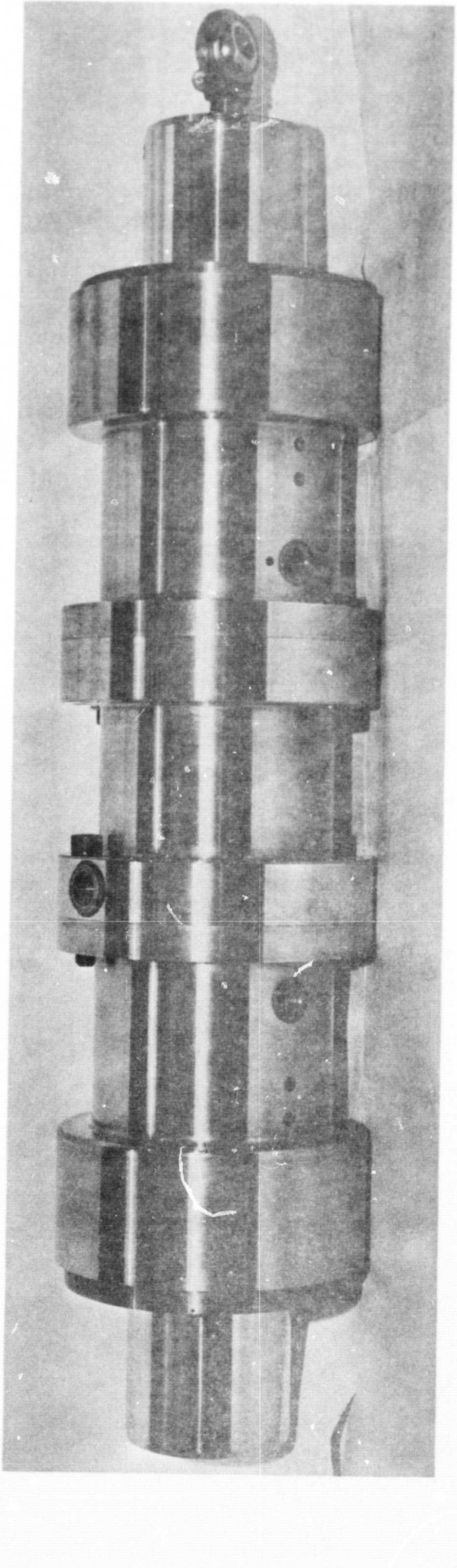


Figure 5-4. Test Actuator

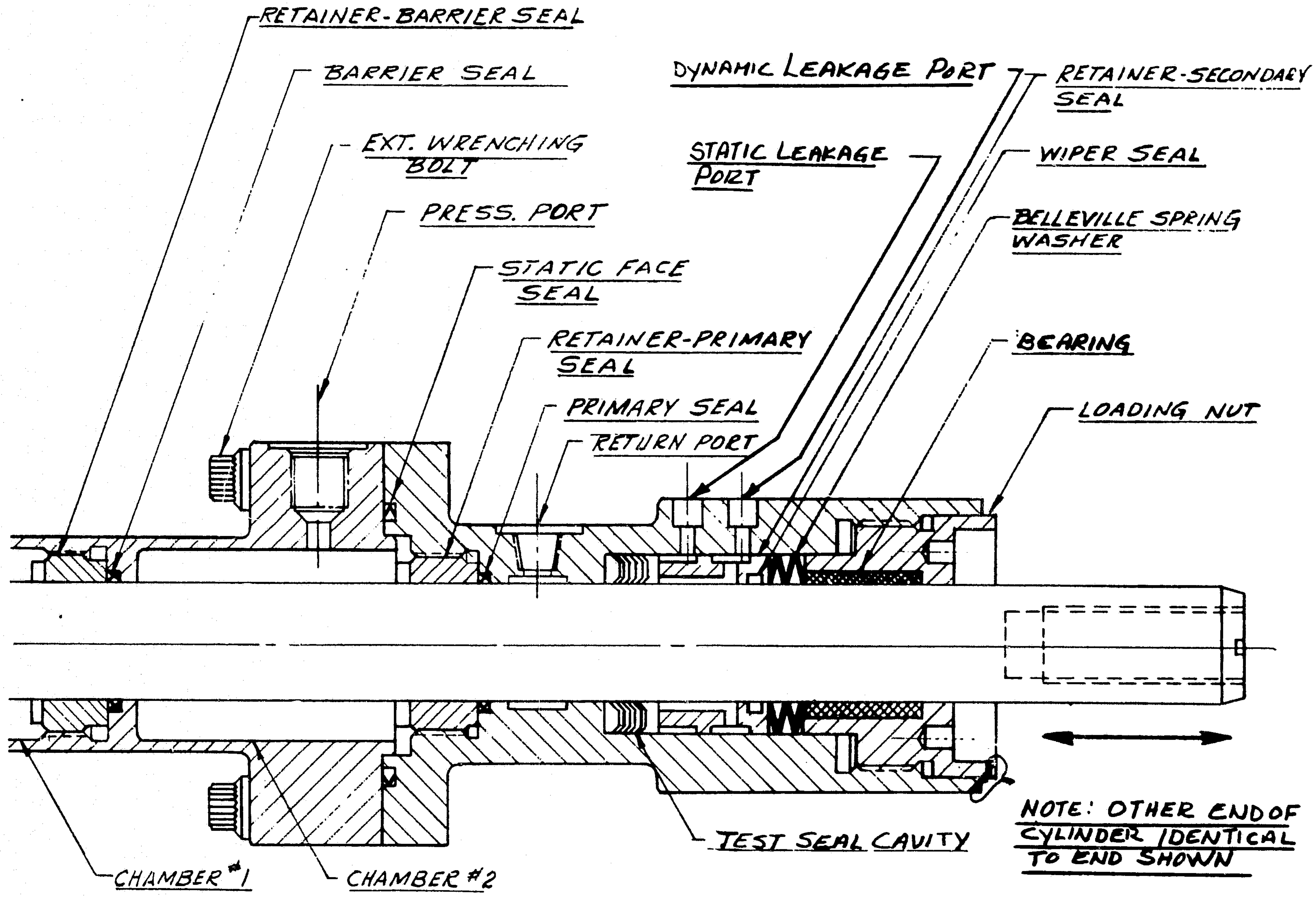


Figure 5-5. One-Inch Rod Seal Test Actuator

and ground to a finish of 4 RMS. Close-fitting graphite bearings are incorporated in each of the end glands to prevent metal-to-metal contact between the rod and gland. The seal cavity is designed to accommodate various seal configurations. Leakage ports are located so that leakage from the static and dynamic portion of the rod seal can be measured separately.

D. RESULTS OF LOW-PRESSURE SEAL TESTS

1. Summary

Results of the low pressure seal tests are summarized in Tables 5-1 and 5-2. In the one-inch rod size, two of the candidate designs (V-seal and reed seal) completed the 150-hour testing with exceptionally low leakage. The cobalt lip seal completed 50 hours each of testing at 400°F and 500°F, and 25 hours at 600°F with negligible leakage. However, the seal cracked (at 133.5 hours) before completing the 150-hour test. The Vascojet 1000 lip seal and the nickel Foametal wedge developed excessive leakage after 86.25 hours and 17.5 hours of testing, respectively. Of the three candidate seals tested in the 3-inch rod size, the V-seal and reed seal completed the 150 hour tests satisfactorily. The third candidate (cobalt molybdenum lip seal) cracked during initial pressure checks and was not tested.

2. Test No. 1 - Design B Polyimide V-Seal (one inch)

Detail design of the V-seal is shown in Figures 4-5, 4-6, and 4-7. The test configuration as shown in Figure 5-6 consisted of three V-seals, a loading ring, backup ring, and four 200-pound each spring washers. The springs were assembled in series and compressed to the 200-pound load, which produced a seal diametral deflection of 0.013 inch. Seal friction at 100 psi fluid pressure was 245 pounds total for both seals.

The seals completed the 400°F and 500°F operation with negligible leakage. At 600°F, seal leakage exceeded the allowable leakage rate of one drop per minute after 21.5 hours of operation. However, testing was continued for the full 50 hours at 600°F at a reduced cycling rate for the short-stroke cycles.

TABLE 5-1. SUMMARY OF LOW-PRESSURE SEAL TESTS - 1 INCH SEALS

Test No.	Seal Config.	Total Hours	Hours at Temp.	Total Cycles	Long Stroke Cycles at Temp.	Short Stroke Cycles at Temp.	Remarks
1	Design B V-Seal Polyimide	185.75	50/400°F 50/500°F 50.2/600°F	1,858,342	29,686/400°F 25,348/500°F 27,953/600°F	592,274/400°F 604,640/500°F 519,591/600°F	Test completed
2	Design D Lip Seal Vascojet 1000	86.25	50/400°F 15.5/500°F	648,190	24,949/400°F 7,891/500°F	443,282/400°F 137,594/500°F	Excessive leakage during 500°F cycling
3	Design I Wedge Seal Nickel Foametal w/BaF_2tCaF_2	17.5	7.5/400°F	100,248	4,225/400°F	75,423/400°F	Excessive leakage
4	Design D Lip Seal Cobalt molybdenum alloy	133.5	50/400°F 50/500°F 25/600°F	1,158,431	23,590/400°F 29,300/500°F 17,400/600°F	428,110/400°F 435,230/500°F 196,700/600°F	Seal cracked during 600°F cycling
5	Design AH Reed Seal Silver alloy and Vascojet 1000	174.5	50/400°F 50/500°F 58/600°F	1,671,131	23,590/400°F 29,300/500°F 36,900/600°F	428,110/400°F 435,230/500°F 503,200/600°F	Test completed

TABLE 5-2. SUMMARY OF LOW-PRESSURE SEAL TESTS - 3 INCH SEALS

Test No.	Seal Config.	Total Hours	Hours at Temp	Total Cycles	Long Stroke Cycles at Temp.	Short Stroke Cycles at Temp.	Remarks
6	Design B		50/400°F		34,500/400°F	412,850/400°F	Test completed
	V-Seal	158.5	50/500°F	1,497,452	25,095/500°F	527,695/500°F	
	Polyimide		50/600°F		22,840/600°F	436,920/600°F	
7	Design D	TEST NOT RUN - SEAL CRACKED DURING PRESSURE CHECK					
	Lip Seal						
	Cobalt molybdenum alloy						
8	Design AH		50/400°F		26,166/400°F	329,304/400°F	Test completed
	Reed Seal	174.0	50/500°F	1,221,400	26,420/500°F	356,211/500°F	
	Silver Alloy and Vascojet 1000		64/600°F		30,523/600°F	436,511/600°F	
	Design B (1)		100/400°F		60,666/400°F	742,154/400°F	Test completed
	V-Seal (2)	332.5	100/500°F	2,718,852	61,515/500°F	883,906/500°F	
	Polyimide		114/600°F		53,363/600°F	873,431/600°F	

(1) Seal Completed Previous Low Pressure Test

(2) Tested in Place of Cobalt Molybdenum Lip Seal in Test With Reed Seal

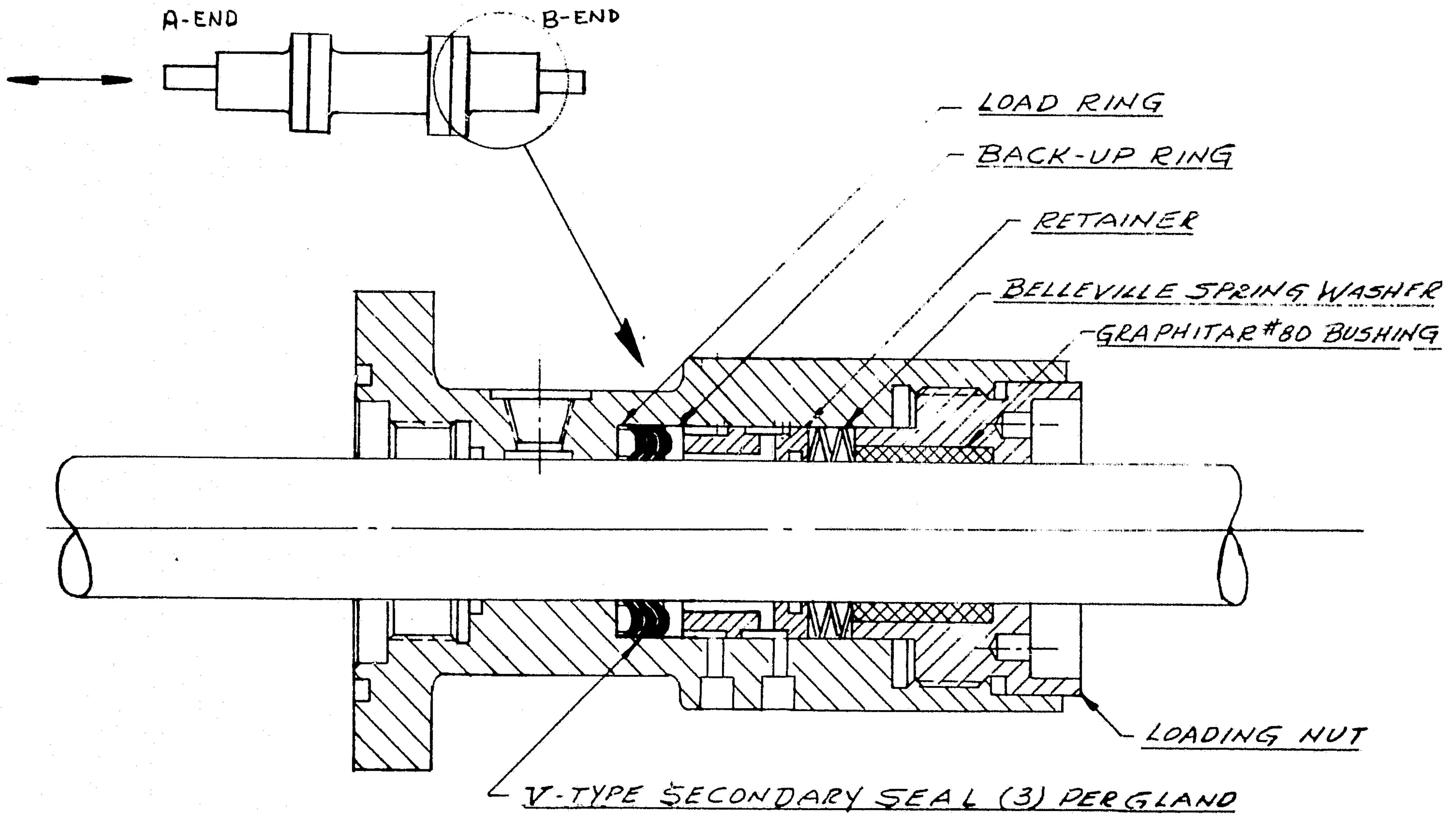


Figure 5-6. Polyimide V-Seal Gland Configuration

The only measurable leakage obtained during the 400°F testing was 0.19 cc from the seal on the B-end of the test actuator. Leakage measured during the 500°F operation consisted of 0.14 cc and 0.3 cc from the static and dynamic portions of the B-end seal, respectively. Leakage of 2 cc was accumulated from the static portion of the A-end seal.

For the first 21.5 hours of operation at 600°F, the accumulated leakage obtained consisted of 1.5 cc and 0.14 cc from the dynamic parts of the A-end and B-end seal, respectively. From that point on, leakage from both seals increased from 2 drops to 21 drops per minute.

Following a final check of the seal friction, the test actuator was disassembled (Figure 5-7) for inspection. Essential findings are discussed in the following paragraphs.

Seal friction increased from 245 pounds at the start of the test to 358 pounds after test. However, 115 pounds were attributed to friction from the Graphitar grade 80 gland bearings. Measurements taken of the bearings showed a decrease in their inside diameters which resulted in a clearance between the rod and bearing of approximately 0.0002 inch. Original bearing clearances were 0.0004 to 0.0006 inch. As the bearings were installed with a shrink fit into the gland housing, the decrease in the inside diameters was believed to be caused by slight stress-relieving of the bearing material. The decrease in the bearing clearance was further aggravated by the presence of caked hydraulic fluid in this area. (See Figure 5-8.)

Inspection of the Polyimide V-seals showed extrusion of the seal on the B-end of the actuator. The inner lip of the up-stream seal extruded into the clearance space formed by the load ring and piston rod. (See Figure 5-9.) This clearance measured approximately 0.014 inch. Although the same clearance gap was present on the A-end, the extrusion experienced by the A-end seal (Figure 5-10) was negligible. A check of the spring washers indicated that the spring load on the A-end (172 pounds) was less than the design load of 200 pounds; this could account for the the minimal extrusion on the A-end. The spring load on the B-end was approximately 209 pounds, which is very close to the design load.

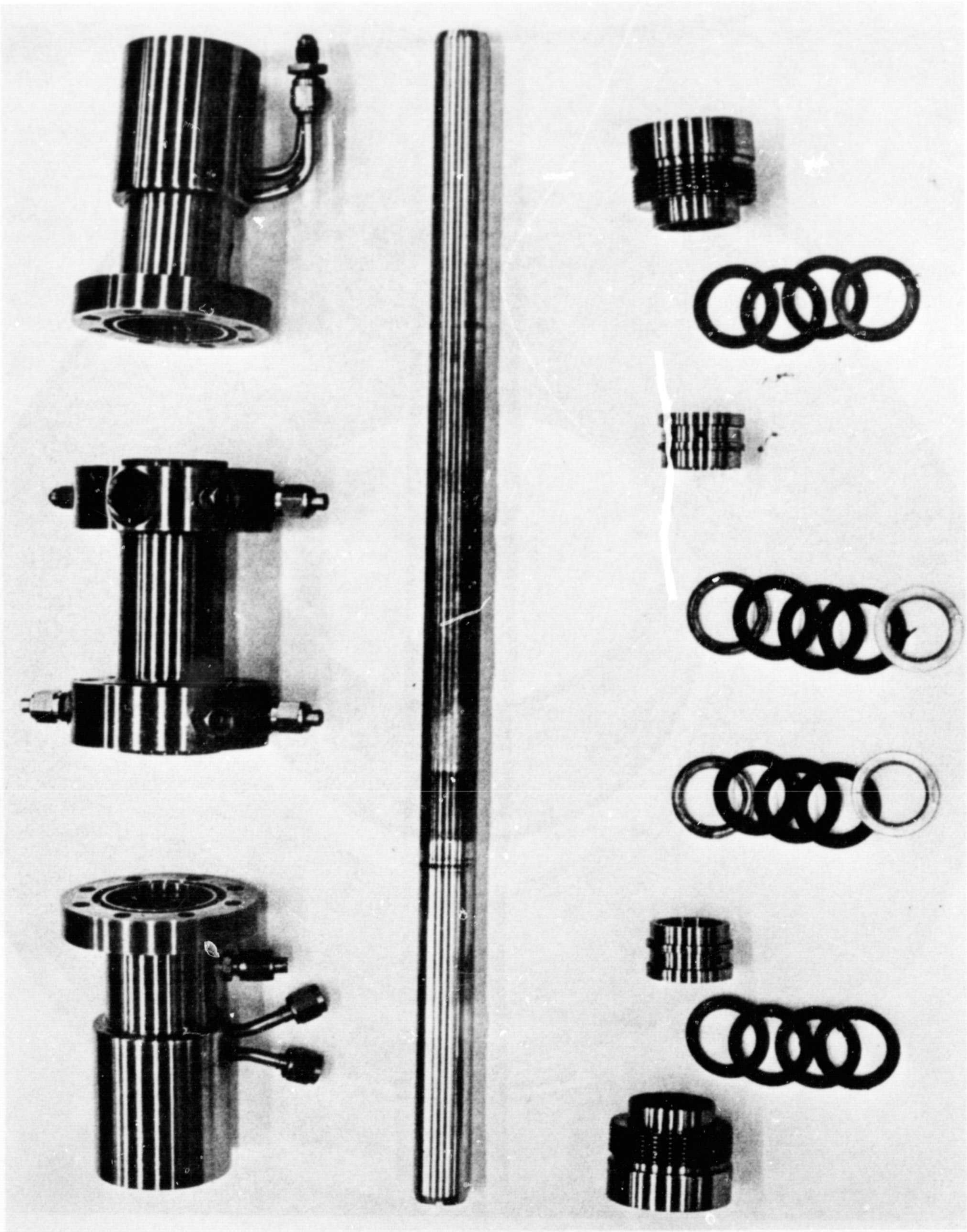


Figure 5-7. One-Inch Test Actuator - Polyimide V-Seal Test

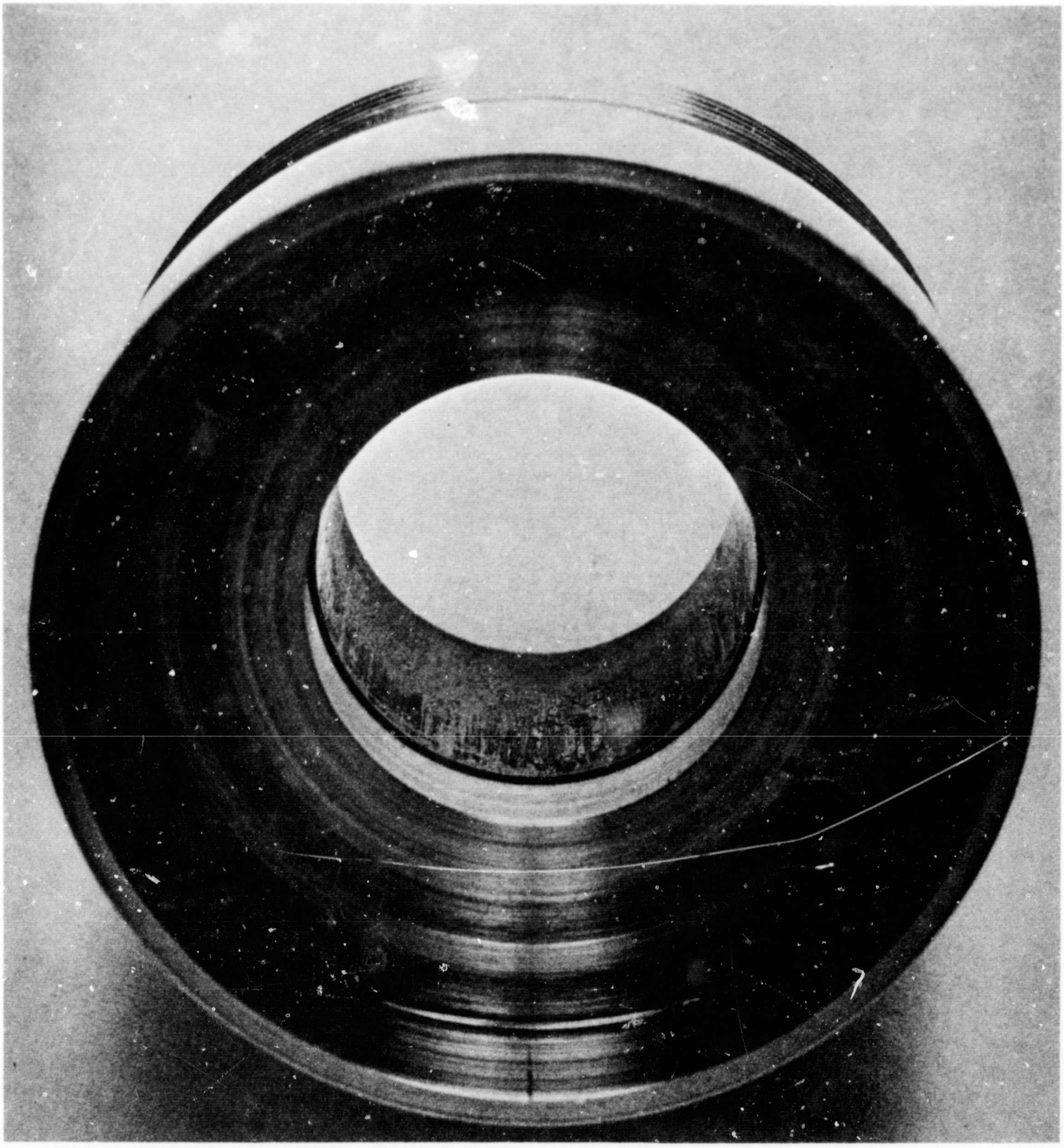


Figure 5-8. Rod End-Gland Bearing

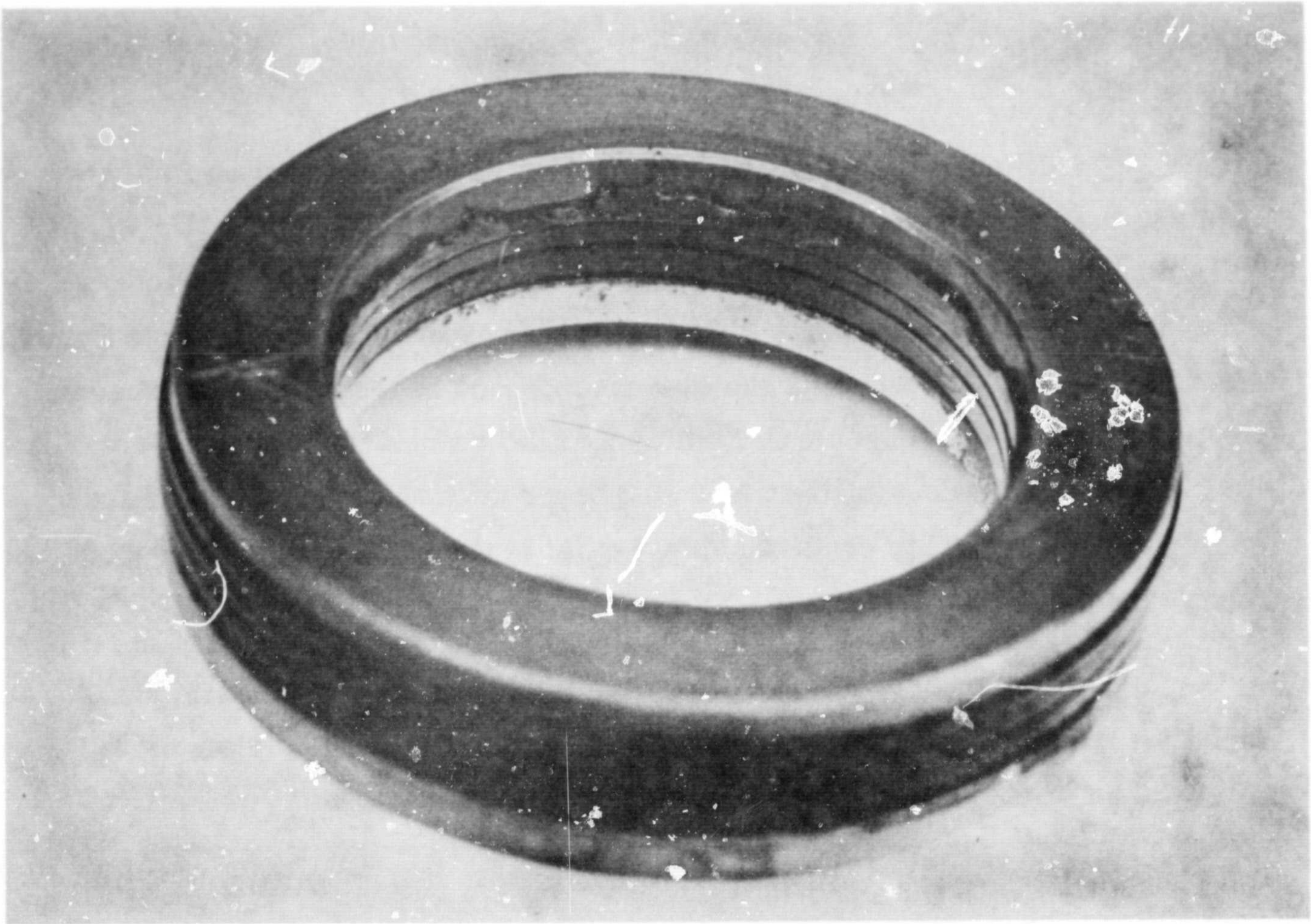


Figure 5-9. Polyimide V-Seal - B-End

Figure 5-10. Polyimide V-Seal - A-End

Although the higher spring load on the B-end resulted in seal extrusion, test data indicated less seal leakage than at the A-end seal, which had a lower spring load. It would appear that the design load of 200 pounds is necessary for effective sealing. The extrusion resulting from this spring load can be minimized by reducing the clearance space between the load ring and piston rod. The decreased clearance may also reduce the seal breakaway friction, as there is a possibility that a wedging condition of the seal was induced during assembly. This condition could exist if the sealing lip were forced into the clearance space.

Further inspection of the seals showed that the dynamic sealing surfaces were highly polished, indicating good contact with the piston rod. Dimensions taken of the seals before and after test were as follows:

	A-END				B-END			
	Before Test		After Test		Before Test		After Test	
Seal No.	I.D.	O.D.	I.D.	O.D.	I.D.	O.D.	I.D.	O.D.
1	.9923	1.5035	.9901	1.489	.9930	1.5025	*.9950	1.491
2	.9915	1.5032	.9902	1.491	.9927	1.5041	.9905	1.492
3	.9924	1.5024	.9910	1.491	.9922	1.5035	.9918	1.491

*Sealing lip badly extruded

With the exception of the No. 1 B-end seal, which was badly extruded (Figure 5-11), all the seals exhibited some decrease in their inside and outside diameters. The shrinkage due to exposure to elevated temperature is believed to be inherent in the Polyimide material. The shrinkage experienced at the inside diameter of the seal could be beneficial at room temperature as it provides a tighter fit around the piston. However, shrinkage of the outer diameter could result in loss of seal contact with the seal cavity at room temperature if adequate spring loading is not maintained.

The observations described above showed that, with the exception of extrusion on one of the three sealing elements on the B-end, all seals were in good condition. Excessive leakage at 600°F appears to be caused by differ-

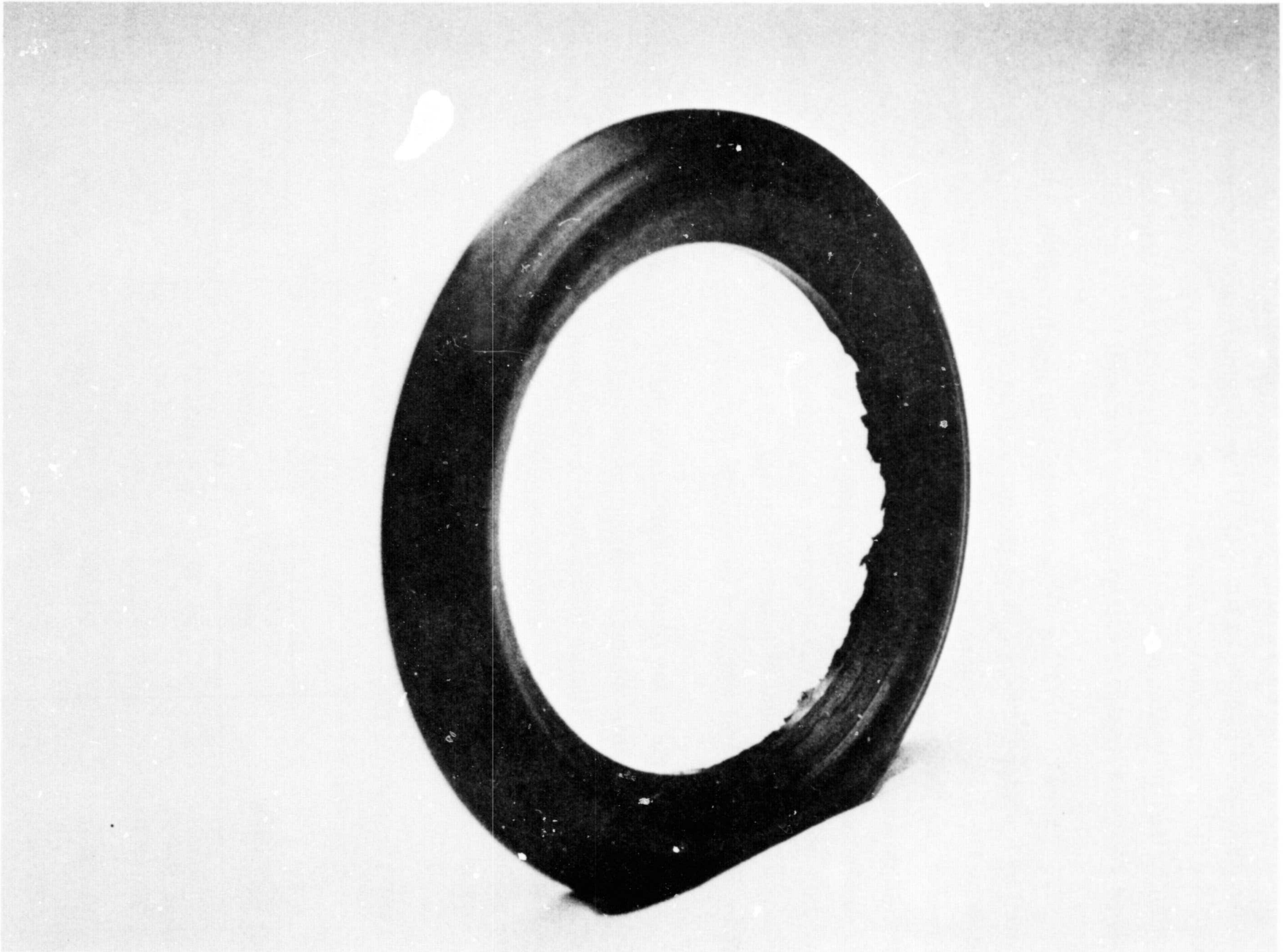


Figure 5-11. Extrusion of Polyimide V-Seal - B-End

ential expansion between the polyimide seals and the 440C pistion rod. Thermal coefficients of expansion (in./in./°F) of Polyimide and 440C are 30×10^{-6} and 5.8×10^{-6}, respectively. The lower spring load on the A-end did not provide sufficient deflection of the seal to compensate for the expansion. A higher leakage rate was therefore experienced by this seal.

Breakdown of the F-50 silicone fluid was indicated. This was evidenced by a sludge-like deposit on the internal part of the test actuator. As shown in Figure 5-12, a tacky coating (greenish in color) was deposited on the surface of the piston rod. This portion of the rod was in complete contact with the fluid during the test. A sample of the coating was scraped off the rod and sent to General Electric Companyfor analysis. Analysis of the coating by X-ray emission was inconclusive. However, the analysis did show that the scrapings contained small amounts of copper (0.3 mg), zinc (0.0002 mg), and a trace amount of iron. Total weight of the sample was approximately 0.4 mg.

Analysis of the fluid samples from the test is summarized in Table 5-3. Viscosity values obtained from the 400°F and 500°F run showed only a slight increase. The change in the acidic condition of the two samples was also minor as compared to the original value. Increases in viscosity and acidity were noticed in the fluid sample obtained from the 600°F run.

TABLE 5-3

LOW-PRESSURE TEST NO. 1 - ANALYSIS OF F-50 SILICONE FLUID

Condition	FLUID SAMPLES		
	50 hr at 400°F	50 hr at 500°F	50 hr at 600°F
Viscosity at 100°F (48.18 cs)	51.09	53.18	72.61
Viscosity at 210°F (15.9 cs)	17.14	17.20	24.96
Acid No. mg KOH/g (0.03)	0.05	0.02	6.2

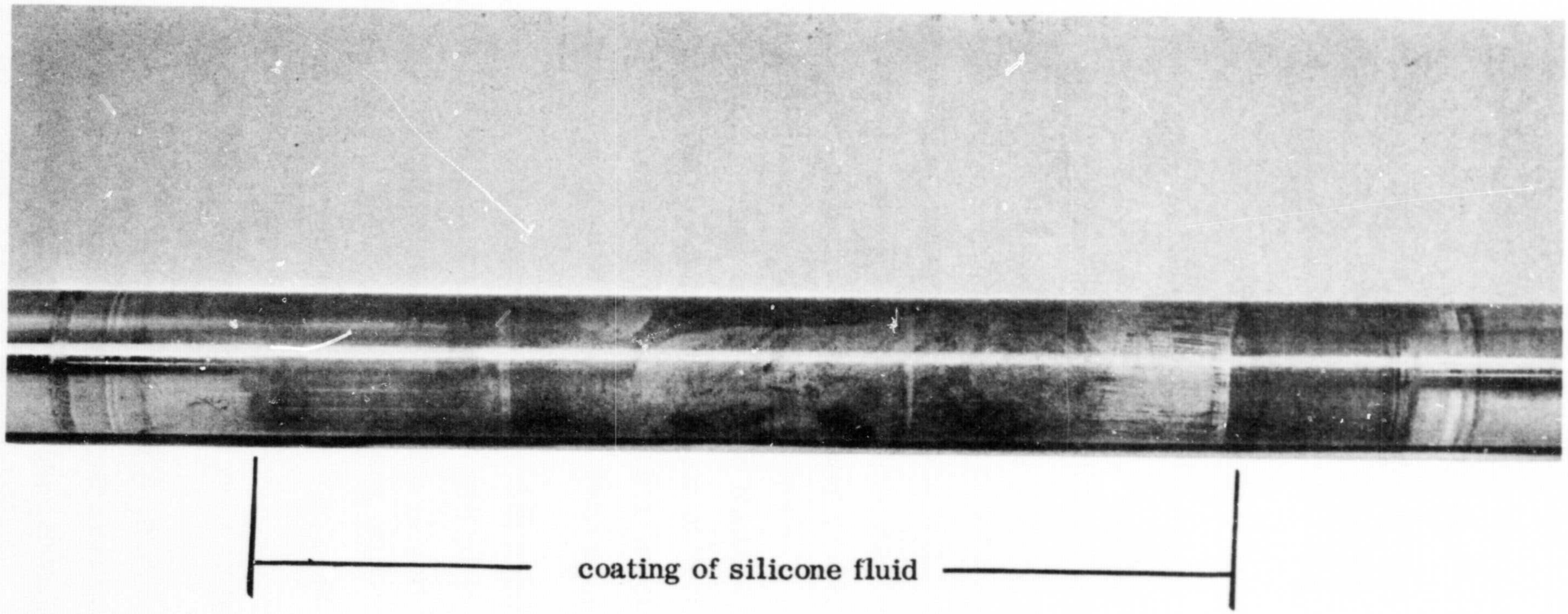


Figure 5-12. Piston Rod - Polyimide V-Seal Test (one-inch)

3. <u>Test No. 2 - Design D - Vascojet 1000 Lip Seal (one inch)</u>

The test configuration and details of the seal are shown in Figures 5-13 and 4-19. The lip seals were assembled onto the piston with an interference fit of 0.004 inch which produced a contact pressure at the seal interface of approximately 1200 psi. With the seals pressurized to 100 psi fluid pressure, the breakaway friction was 142 pounds for both seals.

Testing of this configuration was terminated after 86.25 hours of operation, when excessive leakage developed. As shown in Figures 5-14 and 5-15, the seals completed the 400°F operation with exceptionally low leakage. Total leakage during the 400°F operation was 4.6 cc and 8.9 cc for the A-seal and B-seal, respectively. However, during the 500°F operation, leakage started to increase and exceeded one drop per minute on both seals after 12.75 hours of operation. From that point on, leakage increased rapidly on the B-seal, reaching a leakage rate of 10 drops per minute. Accumulated leakage during the 500°F cycling (20.75 hours) was 17.2 cc and 126 cc for the A-seal and B-seal respectively. The excessive leakage was caused by loss of sealing load that, in turn, was a result of wear on the sealing lips. Wear was first suspected after the 400°F testing, when a check of the seal friction showed that it had decreased from an initial friction of 142 pounds to 30 pounds. Seal breakout friction remaining after test was 20 to 28 pounds for both seals.

The piston rod was examined prior to removal of the test actuator from the cycling rig by disconnecting the piston rod from the drive mechanism and extending it manually until the working portion of the rod in the A-end of the actuator was brought into view. Inspection revealed a wear pattern on the chrome-plated surface of the rod. The width of the wear pattern was approximately the length of the stroke used in the rapid short-stroke cycling portion of the test. A similar wear pattern was also noted on the B-end of the actuator.

At this point, it was felt that seal leakage might have occurred as the seal was sliding over the damaged area of the rod. Based on this premise, it was decided to conduct additional testing to determine if leakage was due primarily to the surface damage on the rod. This testing was accomplished by

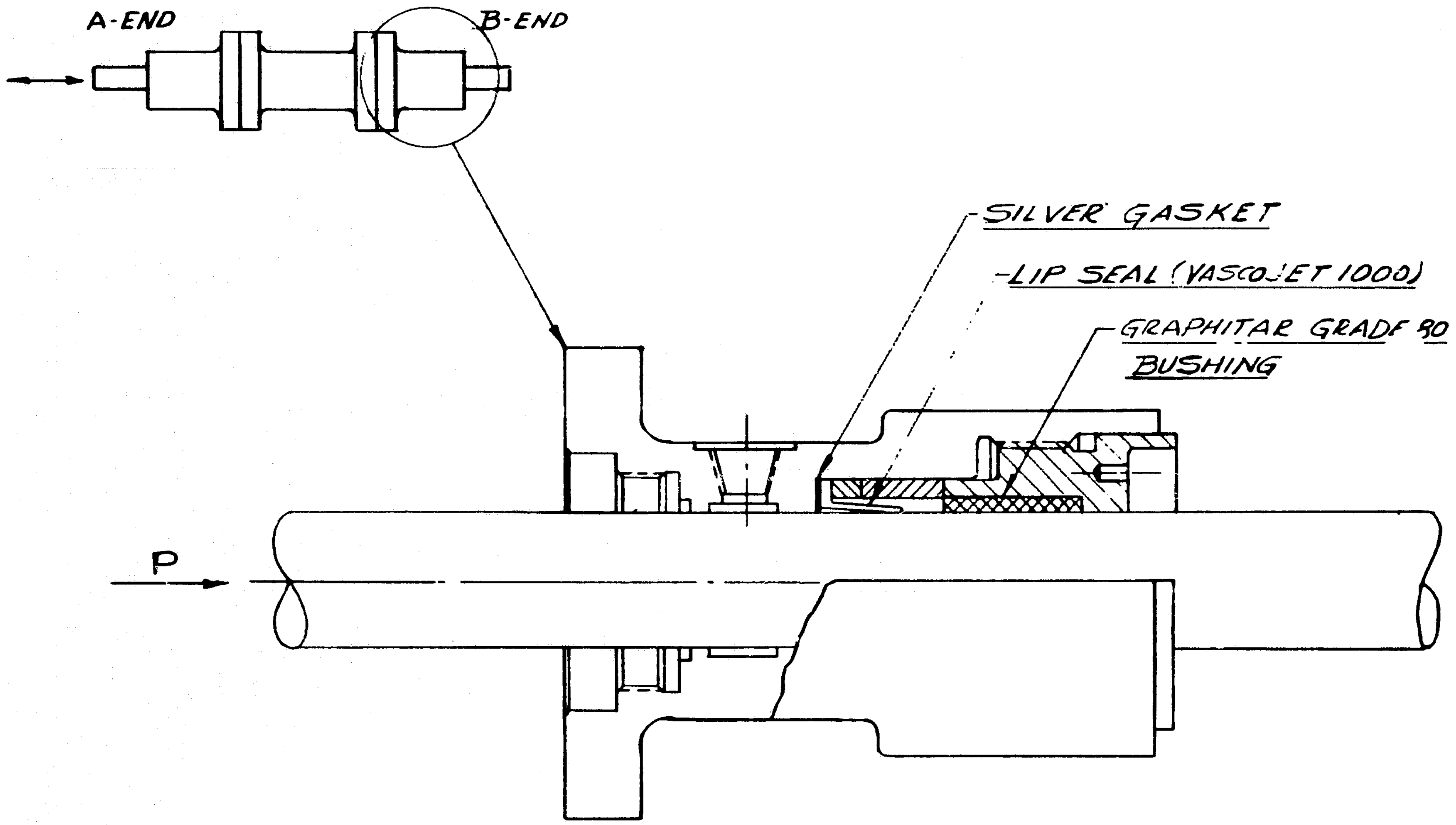


Figure 5-13. Vascojet 1000 Lip Seal Gland Configuration

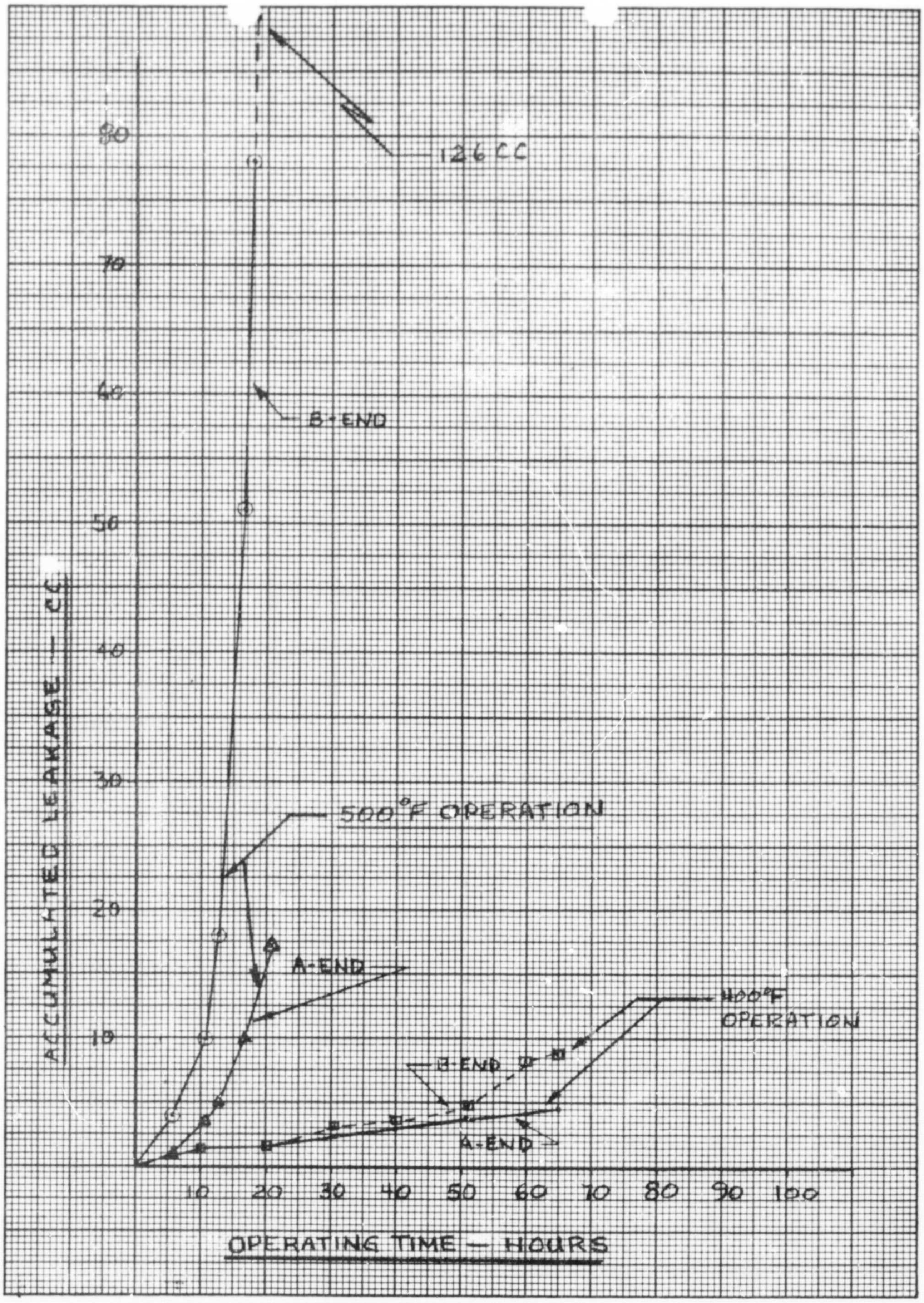


Figure 5-14. Accumulated Leakage Versus Time - Low Pressure Test No. 2 - Vascojet 1000 Lip Seal

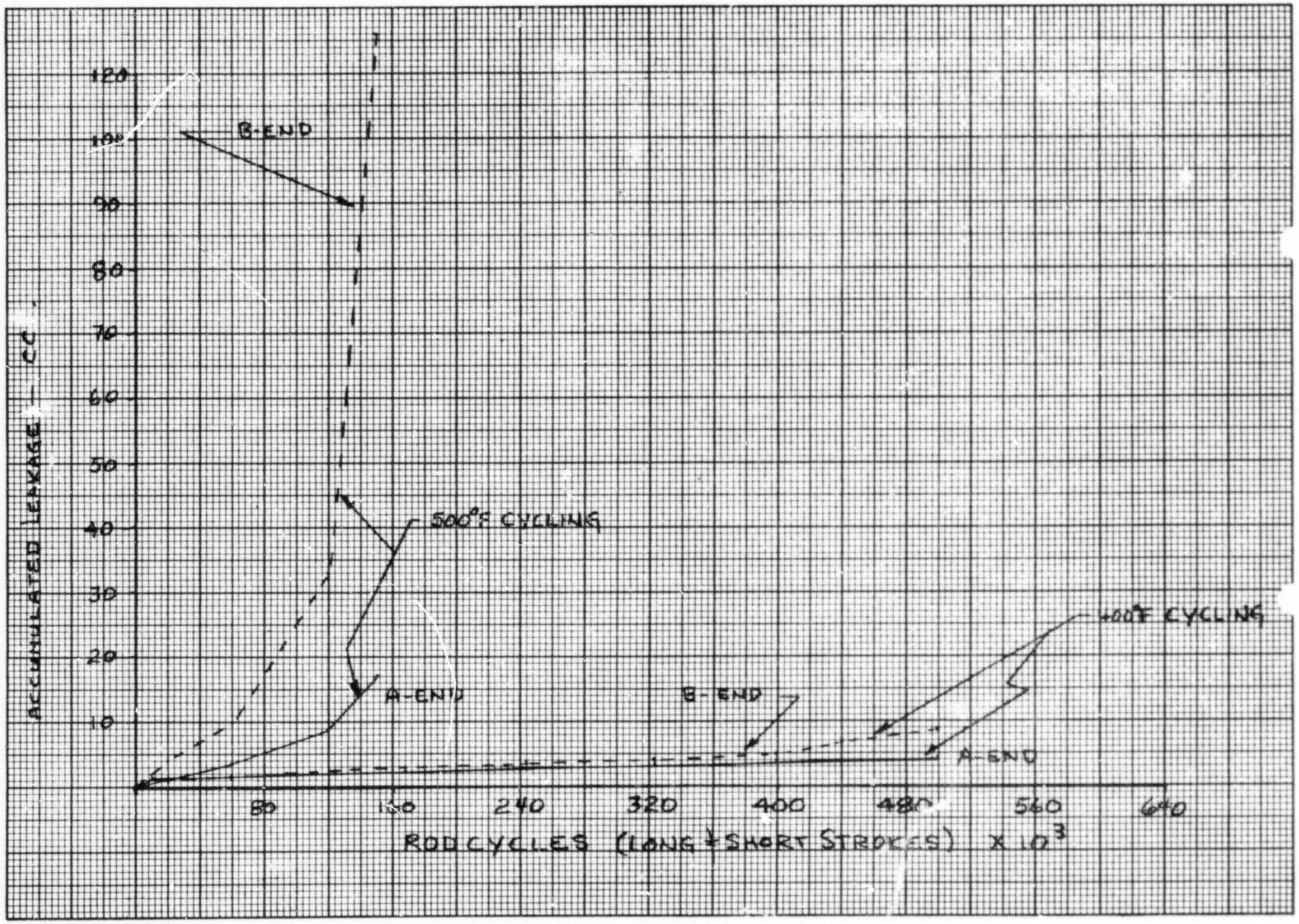


Figure 5-15. Accumulated Leakage Versus Rod Cycles - Low Pressure Test No. 2 - Vascojet 1000 Lip Seal

rigging the actuator so that the seal would be sliding on an unused portion of the rod. During the testing, leakage persisted and reached a total of 10.5 cc on the A-end and 60 cc on the B-end after 18900 cycles. Testing was then discontinued. Inspection of the rod showed that a similar wear pattern had developed on the chrome-plated surface.

Further inspection of the test parts (Figure 5-16) indicated that no catastrophic failure of the chrome plating had occurred. The chrome-plated rod (Figures 5-17 and 5-18) exhibited a highly burnished wear pattern produced by the long-stroke cycles. The absence of heavy wear indicated that an adequate fluid film had been generated under the seal. However, the rapid short-stroke cycling did produce a definite wear on the chrome-plated surface of the rod, due apparently to a lack of lubrication. As shown in Figure 5-19, the wear patterns generated consisted of a series of fine longitudinal scratches. The width of the wear pattern was equivalent to the length of the short-stroke cycle. Wear pattern on the B-end seal is shown in Figure 5-20. Although good contact was achieved over the entire seal circumference (Figures 5-21 and 5-22), there was evidence of uneven seal pressure. This was indicated by a variation in the width of the wear pattern on the seals. The width varied from 0.040 to 0.070 inch on both seals. This condition was believed to be caused by side loading of the piston rod due to normal gland bearing clearances. A slight out-of-roundness of the seals due to heat treating may have also contributed to the uneven wear pattern.

The effective contact pressure remaining on the seals was approximated by pressurizing them with benzene at room temperature until leakage was detected. The fluid pressure recorded at the point when leakage was produced indicated that the instantaneous seal contact load had been exceeded. Under these conditions, leakage on the A-seal was observed at a fluid pressure of 100 psi, and leakage on the B-seal was observed at a fluid pressure of 4 psi. These values, when considered as equivalent seal contact pressures, indicate a substantial reduction from the original effective contact load of 1200 psi.

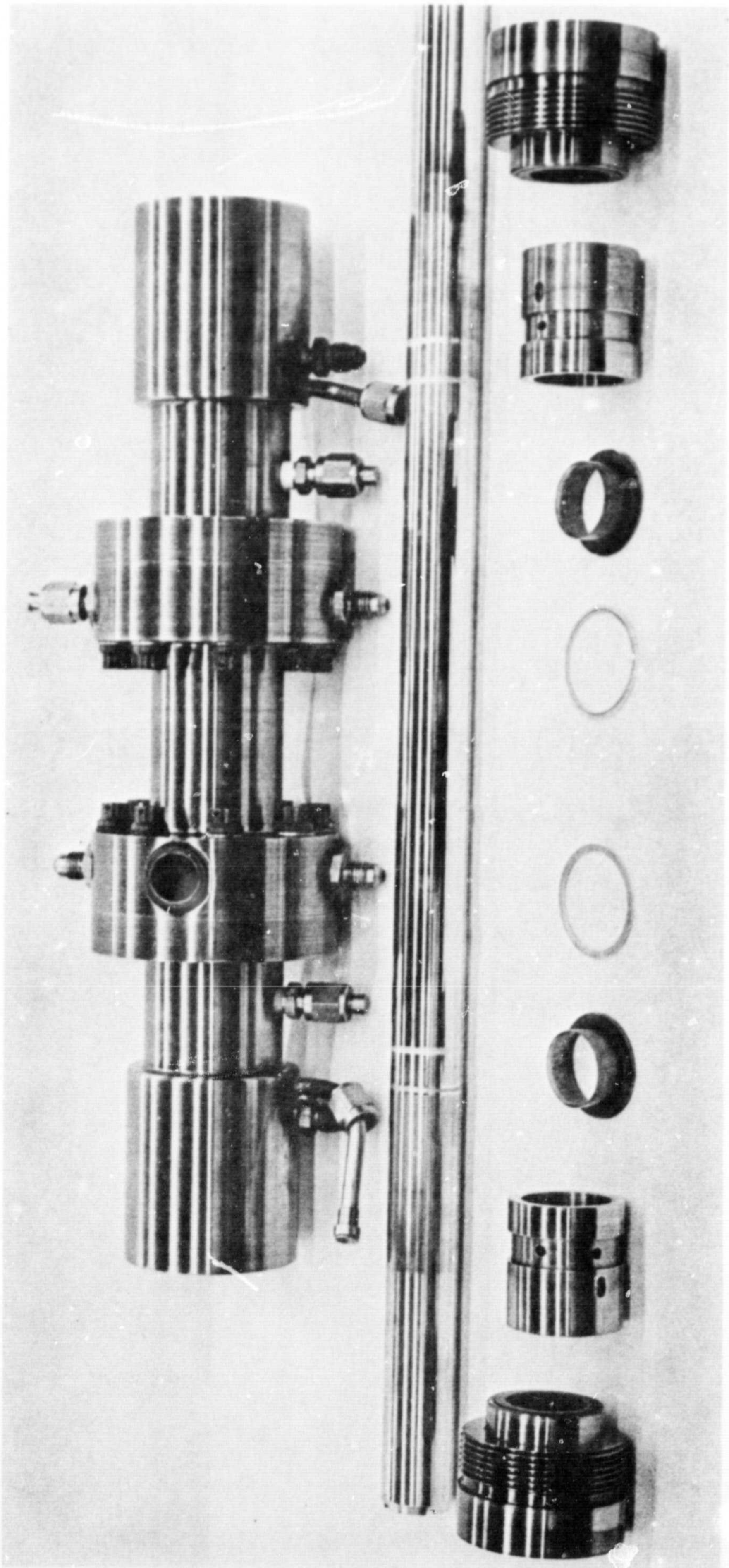


Figure 5-16. One-Inch Test Actuator - Vascojet 1000 Lip Seal Test

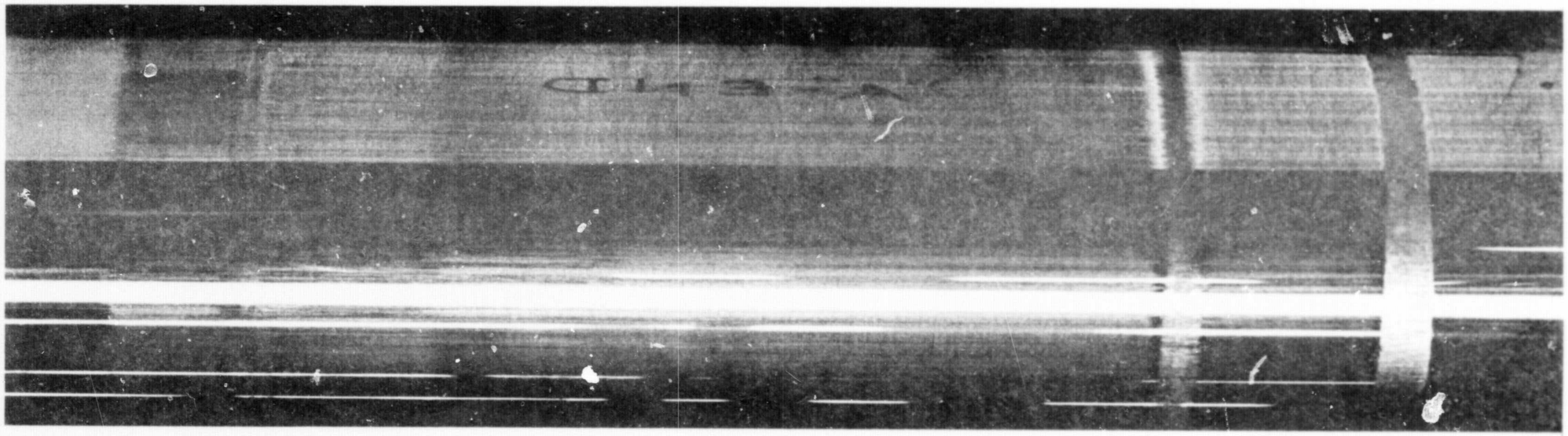


Figure 5-17. Piston Rod A-End - Vascojet 1000 Lip Seal Test

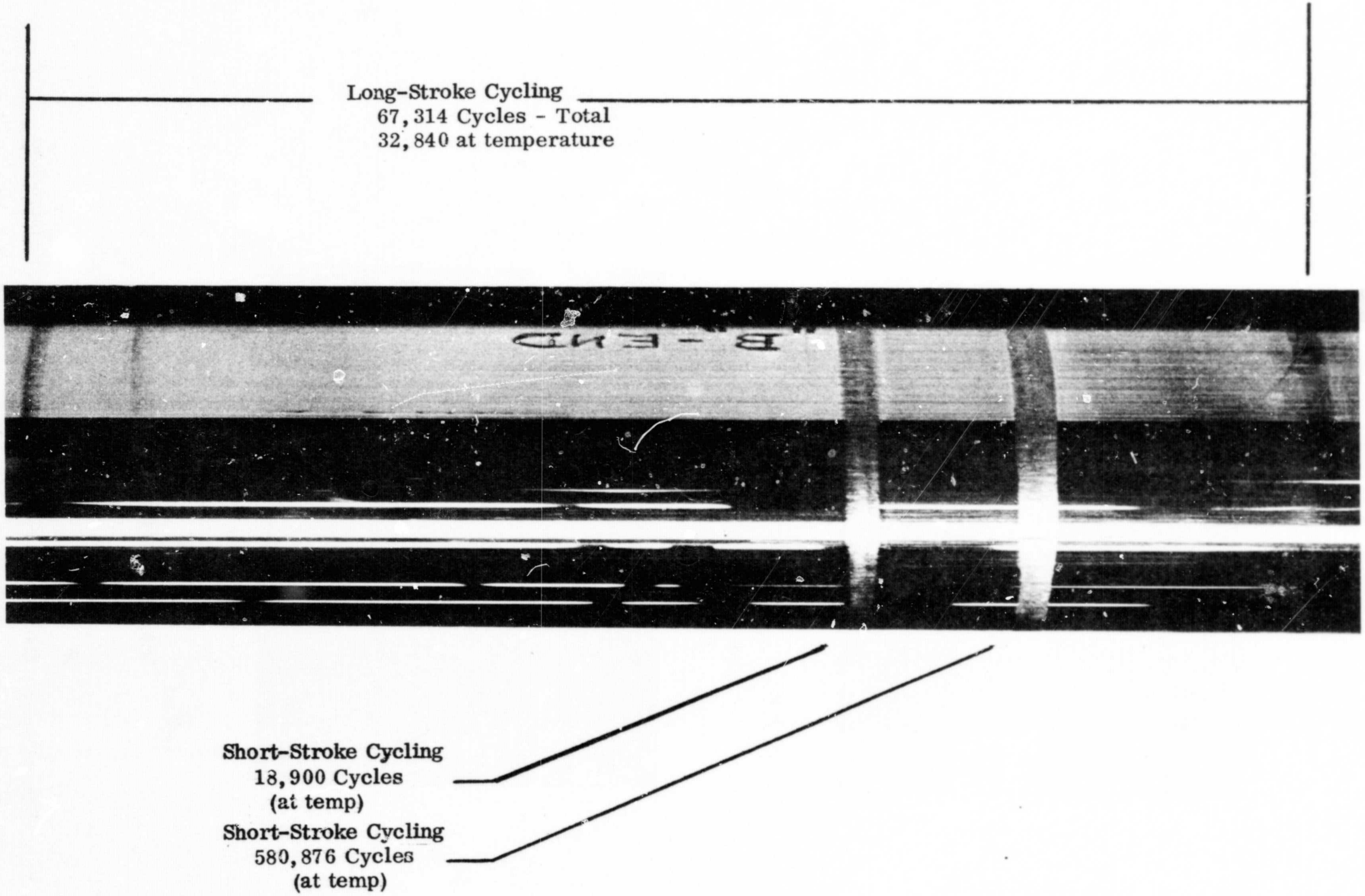
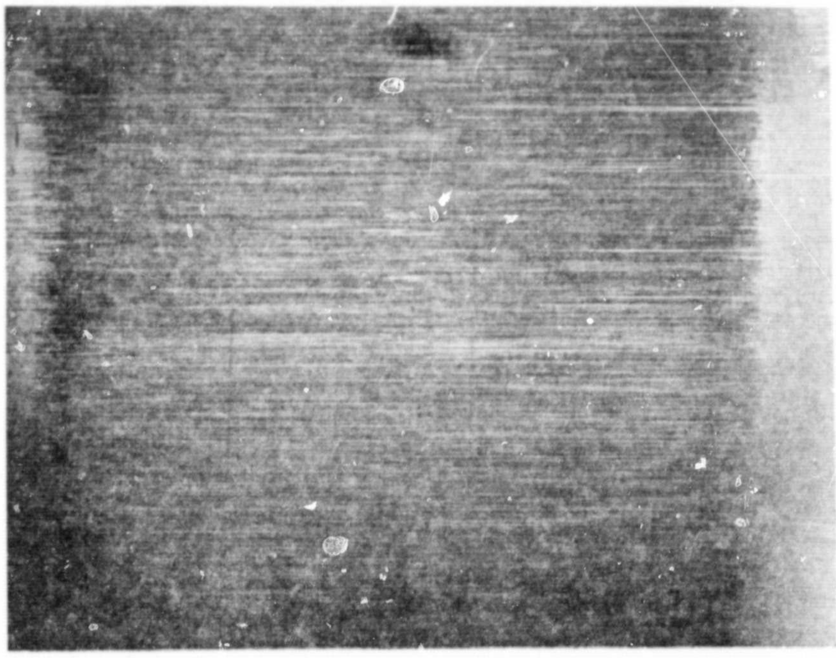
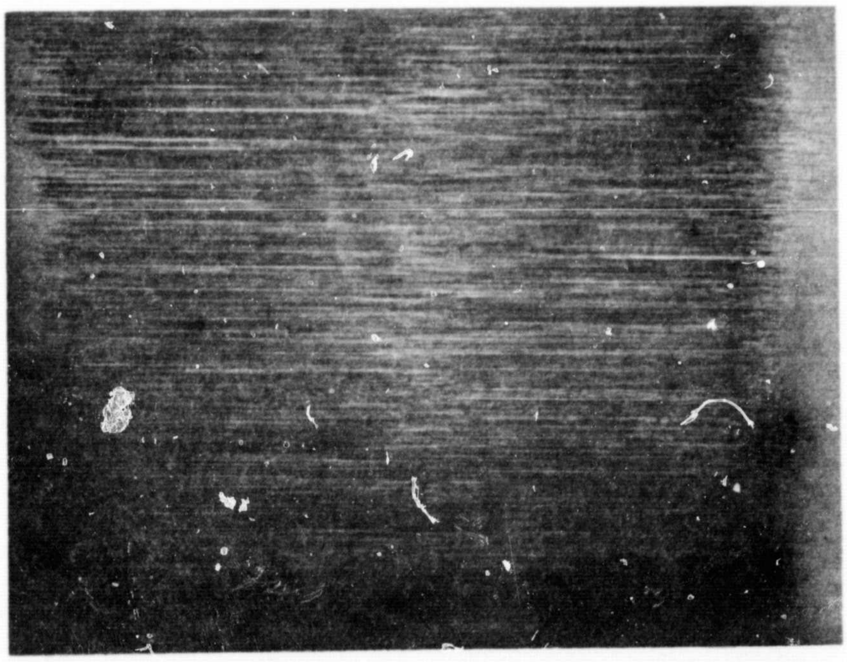


Figure 5-18. Piston Rod B-End - Vascojet 1000 Lip Seal Test



A-End 25X

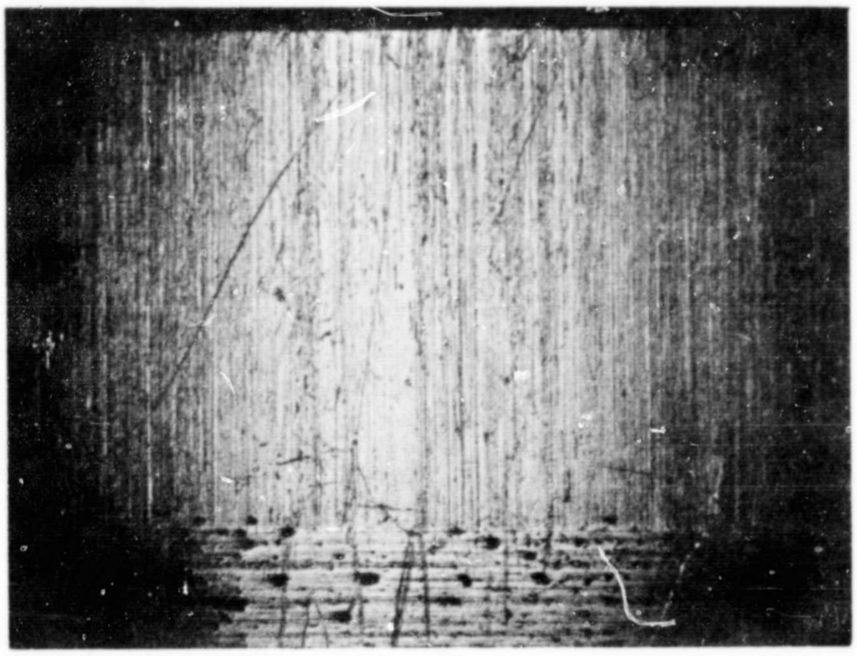
580,876 Short-Stroke Cycles



B-End 25X

580,876 Short-Stroke Cycles

Figure 5-19. Short-Stroke Cycling Wear Pattern on Piston Rod



Sealing Surface

25X

Figure 5-20. Wear Pattern on Vascojet 1000 Lip Seal - B-End

Inspection and measurements taken of the piston rod show that a wear of 0.0002 inch had taken place at the site where the short-stroke cycling was performed. No measurable wear was obtained on the rod surface subject to long-stroke cycling. A check of the chrome-plated surface with a superficial hardness tester revealed a hardness of Rc 64-65.

The wear exhibited by the chrome-plated piston rod and seals was primarily caused by inadequate lubrication during the short-stroke cycling operation. It is believed that seal life can be substantially extended if lubrication can be maintained during short-stroke cycling.

Analysis of the fluid samples obtained from this test is summarized in Table 5-4. Viscosity values obtained for the fluid samples from the 400°F and 500°F runs show a slight increase from the original values. The change in acidity of the 400°F sample was somewhat higher than that determined for the fluid from the 500°F run. However, it should be noted that the fluid in the 500°F run was subjected to only 15.5 hours at that temperature.

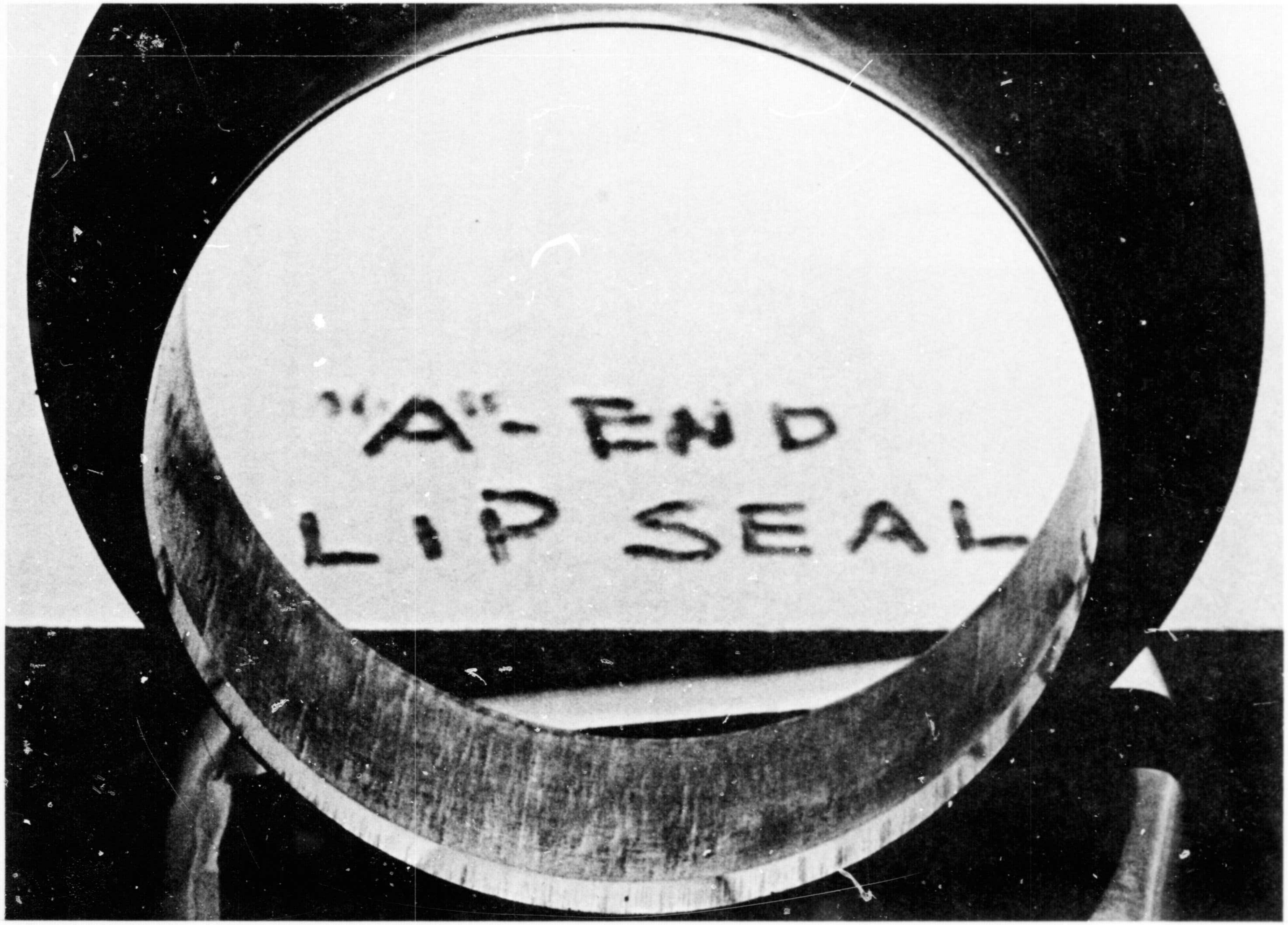


Figure 5-21. Vascojet 1000 Lip Seal - A-End

Figure 5-22. Vascojet 1000 Lip Seal - B-End

TABLE 5-4

LOW-PRESSURE TEST NO. 2 - ANALYSIS OF F-50 SILICONE FLUID

Condition	FLUID SAMPLES	
	50 hr at 400°F	15.5 hr at 500°F
Viscosity at 100°F (48.18 cs)	54.7	52.55
Viscosity at 210°F (15.9 cs)	17.83	17.23
Acid No. mg KOH/g (0.03)	0.11	0.02

Based on the results obtained with this seal configuration, a series of short-duration, short-stroke cycling tests was conducted. The purpose of these tests was to obtain a better determination of the mode of failure and to obtain additional data for possible redesign of the seal. The areas investigated in these tests were 1) incipient wear and 2) the effects of lower seal contact pressure and cycling rate on seal wear.

This testing was conducted with the actuator assembled with a Vascojet 1000 lip seal having a lip thickness of 0.0055 inch and an interference fit of approximately 0.004 inch. This configuration produced a seal contact pressure of 700 to 800 psi as compared to 1200 psi for the seal used in Low-Pressure Test No. 2. In order to duplicate the original test conditions as closely as possible, the piston rod used in Low-Pressure Test No. 2 was used. The actuator was rigged so that the seal would reciprocate on an unused part of the piston rod. Because of a limited supply of lip seals, a set of Polyimide V-seals was installed on the B-end of the actuator. Prior to actual testing, the seals were subjected to 10,000 run-in cycles at a two-inch stroke.

Testing was conducted at room temperature. The piston rod stroke was ± 0.09 inch. Each test was run for a total of 18,000 cycles. For each new run, the piston rod was shifted to a different position. Results of three separate runs are summarized as follows:

a) Run No. 1 - Incipient Wear. This run was made at 300 cpm with the seals pressurized to 100 psi. Inspection of the rod after 18,000 cycles revealed a definite wear pattern on the chrome plate. As shown in Figure 5-23, wear was evidenced by a series of closely spaced longitudinal scratches.

b) Run No. 2 - Random Cycling Rate. This run was made at 100 psi with a random cycling rate that varied from no cycling to 300 cpm. Observations made after 18,000 cycles indicated a slight wear pattern on the chrome plate. The randomly spaced longitudinal scratches (Figure 5-24) were barely visible. It was also noted that the scratches varied in length. The reason for the variation in length was that, at the higher cycling rate, the servo actuator drive experienced some decay in response, which resulted in slight loss of output motion. This condition produced a stroke that was slightly shorter than that obtained at the lower cycling rate.

c) Run No. 3 - Decreased Seal Contact Pressure. This run was made with the seal pressurized to 300 psi instead of 100 psi. The relieving effect of the higher fluid pressure reduced the seal contact load to approximately 400 psi. The cycling rate used was 300 cpm. The wear marks (Figure 5-25) produced after 18,000 cycles were much less pronounced than those exhibited in Run No. 1. However, they were more pronounced than those of Run No. 2.

The results of the above testing show that incipient wear occurs after a relatively short period of seal operation. A comparison of the wear patterns generated shows that the wear is more pronounced under conditions of cycling at

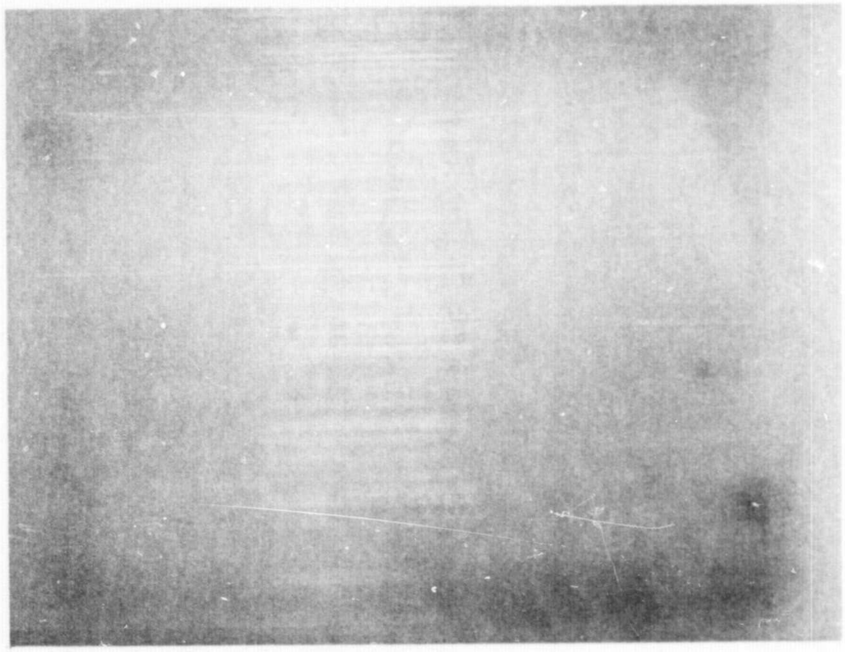


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Figure 5-23. Run No. 1 - Steady Cycling (300 cpm), 100 psi Fluid Pressure

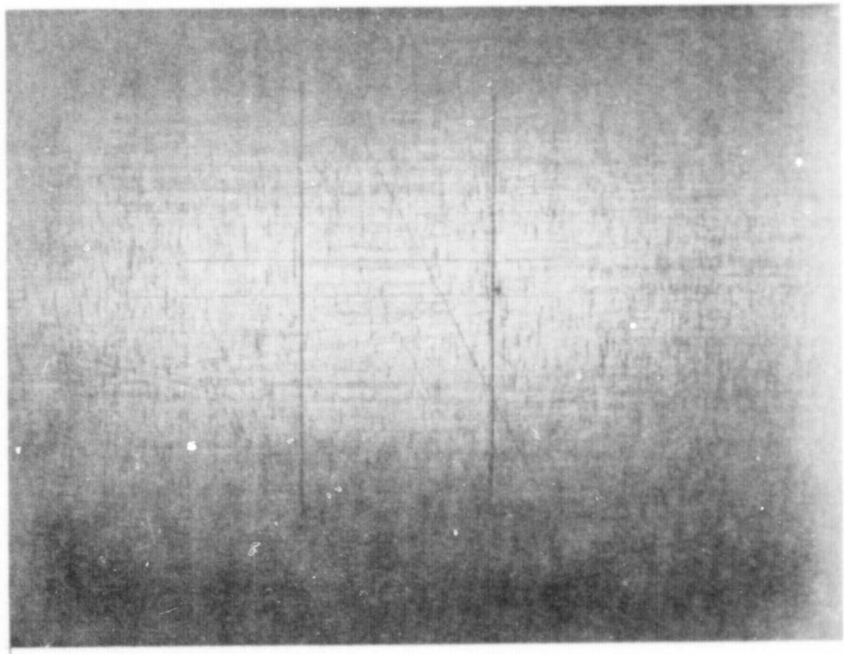

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Figure 5-24. Run No. 2 - Random Cycling (0 to 300 cpm), 100 psi Fluid Pressure

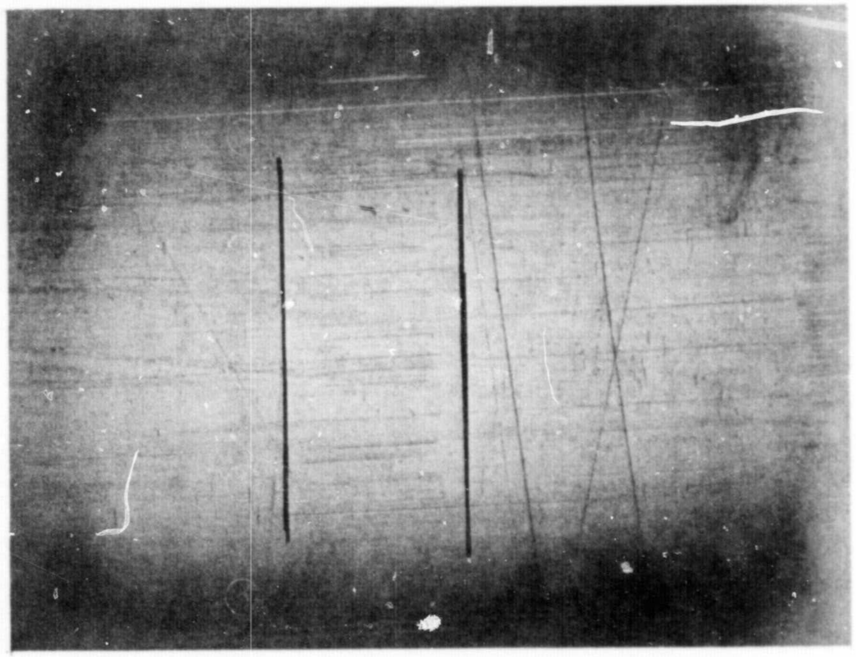

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Figure 5-25. Run No. 3 - Steady Cycling (300 cpm), 300 psi Fluid Pressure constant rate and amplitude at 100 psi. However, under the same cycling conditions but with the seal subjected to a higher fluid pressure (300 psi), the amount of wear is reduced considerably. This is attributable to the lower seal contact load resulting from the higher fluid pressure. It is also possible that the fluid had a greater tendency to leak at the higher pressure, thus providing better lubrication under the seal. Operation of the seal under a random cycle pattern produces the least amount of wear. Although establishing the exact reasons for this would require further investigation, it is believed that the varying cycling rate reduces the amount of local heating at the seal interface. It is also probable that during short periods of no cycling, greater seepage of fluid occurs past the seal and provides a better lubricating film during subsequent operation.

4. Test No. 3 - Design I - Nickel Foametal Wedge Seal (one inch)

The gland configuration for this test is shown in Figure 5-26. Figure 4-22 depicts the design of the wedge seal. In the fabrication of the seal, the surface pores of the nickel Foametal (see Figure 5-27) were left open to expose the impregnant (calcium fluoride-barium fluoride eutectic) to the sliding surface.

For this test, the design load of 120 pounds was used, which produced a circumferential seal load of approximately 50 pounds per inch. The resulting contact pressure at the seal interface was approximately 830 psi. The seal friction resulting from these loading conditions was 50 pounds. As only the A-end of the test actuator was reworked to accept the wedge seal, a set of polyimide V-seals was installed in the B-end.

Testing of the wedge seal was discontinued after 17.5 hours of operation when excessive leakage developed. Leakage of 16 dpm was noticed after 66,700 cycles of operation. At this point an attempt was made to increase the seal load by tightening the seal gland nut. However, this did not result in any appreciable decrease in leakage. Finally, after 100,248 cycles, leakage became excessive and testing was concluded.

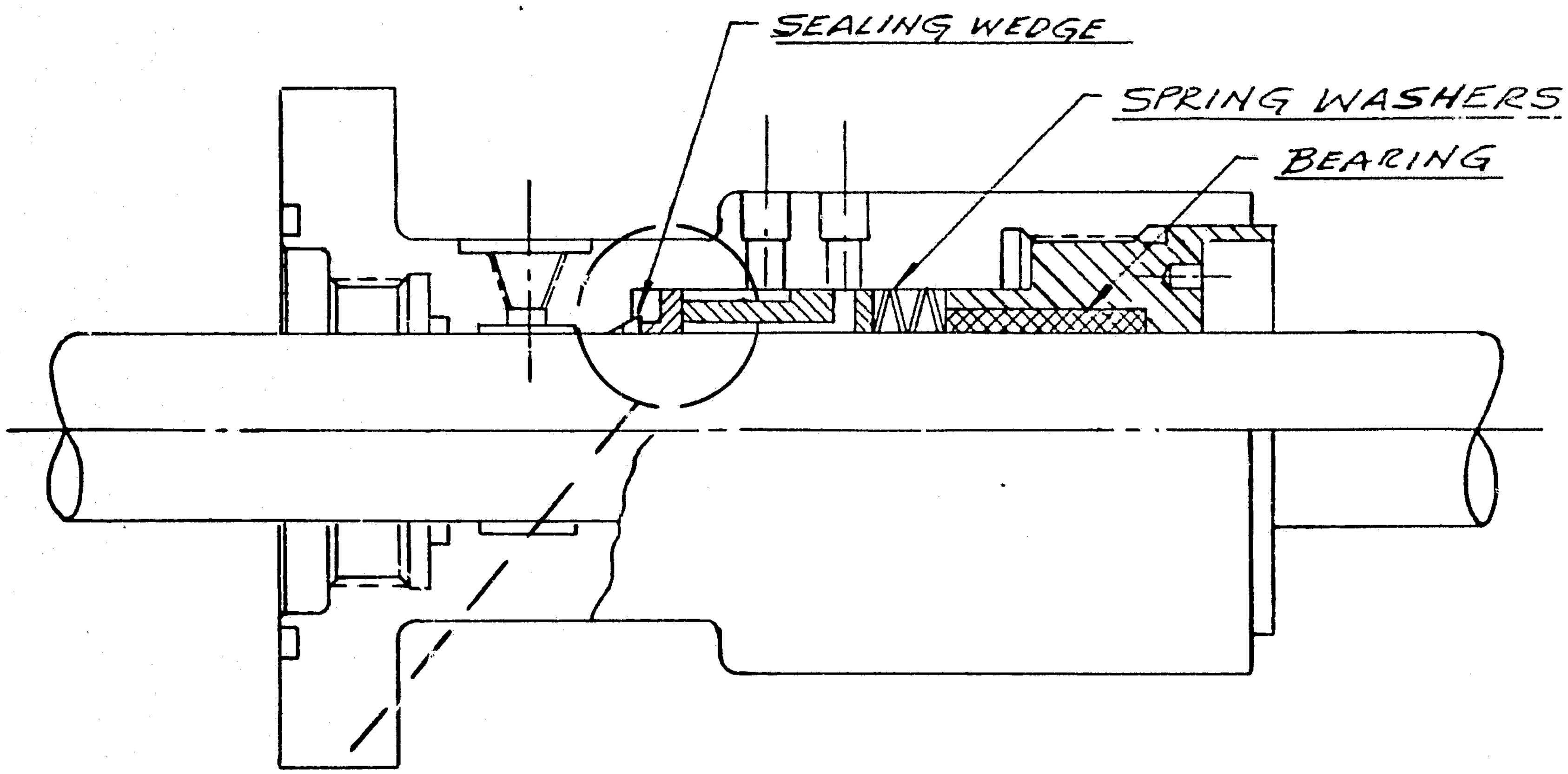

Figure 5-26. Gland Configuration - Nickel Foametal Wedge Seal

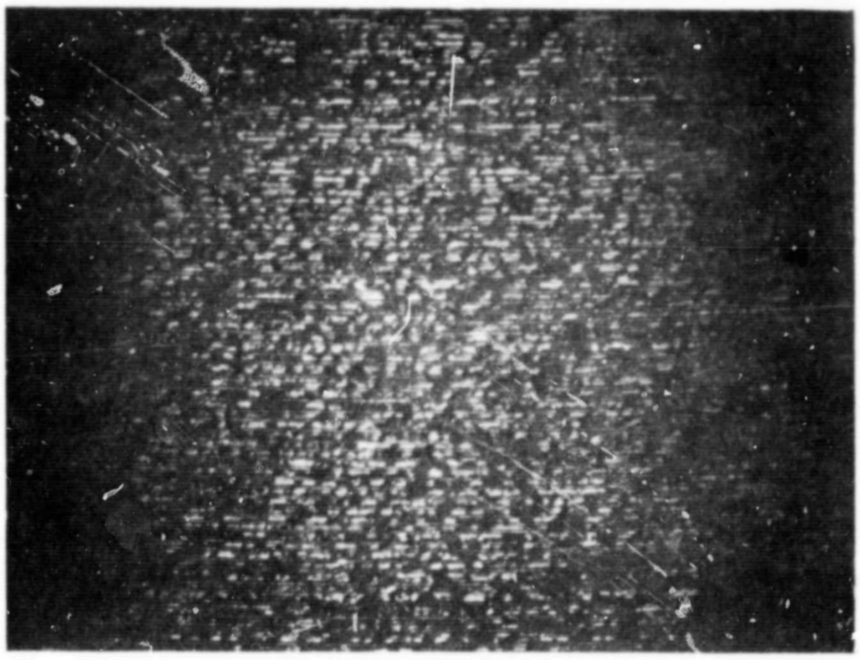

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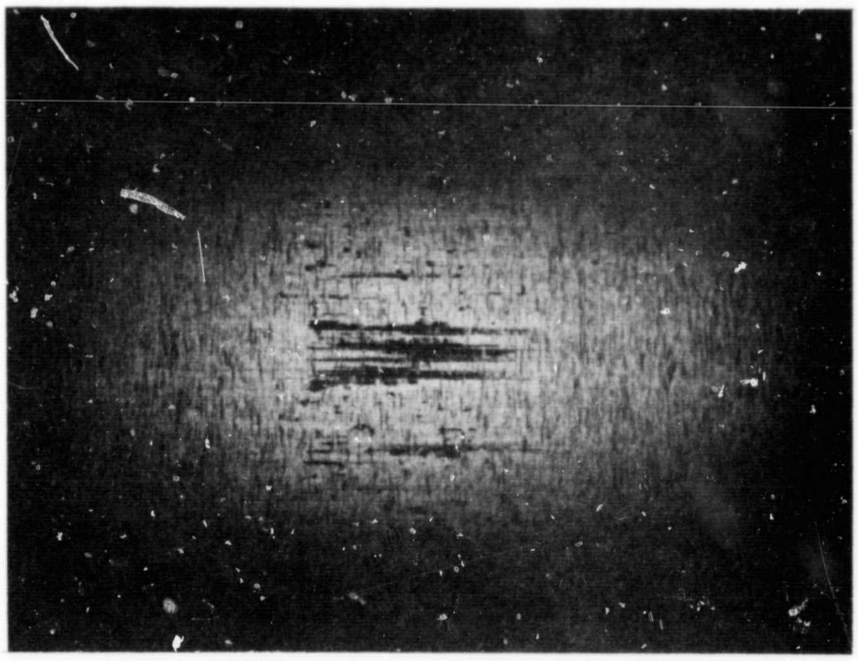
Figure 5-27. Finished Surface on Nickel Foametal Wedge Seal



25X

Figure 5-28. Transfer of Nickel Foametal onto Piston Rod

Inspection of the seal showed heavy wear on approximately 180 degrees of the seal circumference. The uneven wear pattern was believed to be caused by slight lateral displacement of the piston, which resulted in higher bearing loads on one side of the seal. As shown in Figure 5-28, some transfer of the nickel Foametal onto the piston rod was experienced. As the radial clearance between the rod and end gland bearing was only 0.0007 inch, the amount of radial displacement of the rod should have been negligible. But, because of the relative softness of the nickel Foametal, its ability to operate under high bearing loads is limited in the present seal configuration. A possible redesign of this seal considered but not tested is shown in Figure 5-29. This design would minimize the effects of side loading by enabling the wedge seal and the seat to move radially with the piston rod. However, this configuration introduces the need for a static seal between the seat and gland shoulder (see Figure 5-29).

5. Tests Nos. 4 and 5 - Design D and AH - Cobalt Molybdenum Lip Seal and Vascojet 1000/Silver Alloy Reed Seals (one inch)

Evaluation of these two configurations were performed concurrently in the same test actuator. The test configuration for the cobalt lip seal is similar to that shown in Figure 5-13 for the Vascojet lip seal. Figure 5-30 depicts the test configuration for the reed seal. Detail design of the lip seal is shown in Figure 4-20. Figures 4-29 through 4-32 show the details of the reed seal.

The lip seal showed exceptionally low leakage (Figure 5-31) during 50 hours of cycling at 400°F and 50 hours at 500°F. Accumulated leakage during this operation was 4cc. However, during the shutdown following the completion of the 500°F run, 94cc of leakage was collected in a 48-hour period. It was suspected that this leakage was caused by relaxation of the silver gasket static seal located between the flange of the lip seal and the seal cavity. This proved to be the case when the gland nut was tightened approximately 1/4 of a turn in reloading the static seal. Leakage during 25 hours of testing at 600°F totaled 32 cc. An attempt was made to further increase the load on the static seal. Leakage obtained after retightening the static seal increased to 180 dpm. At this point

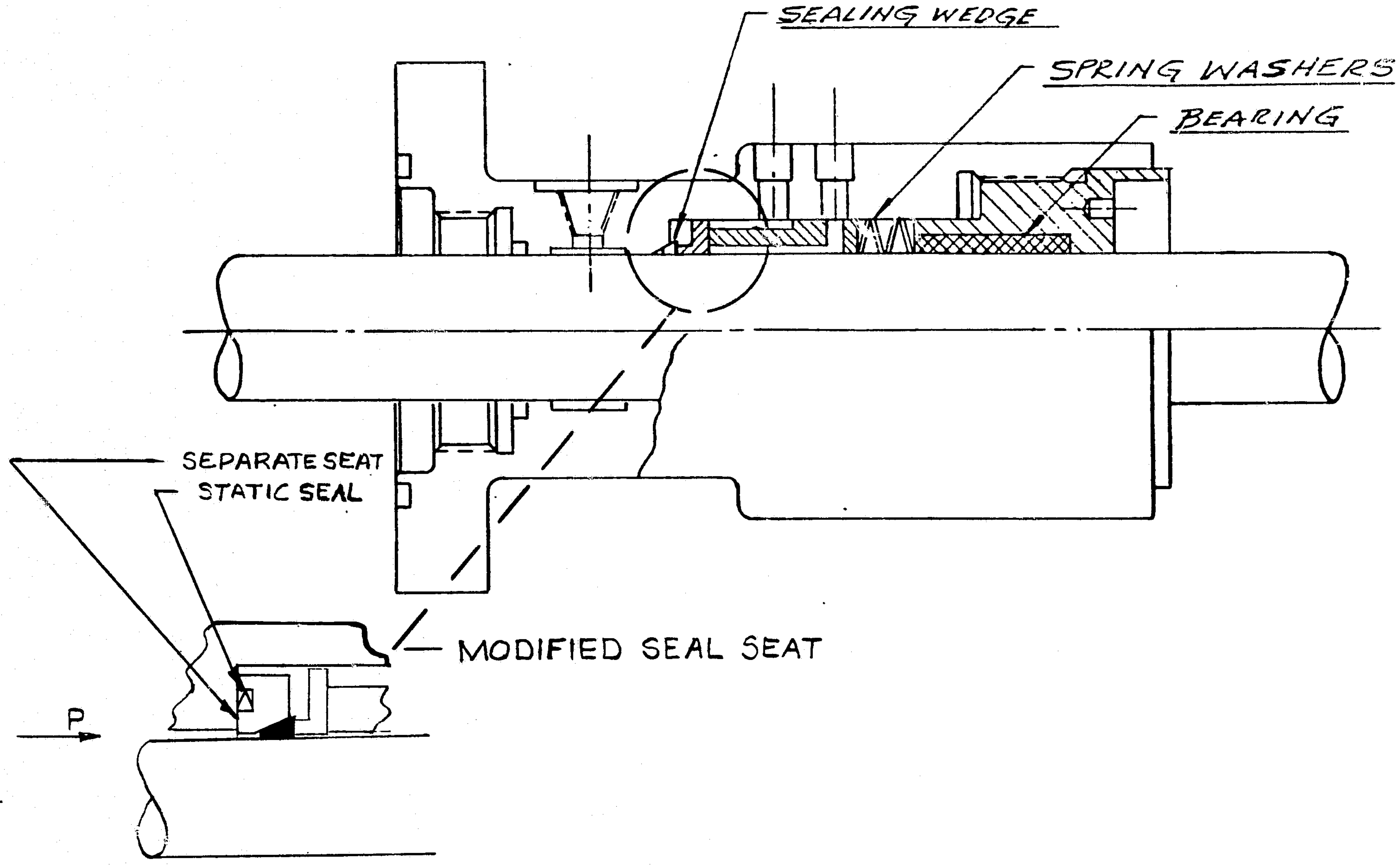


Figure 5-29. Modified Gland Configuration - Nickel Foametal Wedge Seal

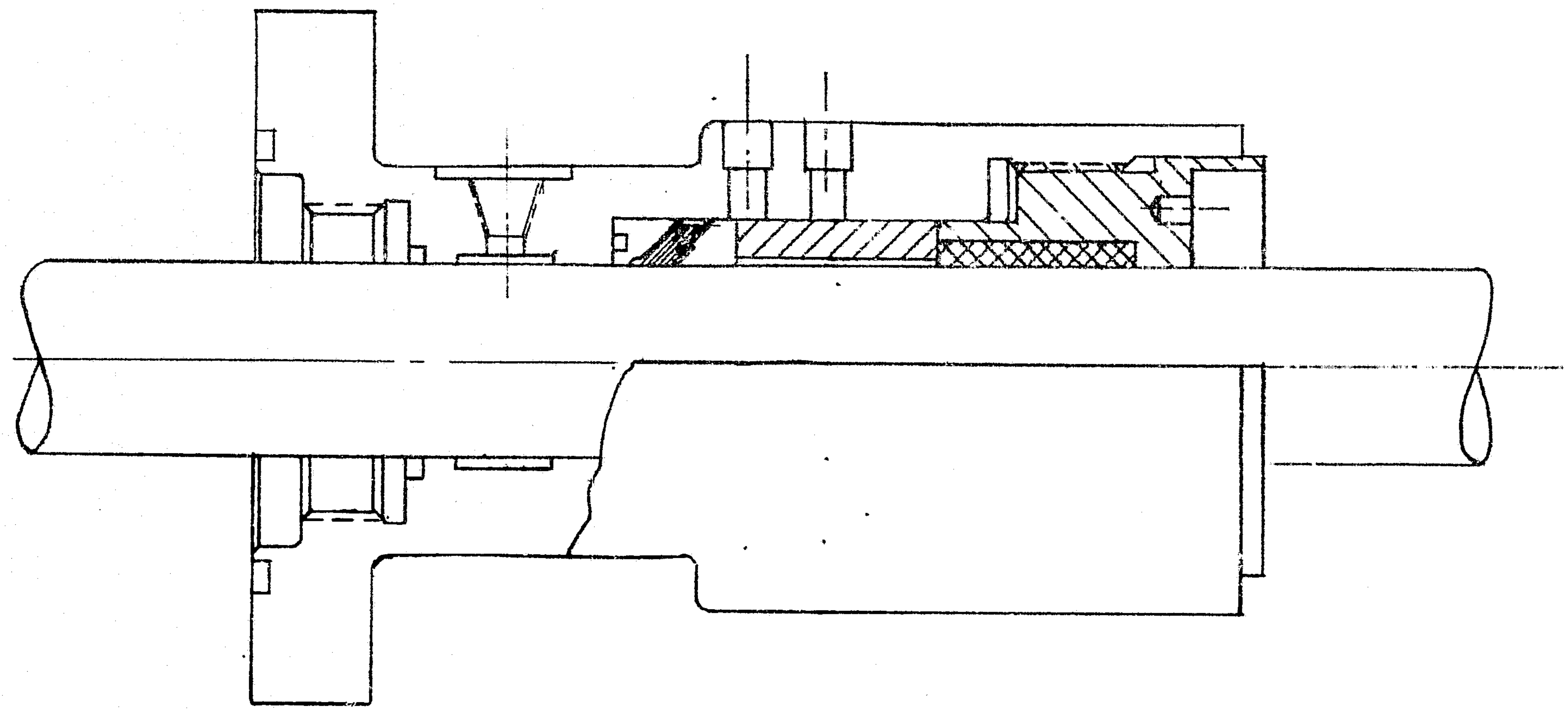


Figure 5-30. Vascojet 1000/Silver Alloy Reed Seal Gland Configuration

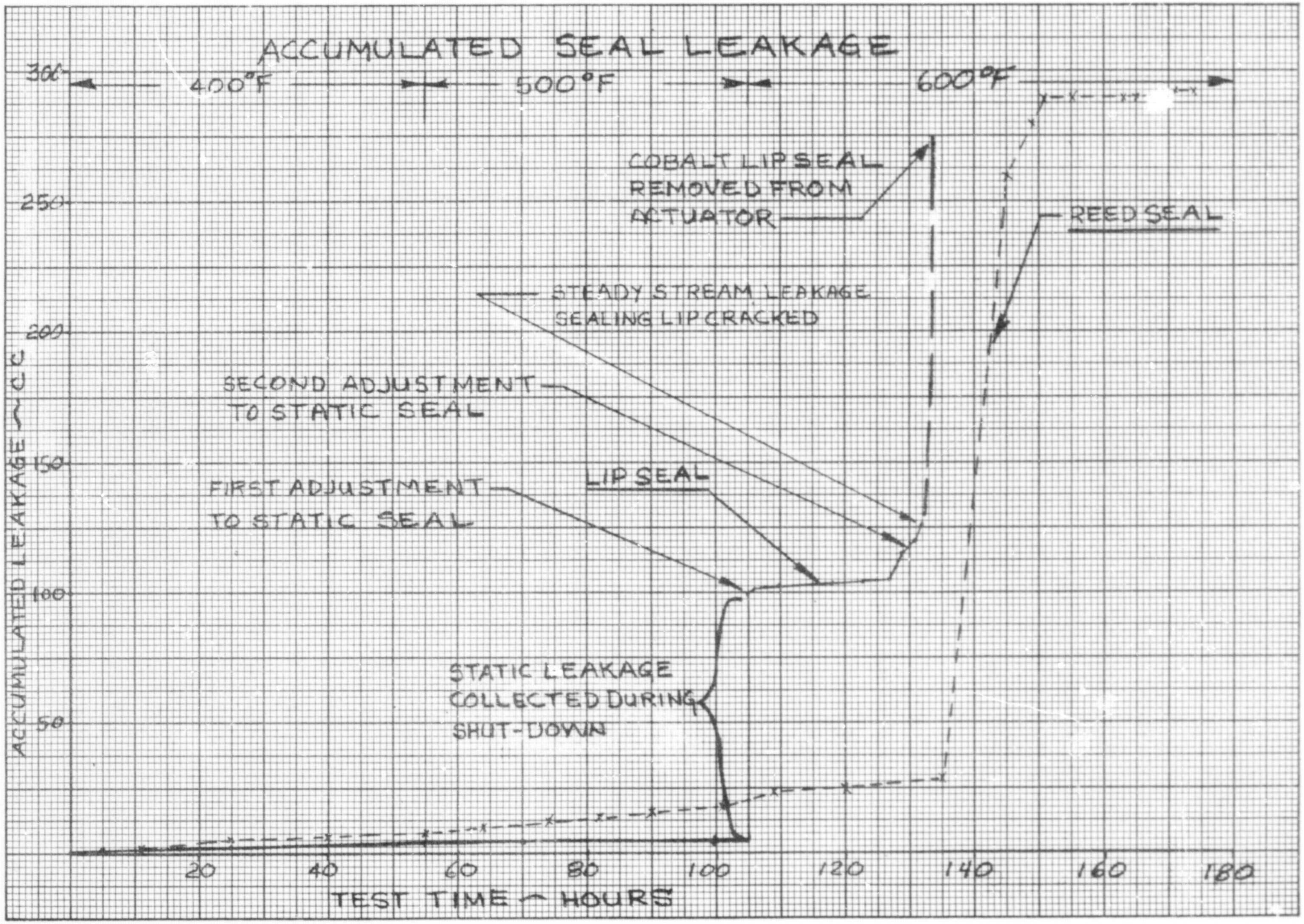


Figure 5-31. Accumulated Seal Leakage

the test was shut down and the lip seal removed from the test actuator. A set of polyimide V-seals was installed in its place to enable further testing of the reed seal.

Inspection of the lip seal, Figure 5-32, showed that the seal had cracked on the sealing lip. The crack that developed on the cobalt lip seal started at the contact edge of the lip and extended back towards the flange approximately 0.2 inch. No definite conclusion has been made as to the reason for the crack. Initially, it was felt that the crack on the seal was caused by excessive side load during the process of tightening the seal gland nut. However, similar cracks have developed on two other cobalt seals during the machining process. With the exception of the crack, the seal was in good condition after the test. As shown in Figure 5-33, the wear pattern on the sealing lip was evenly distributed and highly polished. No evidence of galling or metal pick-up was detected on the wear surface of the seal. The amount of wear was approximated by measuring the thickness of the metal at the wear surface and thickness of metal adjacent to it. As it was assumed that uniform thickness was obtained during grinding, the difference in thickness between the wear surface and the adjacent surface would represent the amount of wear. By this process the thickness of the metal at the wear surface was 0.0005-to-0.0007 inch less than the adjacent surface.

The short-stroke wear pattern on the chrome plated piston rod (Figure 5-34) was similar to that produced by the Vascojet 1000 lip seal. However, it did not appear to be as severe. Measurements taken of the piston rod indicated that approximately 0.0002 inch of wear had taken place at the site where the short-stroke cycles were run. The long-stroke cycling produced a highly burnished finish on the chrome-plated surface of the rod. Measurements taken over this surface indicated a wear of 0.0002-0.0003 inch. Hardness of the chrome plate was Rc 64 and hardness of the cobalt lip seal was Rc 47-52.

The reed seal which was tested concurrently with the lip seal in the same actuator, completed the entire low-pressure test. Slightly higher leakage (Figure 5-31) was experienced with this configuration. Total leakage during 50

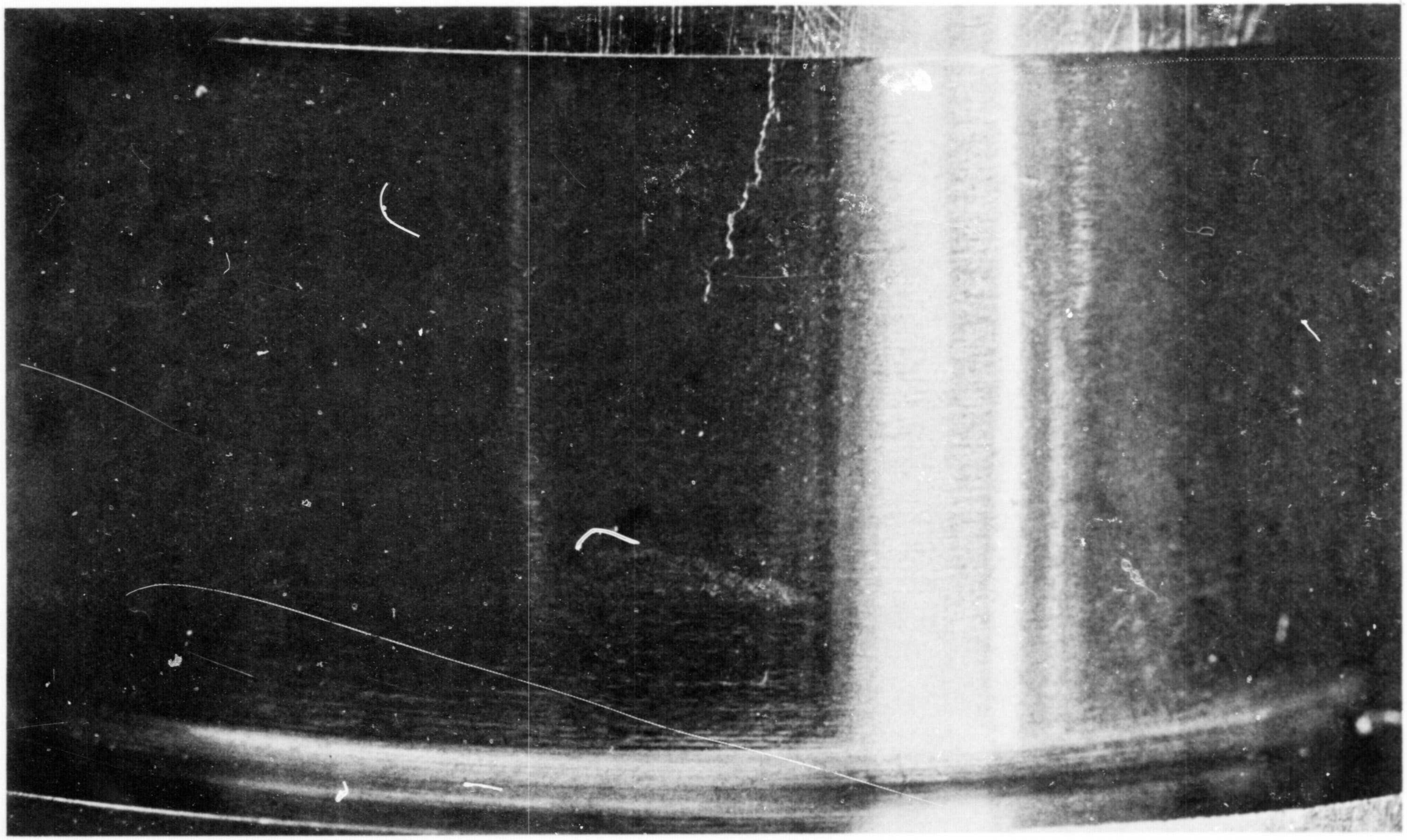


Figure 5-32. Cobalt Molybdenum Lip Seal (one inch)

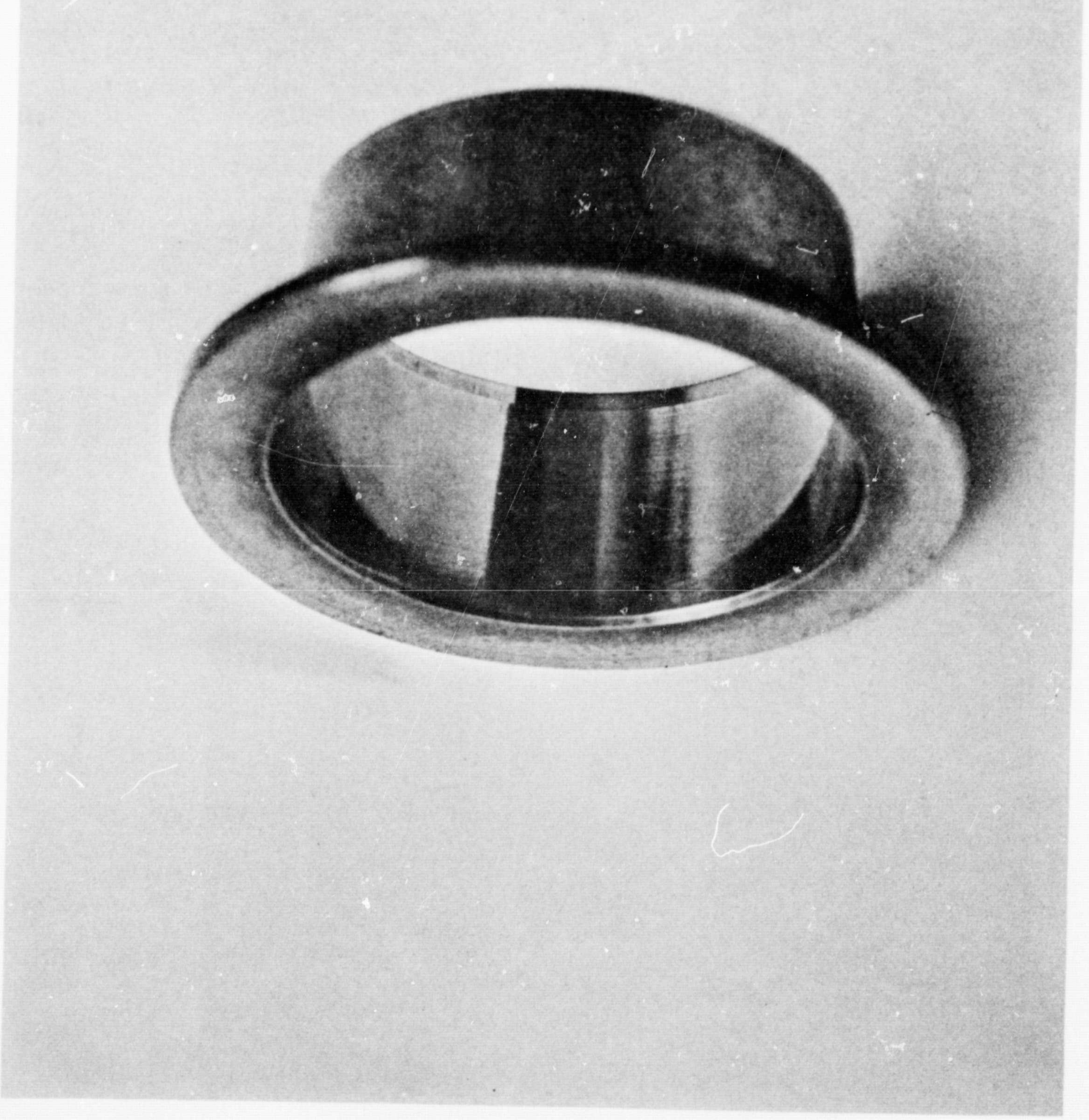


Figure 5-33. Cobalt Molybdenum Lip Seal (one inch)

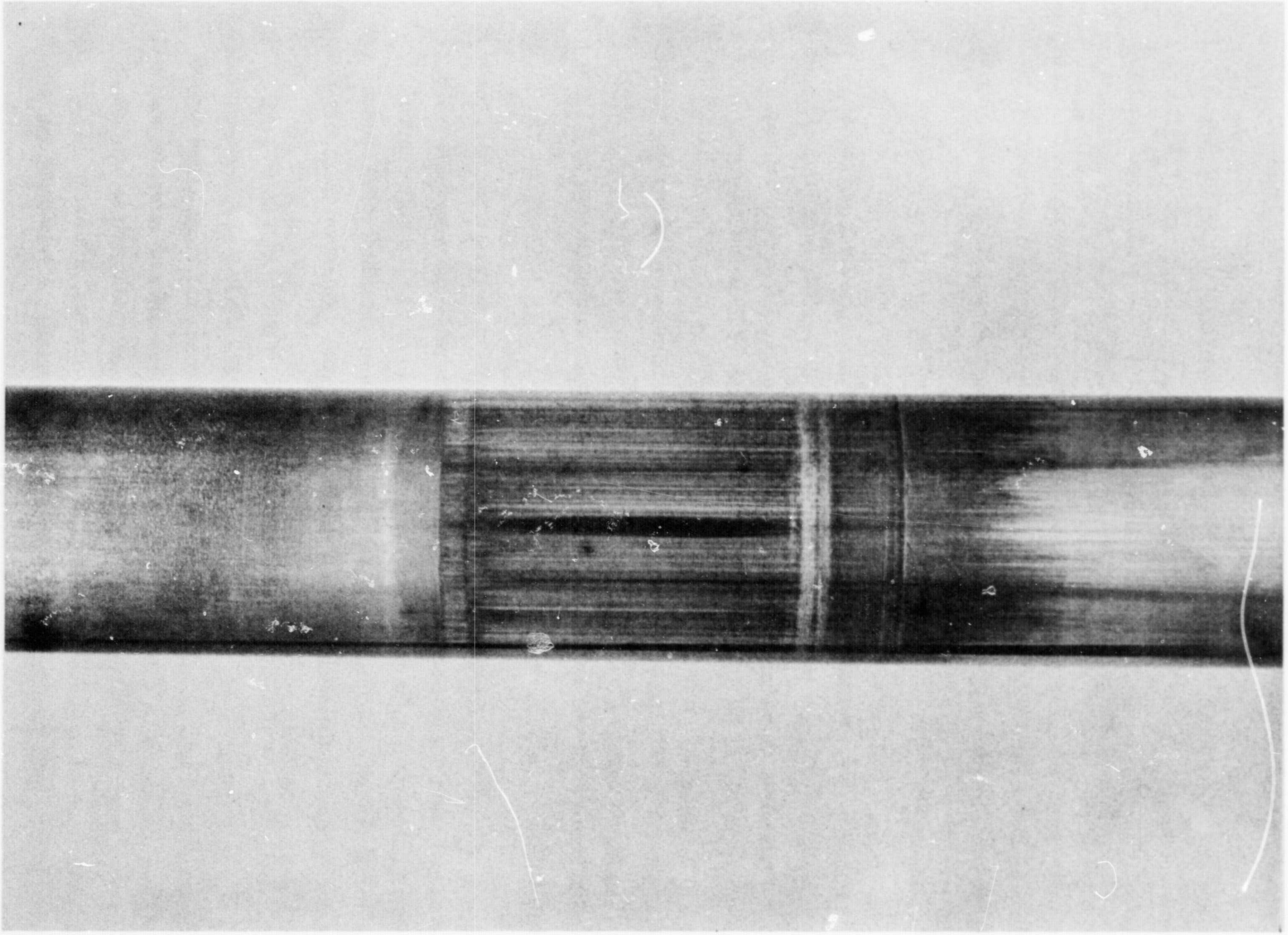


Figure 5-34. Piston Rod (A-End) One-Inch Cobalt Molybderum Lip Seal

hours at 400°F and 50 hours at 500°F was 21 cc. Leakage during the first 25 hours of testing at 600°F was 7.5 cc. An increase in leakage was exhibited by the seal during the latter portion of the test at 600°F. This increased leakage occurred after the removal of the cobalt lip seal from the A-end of the actuator, which may have caused some disturbance of the reed seal. From the 26th hour through the 41st hour of operation, leakage was erratic, ranging from 0 to 25 drops per minute. However, this leakage decreased to practically zero during the last 17 hours of testing at 600°F.

Reconstruction of the events that took place during the removal of the cobalt lip seal revealed that the silver reeds may have been scratched, thus resulting in increased leakage. Damage of the sealing surface could have occurred when the extreme portion of the rod was retracted past the reeds during removal of the lip seal. This portion of the rod, which was exposed to a 600°F ambient temperature, had accumulated a hard deposit of degraded silicone fluid. During subsequent cycling, the scratches on the sealing surface of the reeds apparently healed and good sealing contact was restored. This would account for the reduced leakage experienced towards the end of the 600°F operation.

Sliding friction of the sealing reeds was determined prior to their removal from the test actuator. The friction obtained was 22 pounds, which was a substantial reduction from the original value of 150 pounds recorded prior to testing. Nonetheless, it indicated that some residual interference existed between the sealing reeds and the piston rod. As shown in Figure 5-35 the sealing reeds were in excellent condition.

Dimensions of the inside diameter of the sealing reeds were obtained on a shadowgraph. A comparison of these dimensions and those taken before testing are shown in Table 5-5. The Vascojet reeds show no interference remaining after test. Although the Vascojet reeds were designed to provide an interference fit of 0.0035-inch over the rod on assembly, distortion of the reed during heat treating made it difficult ot ascertain the amount of initial interference.

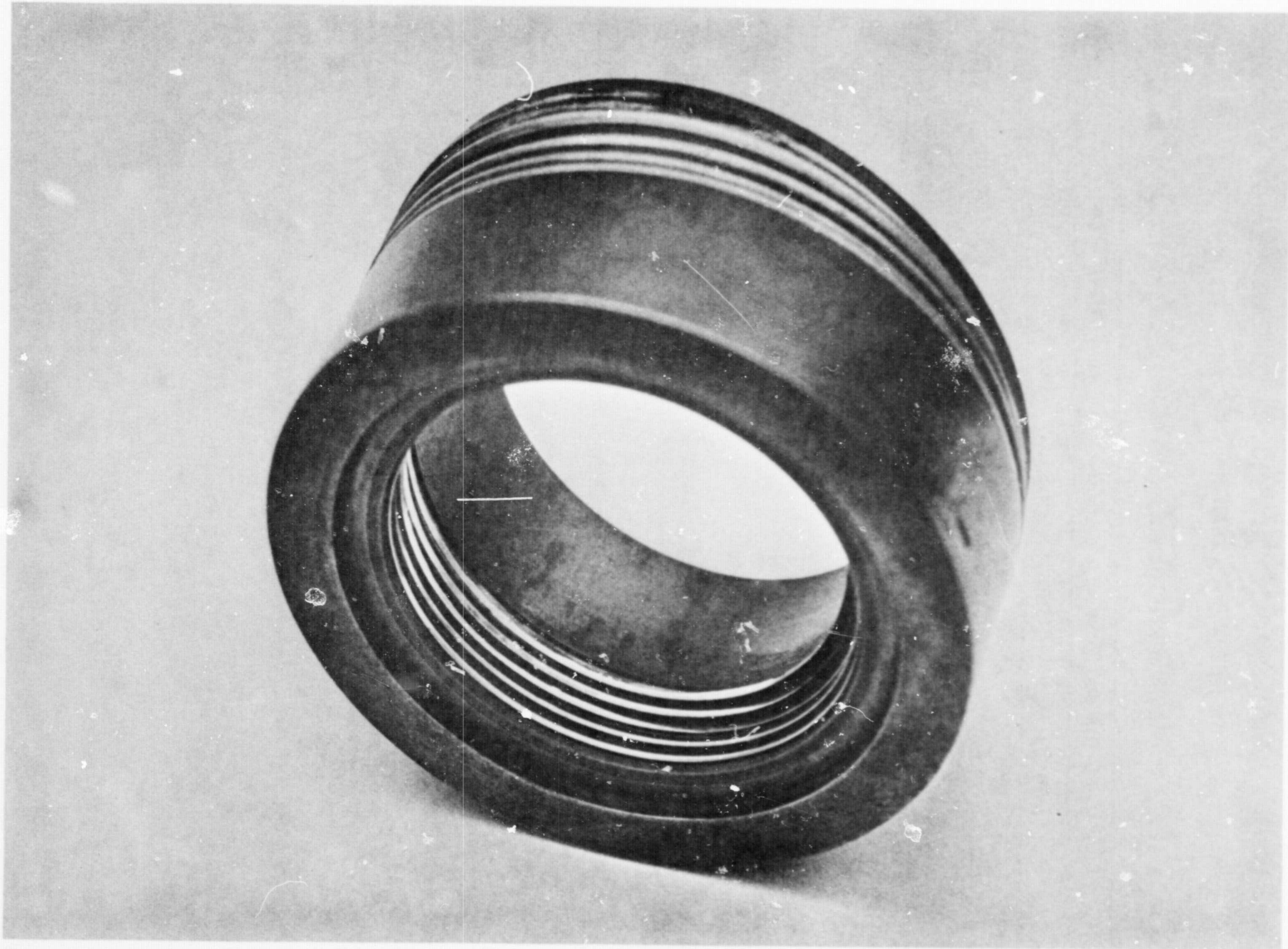


Figure 5-35. Vascojet 1000/Silver Alloy Reed Seal (one inch)

TABLE 5-5

REED SEAL DIMENSIONS

Piston Rod Diameter - 0.9975

Reed No.	Before Test		After Test	
	Seal I.D.	Interference	Seal I.D.	Interference
1 (Silver)	.9949	.0026	.9975	-
2 (Vascojet)	*.9975	0	.9975	-
3 (Silver)	.9949	.0026	.9973	.0002
4 (Vascojet)	*.9967	.0008	.9975	-
5 (Silver)	.9949	.0026	.9973	.0002
6 (Vascojet)	*.9971	.0004	.9980	-
7 (Silver)	.9949	.0026	.9968	.0007
8 (Vascojet)	*.9968	.0007	.9975	-
9 (Silver)	.9949	.0026	.9972	.0003

* Due to distortion resulting from heat treating, these are average diametral dimensions.

The silver reeds all provided an initial interference fit of 0.0026-inch. The interference remaining after test varied from zero to 0.0007-inch. Greater wear was exhibited by the silver reeds that were closer to the fluid pressure. This was believed to be attributable to the higher contact stresses on the reeds due to pressure energizing. As the reeds were located further away from the fluid the effects of pressure energizing were less pronounced.

Dimensional check of the piston rod (Figure 5-36) showed no measurable wear due to the long-stroke cycling. An increase (0.003-inch) in diameter was detected on the portion of the rod that was subjected to short-stroke cycling. This was believed to be due to a build-up of silver from the silver reeds. This build-up of material was easily rubbed off with crocus cloth.

6. <u>Test No. 6 - Design D - Polyimide V-Seal (three inch)</u>

The gland configuration for the three-inch polyimide V-seal is similar to that shown in Figure 5-6 for the one-inch V-seal. Details of the three-inch V-seal are shown in Figures 5-37 and 5-38. The seals were assembled with a spring load of 1200 pounds. Seal friction was approximately 360 pounds for two seals.

Results of this test are summarized in Table 5-2; both seals completed the 150-hour test satisfactorily. Very low leakage was obtained on the B-end seal: total leakage for this seal was 13 cc during the entire testing period. Leakage from the A-end seal was practically zero during the 400°F and 500°F operation. However, leakage from the A-end seal during the 600°F operation was considerably higher. For the first 30 hours at 600°F, 18.5 cc of leakage was accumulated and, during the final 20 hours at 600°F, 421.5 cc was collected. Seal friction measured after testing was 210 pounds as compared to the original friction load of 360 pounds.

Inspection showed the B-end seal (Figure 5-39) to be in excellent condition. The high leakage experienced by the A-end seal was attributable to tool chatter marks (Figure 5-40) which produced a wavy sealing surface. This resulted in poor contact between the seal and the rod.

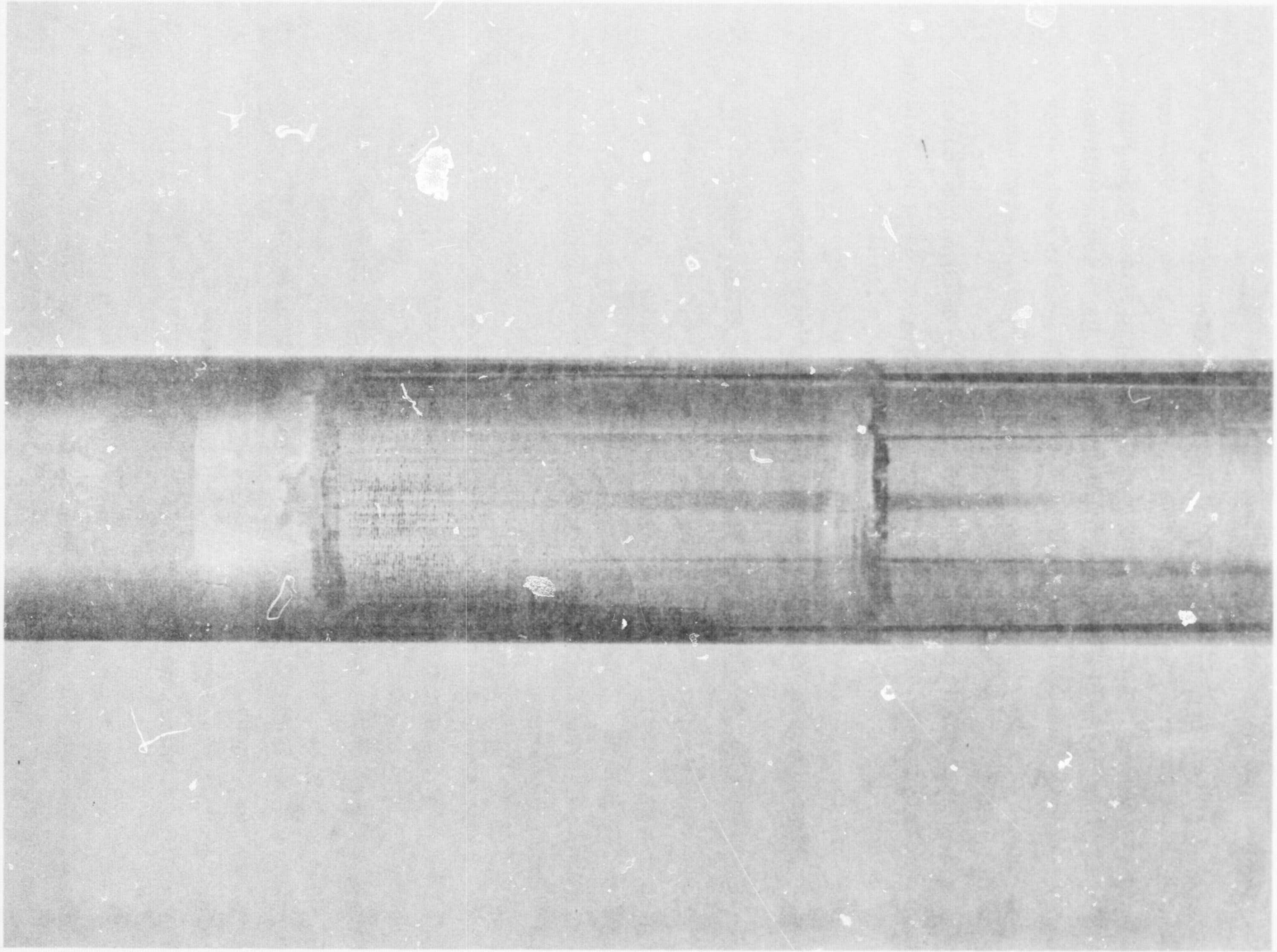


Figure 5-36. Piston Rod (B-End) One-Inch Vascojet 1000/Silver Alloy Reed Seal

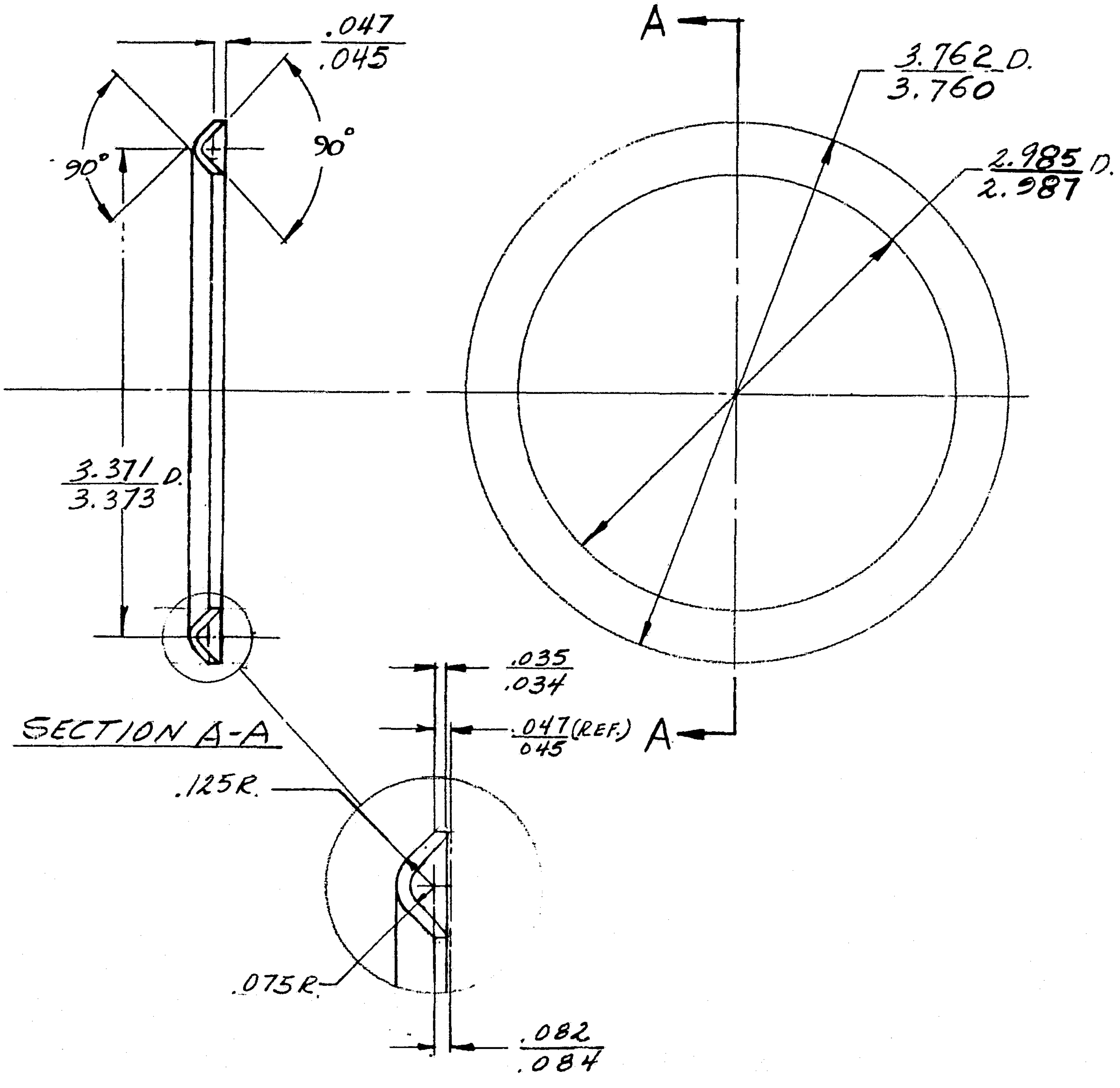
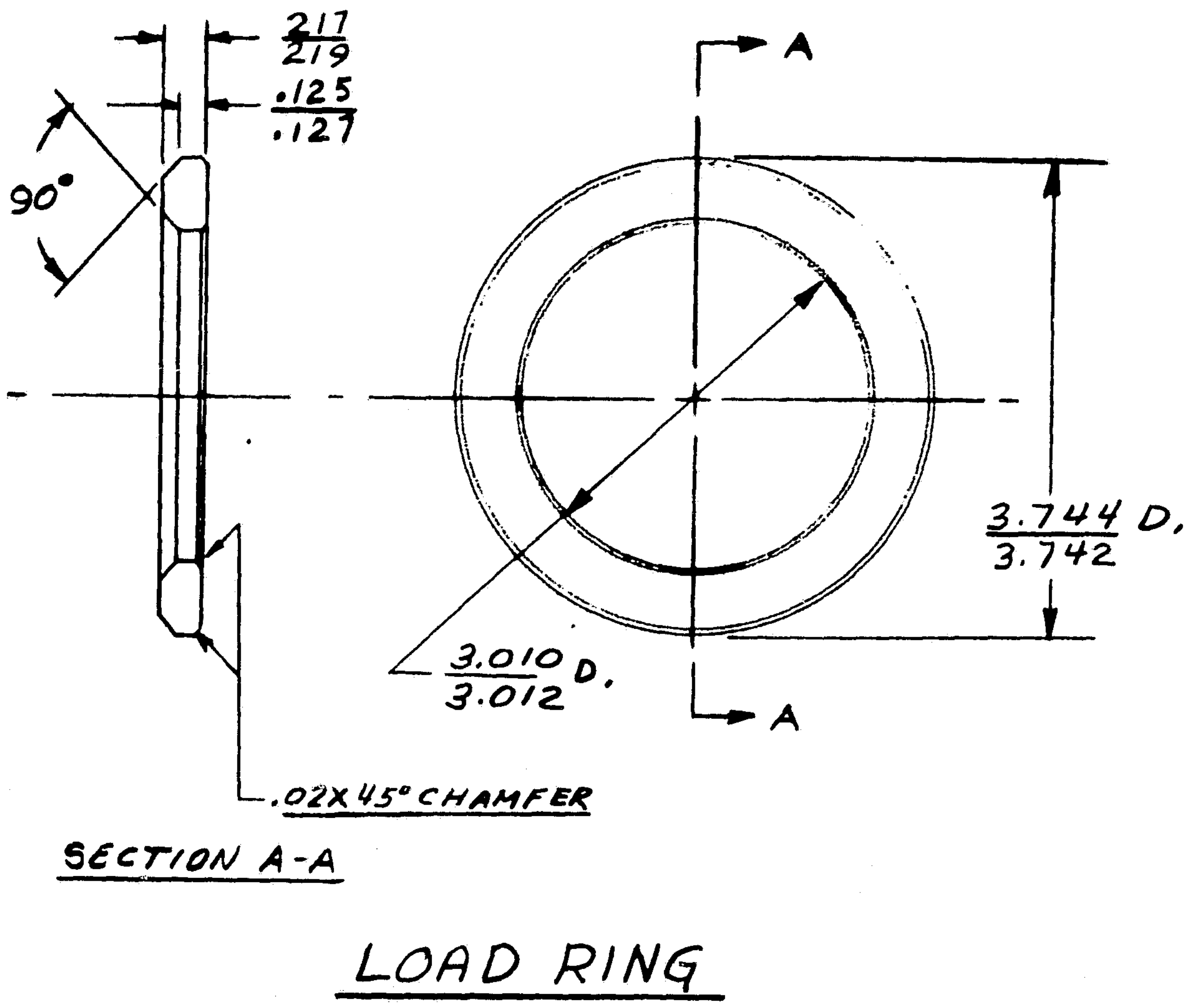


Figure 5-37. V-Seal, Polyimide - Three-Inch Diameter



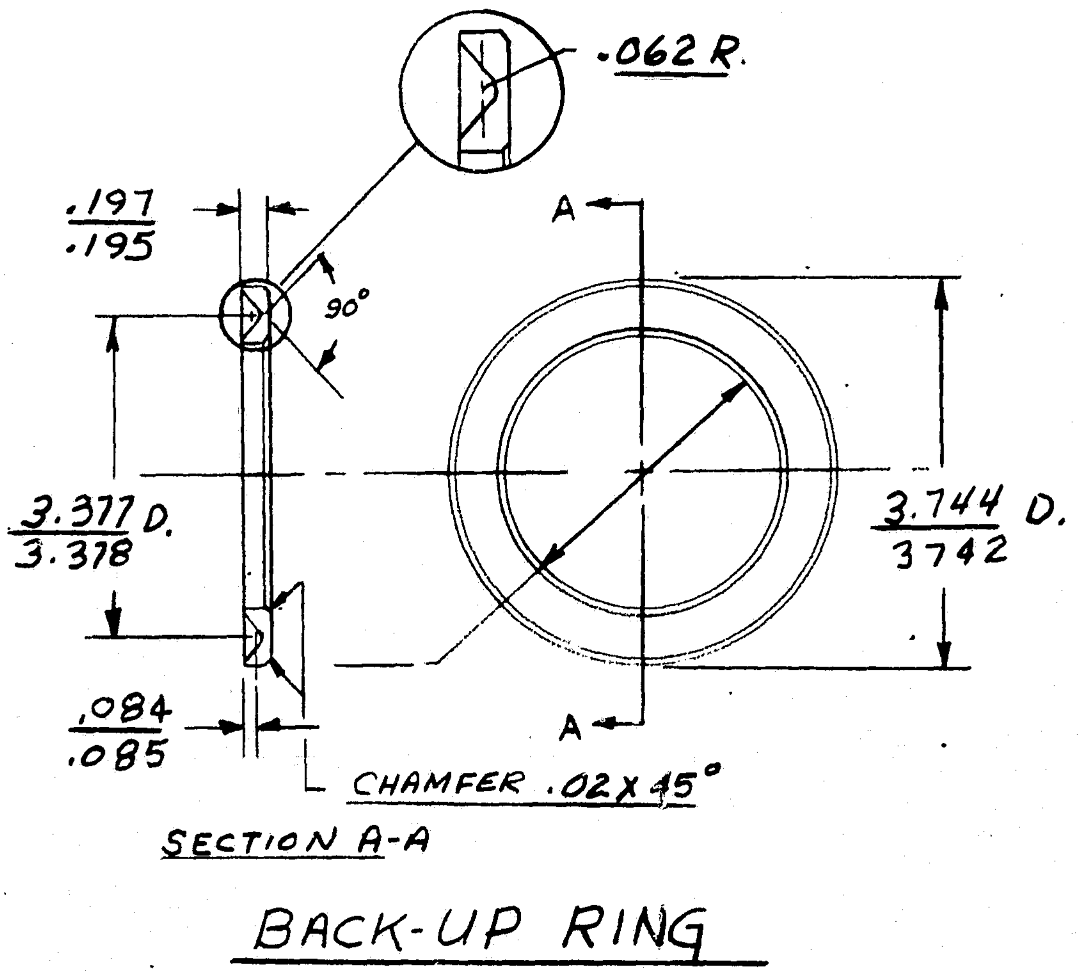


Figure 5-38. Load Ring and Backup Ring

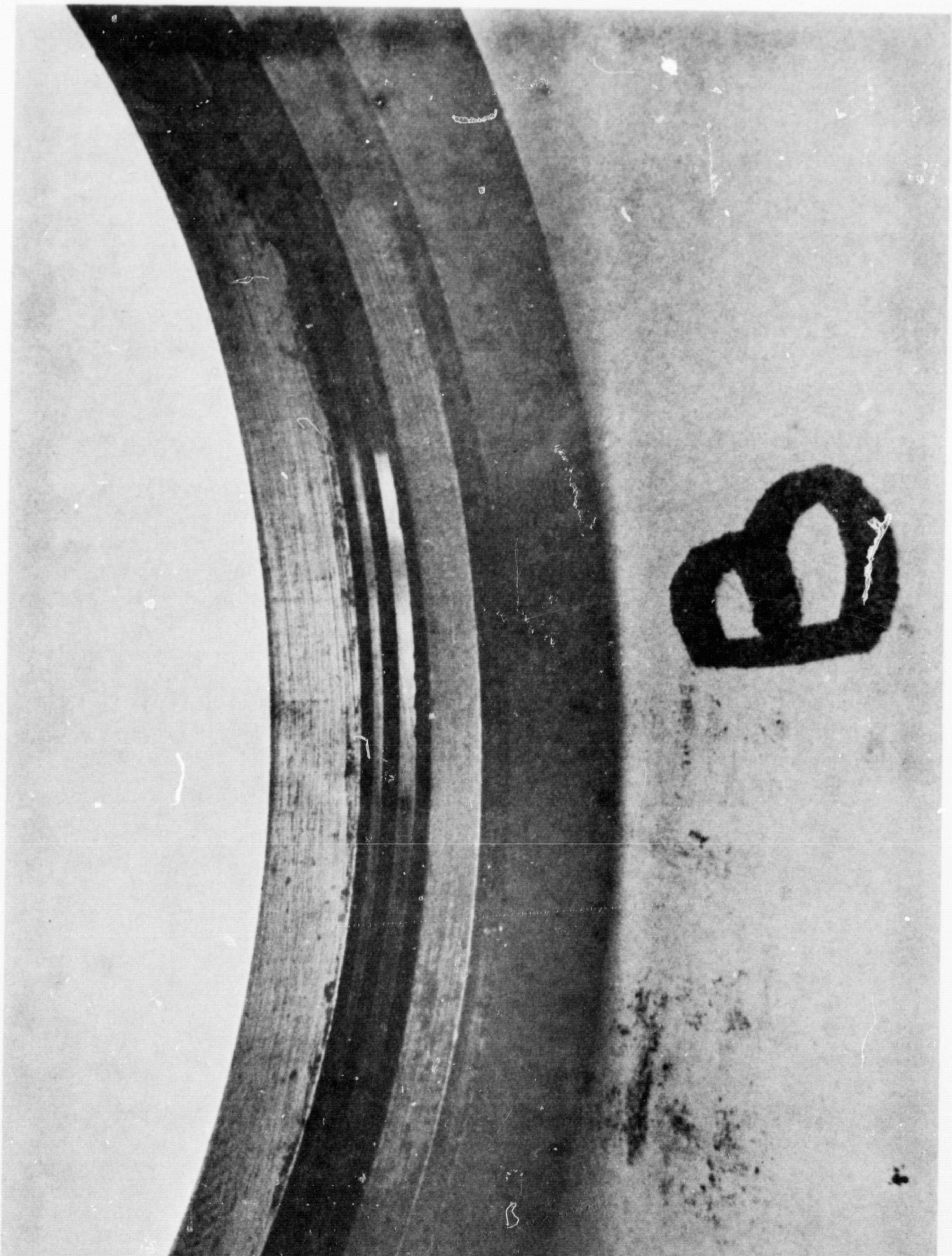


Figure 5-39. Three-Inch Polyimide V-Seal (B-End)

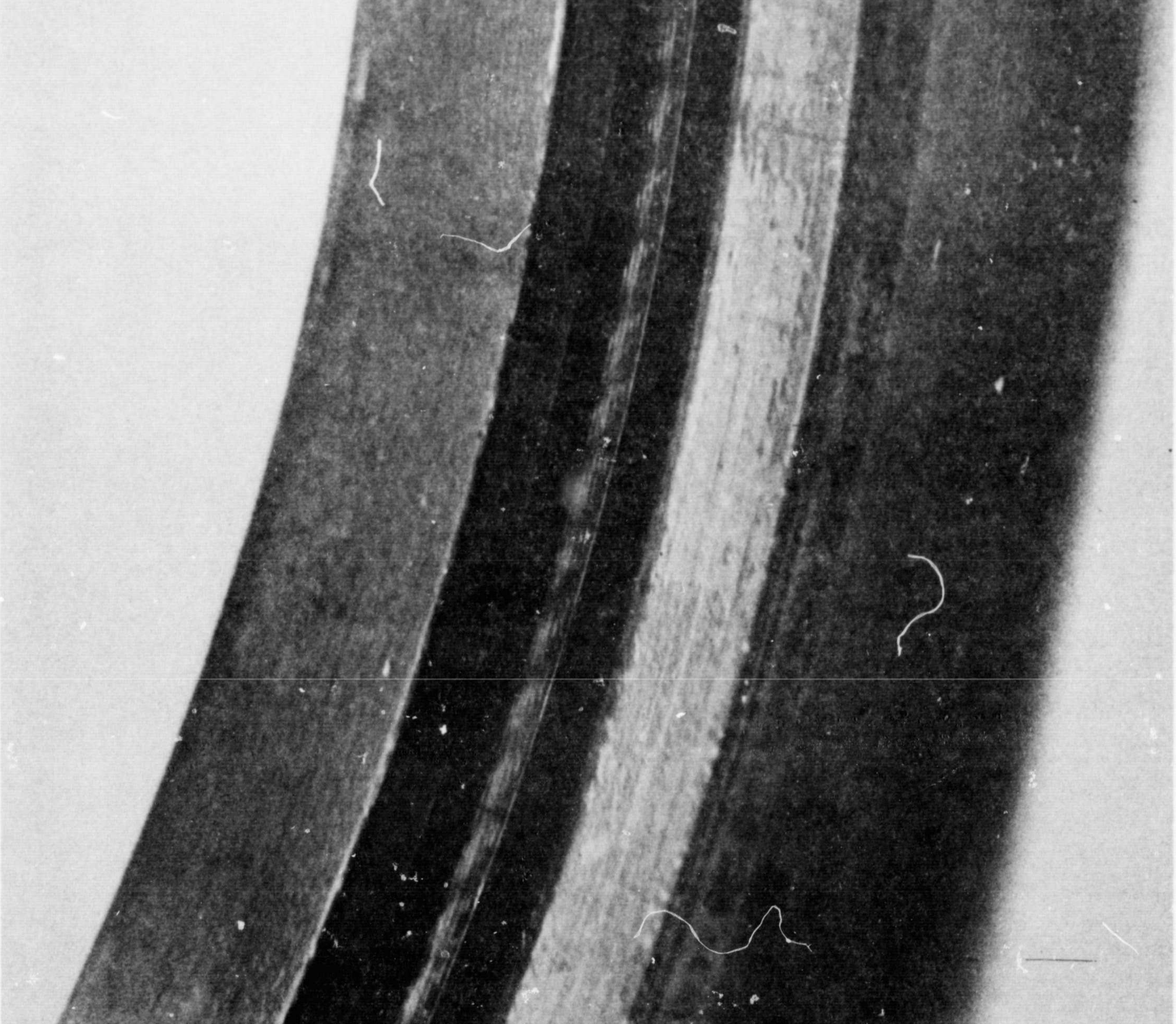


Figure 5-40. Three-Inch Polyimide V-Seal (A-End)

The piston rod (Figures 5-41 and 5-42) was in good condition except for the highly polished surface produced by the short-stroke cycles. A slippery coating was noted on the piston rod surface. This coating appeared to be a transfer of the polyimide material onto the rod surface, which may have contributed to the lower seal friction obtained after the test.

7. Test No. 7 - Design D - Cobalt Molybdenum Alloy Seal (three inch); Test No. 8 - Design AH - Vascojet 1000/Silver Alloy Reed Seals (three inch)

Design details of the above seals are shown in Figures 5-43 through 5-47. Gland configuration for the lip seal and reed seal are similar to those shown in Figures 5-13 and 5-30, respectively.

The above seal configurations were evaluated concurrently in the same test actuator. However, the lip seal was removed from the test actuator due to excessive leakage during the 100 psi pressure check prior to actual testing. Testing of the reed seal was then accomplished by replacing the lip seal with a set of Polyimide V-seals from Test No. 6.

The leakage (20 dpm) exhibited by the lip seal during the pressure check was determined to have been caused by a crack at the contact surface of the seal (see Figure 5-48). This was somewhat expected because a thin line crack was noticed on the outside surface of the sealing lip prior to testing. At the time, the crack did not appear to have developed through the sectional thickness of the lip. As this seal was the only seal that completed the fabrication cycle without material failure, it was decided to proceed with the testing. It was apparent that propagation of the crack occurred when the seal was assembled onto the piston rod.

Successful fabrication of a seal specimen has been difficult, due to cracking of the sealing lip during final machining. It was believed that the cracks result from flows or inclusions in this particular batch of material. Consequently, further work on this seal was discontinued during this phase of the program.

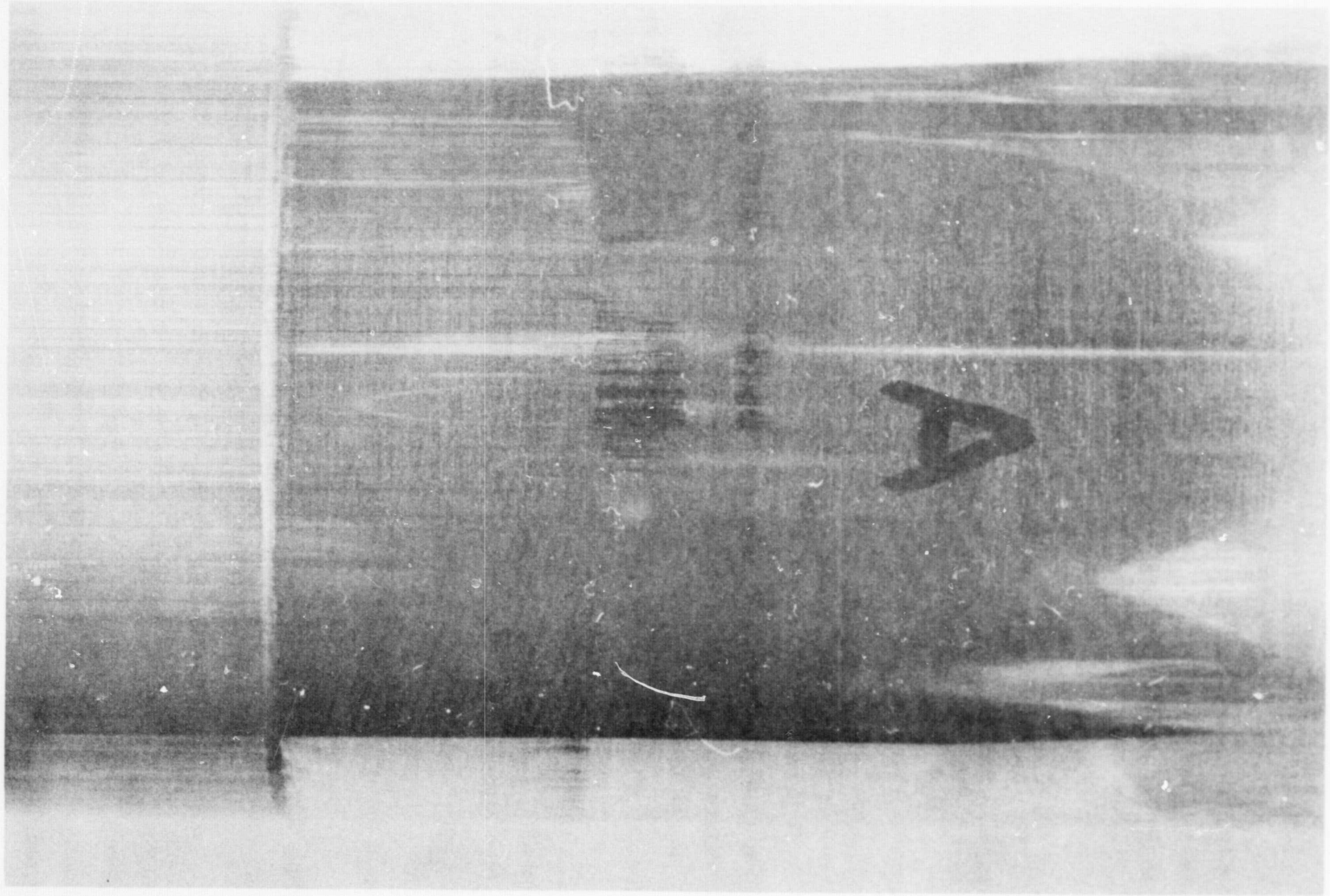


Figure 5-41. Piston Rod (A-End) Three-Inch Polyimide V-Seal

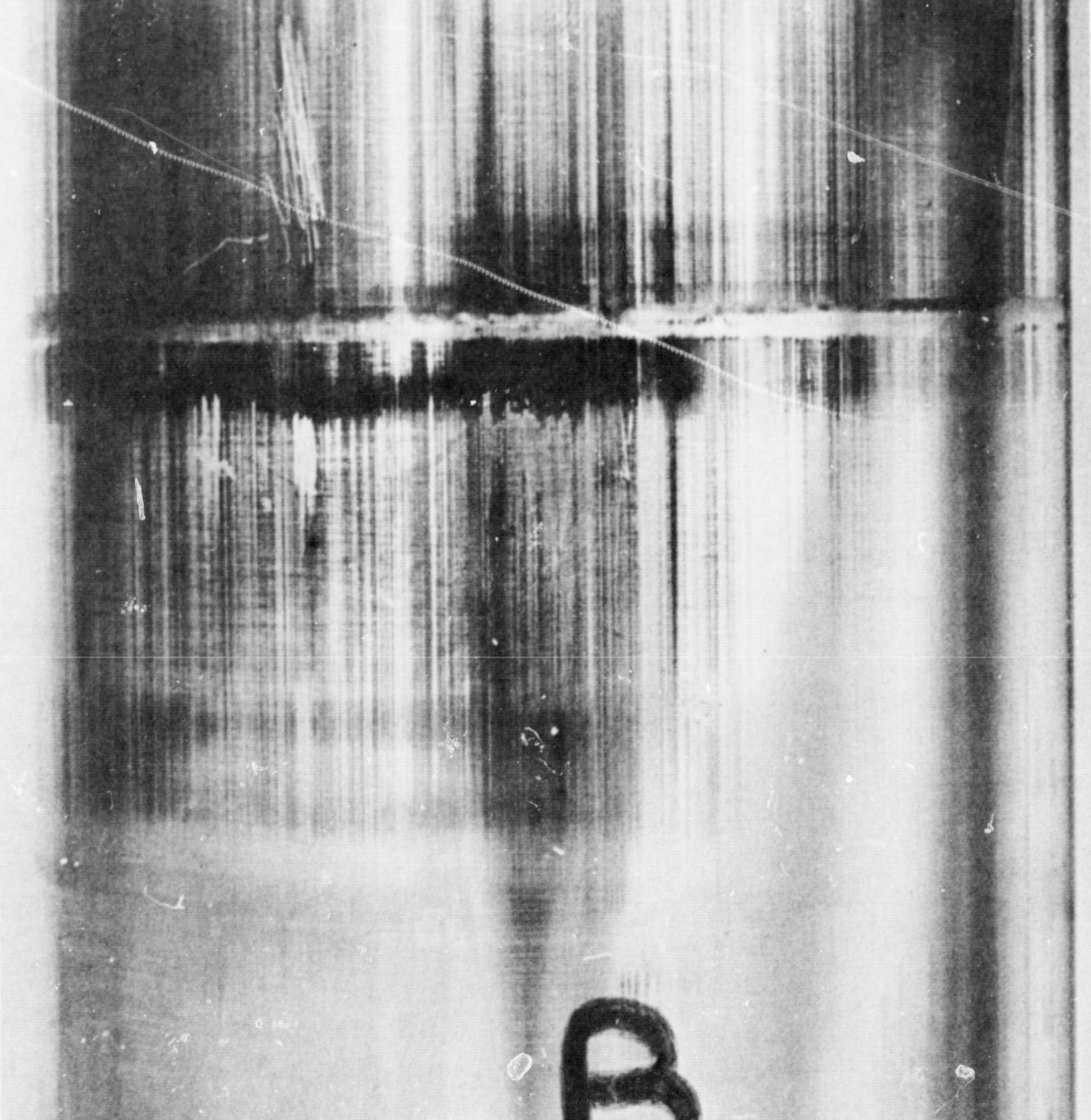


Figure 5-42. Piston Rod (B-End) Three-Inch Polyimide V-Seal

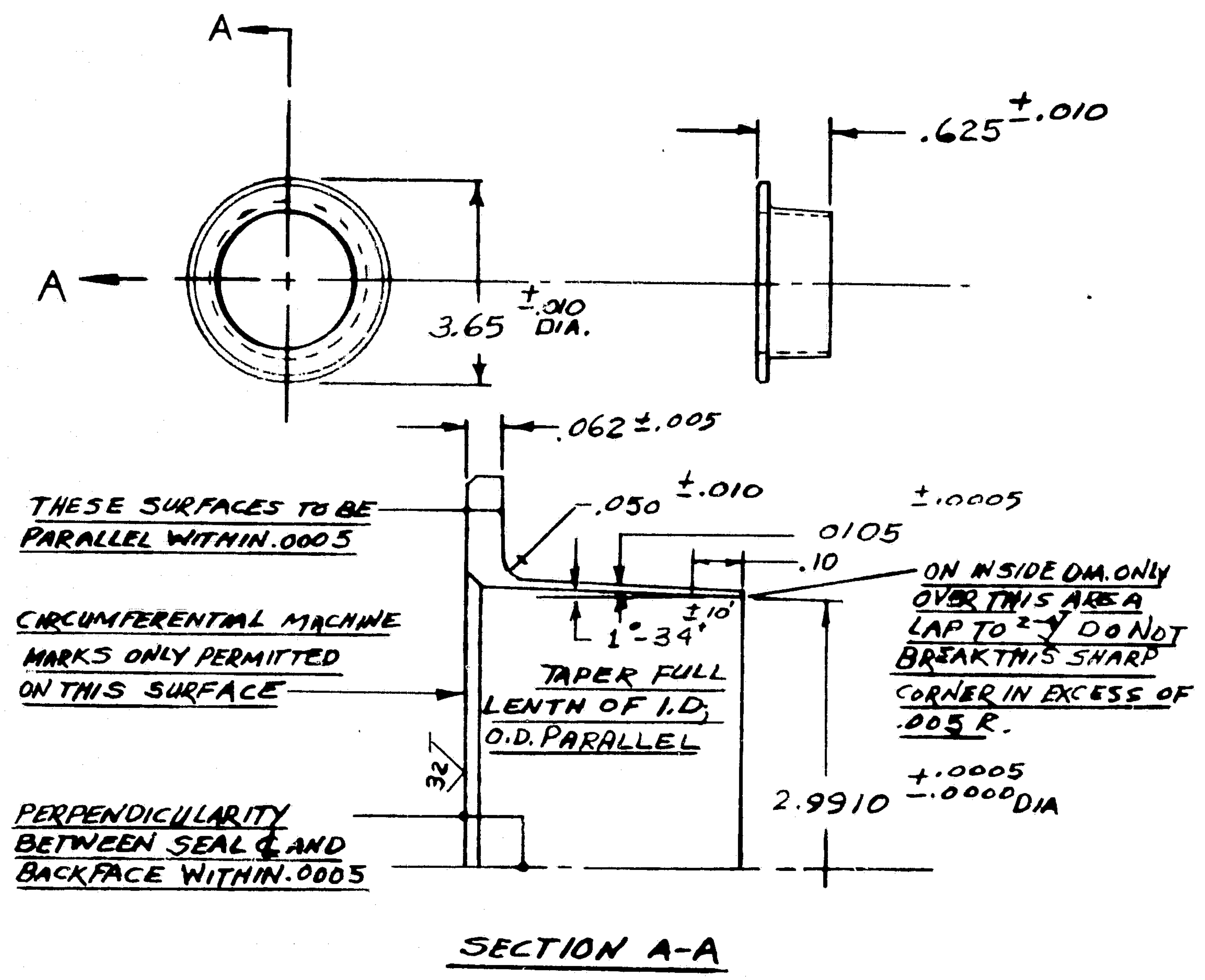
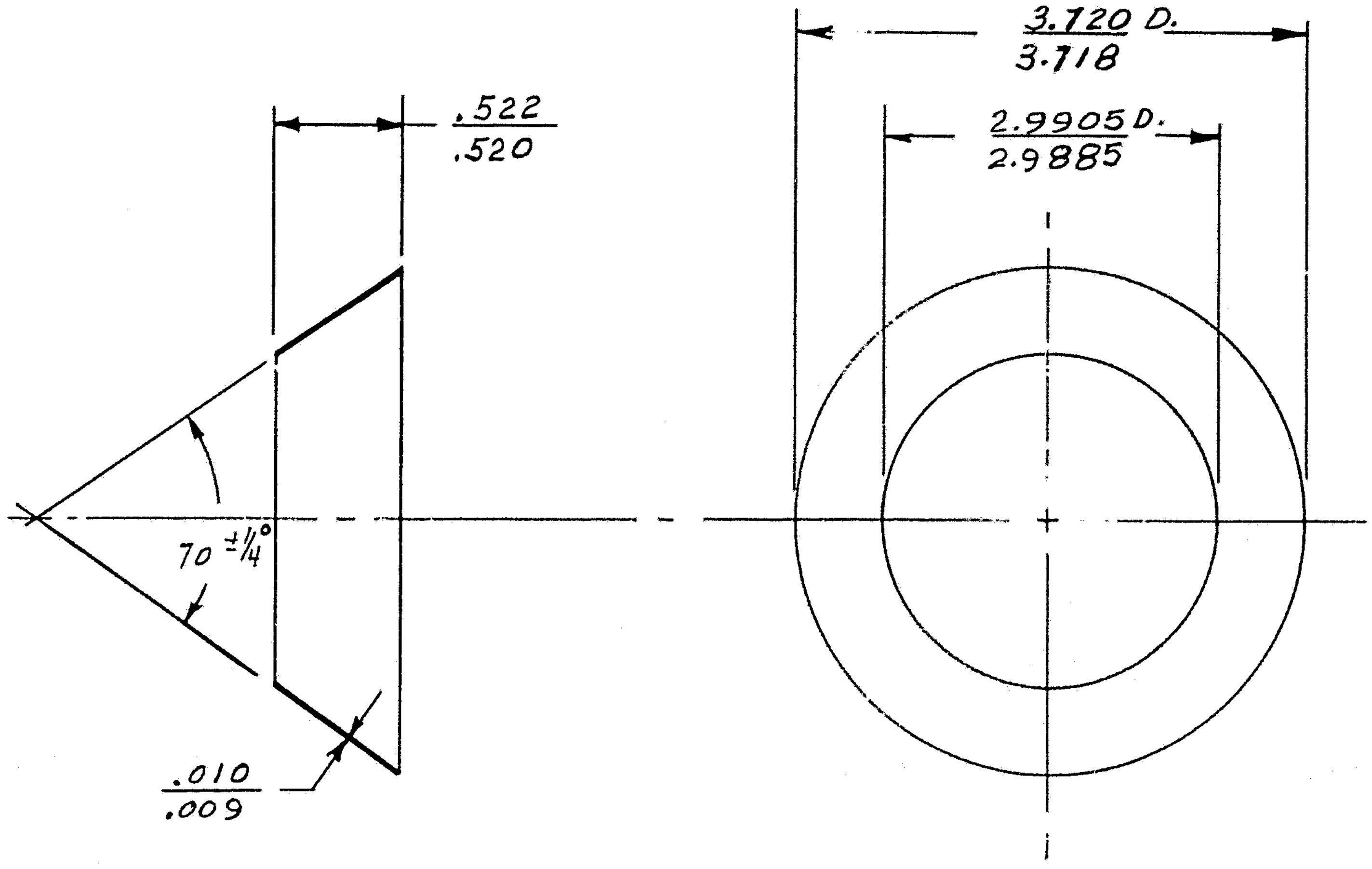


Figure 5-43. Three-Inch Cobalt Molybdenum Lip Seal



NOTE:
HEAT TREAT PART AS FOLLOWS:
(1) PREHEAT AT 1500°F., (2) HOLD AT 1850°F. FOR 20 TO 30 MIN. IN NEUTRAL ATMOSPHERE AND COOL TO RM. TEMP. RATE OF COOLING FR. 1850°F. TO 1200°F. IN 60 SEC. (3) TEMPER (IN NEUTRAL ATMOS.) IMMEDIATELY UPON REACHING RM. TEMP. FOR 3 HRS. AT 1050°F., AIR COOL AND RETEMPER FOR ADDITIONAL 3 HRS. AT 1050°F. AND AIR COOL. (BRIGHT SURFACE REQ'D.) TEST BUTTON HARDNESS TO BE R_c 48-52.

Figure 5-44. Reed Seal, Vascojet 1000

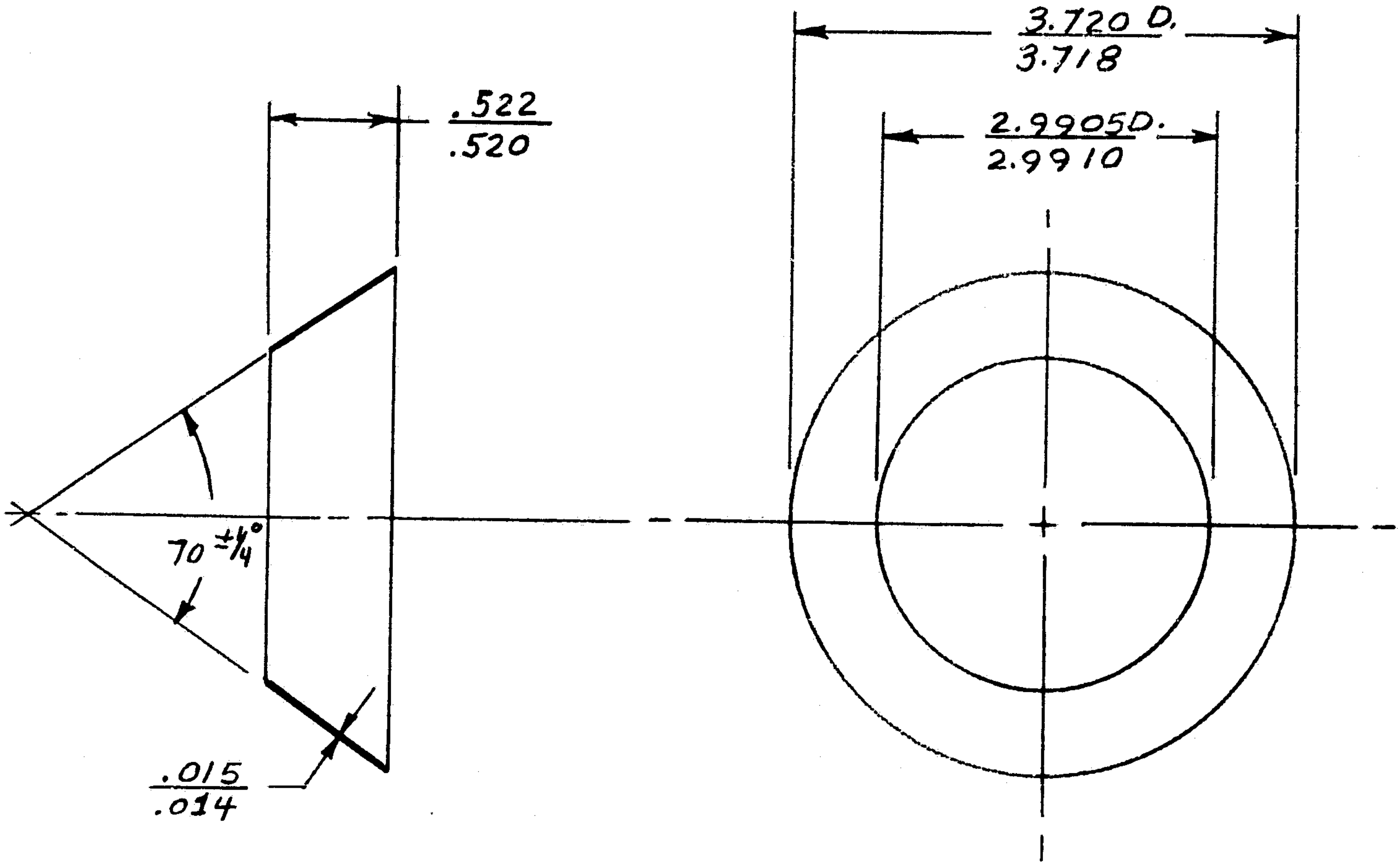


Figure 5-45. Reed Seal, Silver Alloy (72% Ag + 28% Cu)

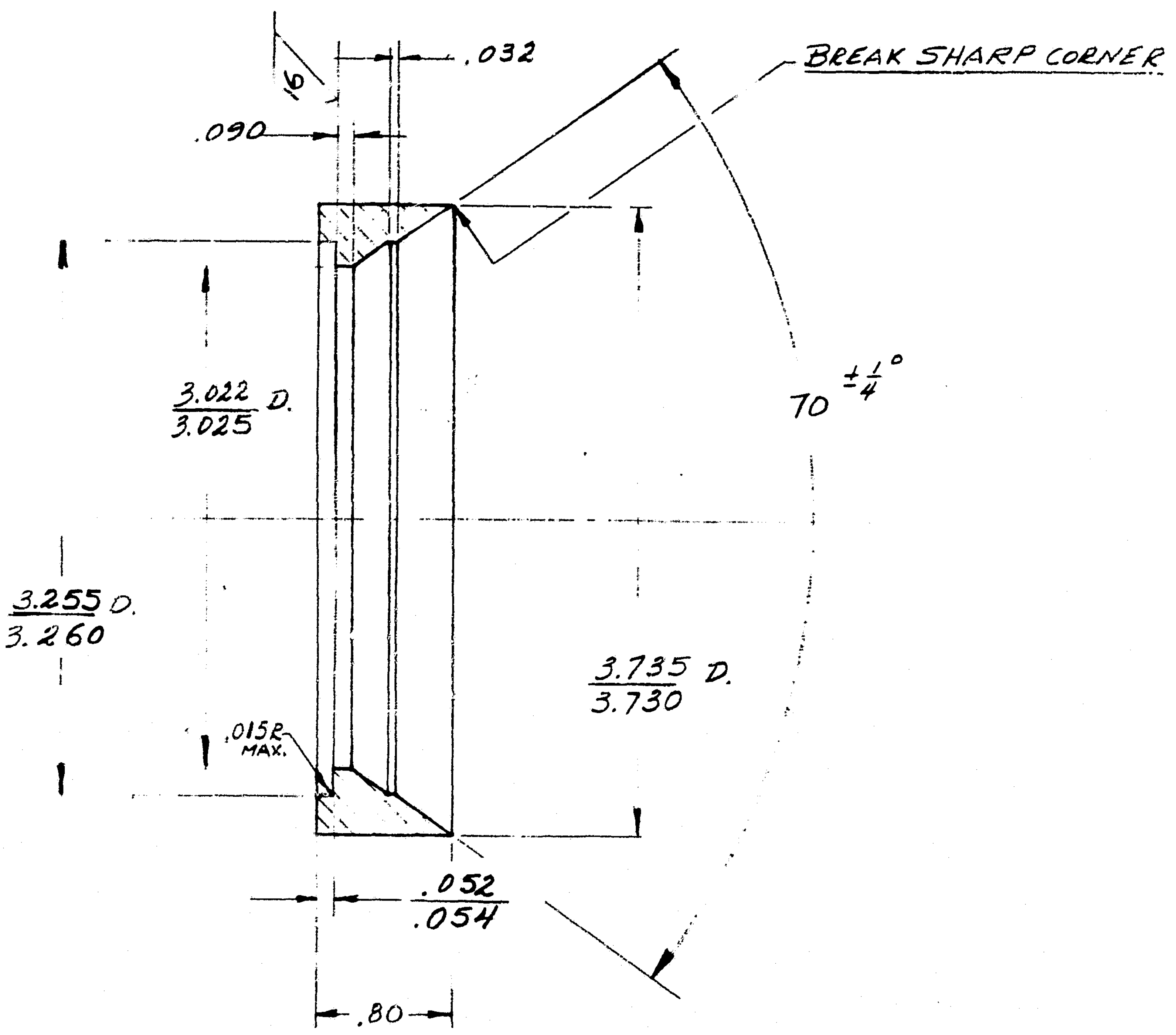


Figure 5-46. Female Adapter, Reed Seal - Three-Inch Rod - Material, 17-4 PH CRS

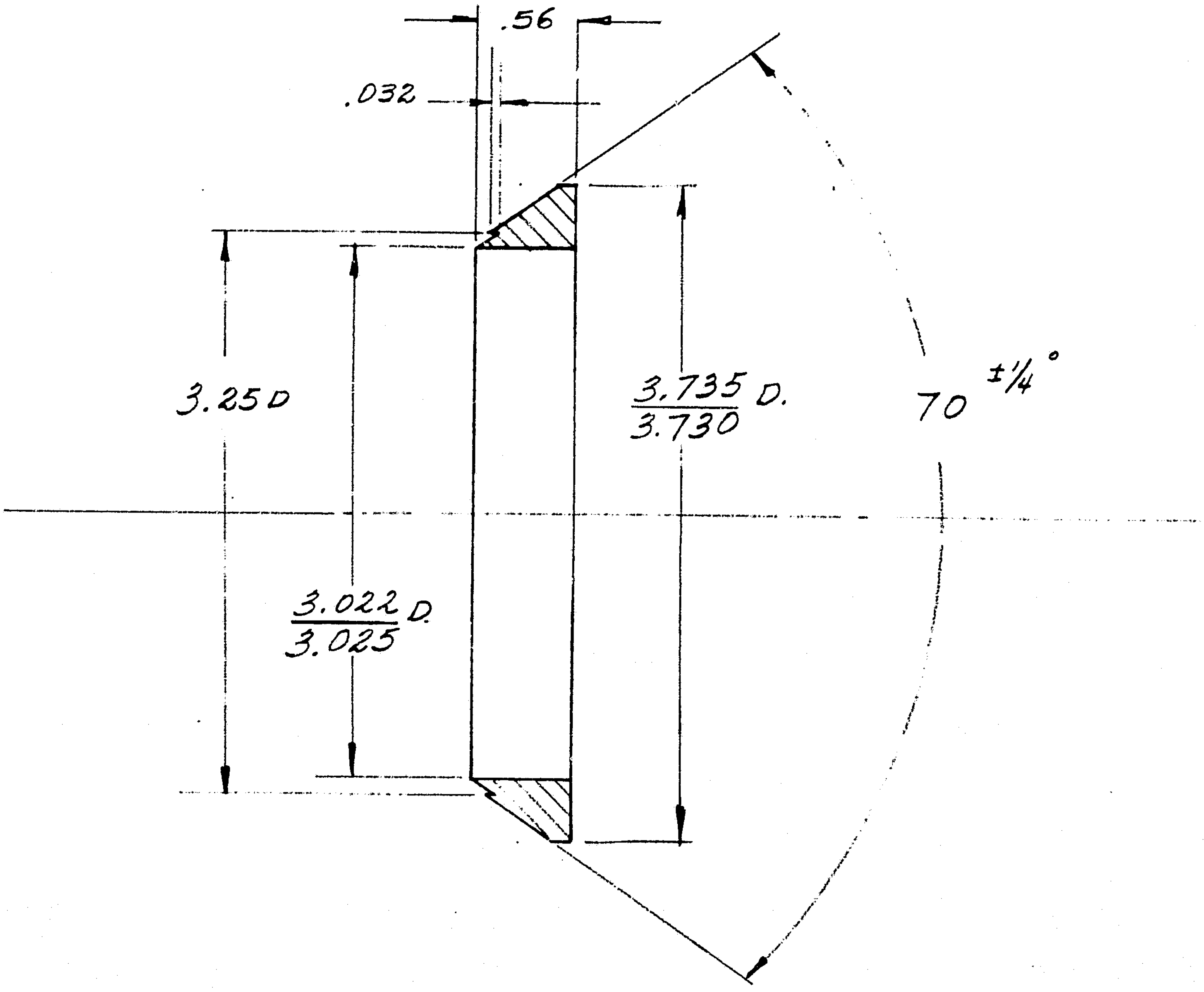


Figure 5-47. Male Adapter, Reed Seal - Three-Inch Rod - Material, 17-4PH CRS

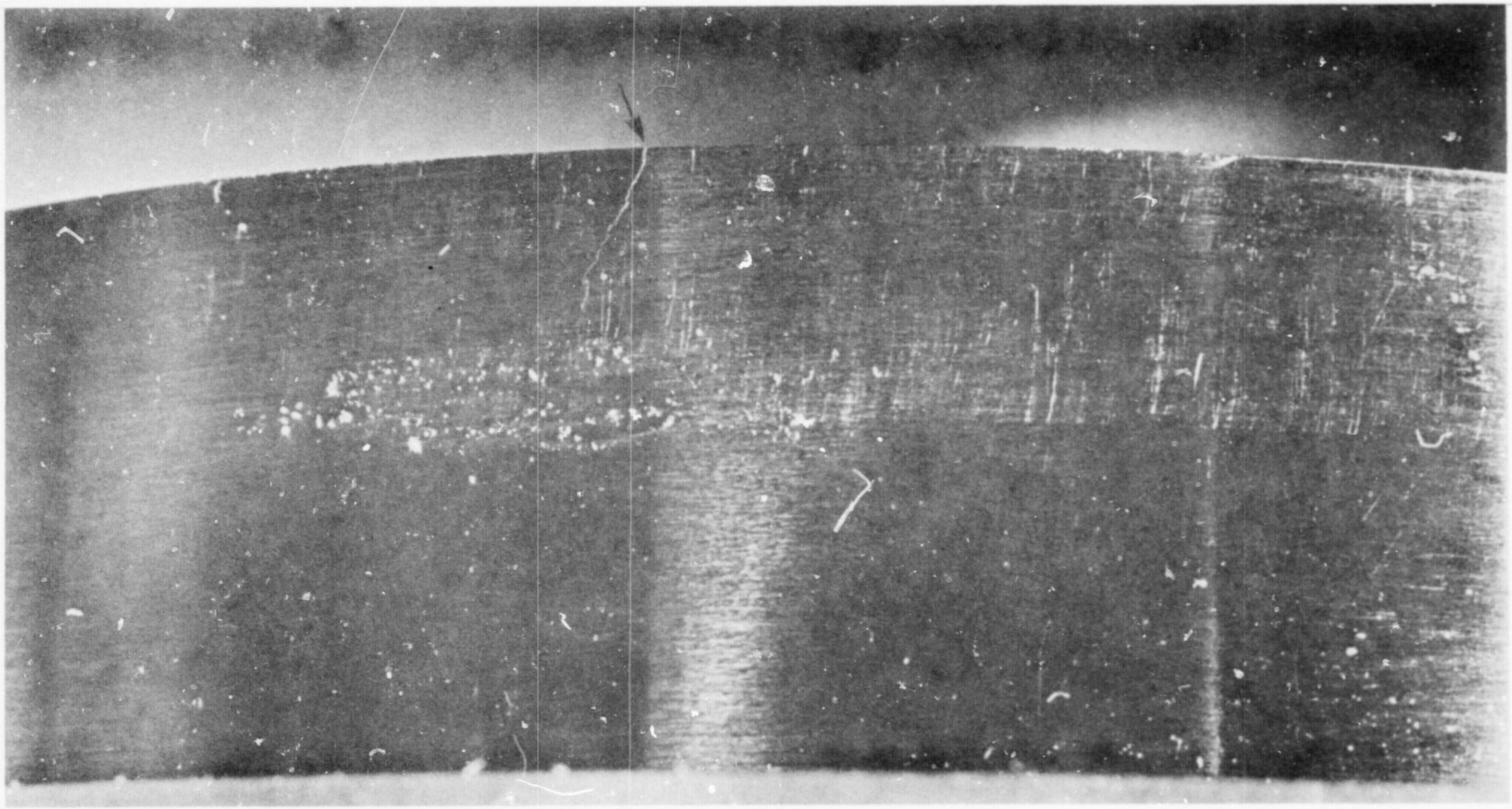


Figure 5-48. Three-Inch Cobalt Molydenum Lip Seal

The reed seal (which consists of four Vascojet 1000 reeds interspersed with four silver alloy reeds) successfully completed the 150-hour low pressure test. Total leakage collected during a total of 174 hours of operation was 60 cc. As reflected in the total number of cycles completed (Table 5-2), short-stroke cycling was performed at a lower rate than usual. The reduced cycling rate (100-200 cpm) was necessary because of a worn bushing in the driving actuator, which produced considerable backlash in the rig.

Inspection shows that the sealing reeds (Figure 5-49) were in good condition. The silver alloy reeds exhibited good contact with the piston rod and showed a retention of 0.002 to 0.003 inch of their original (0.005 to 0.006 inch) interference. (See Table 5-6.) Two of the four Vascojet reeds showed uneven contact with the piston rod. This condition was believed to be the result of out-of-roundness of the reeds due to heat treating, as shown in Table 5-6. These two reeds showed virtually no preload (interference) remaining after test. However, the No. 2 and 4 Vascojet reeds exhibited approximately 0.003-inch interference after test.

The set of polyimide V-seals that was used to replace the cracked cobalt lip seal completed the test with essentially zero leakage. Total leakage collected during the entire test was 1.5 cc. This set of seals was previously subjected to the complete low-pressure evaluation in Test No. 6. Consequently the seal has undergone a total of 332.5 hours of operation. Condition of the seal was good except for a circumferential crack (Figure 5-50) on the seal located closest to pressure. As shown in Figure 5-51, a small portion of the sealing lip was broken off from the seal located furthest from pressure.

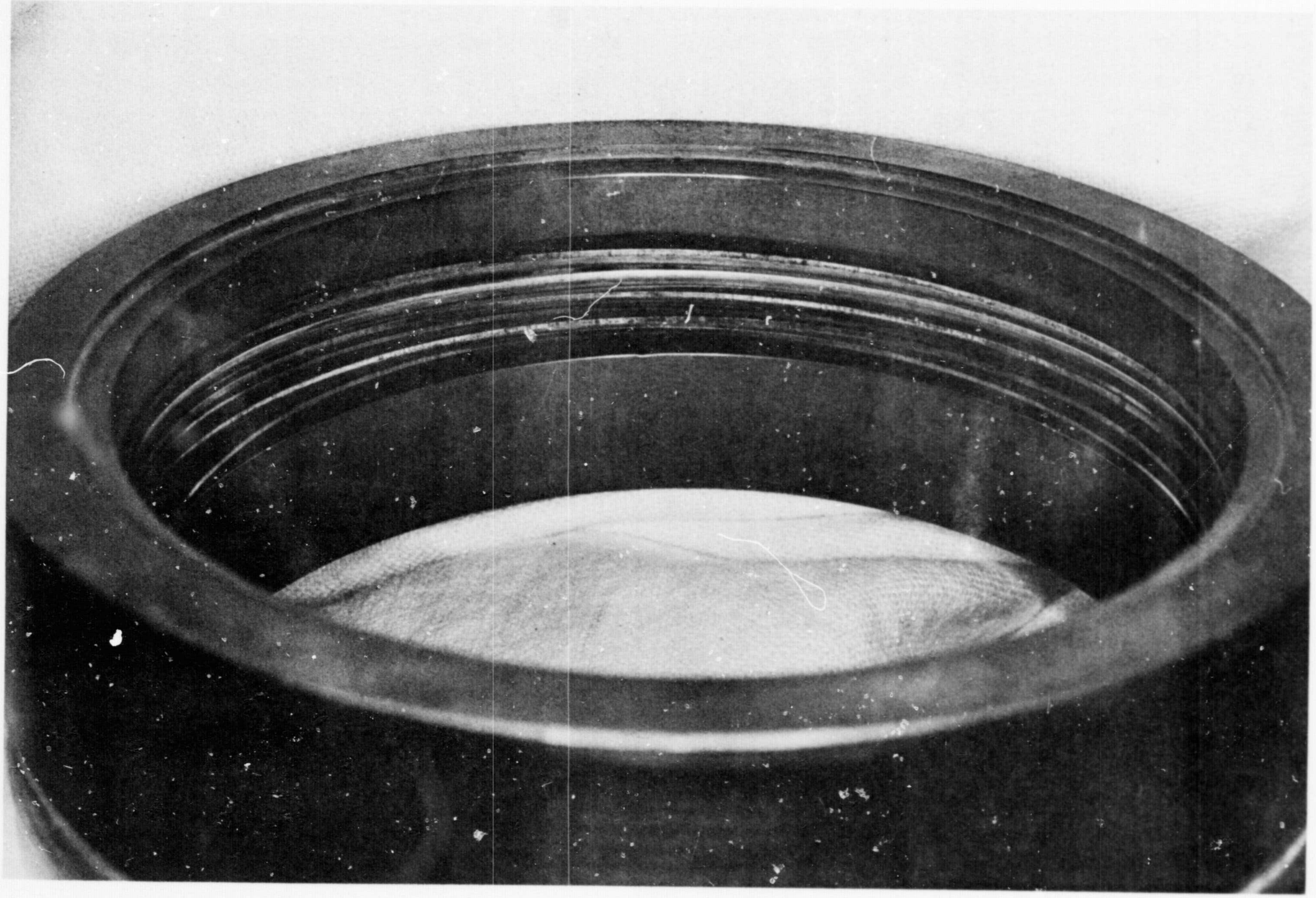


Figure 5-49. Three-Inch Vascojet 1000/Silver Alloy Reed Seal

TABLE 5-6

REED SEAL DIMENSIONS

Piston Rod Diameter - 2.9970 - Inch

Design interference - Vascojet reeds - 0.007 - 0.008 - In.
- Silver reeds - 0.005 - 0.006 - In.

Reed No.	Before Test		After Test	
	*Seal I. D.	Interference	*Seal I. D.	Interference
1 (Silver)	2.992	0.005	2.995	0.002
2 (Vascojet)	2.990	0.007	2.994	0.003
3 (Silver)	2.991	0.006	2.994	0.003
4 (Vascojet)	2.991	0.006	2.994	0.003
5 (Silver)	2.993	0.004	2.995	0.002
6 (Vascojet)	2.993	0.004	2.996	0.001
7 (Silver)	2.990	0.007	2.994	0.003
8 (Vascojet)	2.994	0.007	2.997	0.000

*Average dimensions

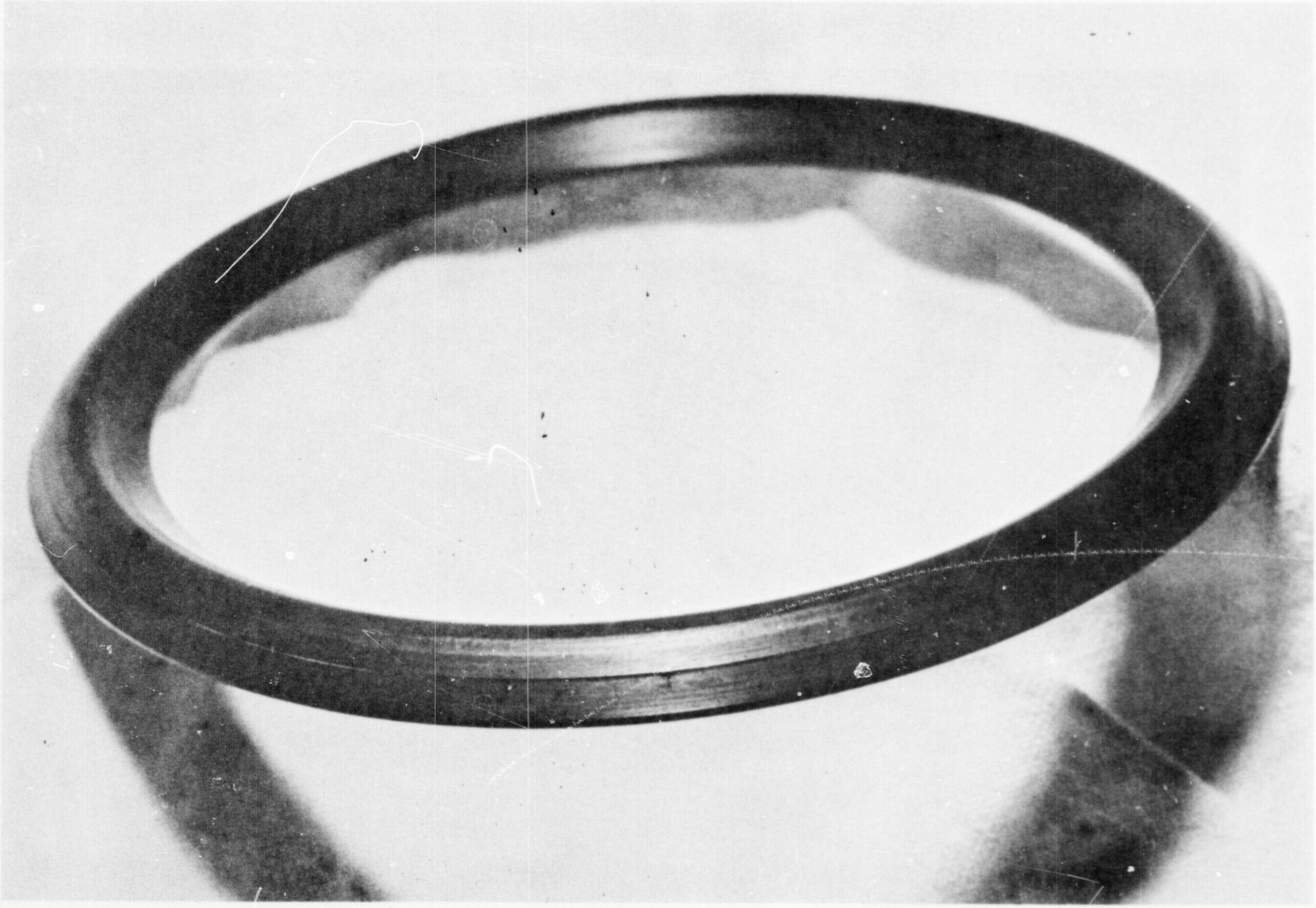


Figure 5-50. Cracked Polyimide V-Seal (three inch)

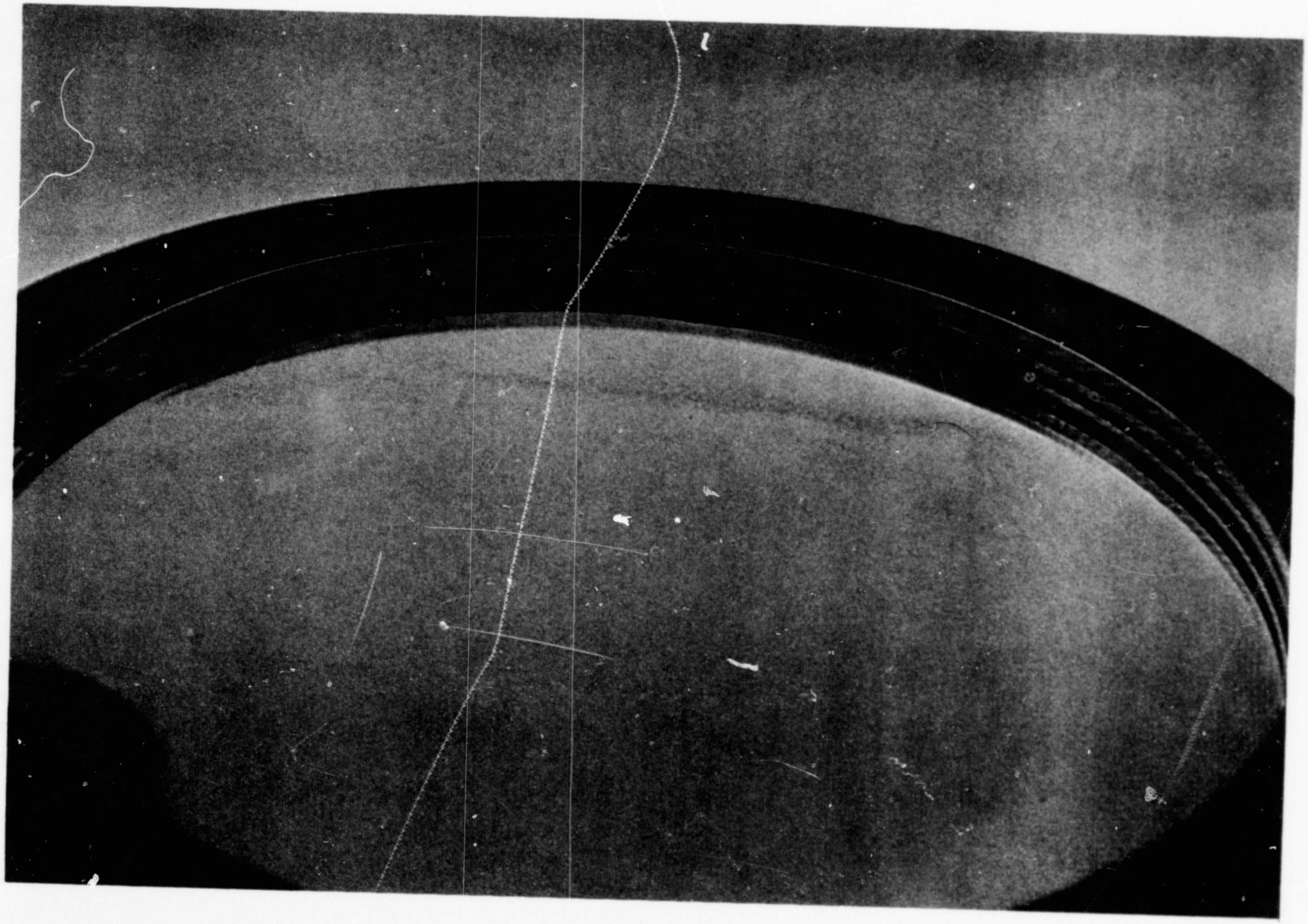


Figure 5-51. Damaged Sealing Lip Polyimide V-Seal (three inch)

SECTION VI
ENDURANCE TESTING

A. GENERAL

Based on the results obtained in the low-pressure test phase, the polyimide V-seal (Design B), cobalt molybdenum lip seal (Design D), and Vascojet 1000/silver alloy reed seal (Design AH) were selected for endurance testing. These configurations were evaluated in the one-inch and three-inch rod sizes. Polyimide pressure balanced split sealing rings were selected for the high-pressure first-stage and were combined with the second stage to make up the complete two-stage seal assembly. In the endurance test, the first-stage seals were subjected to pressures to 4000 psi. The second-stage seals were pressurized to a nominal 100 psi. Testing was conducted at temperatures to 500°F using F-50 silicone as the test fluid. The seals were tested to failure or completion of 3000 hours. Seal failure was defined as leakage in excess of one drop per minute or a two-fold increase in seal friction.

B. TEST PARAMETERS

The endurance test duty cycle is shown in Figure 6-1. The different pressure levels are representative of a flight control actuator. Short excursions from actuator load neutral, such as occur during cruise, comprise the major portion of the profile. Near the neutral positions, both sides of the actuator piston sense approximately one-half system pressure. The maximum and minimum pressures occur during low-speed flight, when large control surface loads and excursions are encountered.

The time required to complete the sequence of events depicted in Figure 6-1 (which represents one flight mission) is approximately 180 minutes. During this time period, the fluid in the actuators was heated to 500°F in approximately 35 minutes, held at 500°F for 115 minutes, and then cooled down to approximately 100°F before the start of the next mission.

The sequence of operation for each mission was as follows:

1) Heat fluid in actuators to 500°F in approximately 35 minutes

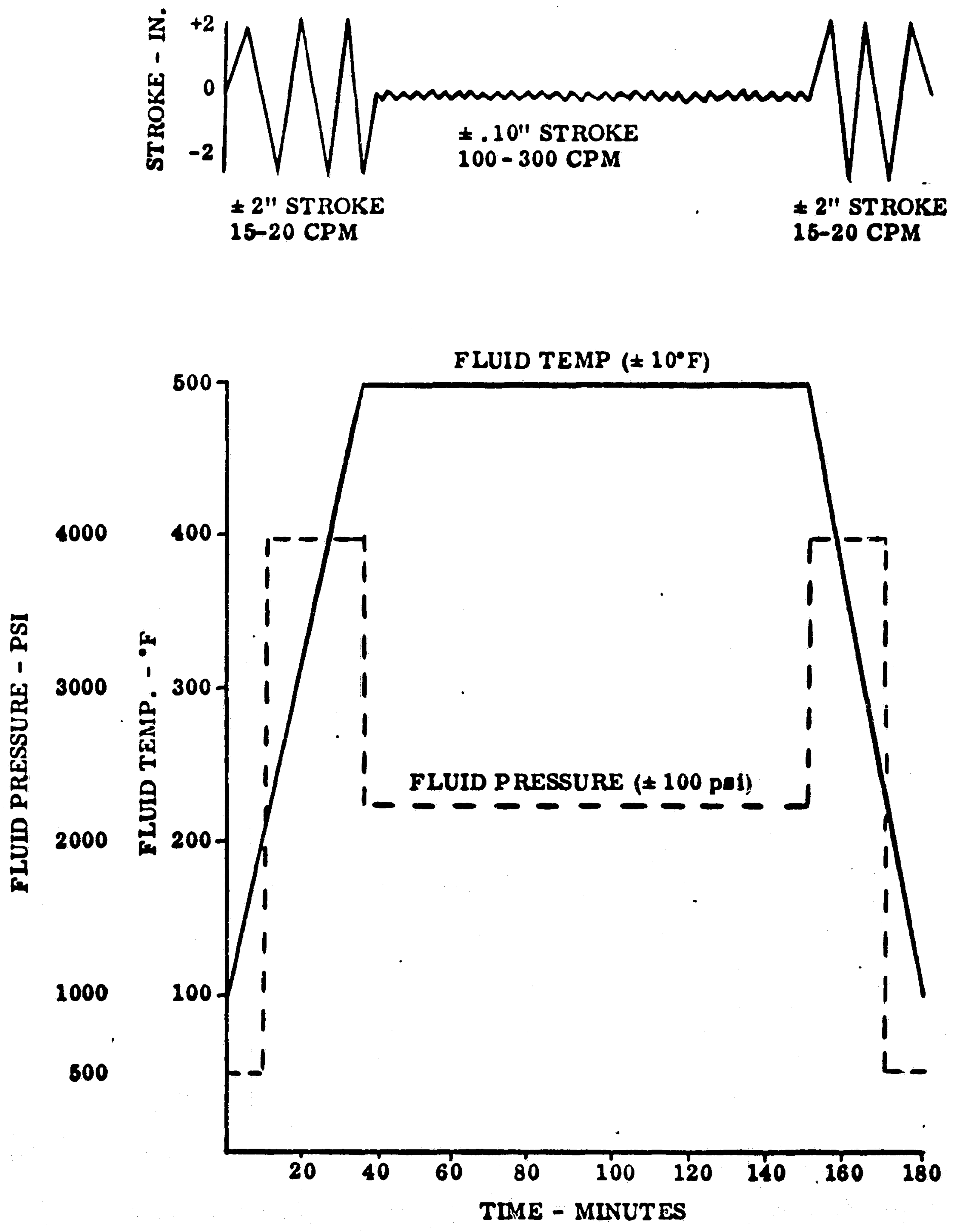


Figure 6-1. Endurance Test Duty Cycle

2) Conduct long-stroke cycling (± 2" stroke at 15-20 cpm) at 500 psi for approximately ten minutes

3) Continue long-stroke cycling (same stroke and rate) and increase pressure in actuators to 4000 psi. Maintain operation for approximately 25 minutes or until fluid temperature reaches 500°F

4) With actuator fluid temperature at 500°F, change to short-stroke cycling mode (± 0.10" @ 100-300 cpm). Maintain operation for approximately 115 minutes at fluid pressure of 2200-2500 psi

5) At completion of 4), start cool-down of fluid to 100°F

6) During cool-down change to long-stroke cycling (± 2" @ 15-20 cpm) at 4000 psi for approximately 20 minutes

7) Lower fluid pressure to 500 psi and maintain same cyclic mode for approximately 10 minutes

C. TEST EQUIPMENT AND PROCEDURE

The cycling rig as shown in Figure 6-2 is basically the same as that used in the low-pressure test phase. A schematic of the hydraulic circuit is shown in Figure 6-3. The system consists of a high-pressure pump, a fluid reservoir, flowmeters, and shutoff valves. The pump delivers fluid at 4000 psi to the three test actuators. Return lines leading from the A and B-end of each actuator are pressurized to 100 psi. These lines are situated between the first-and second-stage seals and are used to return first-stage seal leakage back to the reservoir via the flowmeters. Automatic shutoff valves are located on each of the return lines. During operation these valves are in the normally opened position. When the flowmeters indicate excessive first-stage seal leakage, the valves are closed in a sequential manner to isolate the seal that is leaking excessively. When the valve is closed, leakage by-passes the flowmeter and returns directly to the reservoir.

D. TEST CONFIGURATION

Details of the one-inch seal configurations tested were essentially the same as those shown in Figures 4-5, 4-6, 4-7, 4-20, and 4-29 through 4-32, which were used in the low-pressure test phase. For the V-seal (Figure 4-5) the inside included angle of the "V" was changed from 96° to 90° which corresponds to the included angle of the outer surface of the "V". This change was made to provide better support of the inner and outer lips of the V-seal. Details of the three-inch seals tested are shown in Figures 5-37, 5-38, and 5-43 through 5-47.

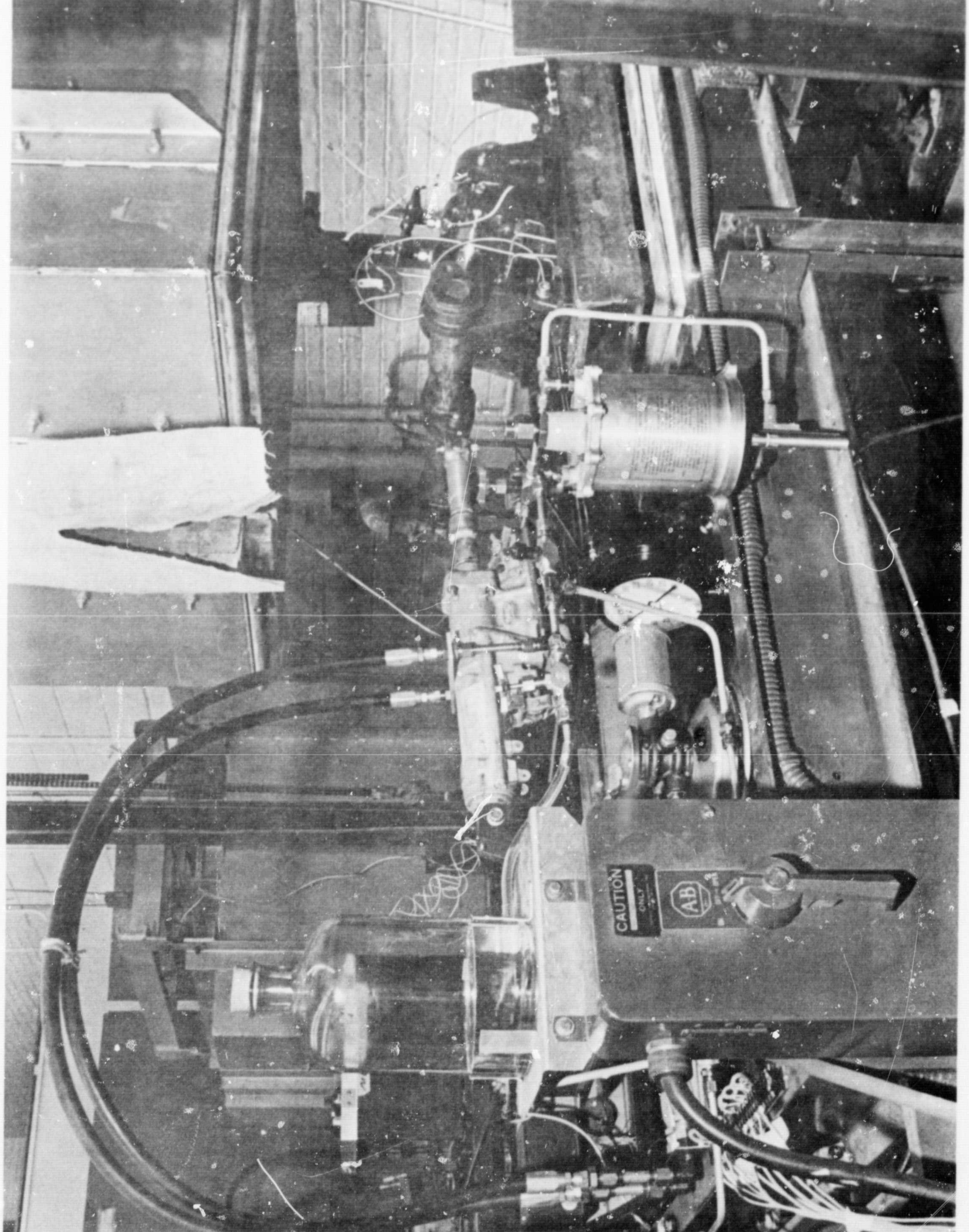


Figure 6-2. Actuator Cycling Rig

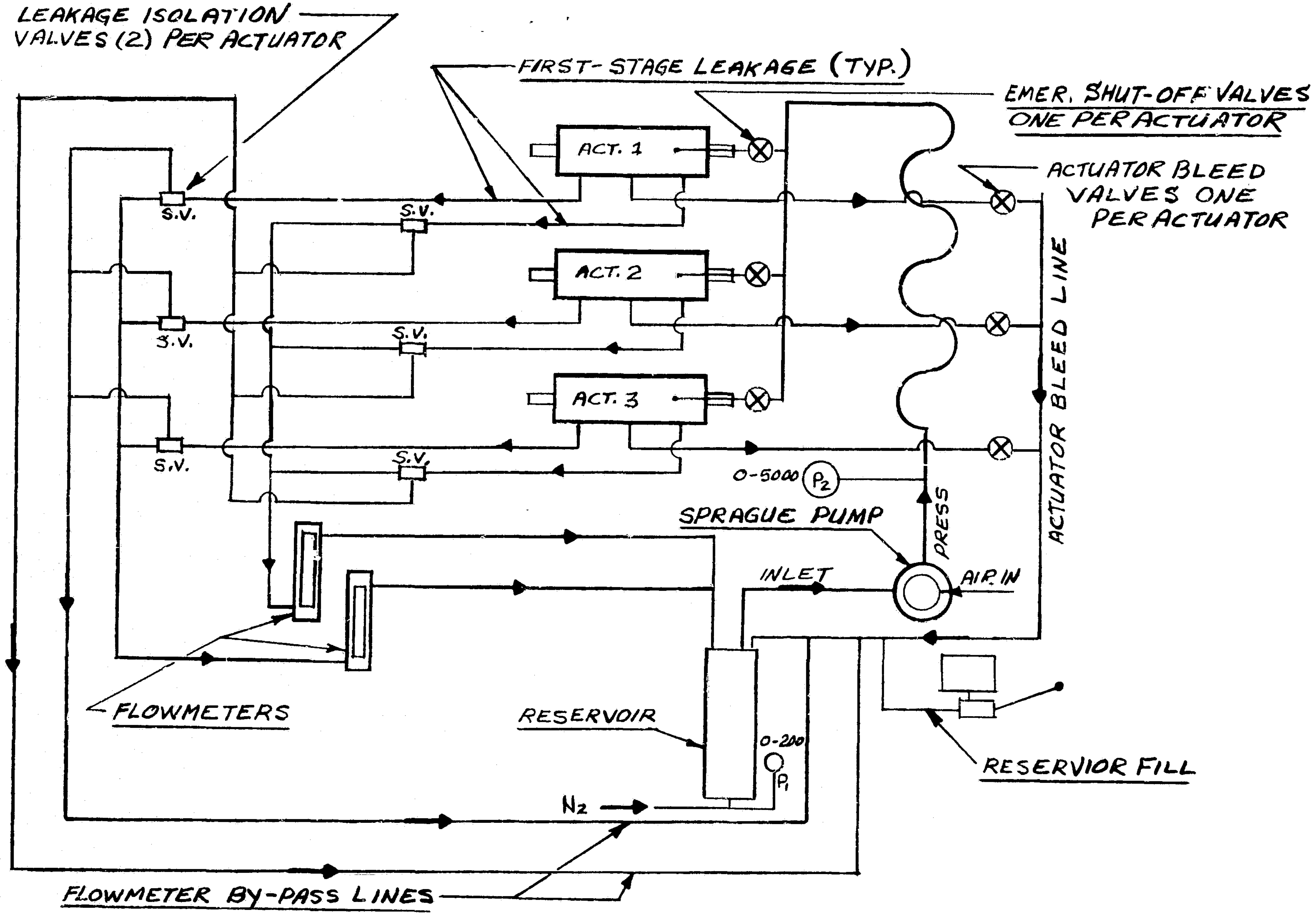


Figure 6-3. Hydraulic Circuit Schematic

TABLE 6-1. DETAILS OF ONE-INCH ROD SEALS USED IN ENDU

Seal Configuration			Piston Rod Diam.	Rod/Gland Bearing Clearance	Sea
Test Actuator	First-Stage (4000 psi)	Second-Stage (100 psi)			First-St
1-1	Polyimide pressure balance split sealing ring	Design B-Polyimide V-seals. 3 V-seals per gland seal installed on rod with approx. 0.002-0.003-inch interference. Spring load on seals, 200 pounds.	0.9974	0.0007 0.0011	A-end 1 B-end 1
2-2	Same as above	Design D - cobalt molybdenum lip seal. Seal assembled with interference fit of approx. 0.0013-0.0014-inch was modified to 0.0007-0.0008 inch.	0.9978	0.0009 0.0012	A-end 1 B-end 1
2-3	Same as above	Design AH-Vascojet 1000/ silver alloy reed seals. 4 Vascojet reeds and 4 silver reeds per gland. Vascojet reeds assembled with 0.0036-inch interference. Silver reeds with 0.0025-inch interference.	0.998	0.007	A-end 1 B-end 2

RANCE TESTS

Leakage - dpm		Seal Friction - pounds		
ge	Second-Stage	First-Stage (@ 4000 psi)	Second-Stage (@ 100 psi)	Total
	0 0	71	124	200
	0 0	71	80	145
	0 0	78	170	250

TABLE 6-2. DETAILS OF THREE-INCH ROD

Seal Configuration			Piston Rod Diam.	Ro Bea Cle
Test Actuator	First-Stage (4000 psi)	Second-Stage (100 psi)		
3-1	Polyimide pressure-balanced, split sealing rings	Design B Polyimide V-seals 3 V-seals per gland Seals installed on rod with approx. 0.012 interference. Spring load on seals 1200 pounds	2.9970	0. 0.
3-2	Same as above	Design D - cobalt molybdenum alloy lip seal (B-end only) Lip seal assembled with 0.004-0.005-inch interference. B-end assembled Assembled with Polyimide V-seal.	2.9975	0.
3-3	Same as above	Design AH-Vascojet 1000/silver alloy reed seals. 4 silver and 4 Vascojet reeds per gland. Silver reeds assembled with 0.006-0.007-inch interference. Vascojet reeds assembled with 0.007-0.009-inch interference.	2.9970	0.

LS USED IN ENDURANCE TESTS

nd ce	Seal Leakage (dpm)		Seal Friction - Pounds		
	First-Stage (4000 psi)	Second-Stage (100 psi)	First-Stage (@ 4000 psi)	Second-Stage (@ 100 psi)	Total
-	A-end 1	0 0	310	124	446
	A-end 12 dpm B-end 8 dpm	0 0	300	140	450
	A-end 10 B-end 10	0 0	300	600	860

Tables 6-1 and 6-2 show the assembly details and the leakage and friction characteristics of each seal prior to test. Basic details of the first-stage seal used in all the actuators are shown in Figures 3-14 and 4-37.

E. RESULTS OF ENDURANCE TESTING

Performance of the seals during the endurance testing is summarized in Table 6-3. The one-inch-size polyimide V-seals exhibited the most outstanding performance of all the seals tested. The A-end and B-end seals met the minimum leakage requirement of less than one drop per minute for 1,322 and 1,358 hours, respectively, while accumulating over nine million cycles. The three-inch V-seals met the leakage requirements during 664 hours and 805 hours of operation. Testing of both the one-inch and three-inch V-seals were concluded because of excessive leakage. This condition was caused by circumferential and/or radial cracks on the polyimide material.

The one-inch Vascojet 1000/silver alloy reed seals produced an endurance life of 574 hours and 433 hours for A-end and B-end seals, respectively. An endurance life of 428 hours was obtained with the three-inch reed seals. Leakage of the seals was caused by wear of the chrome-plated rod during short-stroke cycling.

One each of the one-inch and three-inch cobalt lip seals were successfully tested. Operating life of these seals were 170 hours and 279 hours for the one-inch and three-inch seals, respectively. Leakage exceeded one drop per minute because of wear on the chrome plated rod produced during the short-stroke cycle operation.

These results demonstrated the feasibility of each seal design developed during the program. The polyimide material appears most promising. However, further investigation is required to determine the cause of material failure. Although the hard-metal seals are insensitive to the temperature environment, their ability to operate under marginal lubrication conditions is limited. However, it is believed that use of harder coatings for the mating rod surface may produce a substantial increase in seal life.

TABLE 6-3. SUMMARY OF ENDURANCE TEST RESULTS

Seal Configuration	Total Test Time (hr)	Time at 500°F (hr)	Total Cycles	Cycles at 500°F	Thermal Cycles	Accumulated Leakage (CC) A-end	Accumulated Leakage (CC) B-end
Design B (one-inch)							
Polyimide V-seal A-end	1,322	617.5	9,773,636	8,771,871	301	365	
B-end	1,358.2	631.5	9,987,325	8,921,071	315		196
Design B (three-inch)							
Polyimide V-seal A-end	664.2	275.5	4,526,433	4,087,352	139	395	
B-end	805.9	354	5,814,219	4,960,596	182		893
Design AH (one-inch) Vascojet 1000/silver alloy							
Reed Seal - A-end	574	252	3,612,351	3,112,040	124	483	
B-end	433	185.5	2,590,075	2,205,882	89		270
Design AH (three-inch) Vascojet 1000/silver alloy							
Reed Seal - A-end	428	178.8	2,904,513	2,480,909	91	217	
B-end	428	178.8	2,904,513	2,480,909	91		370
** Design D (one-inch)							
Cobalt Moly Lip Seal B-end	170.8	77	1,353,314	1,039,660	38		364
Polyimide V-seal A-end	170.8	77	1,353,314	1,039,660	38	2.5	
* Design D (three-inch)							
Cobalt Moly Lip Seal B-end	279	132	2,110,048	1,829,347	68		692
Polyimide V-seal A-end	279	132	2,110,048	1,829,347	68	19	

* One lip seal tested in 3-inch size

**A-end lip seal cracked and was replaced with Polyimide V-seal

Detailed discussions of each seal configuration tested follow.

1. <u>One-Inch Polyimide V-Seal (Design B)</u>

Leakage from these seals was negligible up to approximately 1200 hours of operation. As shown in Figure 6-4, accumulated leakage after 1267 hours was 40cc and 15cc for the A-end and B-end seals, respectively. From that point on, leakage increased quite rapidly. Testing of the A-end seal was concluded after 1332 hours when leakage of 10 drops per minute was experienced. Testing of the B-end seal was terminated after 1358 hours when leakage exceeded one drop per minute. Leakage during the final four hours of testing was approximately six drops per minute and reached as high as 40 drops per minute at the conclusion of testing. Excessive leakage in both cases was determined to have been caused by cracks in the polyimide material, which provided the leakage path for the fluid.

Figures 6-5 and 6-6, depicts the condition of the seals after the endurance test. Circumferential and/or radial cracks were noted on all the seals. Typical failure of the material is shown in Figures 6-7, 6-8, and 6-9. The seal that was situated closest to the fluid pressure (Figure 6-7) appears to have experienced the most damage. This seal was cracked in the radial and circumferential direction. The middle seal (Figure 6-8) exhibited a radial crack only. The last seal (Figure 6-9) in the stack exhibited the least damage.

Based on the condition of the seals, it appears that seal damage decreases when the seals are not in direct contact with the fluid. This seems to indicate that the F-50 silicone fluid may have contributed to the cracking of the material. Another possible cause of material failure is the high coefficient of thermal expansion of the polyimide material. The design of the seal assembly is such that the seal is confined in the gland cavity, which restrains it from expanding at high temperatures. This condition results in a build up of high stresses in the material. As the seals are preloaded at assembly by the spring washers, the additional stresses caused by thermal expansion tends to aggravate the condition. These thermal stresses are cyclic because the seals are subjected to a heating and cooling cycle. This condition could cause fatigue of the material, resulting in cracking.

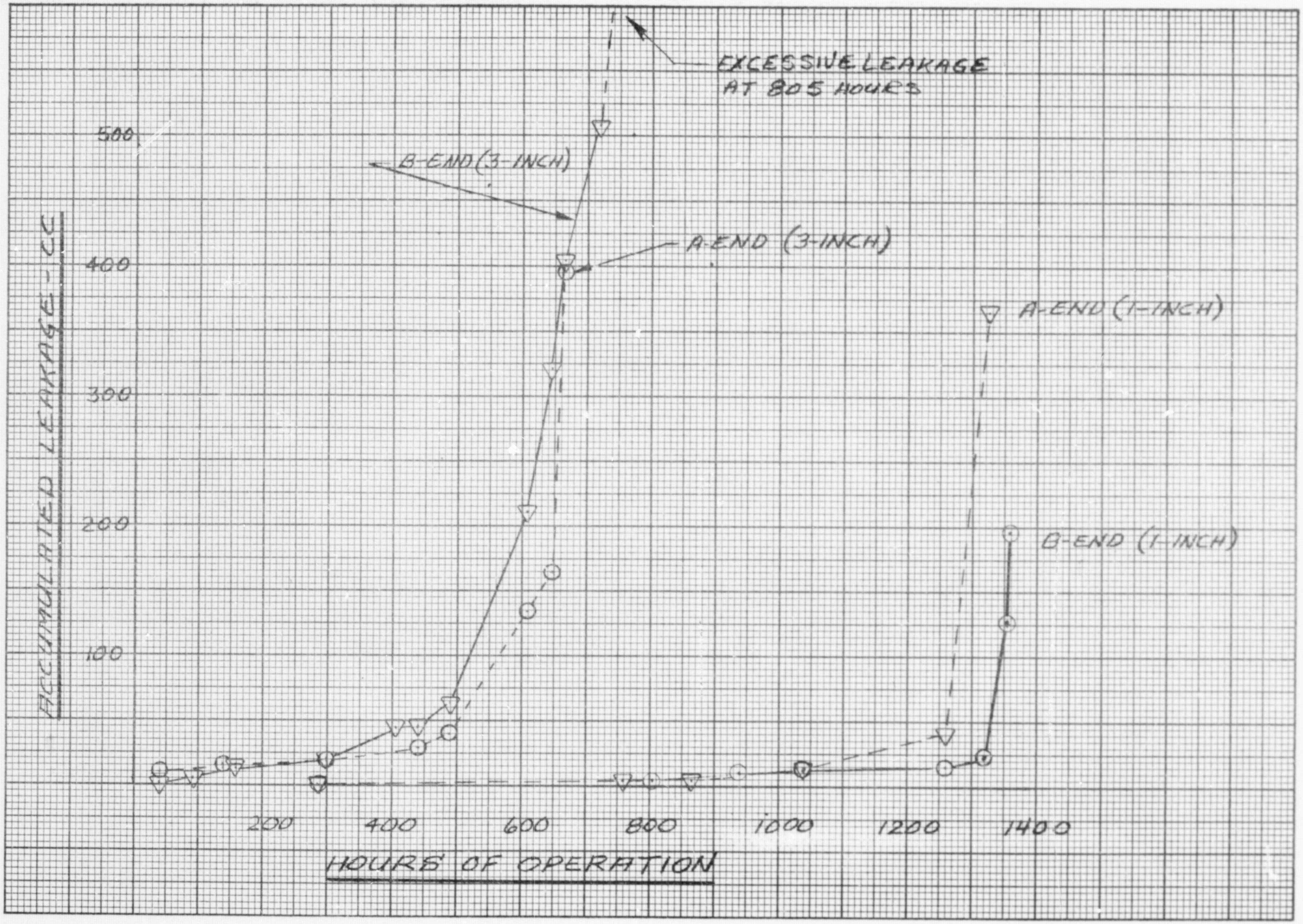


Figure 6-4. Accumulated Leakage versus Time, V-Seals

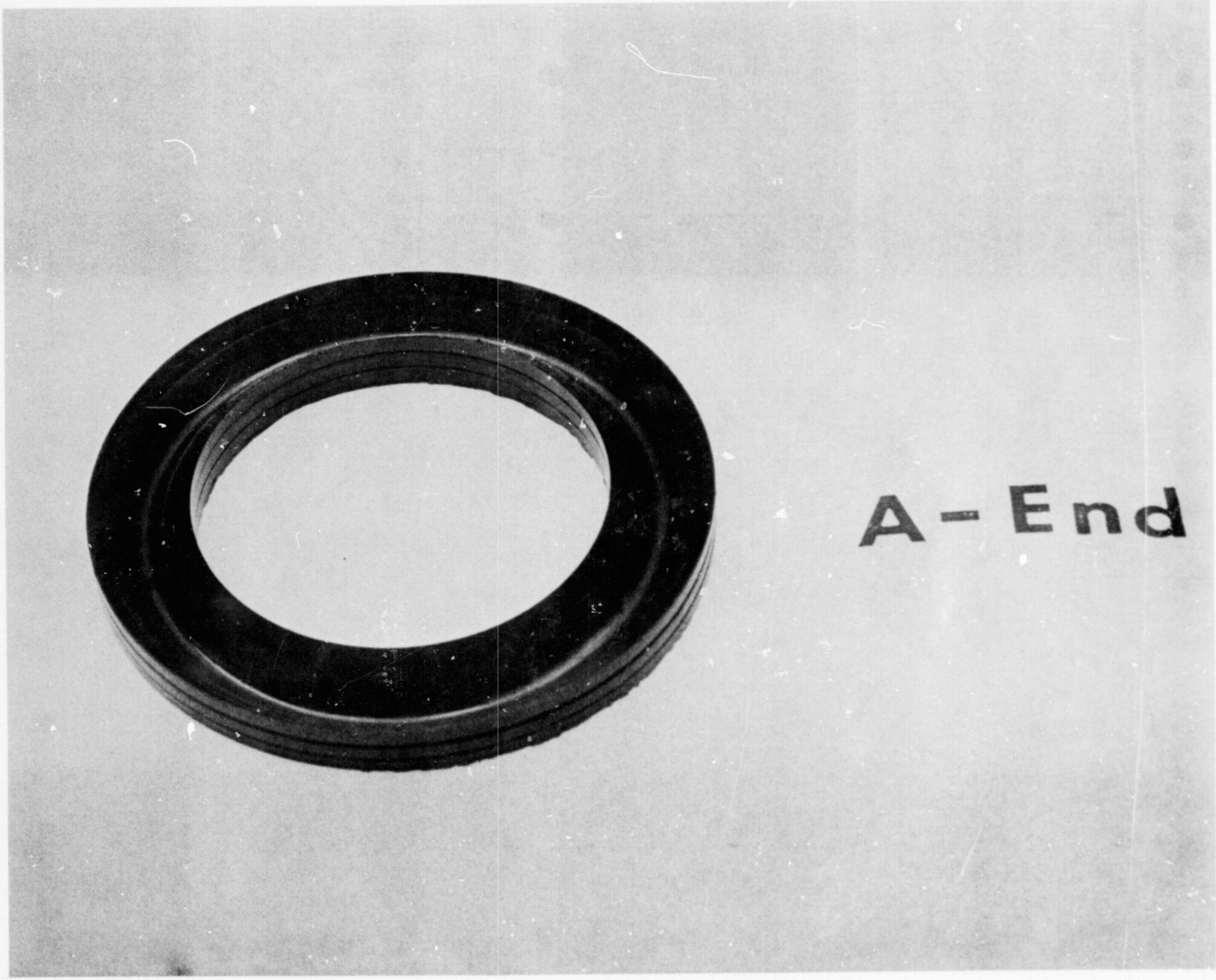


Figure 6-5. One-Inch Polyimide V-Seal, A-End

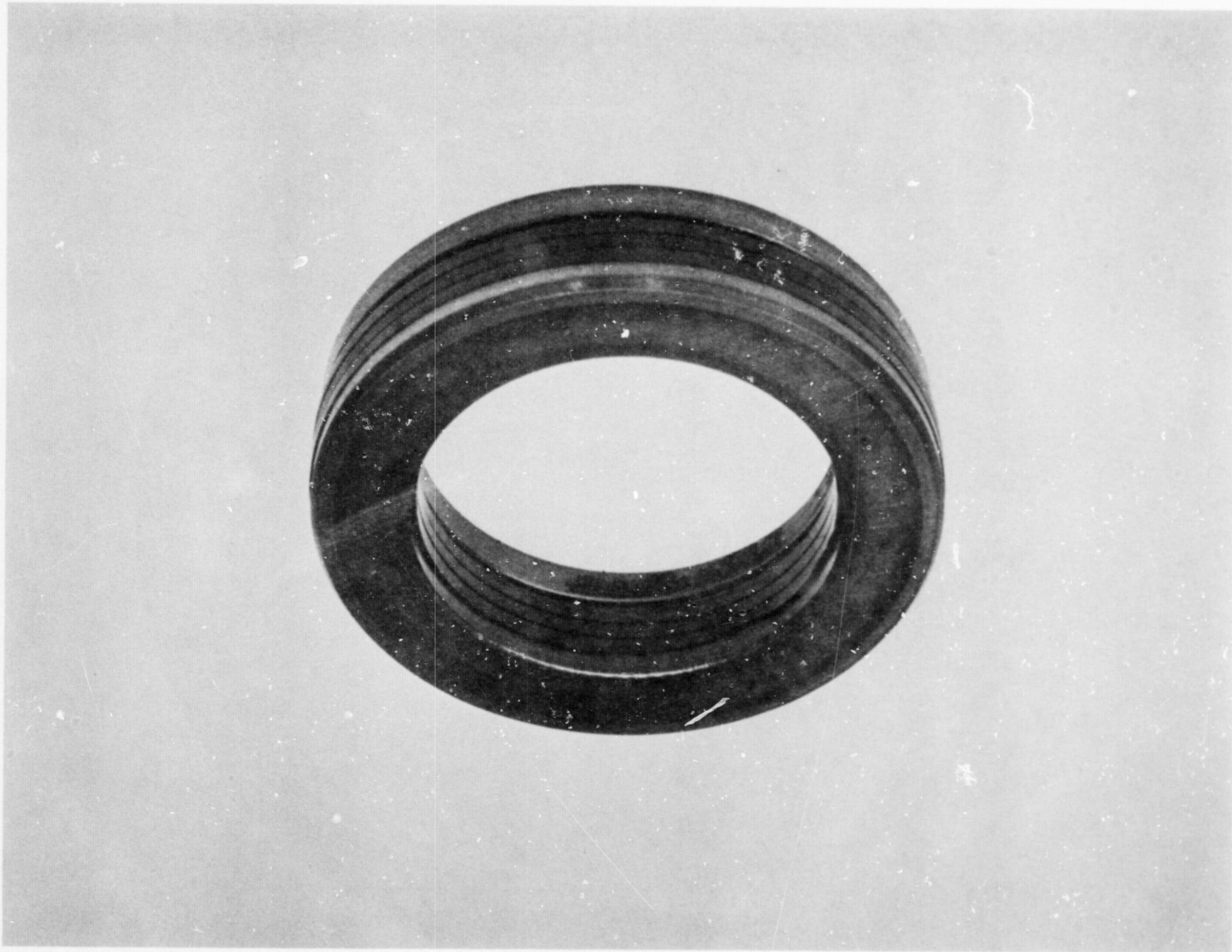


Figure 6-6. One-Inch Polyimide V-Seal, B-End

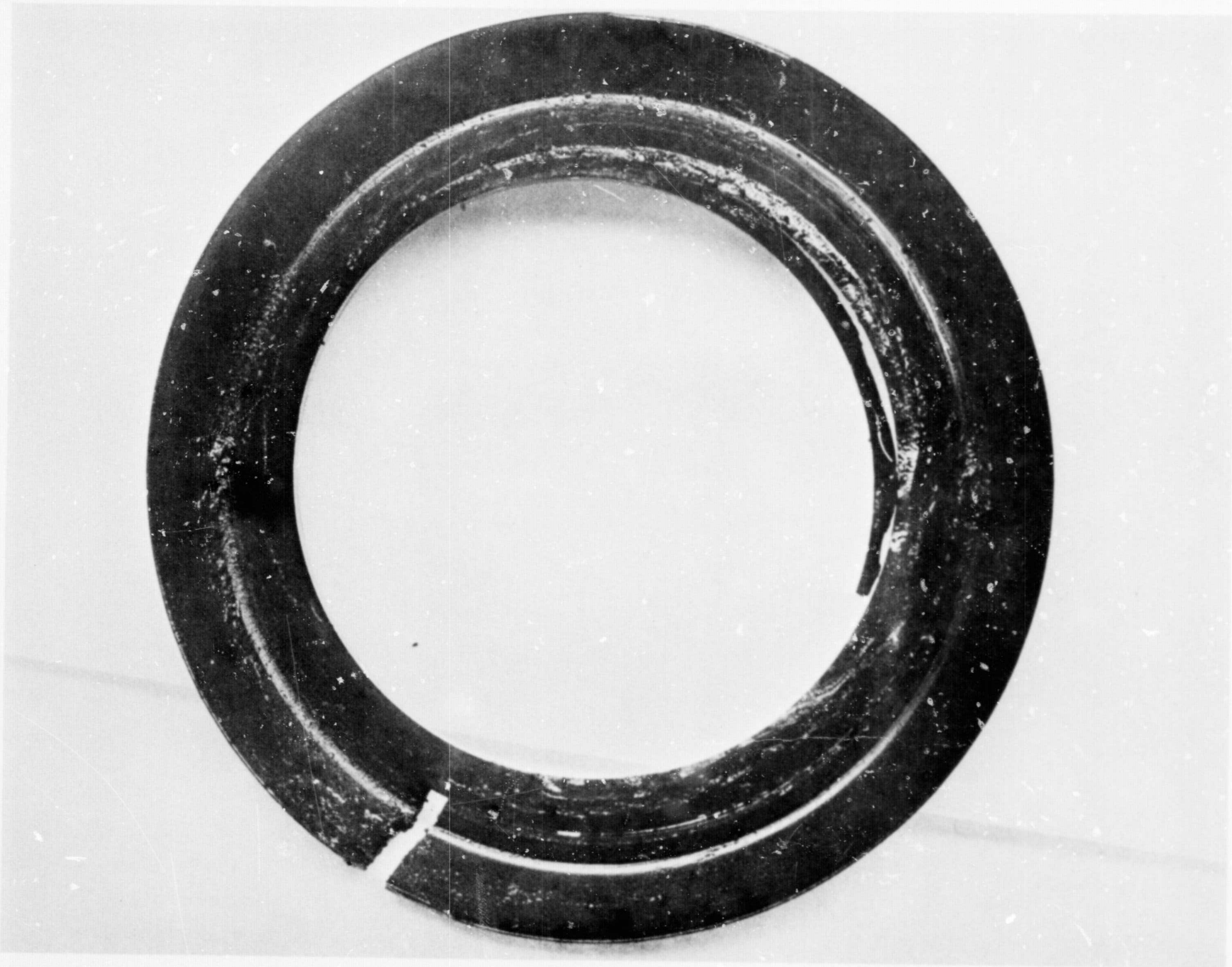


Figure 6-7. One-Inch Polyimide V-Seal, B-End, No. 1

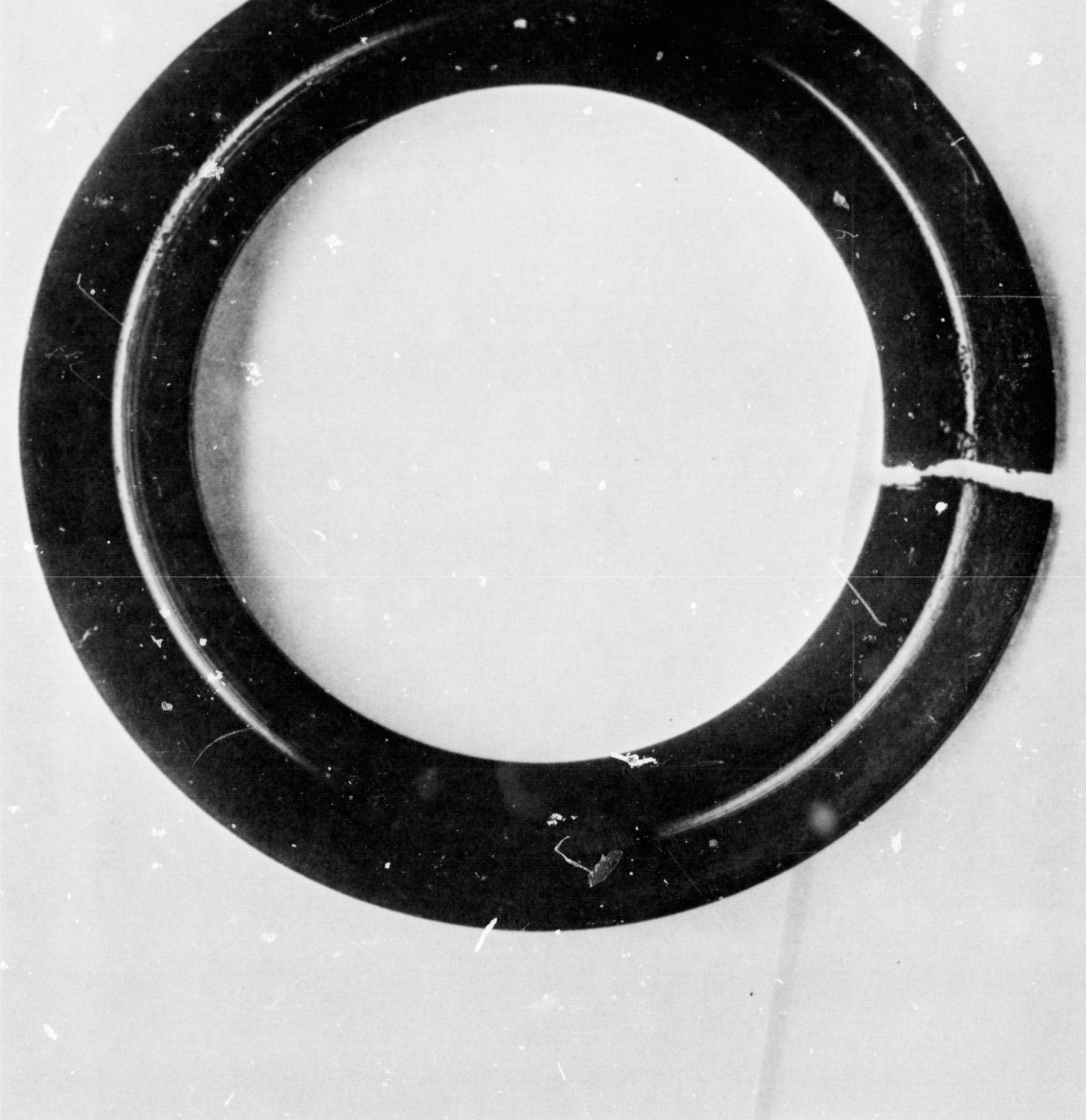


Figure 6-8. One-Inch Polyimide V-Seal, B-End, No. 2

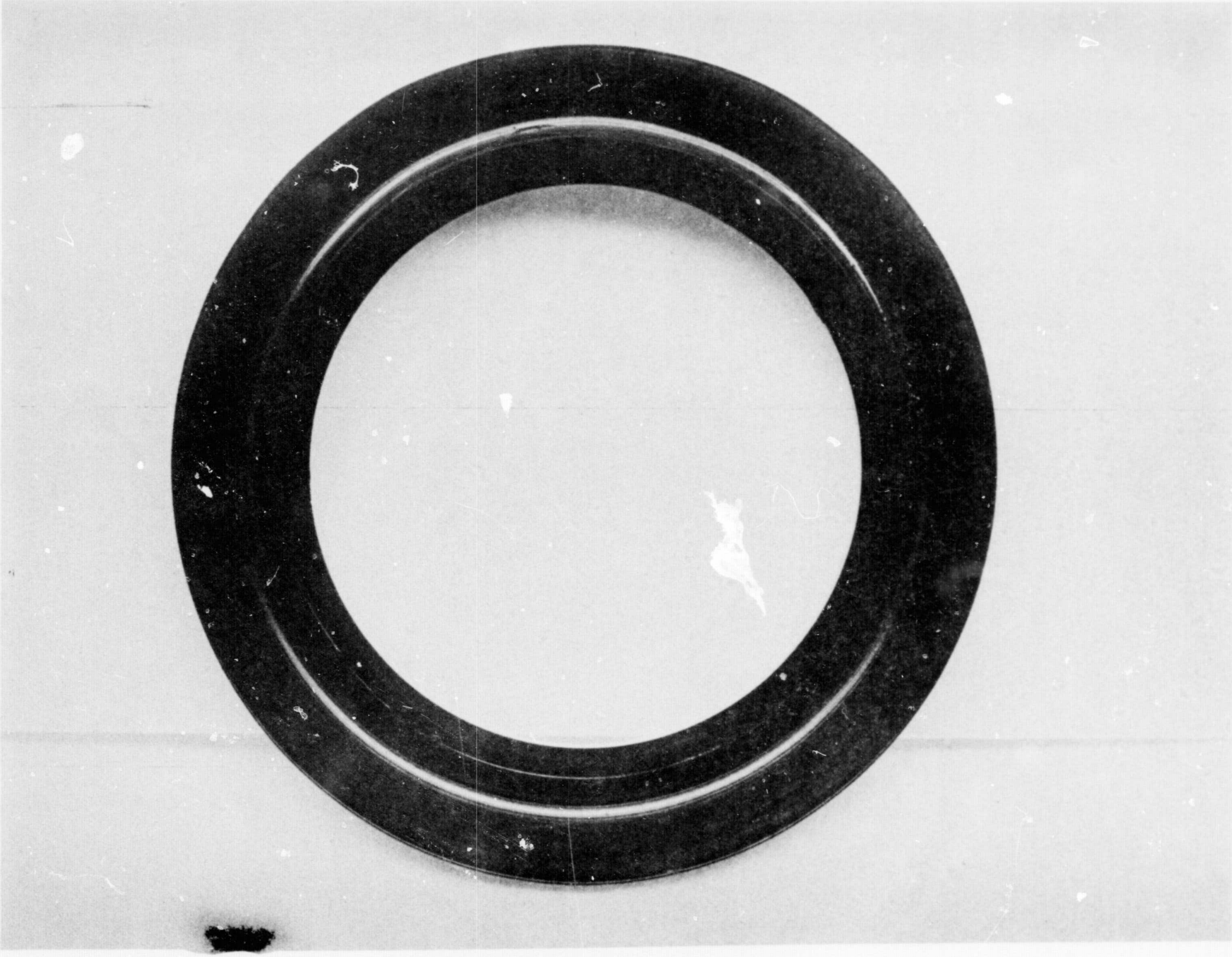


Figure 6-9. One-Inch Polyimide V-Seal, B-End, No. 3

Measurements were taken of the seals after test and are compared in Table 6-4 with the original dimensions. Although some changes in dimensions are indicated, it is difficult to consider these changes as a measure of wear because of the inherent shrinkage and creep characteristics of the material. For instance, the inner diameter of the seals showed only slight increase from the

TABLE 6-4. SEAL DIMENSIONS - ONE-INCH V-SEALS
ROD DIAMETER - 0.9974

End	Seal No.	Before Test		Interference	After Test		Interference
		I.D.	O.D.		I.D.	O.D.	
A	1	0.9954	1.502	0.002	0.9965	1.494	0.0009
	2	0.9950	1.501	0.0024	0.996	1.493	0.001
	3	0.9945	1.502	0.0029	0.9945	1.492	0.0029
B	1	0.9940	1.504	0.0034	0.9955	1.497	0.0019
	2	0.9945	1.502	0.0029	0.9970	1.500	0.0004
	3	0.9940	1.502	0.0034	0.9960	1.498	0.0014

original diameter, yet visual inspection of the seals indicated that wear of the material had occurred. This condition is better illustrated by considering the outer diameter of the seal. This surface acts as a static seal with little or no relative motion. Still this diameter showed a decrease of 0.008 to 0.010-inch as compared to the original diamensions.

With the exception of slight pitting of the chrome plate, the piston rod appears to be in good condition. The long-stroke cycling operation produced no measureable wear on the chrome plated surface. However, a slight wear pattern was produced during short-stroke cycling (see Figures 6-10 and 6-11). Wear in these areas was determined with a surface analyzer and was found to be approximately 0.00001-inch and 0.00003-inch on the A-end and B-end of the rod, respectively.

Replacement of the high-pressure first-stage seal on the A-end was required when excessive leakage developed after 570 hours (4,580,000 cycles) of operation. Leakage from this seal increased from 5cc per minute to 20cc per minute. The high leakage was caused by a break at the step joint (Figure 6-12) of the sealing ring. This type of failure was not unusual, as the Polyimide material

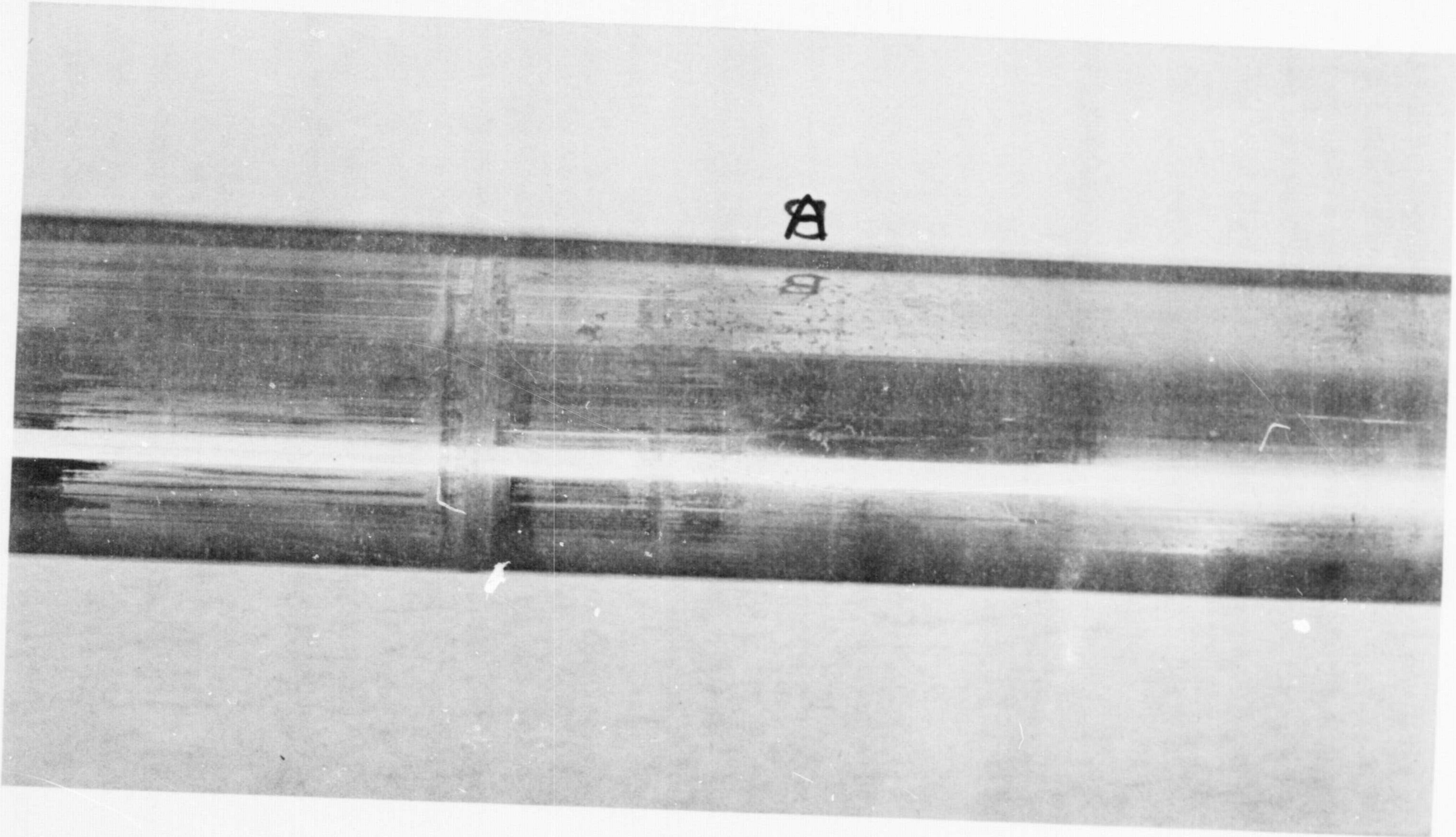


Figure 6-10. Piston Rod, A-End, One-Inch Polyimide V-Seal

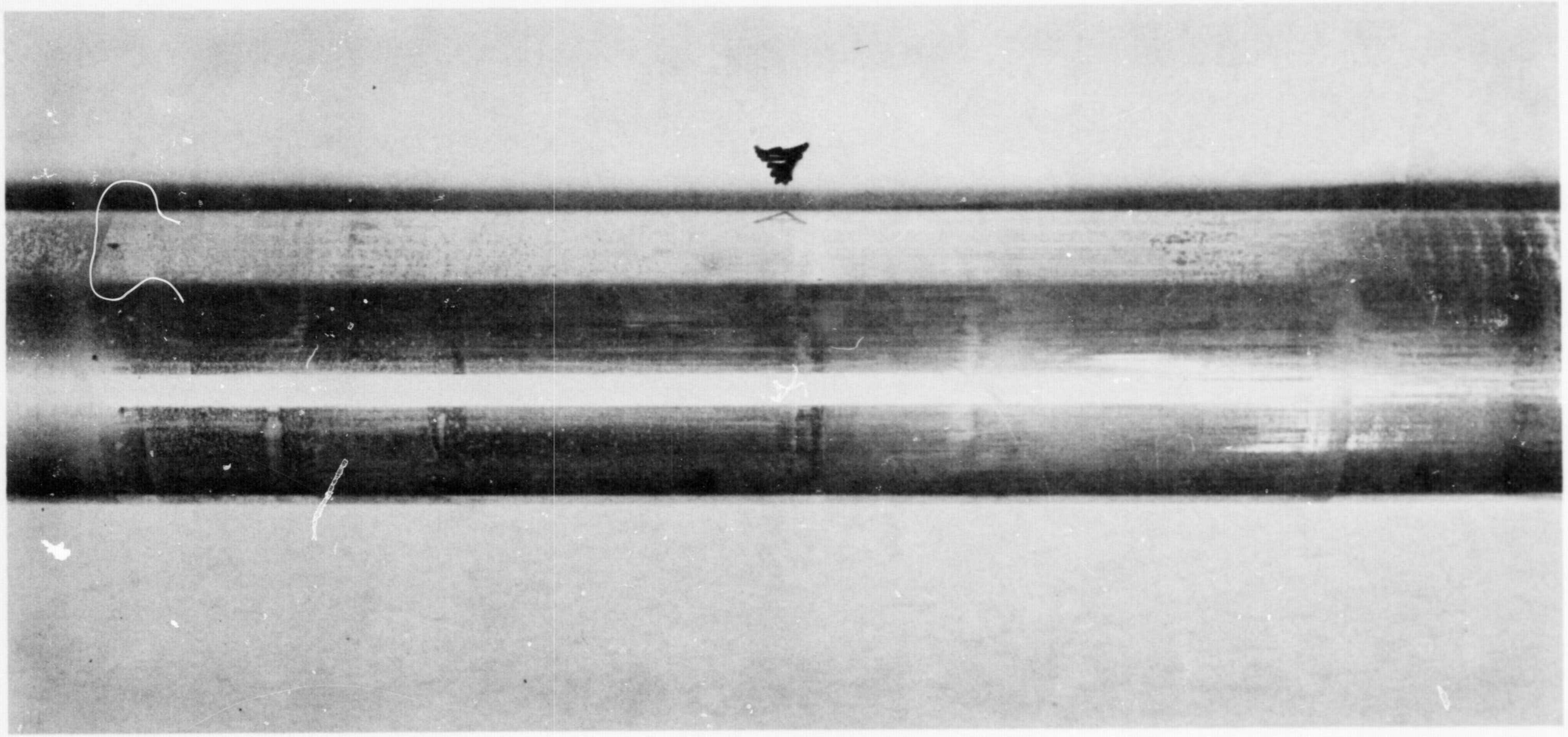


Figure 6-11. Piston Rod, B-End, One-Inch Polyimide V-Seal

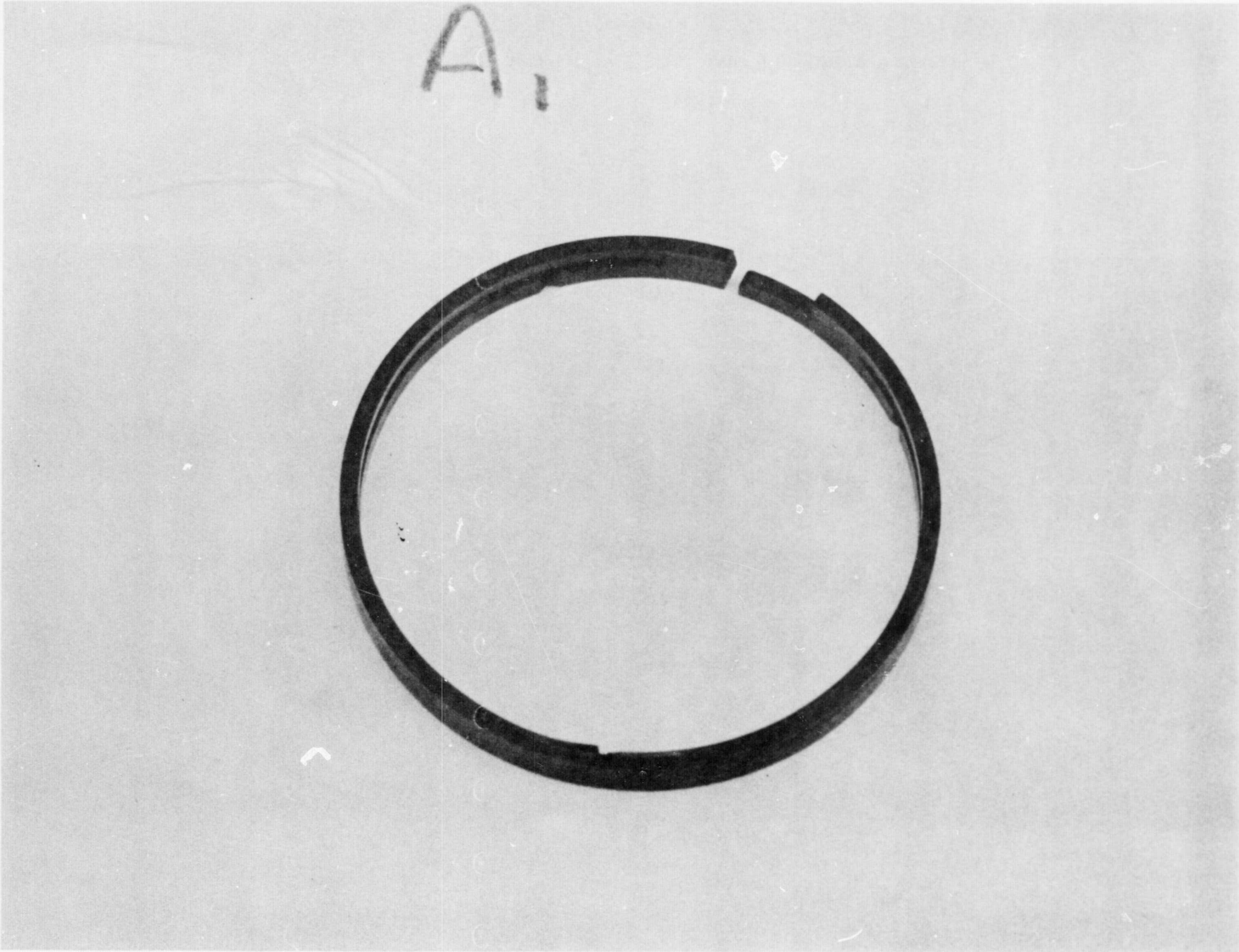


Figure 6-12. First-Stage Seal (A-End), Actuator 1-1

is quite notch sensitive. Because of the limited supply of the balanced-type sealing ring, an unbalanced-type tooling ring was used as replacement for the failed seal. Performance of this seal (Figure 6-13) was satisfactory during the rest of the endurance test. Leakage generally ranged from 5cc to 12cc per minute.

The B-end first-stage seal (Figure 6-14) exhibited leakage rates of 2 to 12cc per minute during 940 hours of operation. From that point on, leakage was as high as 20cc per minute. However, as this leakage did not affect operation of the hydraulic circuit, it was decided not to replace this seal.

It was noted that, occasionally throughout the testing, leakage exceeding 20cc per minute was exhibited by the first-stage seals. This condition occurs for a very short period only and is believed to be due to the seal lifting off the gland shoulder momentarily and allowing fluid to flow by. This unseating of the seal is probably caused by the unbalancing force on the seal during operation. Observations made during the testing shows that this condition usually occurs during the low-pressure (500 psi) segment of the test spectrum. At the lower pressures, a condition could exist wherein the friction force between the seal and piston rod is greater than axial force holding the sealing ring against the gland shoulder. Under these conditions, the piston rod would tend to drag the seal away from the gland shoulder. The above condition was experienced occasionally by all the actuators during the endurance test. The simple solution to this problem in the future is to incorporate a wave spring in the seal cavity to maintain a biasing load in the axial direction.

2. Three-Inch Polyimide V-Seal (Design B)

Testing of the A-end seal was concluded after 664.2 hours. Although average leakage from this seal during the final nine hours of testing was 2 drops per minute sporadically, the leakage ran as high as 35 dpm. The B-end seal was operated for 805.9 hours before testing was finally concluded. Leakage of 1 dpm was exhibited by this seal after 722 hours of testing and gradually increased to approximately 8 dpm at the conclusion of the test. Both seals exhibited low leakage (Figure 6-4) prior to their failure.

The mode of failure of these seals was similar to that experienced by the one-inch V-seals. As shown in Figure 6-15, the A-end seal exhibited circum-

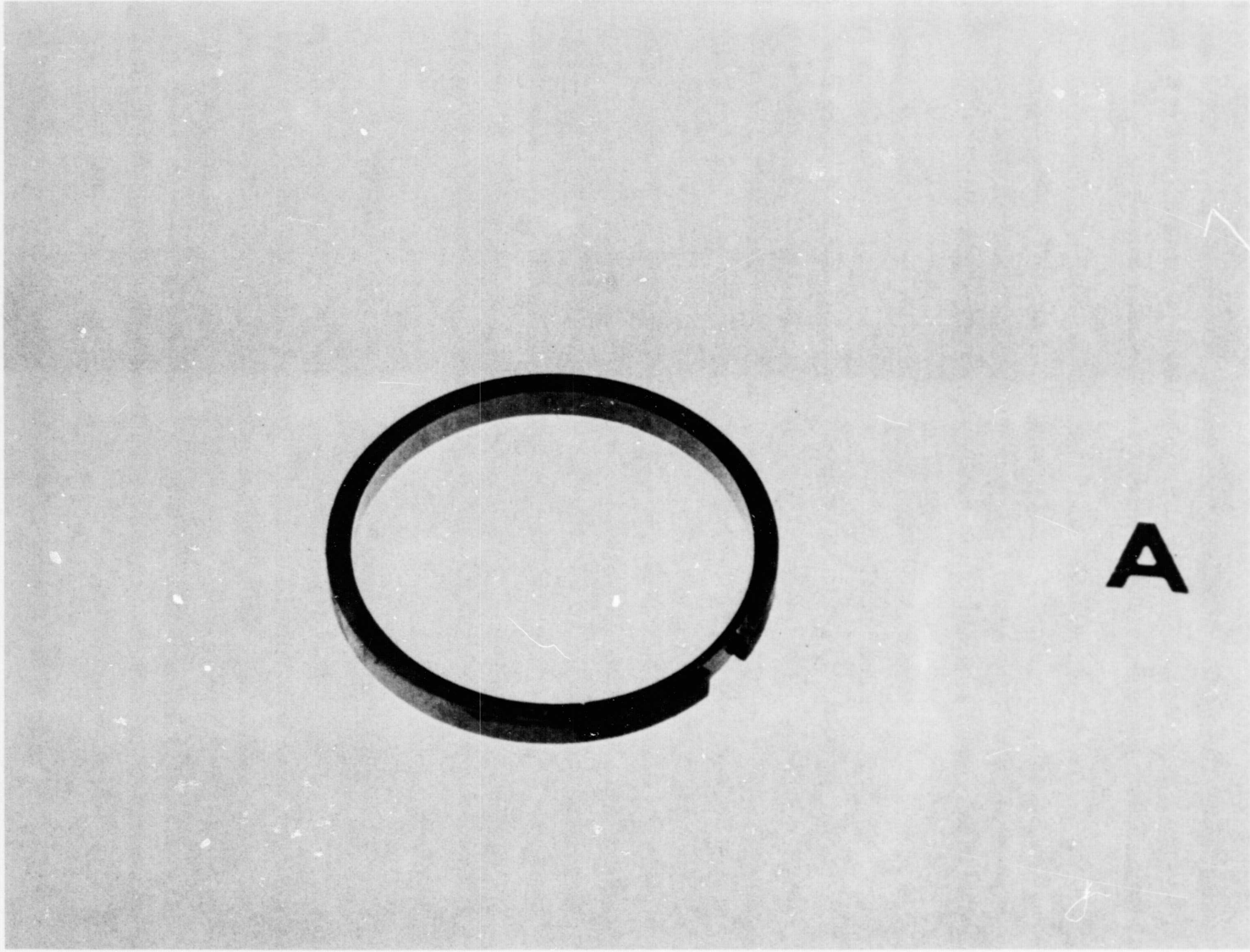


Figure 6-13. First-Stage Seal (A-End), Actuator 1-1

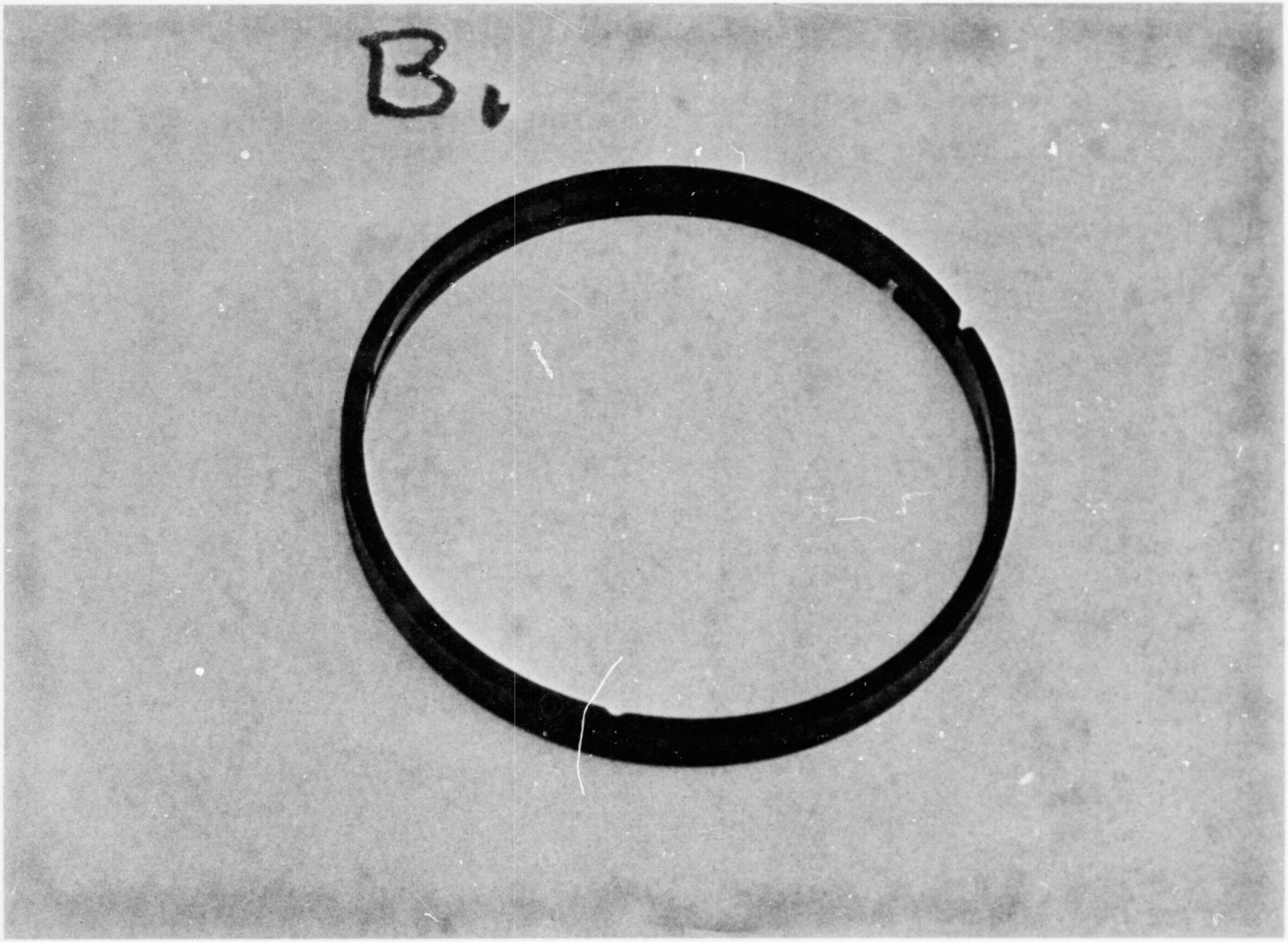


Figure 6-14. First-Stage Seal (B-End), Actuator 1-1

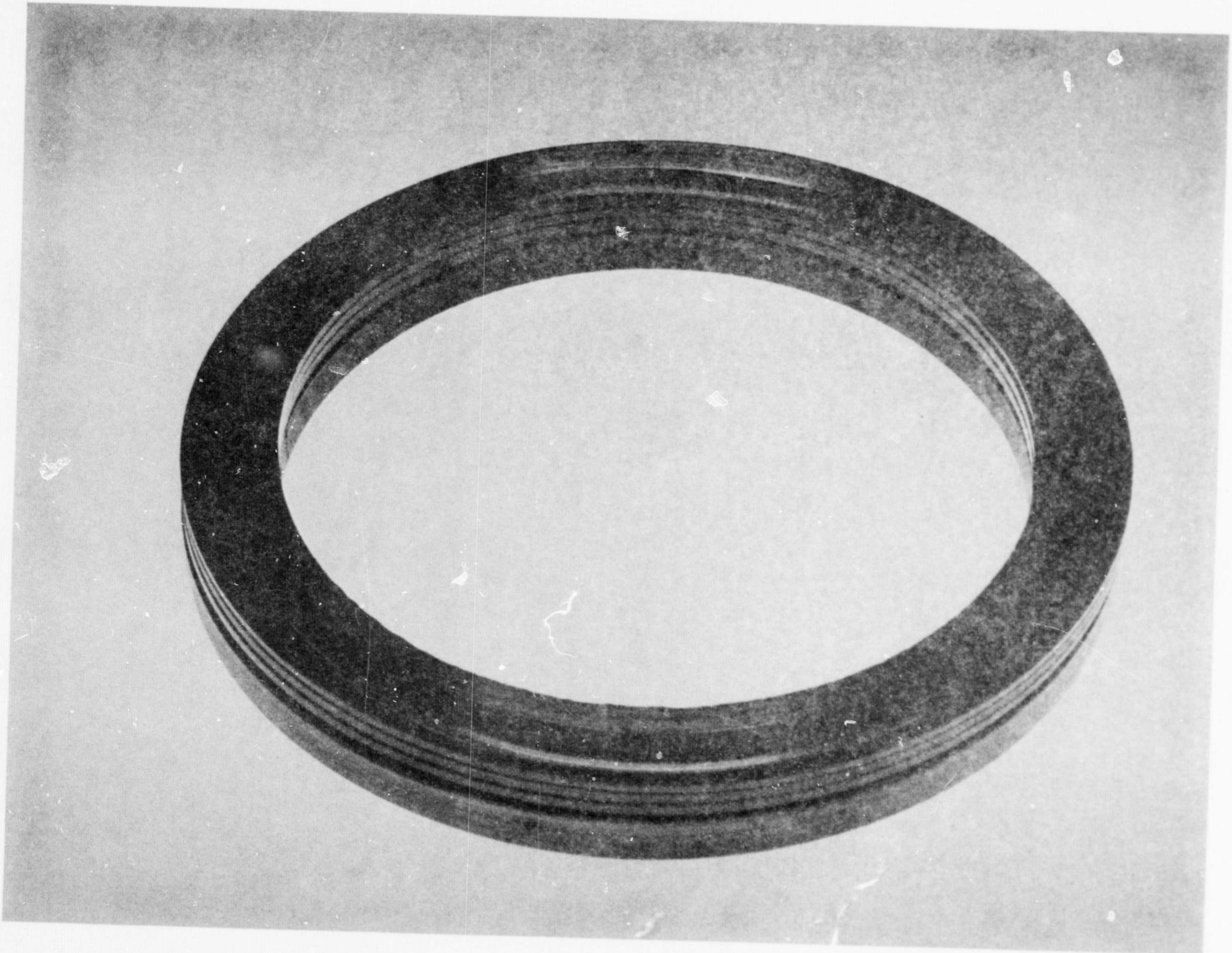


Figure 6-15. Three-Inch Polyimide V-Seal, A-End

ferential cracks on the inside surface of the "V" and also along the sealing surface of the seal. As shown in Figure 6-16, a portion of the sealing lip of the third V-seal was broken off. Figures 6-17, 6-18, and 6-19 further depict the condition of the B-end seals. The A-end seals exhibited similar failure of the material. Seal dimensions taken before and after testing are compared in Table 6-5.

TABLE 6-5. SEAL DIMENSIONS - THREE-INCH V-SEALS
ROD DIAMETER - 2.9970

End	Seal No.	Before Test I.D.	Before Test O.D.	Interference	After Test I.D.	After Test O.D.	Interference
A	1	2.985	3.759	0.012	2.999	3.735	-
	2	2.984	3.760	0.013	3.002	3.760	-
	3	2.987	3.761	0.010	2.999	3.740	-
B	1	2.987	3.761	0.010	2.996	3.735	0.001
	2	2.985	3.760	0.012	3.000	3.735	-
	3	2.984	3.760	0.013	2.9947	3.740	0.0023

Wear patterns produced by the short-stroke cycling operation were visible on the A-end and B-end of the piston rod (see Figure 6-20 and 6-21). The apparent wear on the chrome plate, as measured with a surface analyzer was approximately 0.00002 and 0.000014 to 0.000016-inch for the A-end and B-end, respectively. No measureable wear was detected on the chrome-plated surface contacted by the seals during long-stroke cycling. A light wear pattern was also generated on the piston rod surface by the first-stage seals during short-stroke cycling. However, no measureable wear could be detected.

Performance of the high-pressure first-stage seals was satisfactory. Leakage rate of approximately 20cc per seal was maintained for about 500 hours of testing. However, leakage then gradually increased to 50-60cc per minute (per seal).

Inspection of the first-stage seals showed that both seals (Figures 6-22 and 6-23) had breakage of the ring joints. Joint failure was attributable to the high notch-sensitivity of the material. This failure also accounts for the increased seal leakage experienced during the latter part of the endurance test.

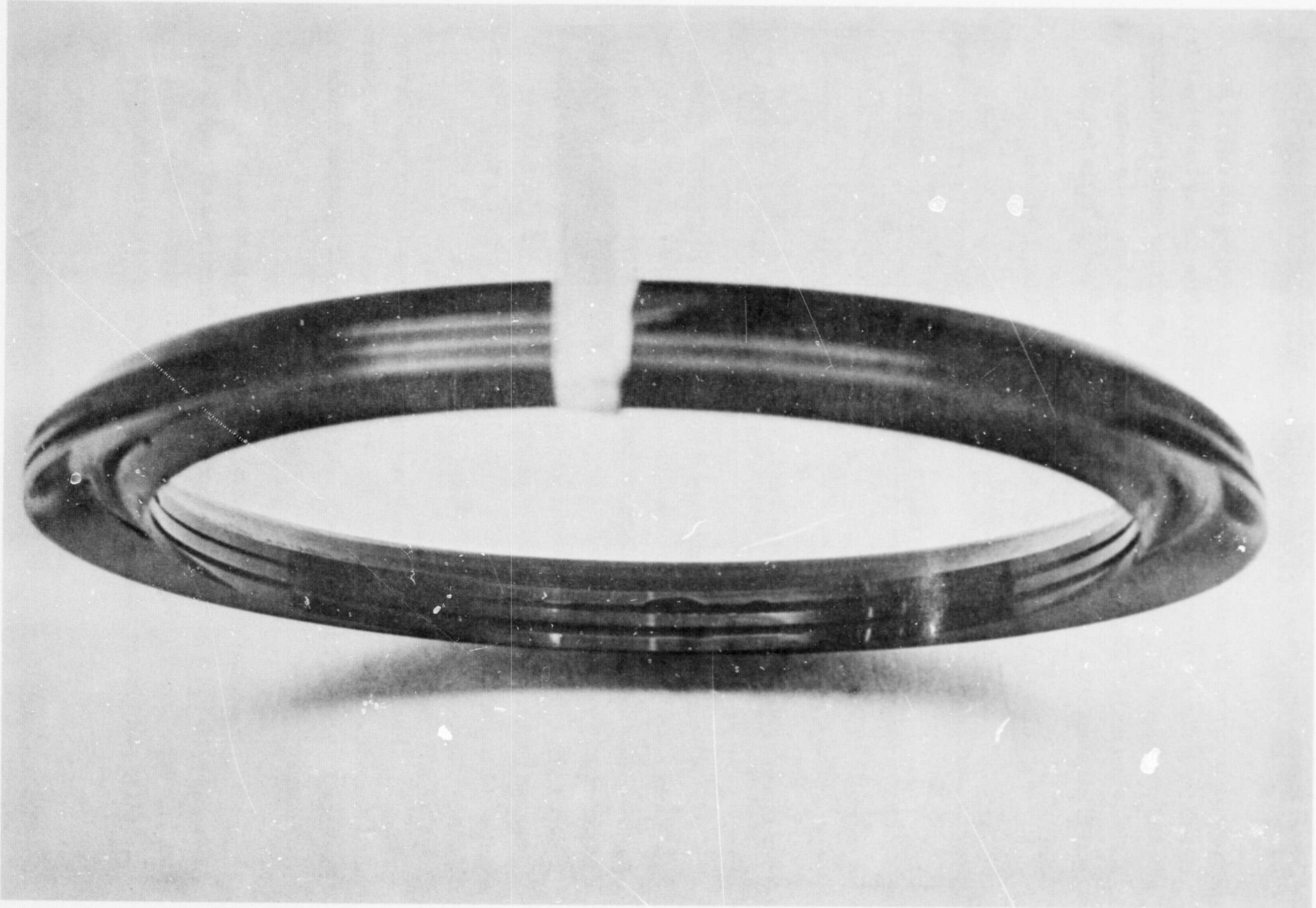


Figure 6-16. Three-Inch Polyimide V-Seal, B-End

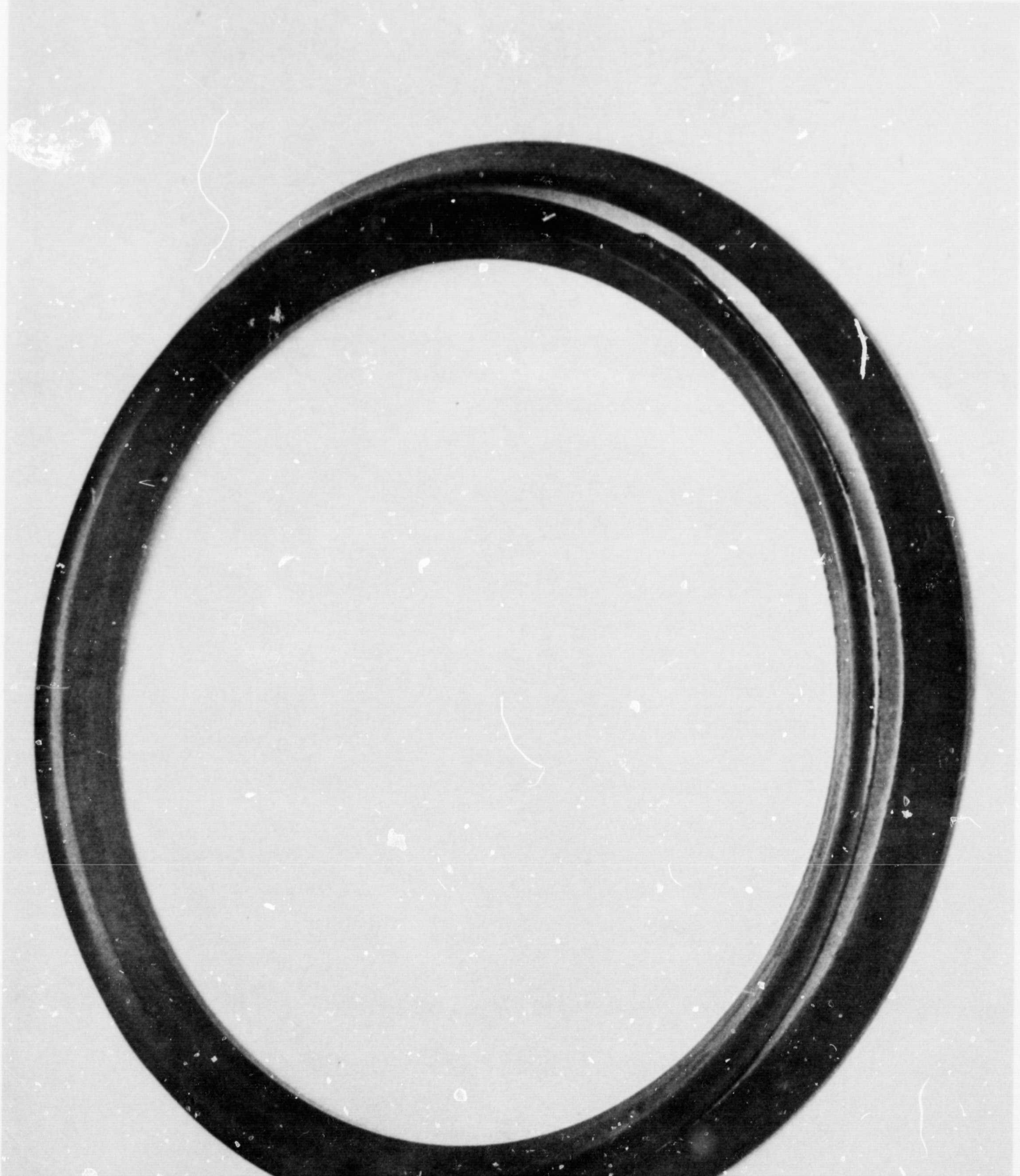


Figure 6-17. Three-Inch Polyimide V-Seal, A-End (Circumferential Cracks)

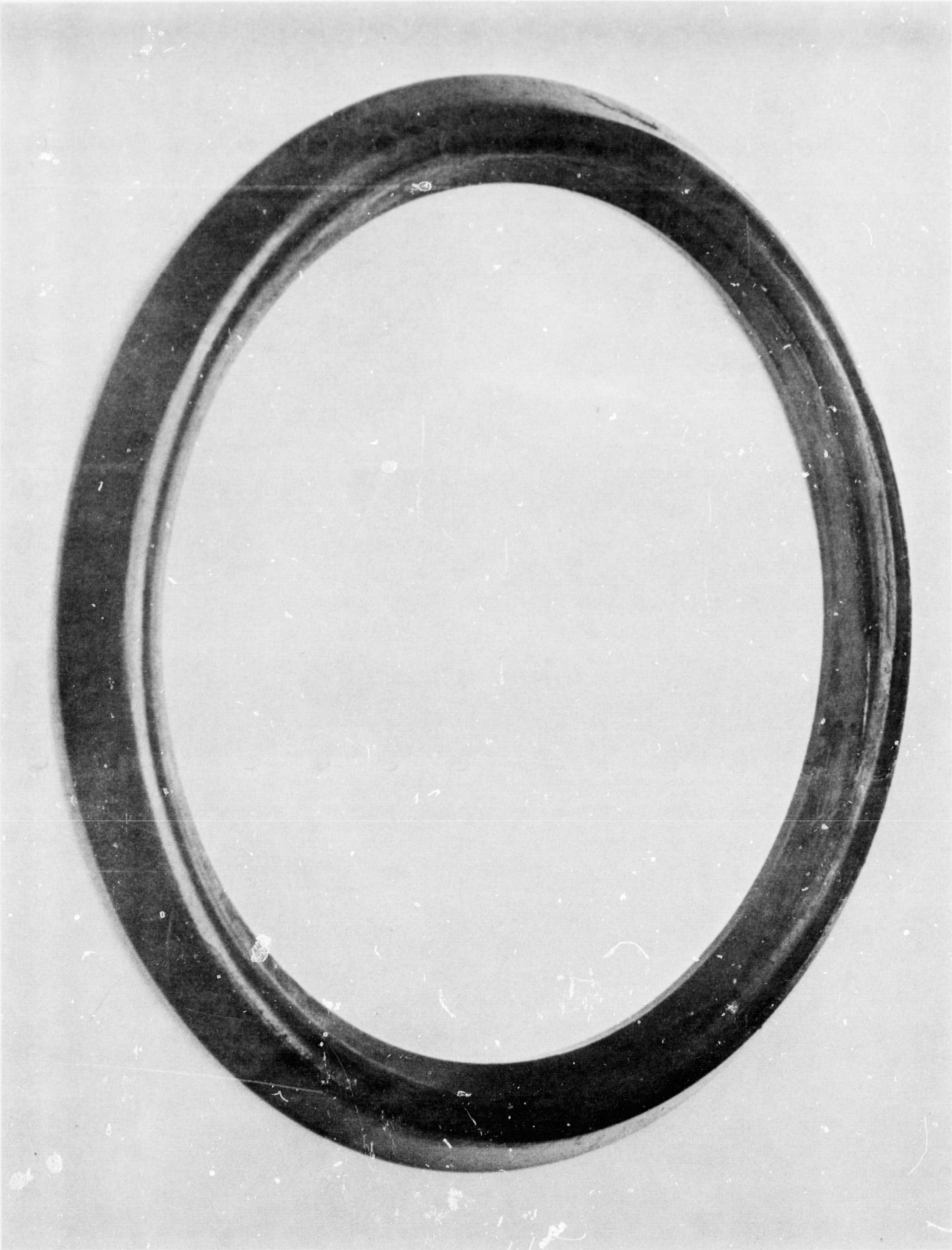


Figure 6-18. Three-Inch Polyimide V-Seal, B-End (Circumferential Cracks)

Figure 6-19. Three-Inch Polyimide V-Seal, B-End (Circumferential Cracks)

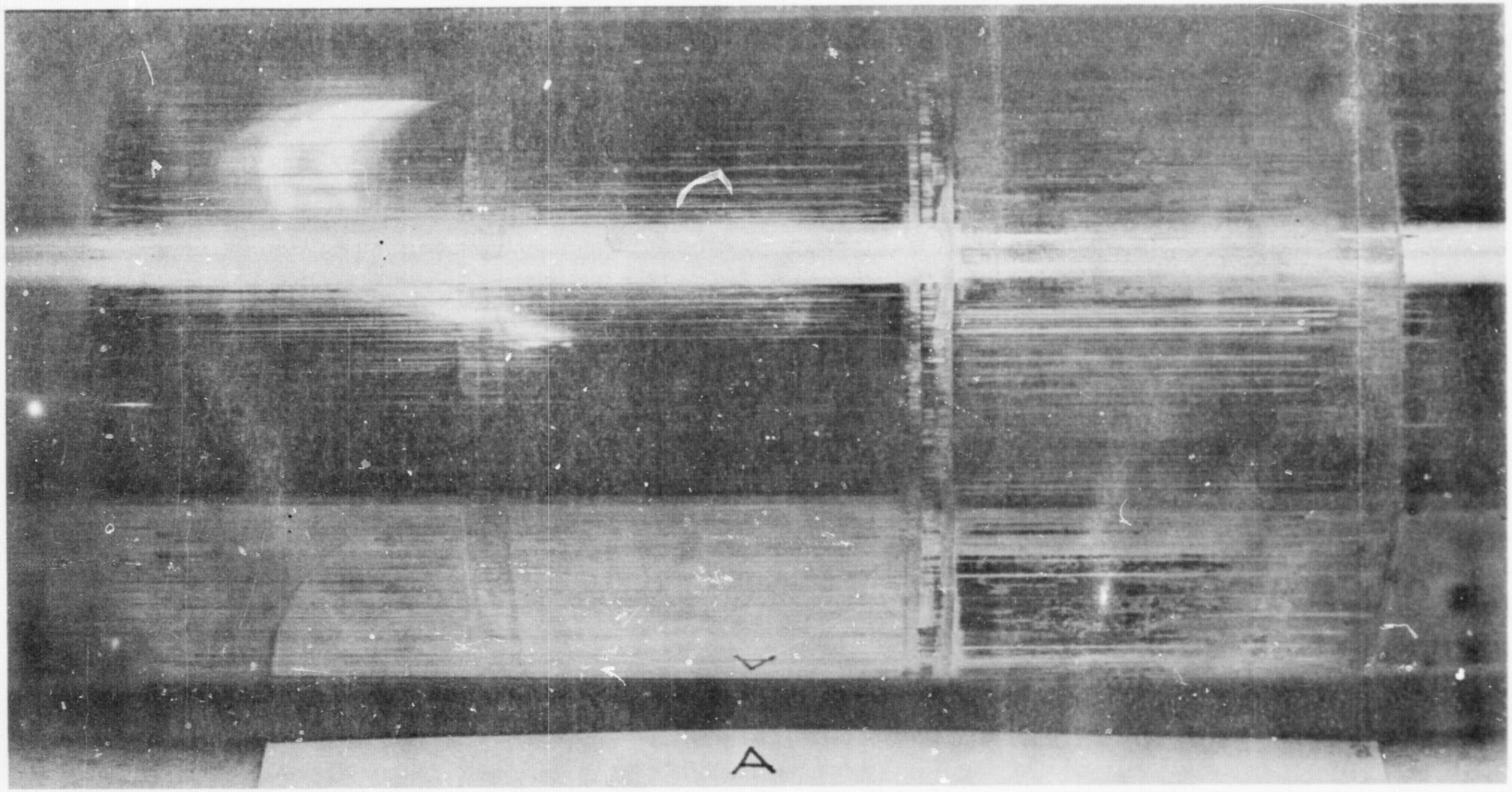
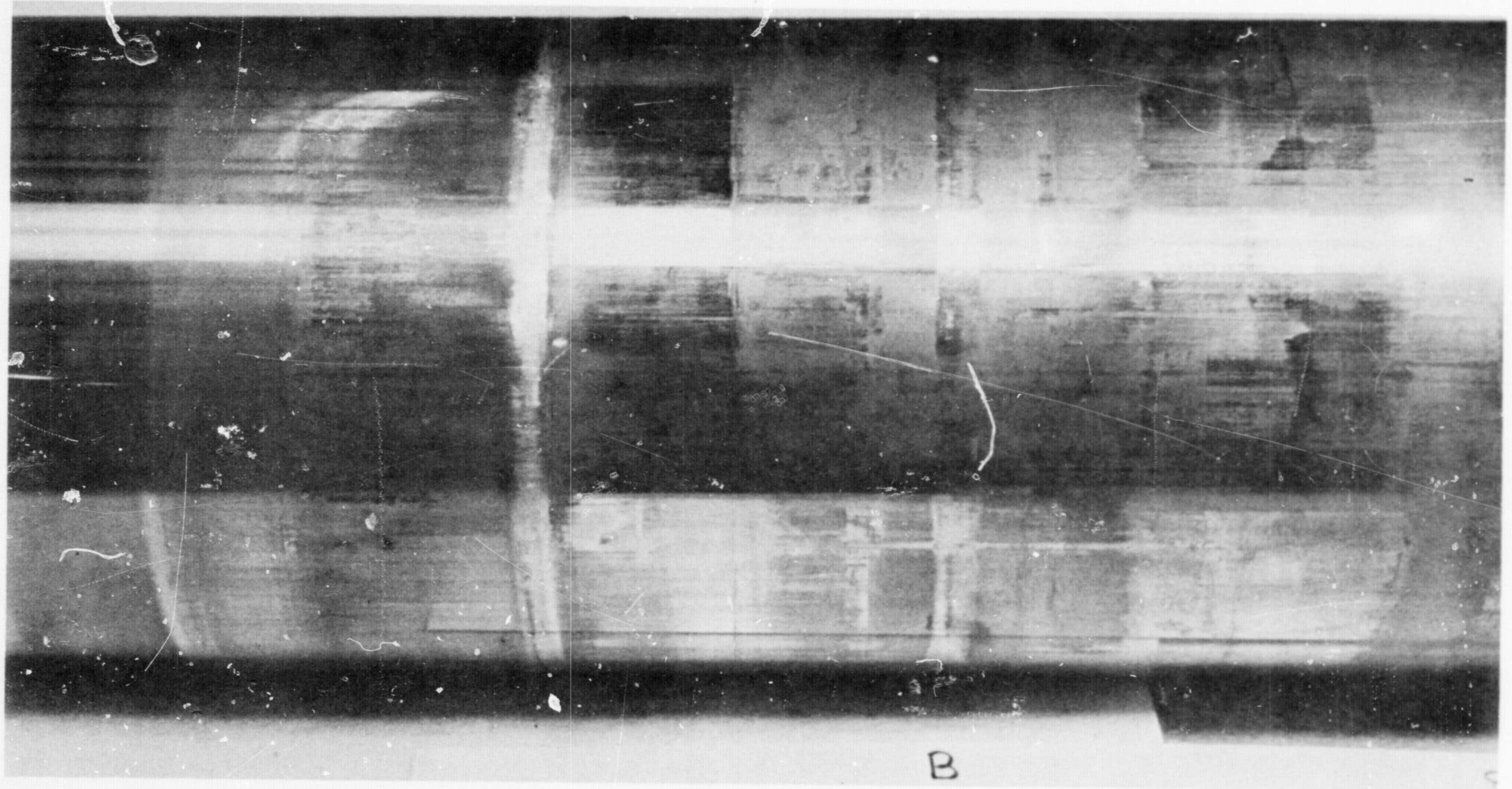


Figure 6-20. Piston Rod, A-End, Three-Inch Polyimide V-Seal



Fgirue 6-21. Piston Rod, B-End, Three-Inch Polyimide V-Seal

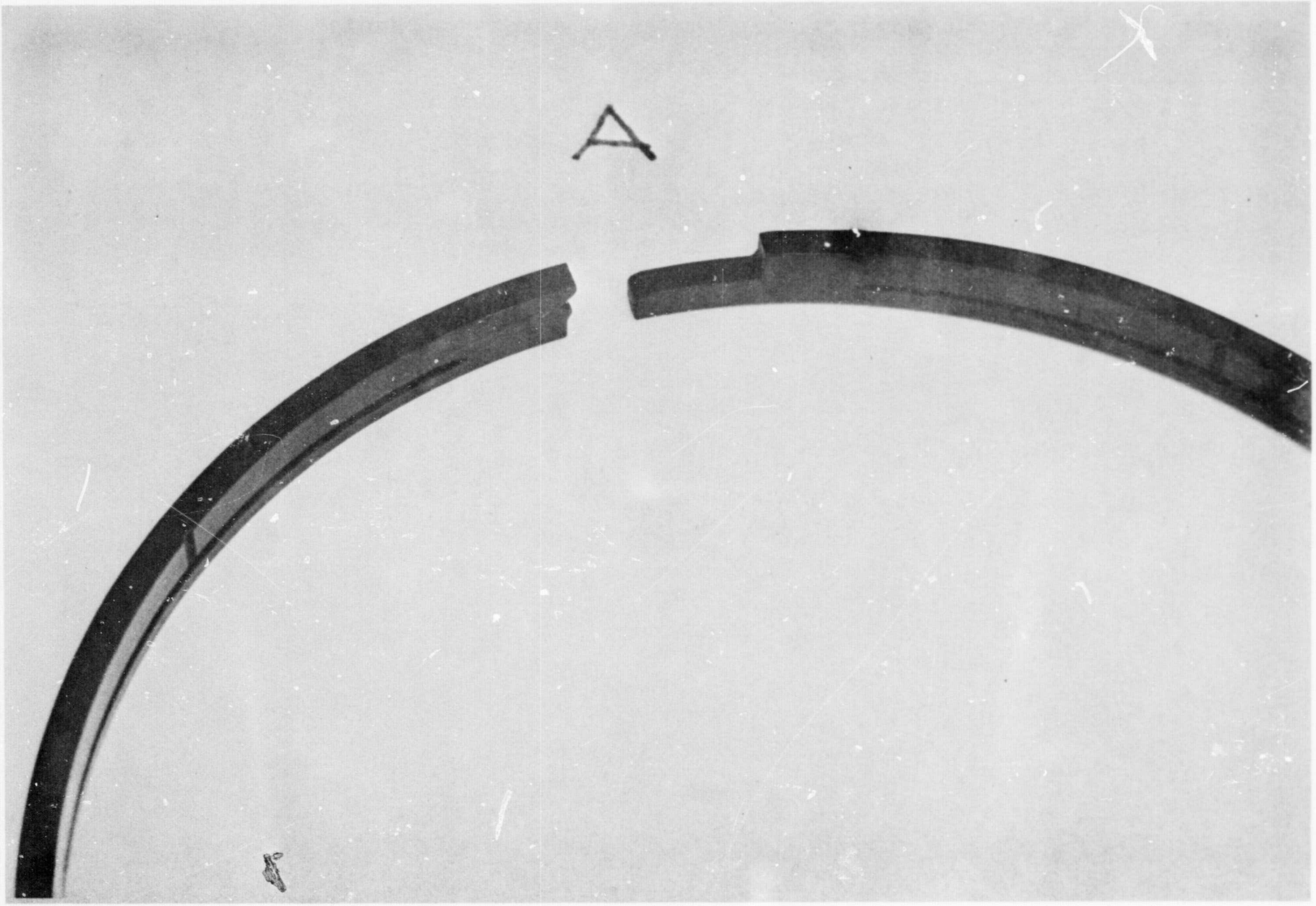


Figure 6-22. First-Stage Seal (A-End), Actuator 3-1

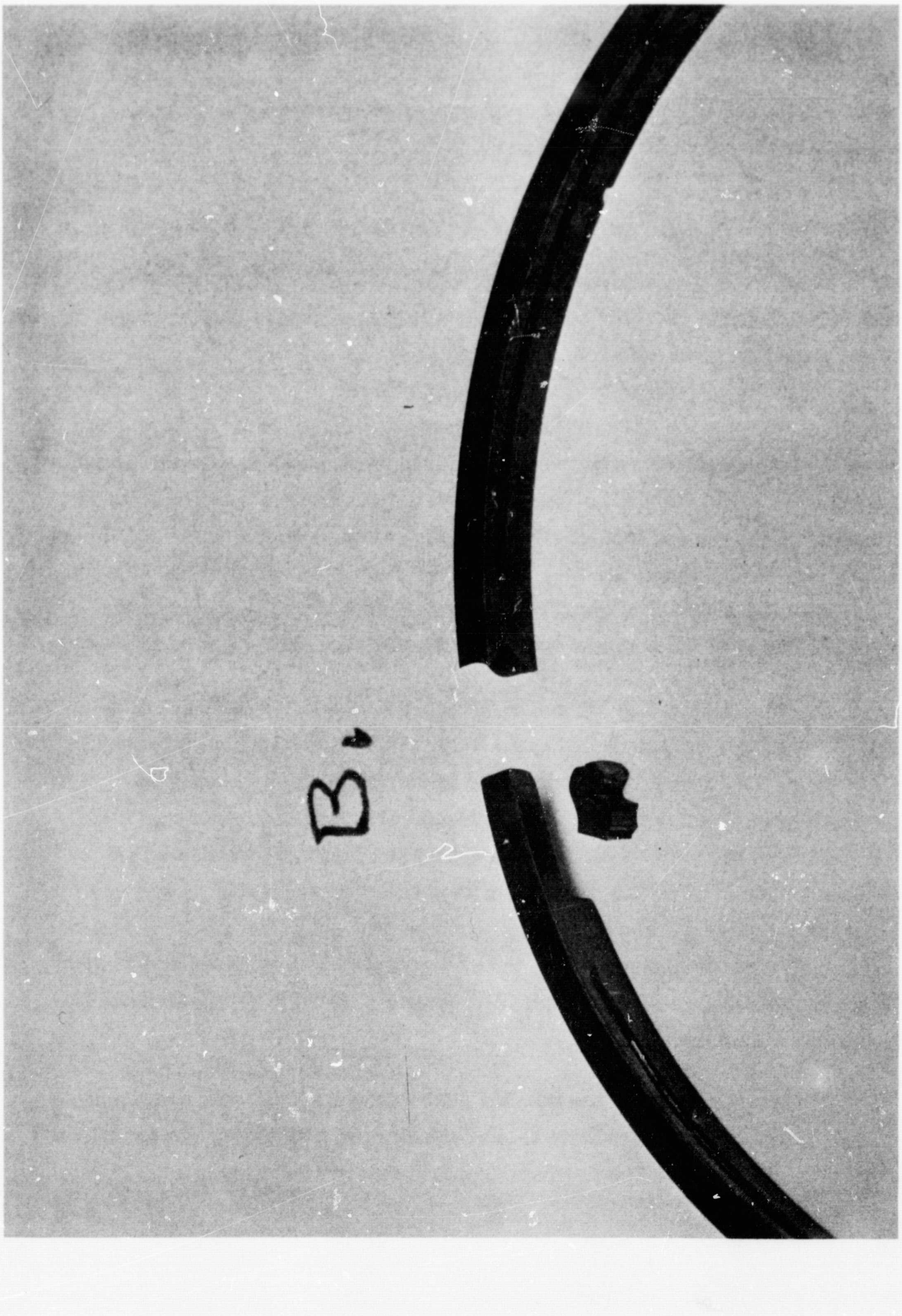


Figure 6-23. First-Stage Seal (B-End), Actuator 3-1

3. <u>One-Inch Cobalt Molybdenum Lip Seal (Design D)</u>

Four of these seals were fabricated for test. Two of the seals split during assembly onto the piston rod. Splitting of the third seal occurred after 290,160 cycles of operation accompanied by excessive leakage. As shown in Figure 6-24, the crack occurred at the sealing lip. The wear pattern on the seal was unevenly distributed. This condition indicated that the seal may have cracked during assembly. This would have resulted in a poorly distributed contact load.

Although these seals were designed to a conservative stress level (less than 50% of yield), cracking of the material appears inherent in this configuration. Based on these failures, the fourth seal was modified to provide a minimum stretch (interference) of 0.0007 to 0.0008 inch. This represents approximately one-half the stretch used in the seals that cracked. In order to obtain additional wear compensation on the modified seal, a Vascojet 1000 sleeve (Figure 6-25) was assembled over the cobalt lip. The Vascojet sleeve was designed to produce a spring load of 25 pounds per inch of circumference when assembled onto the lip seal. As this load acts to compress the seal against the piston, cracking of the seal did not occur. Testing of this configuration was accomplished on the B-end of the actuator. The A-end of the actuator was assembled with a set of polyimide V-seals.

Performance of this lip seal configuration was quite satisfactory during 99.8 hours of testing. As shown in Figure 6-26, total leakage during this time period was 108cc. However, leakage started to increase rapidly beyond this point. Although inspection of the seal showed no evidence of damage, it was noted that the wear pattern on the chrome plating was quite prominent. However, as excessive leakage occurred during cool-down from high temperatures, it was suspected that leakage may have been caused by permanent set of the silver gasket static seal. Based on this assumption, the silver gasket was removed and the seal cavity modified to accept a silver-plated V-type seal (Figure 6-27) made from 17-4PH corrosion resistant steel.

Following the above rework, the seal was operated for an additional 70 hours before testing was terminated. For the first 39 hours of operation following the rework, leakage was 2cc per hour (approximately 40 drops/hour). This

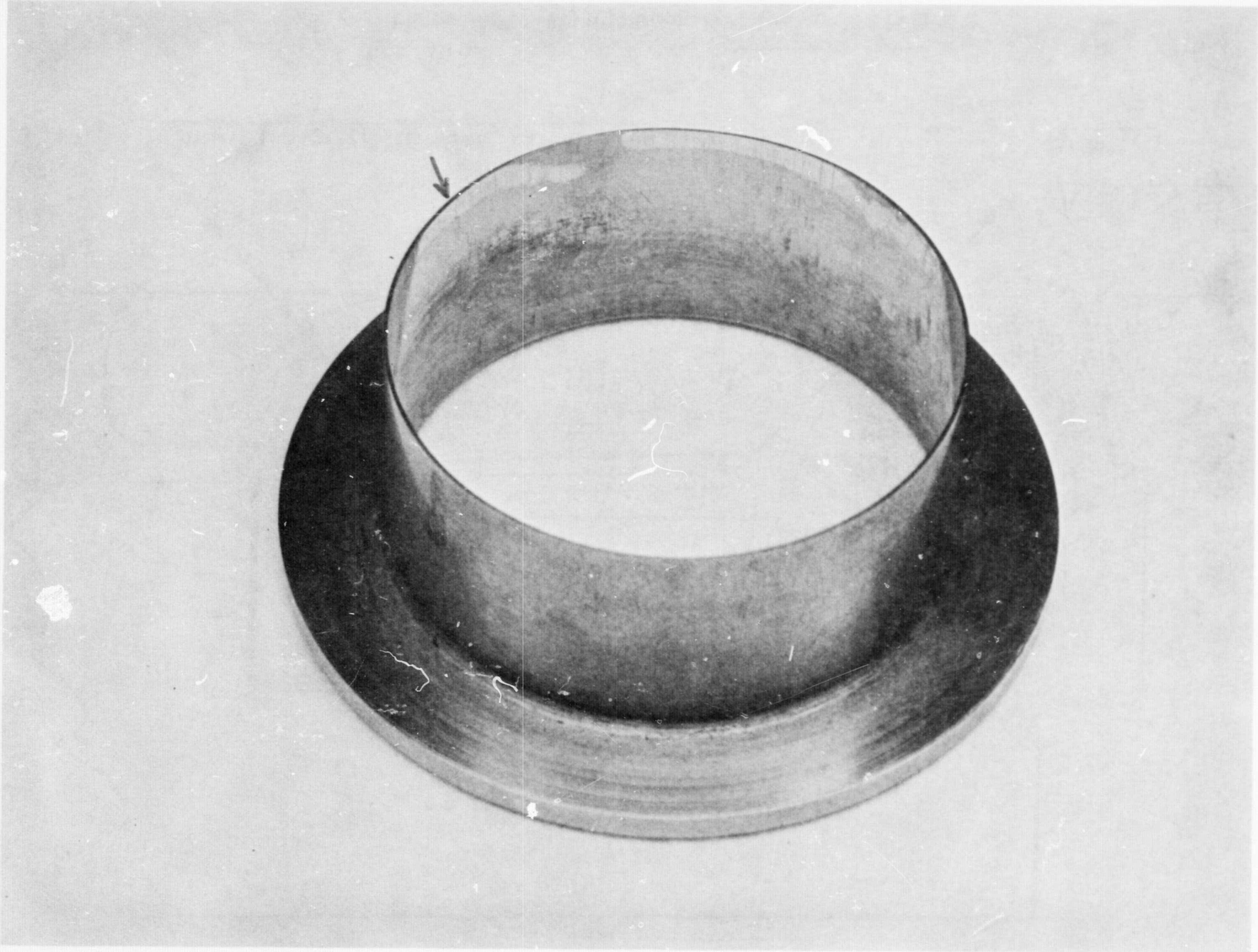


Figure 6-24. One-Inch Cobalt Molybdenum Lip Seal, A-End

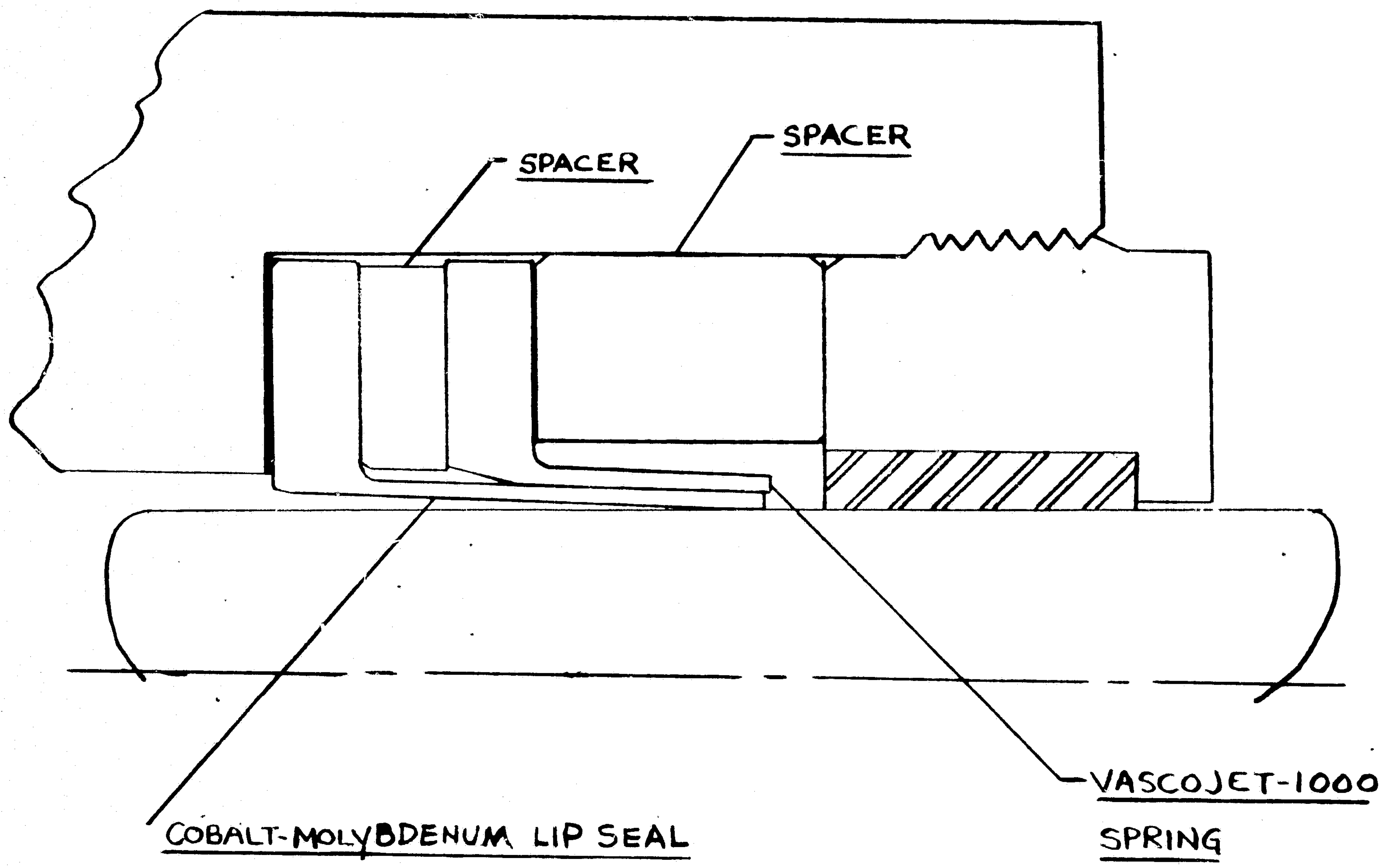


Figure 6-25. Modified Cobalt Molybdenum Lip Seal

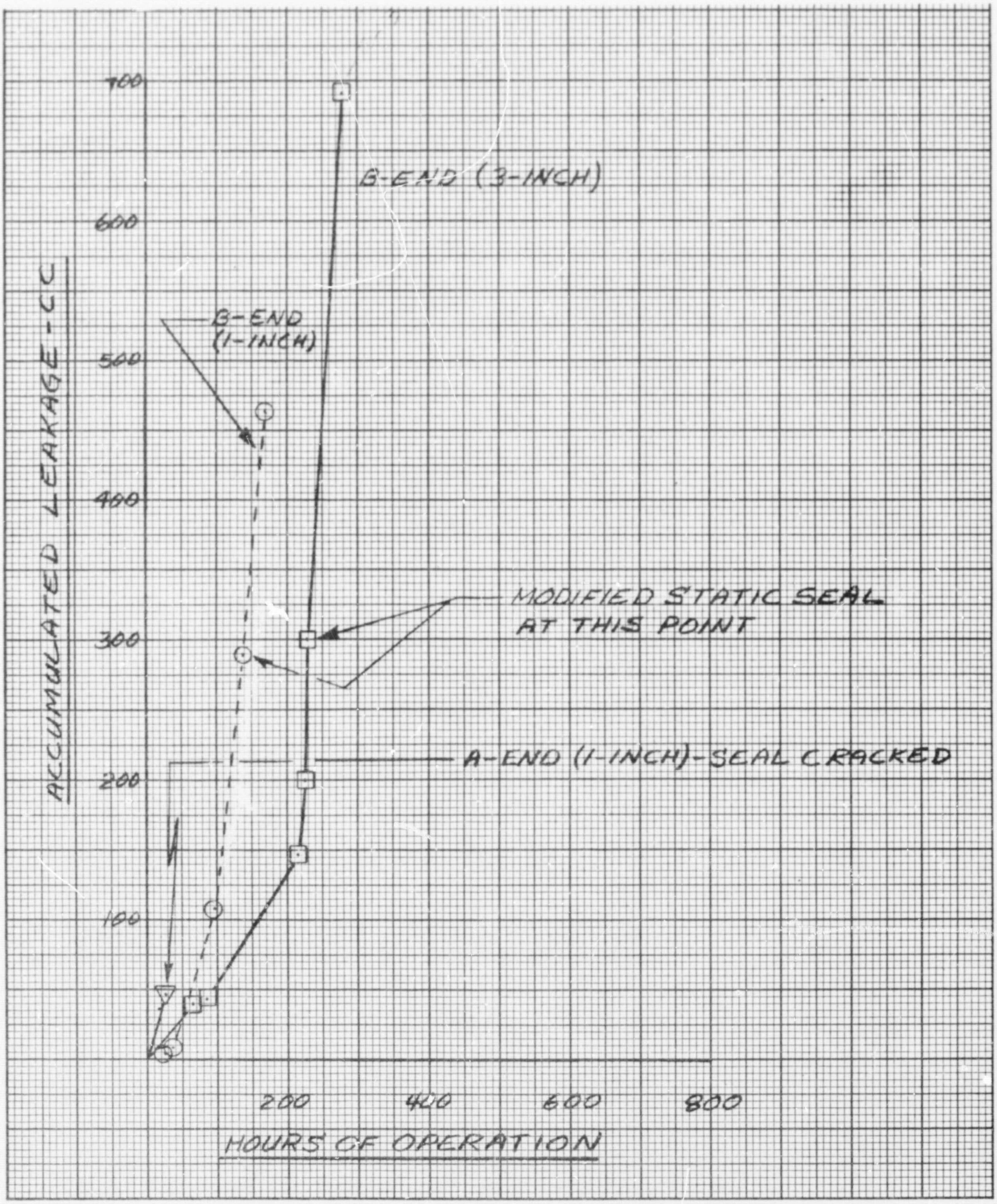


Figure 6-26. Accumulated Leakage versus Time, Lip Seals

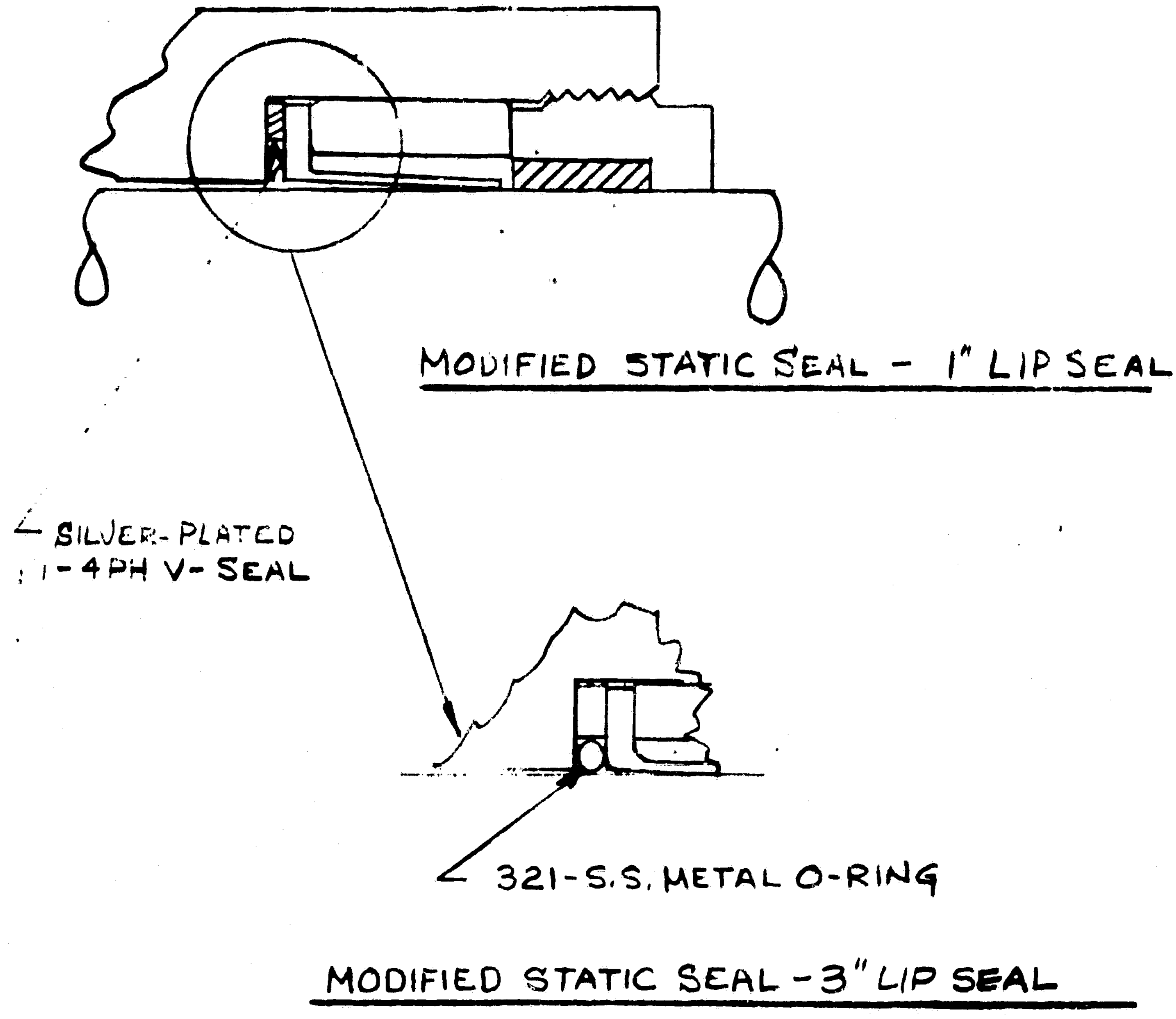


Figure 6-27. Modified Static Seal, One-Inch and Three-Inch Lip Seal

decrease in leakage indicated that the excessive leakage previously experienced was caused by the static seal. However, during the final 31 hours of testing, leakage gradually increased to 3 to 5 drops per minute and testing was terminated. Total operating time on this seal was 170.8 hours.

Inspection of the lip (Figure 6-28), showed that good contact was achieved with the piston rod. The variation in the width of the contact surface on the sealing lip indicated that some side load was experienced by the seal. Failure was caused primarily by wear of the seal beyond the compensating range of the Vascojet 1000 spring. The amount of wear at the sealing lip was determined to be approximately 0.0015-inch. This value was obtained by measuring the thickness of the lip at the contact surface and measuring the thickness adjacent to the contact surface. Assuming constant lip thickness was obtained during fabrication, the difference between the two measurements represents the amount of wear.

The piston rod exhibited a wear pattern produced by the short-stroke cycling (Figure 6-29). A wear of 0.00005 inch was detected at this site. The long-stroke cycling produced no measure wear on the chrome plated rod surface.

Performance of the high-pressure first-stage seals (Figures 6-30 and 6-31) was satisfactory during the entire testing. Leakage past these seals was approximately 5 to 12cc per minute per seal.

4. Three-Inch Cobalt Molybdenum Lip Seal (Design D)

Endurance testing of the three-inch cobalt molybdenum lip seal was previously abandoned because of manufacturing difficulties. However, the contractor was successful in a final attempt to fabricate a workable seal. Using a rather conservative design, the seal was assembled with an interference fit of approximately 0.005-inch interference. As only one lip seal was made for the test, Polyimide V-seals were used on the A-end of the actuator.

Performance of the lip seal configuration was satisfactory up to 227 hours of operation. Seal leakage during this time period was less than 1cc per hour. Accumulated leakage during the same period was 200cc (Figure 6-26), which was surprisingly low for a three-inch diameter seal. However, leakage

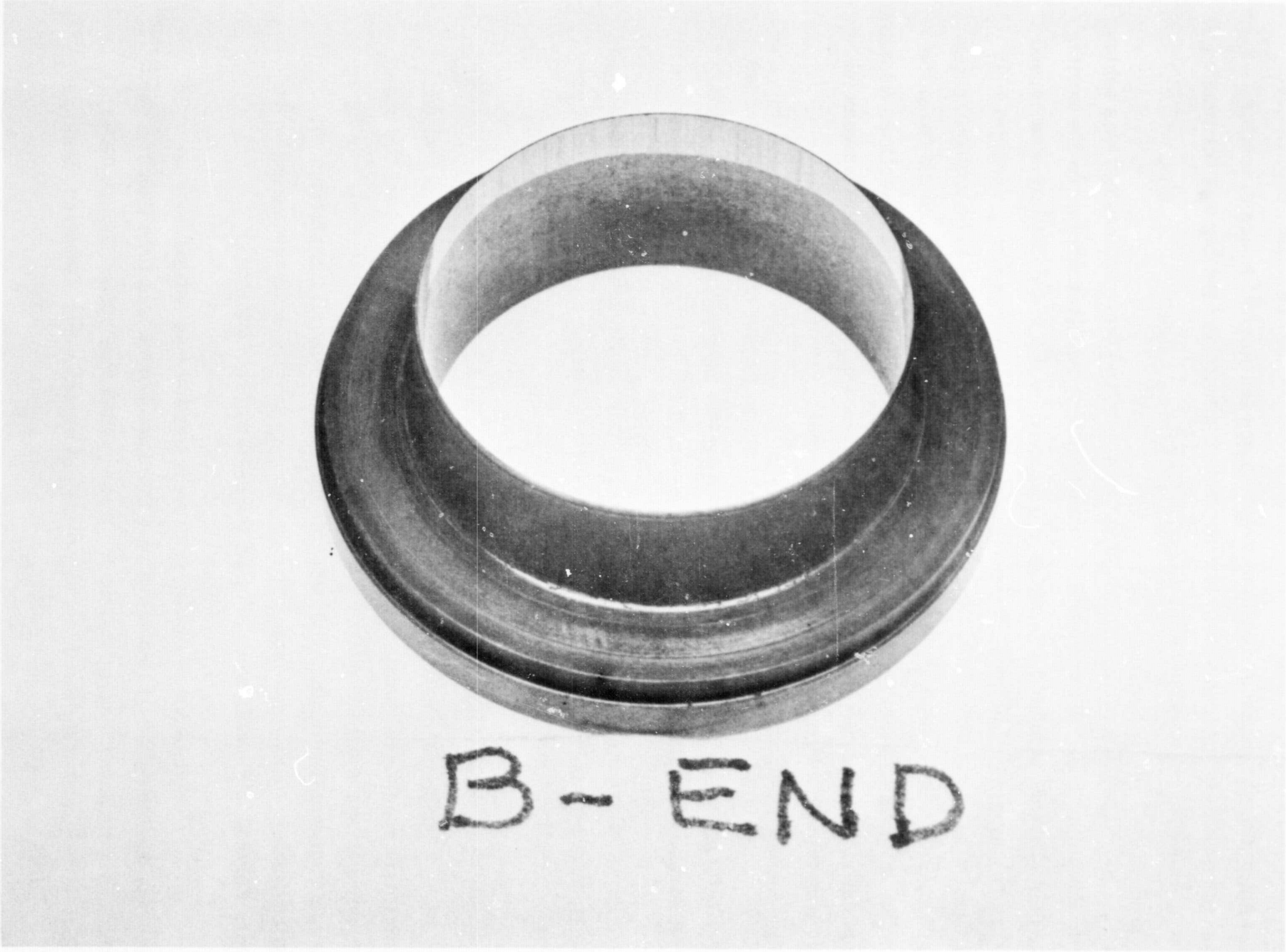


Figure 6-28. One-Inch Cobalt Molybdenum Alloy Lip Seal

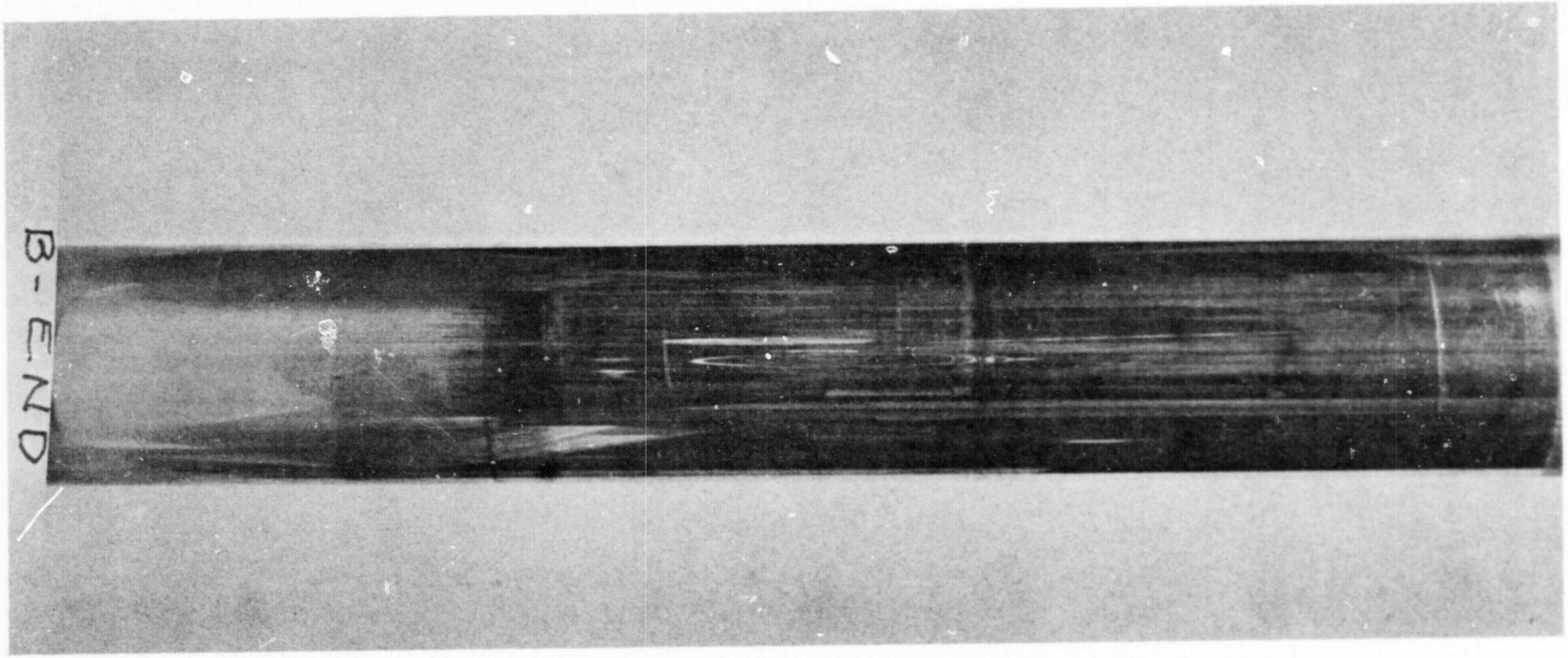


Figure 6-29. Piston Rod, B-End, One-Inch Cobalt Molybdenum Lip Seal

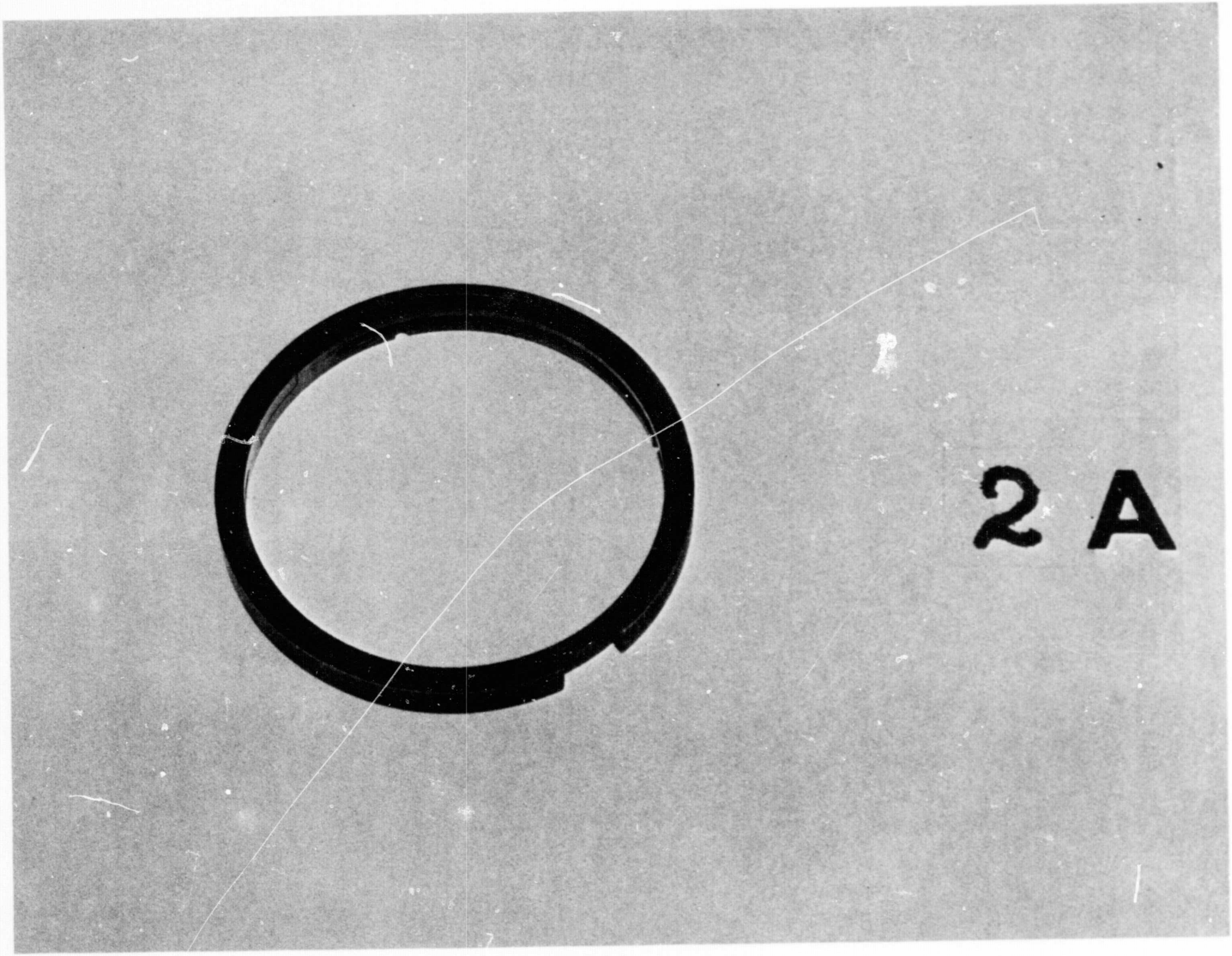


Figure 6-30. First-Stage Seal (A-End), Actuator 1-2

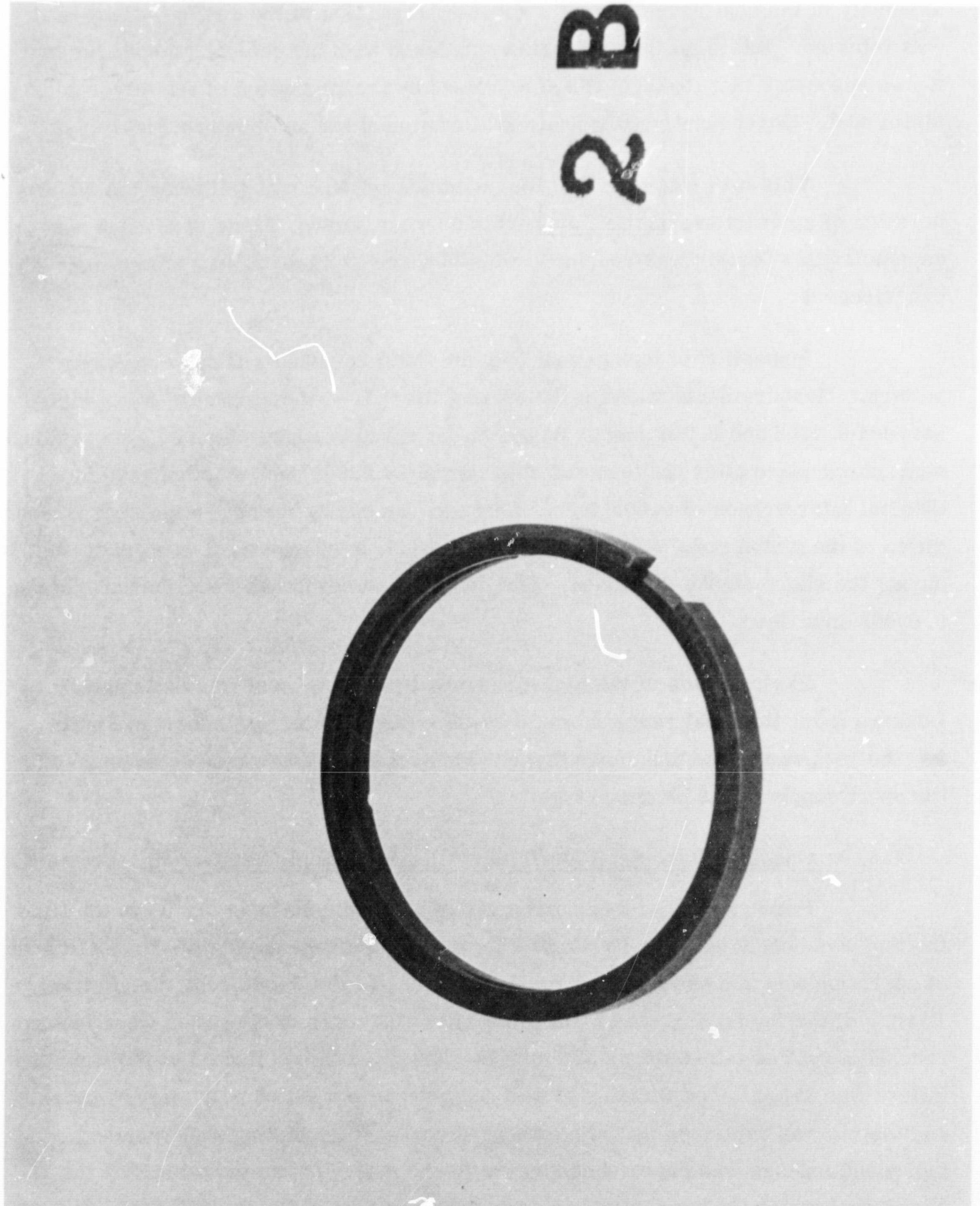


Figure 6-31. First-Stage Seal (B-End), Actuator 1-2

showed a rapid increase (+ 100cc) during the next 20 hours of operation. Disassembly of the seal revealed that a substantial portion of the seal's contact load was retained. Based on the problem encountered with the one-inch cobalt lip seal, it was suspected that leakage was also caused by the relaxation of the silver gasket static seal. Consequently, the static seal was modified as shown in Figure 6-27.

This seal was operated for an additional 50 hours following the incorporation of a new static seal. During this 50-hour period, 392cc of leakage was accumulated. Testing was concluded when leakage of 20 drops per minute was experienced.

Inspection of the lip seal (Figure 6-32) revealed a fairly even wear pattern. Measurements taken of the sealing lip showed the amount of wear varied between 0.0005 and 0.001 inch. Based on the measurements obtained, the residual seal interference after the test was approximately 0.003 inch as compared to the original interference of 0.005 inch. Leakage was due primarily to the scored condition of the piston rod. As shown in Figure 6-33, a wear pattern was generated during the short-stroke operation. The depth of the scratches were measured to 0.00003-inch deep.

Performance of the high-pressure first-stage seal was satisfactory. Leakage from this seal range from 15-to-30cc per minute. As shown in Figure 34, the seal was of the unbalanced type. This configuration was used because of the short supply of the balanced type.

5. One-Inch Vascojet 1000/Silver Alloy Reed Seal (Design AH)

Performance of this configuration was quite satisfactory up to the time the test was concluded. As shown in Figure 6-35, leakage during the first 410 hours of operation was 3cc and 90cc for the A-end seal and the B-end seal, respectively. Testing of the B-end seal was terminated after 433 hours of operation when leakage reached a rate of 4-to-5 drops per minute. Total leakage collected at the time of failure was 270cc. The failed seal was replaced with a set of polyimide V-seals and testing was continued on the A-end reed seal. When testing was resumed, increased leakage was experienced by the A-end seal. It was probable that the A-end seal may have been damaged during the removal of the B-end seal. Any

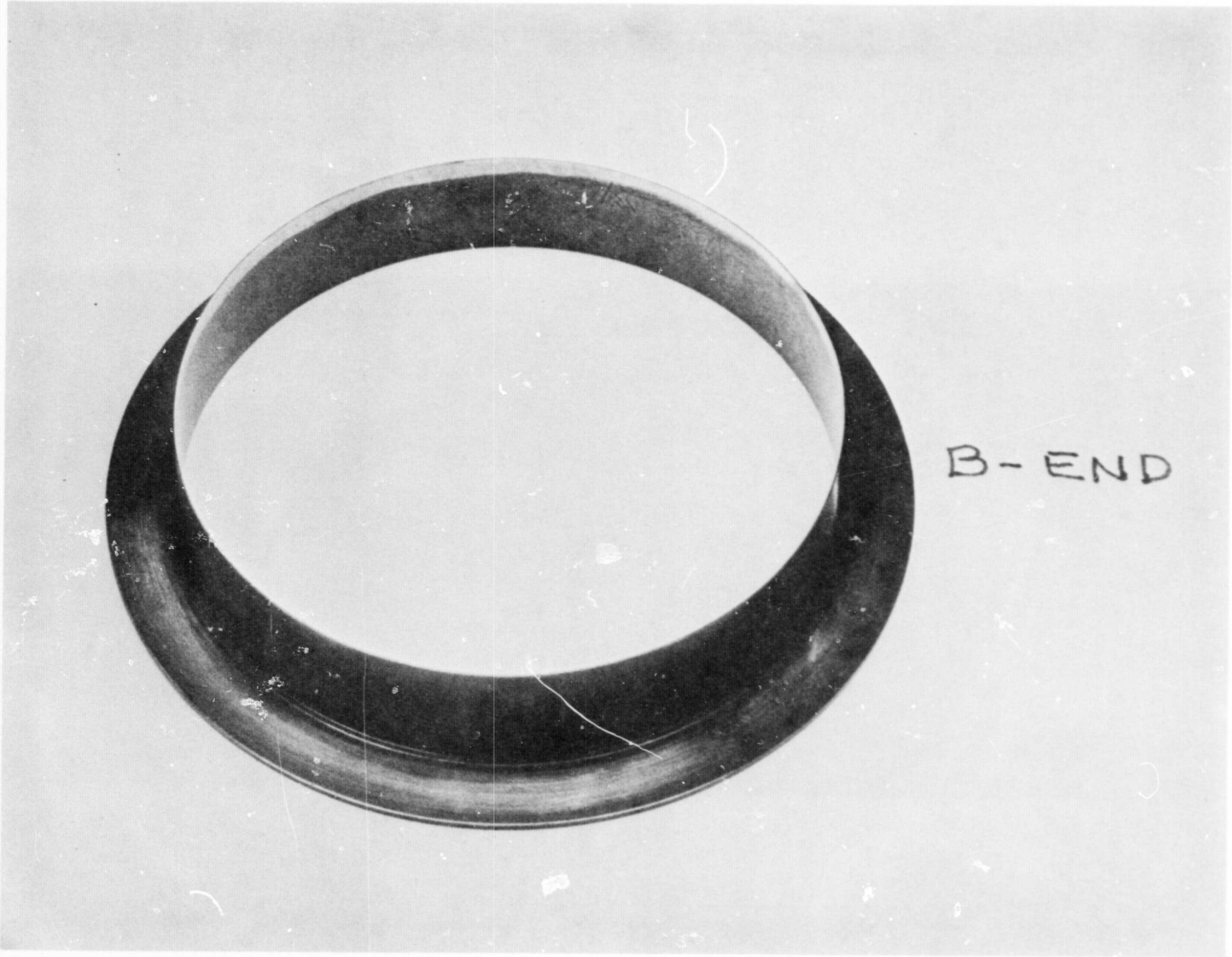


Figure 6-32. Three-Inch Cobalt Molybdenum Lip Seal, B-End

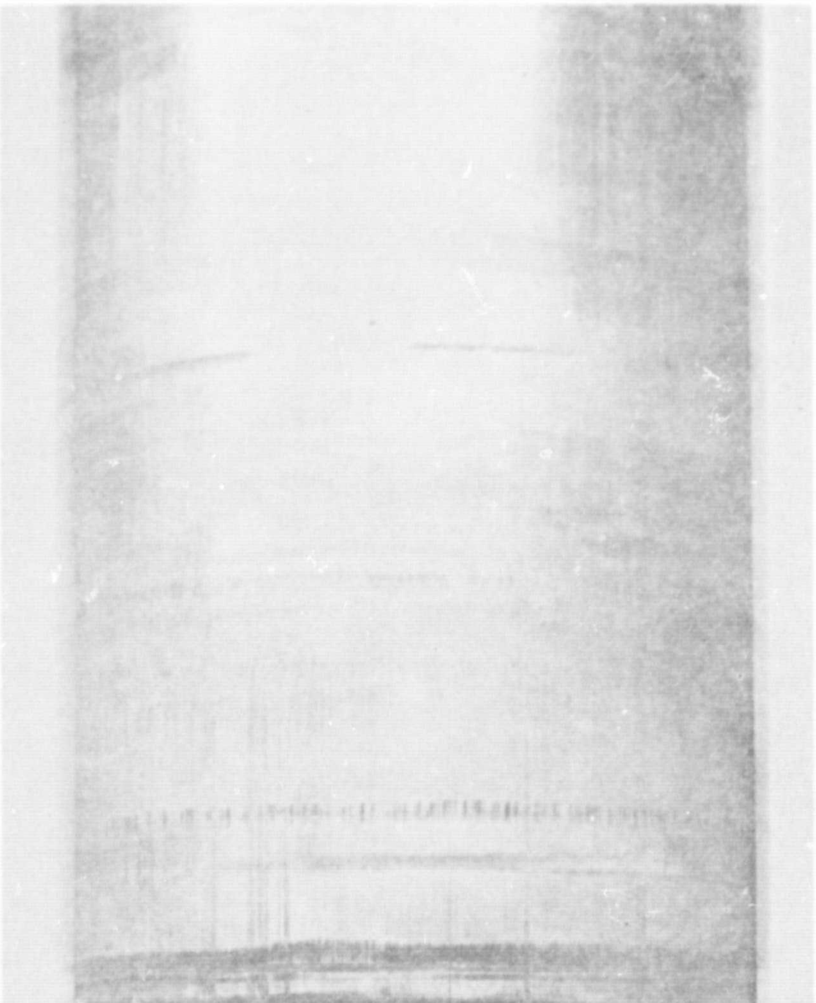


Figure 6-33. Piston Rod, B-End, Three-Inch Cobalt Molybdenum Lip Seal

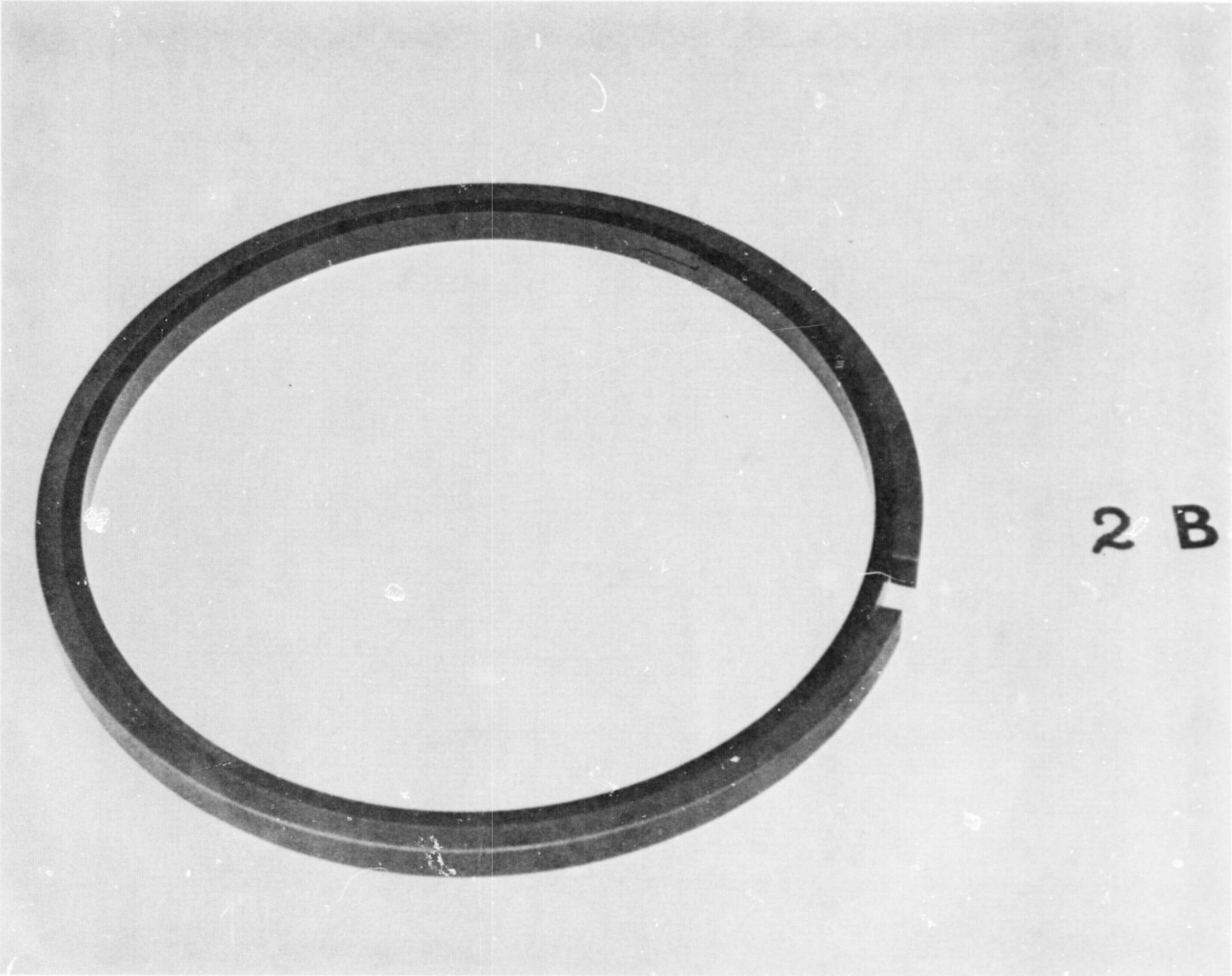


Figure 6-34. First-Stage Seal (B-End), Actuator 3-2

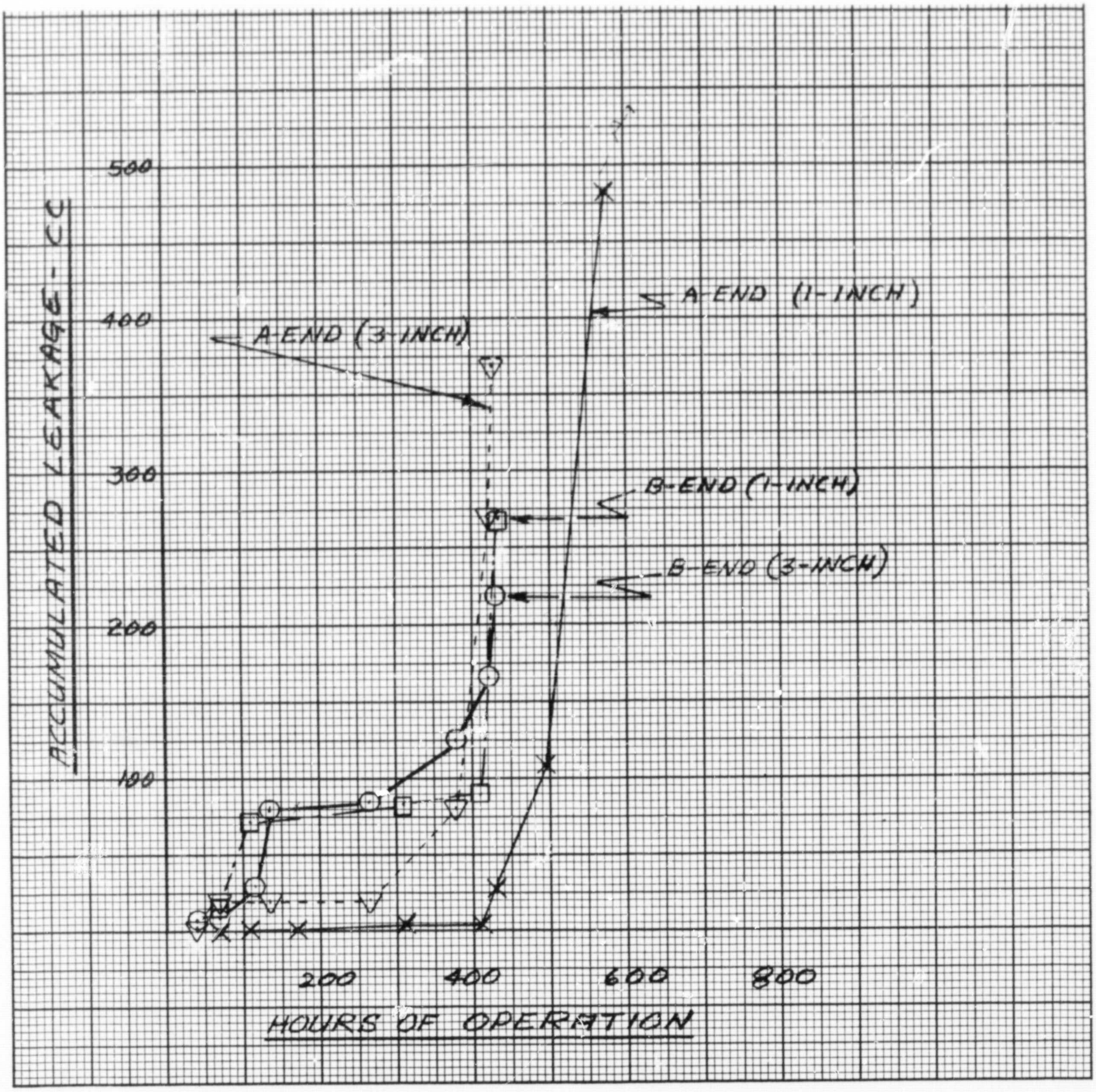


Figure 6-35. Accumulated Leakage versus Time, Reed Seals

rotation of the piston rod could alter the contact pattern at the seal-rod interface, thereby requiring the seal to undergo a wearing in process to achieve good contact again. Testing of the A-end seal was concluded after 574 hours when leakage reached 2-to-8 drops per minute. Accumulated leakage on this seal was 483cc.

The A-end seal, as shown in Figure 6-36 appears to be in good condition. Several of the reeds from the B-end (Figure 6-37) were damaged during removal from the actuator. However, the seals appeared to be in good condition. A comparison of the I. D. dimensions taken before and after test is shown in Table 6-6. With this configuration, the I. D. dimension is the most critical measurement as this determines the range of wear compensation and seal contact load. As shown in Table 6-6, most of the reeds retained some of their original interference fit.

TABLE 6-6. ONE-INCH VASCOJET 1000/SILVER ALLOY REED SEAL (DESIGN AH) SEAL DIMENSIONS BEFORE AND AFTER TEST

ROD DIAMETER - 0.998-INCH

End	Seal No.	Before Test		After Test	
		I. D.	Interference	I. D.	Interference
A	1-S	0.996	0.002	1.002	-
	2-V	0.995	0.003	0.997	0.001
	3-S	0.9952	0.0028	0.996	0.002
	4-V	0.9942	0.0038	0.9955	0.0025
	5-S	0.995	0.003	0.9955	0.0025
	6-V	0.9945	0.0035	0.9955	0.0025
	7-S	0.9955	0.0025	0.9965	0.0015
	8-V	0.9947	0.0033	0.9955	0.0025
B	1-S	0.9958	0.0022	0.9967	0.0013
	2-V	0.9955	0.0025	0.9985	-
	3-S	0.9952	0.0028	0.996	0.002
	4-V	0.995	0.003	0.999	-
	5-S	0.9955	0.0025	0.997	0.001
	6-V	0.995	0.003	0.997	0.001
	7-S	0.9952	0.0028	0.996	0.002
	8-V	0.9945	0.0035	0.9965.	0.0015

S - Silver alloy
V - Vascojet 1000

Inspection of the piston rod indicated that excessive seal leakage was due to the heavy wear pattern (Figures 6-38 and 6-39) on the chrome plating. The

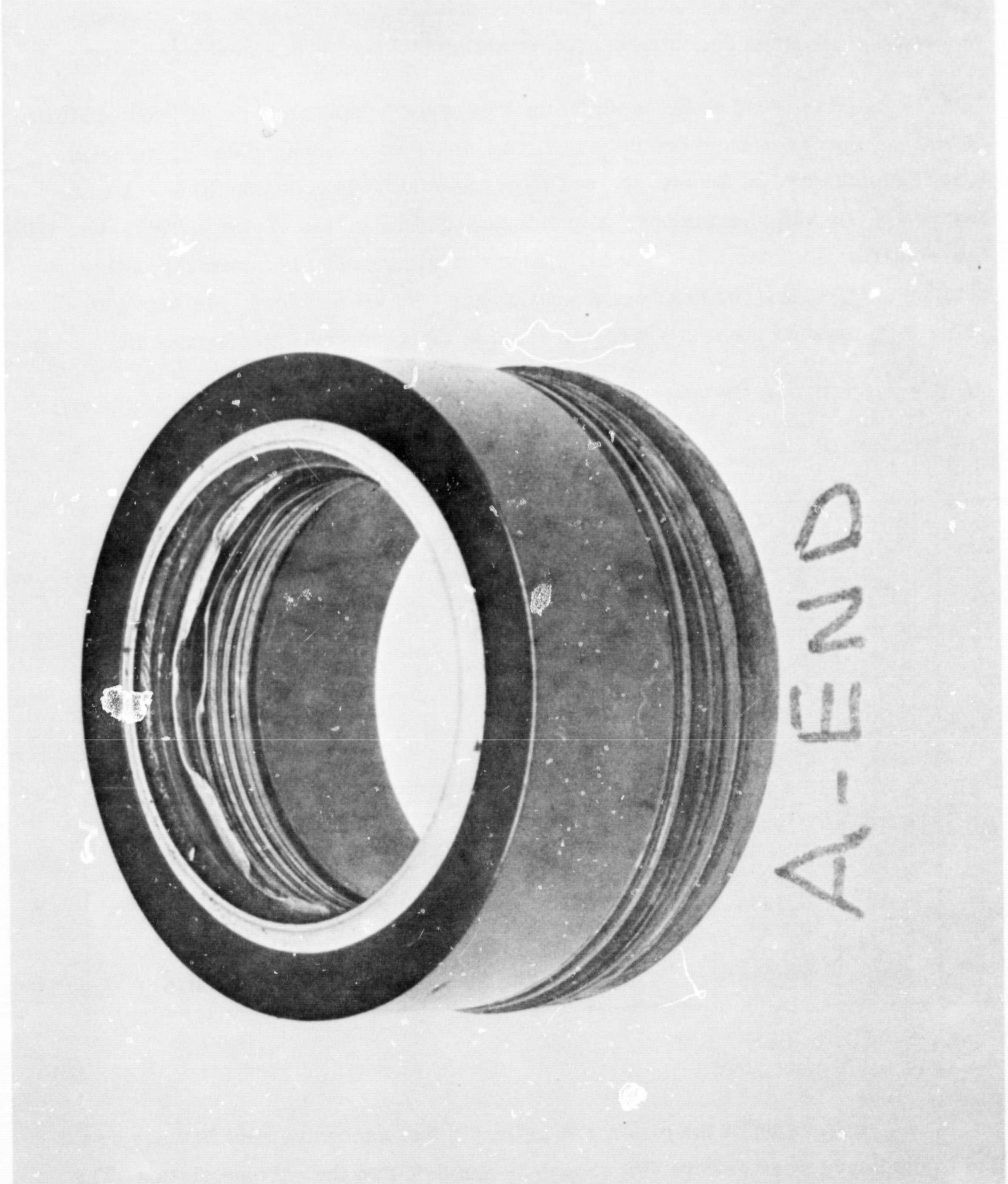


Figure 6-36. One-Inch Reed Seal, A-End (Damaged during disassembly)

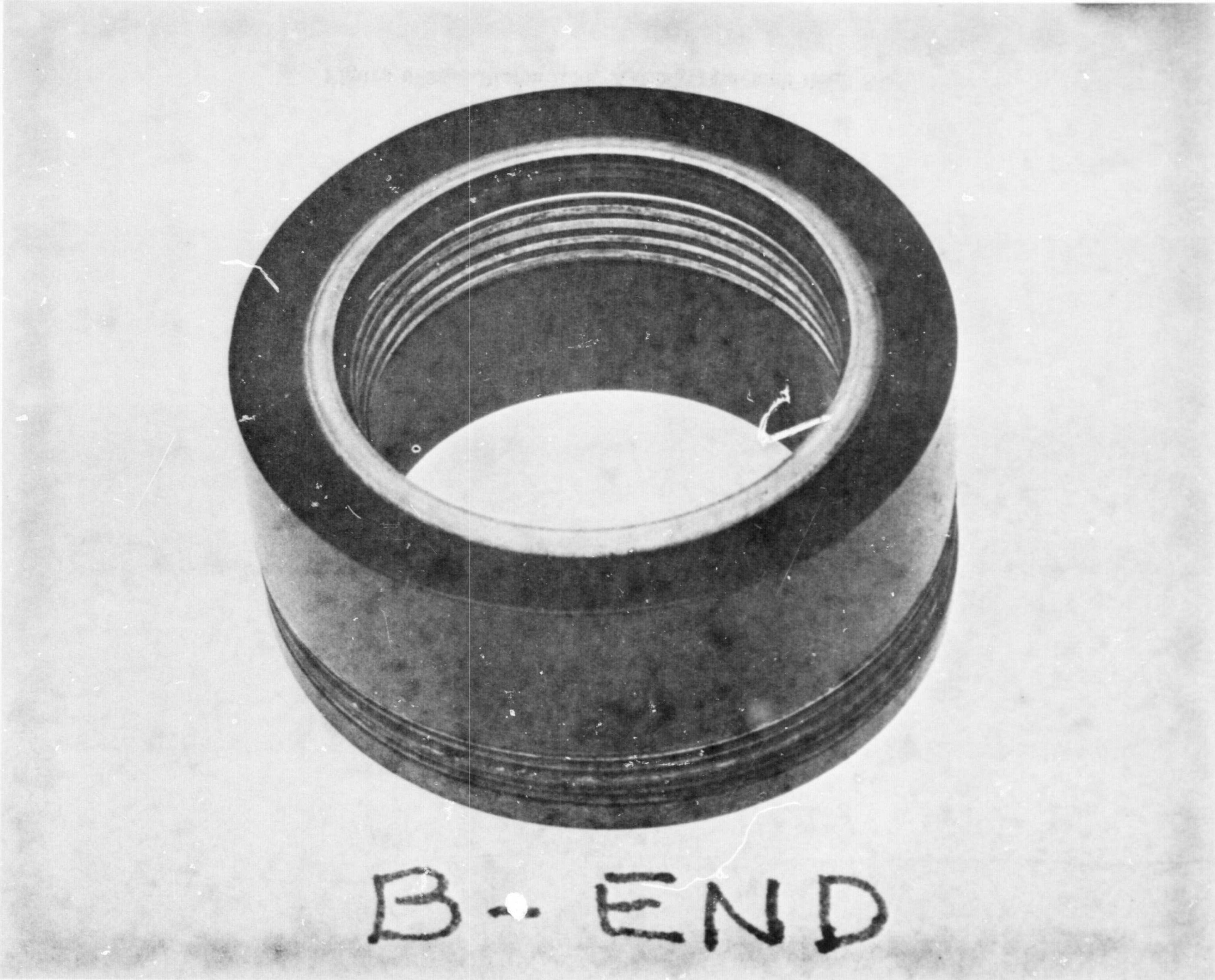


Figure 6-37. One-Inch Reed Seal, B-End

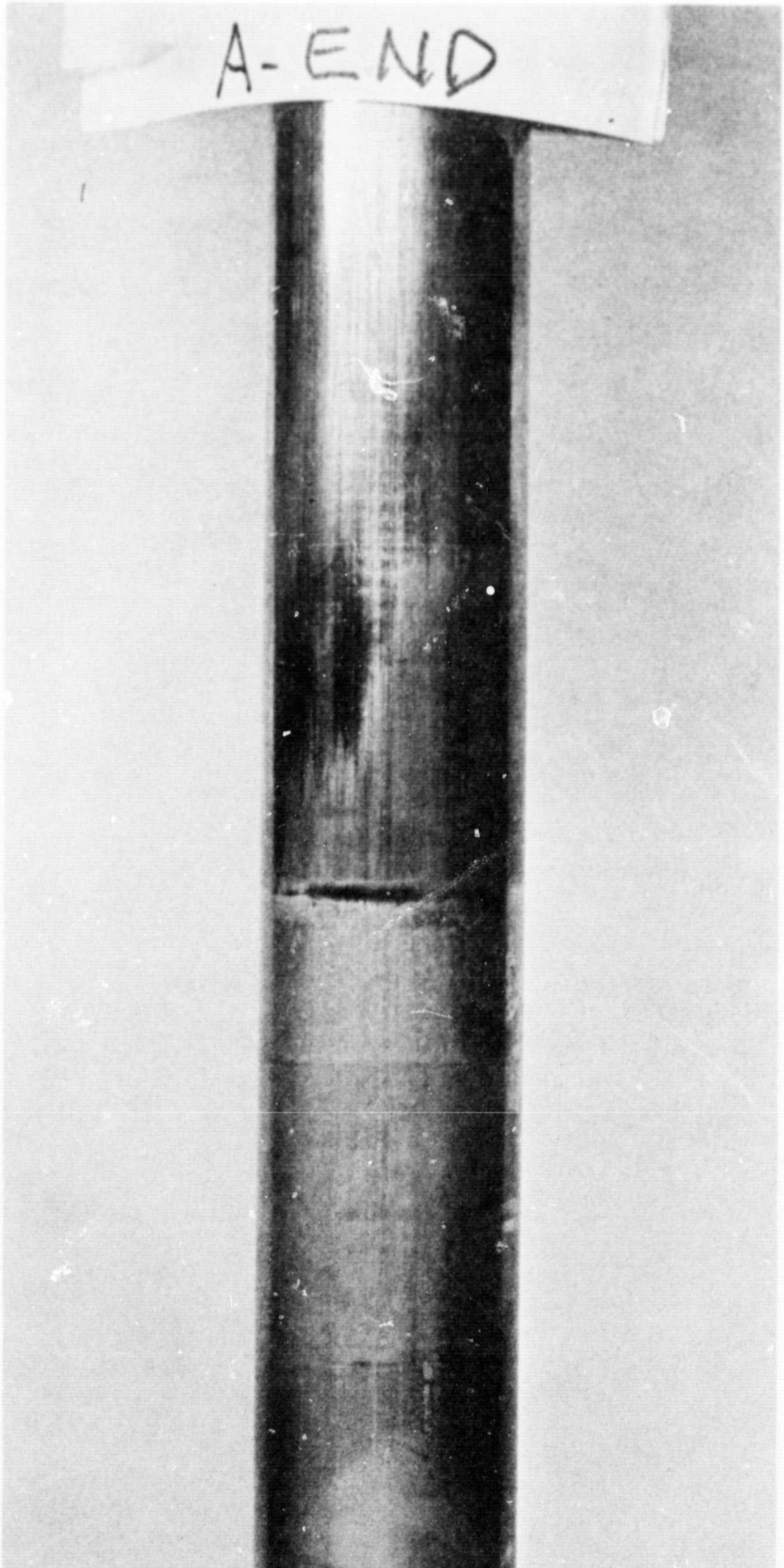


Figure 6-38. Piston Rod, A-End, One-Inch Reed Seal

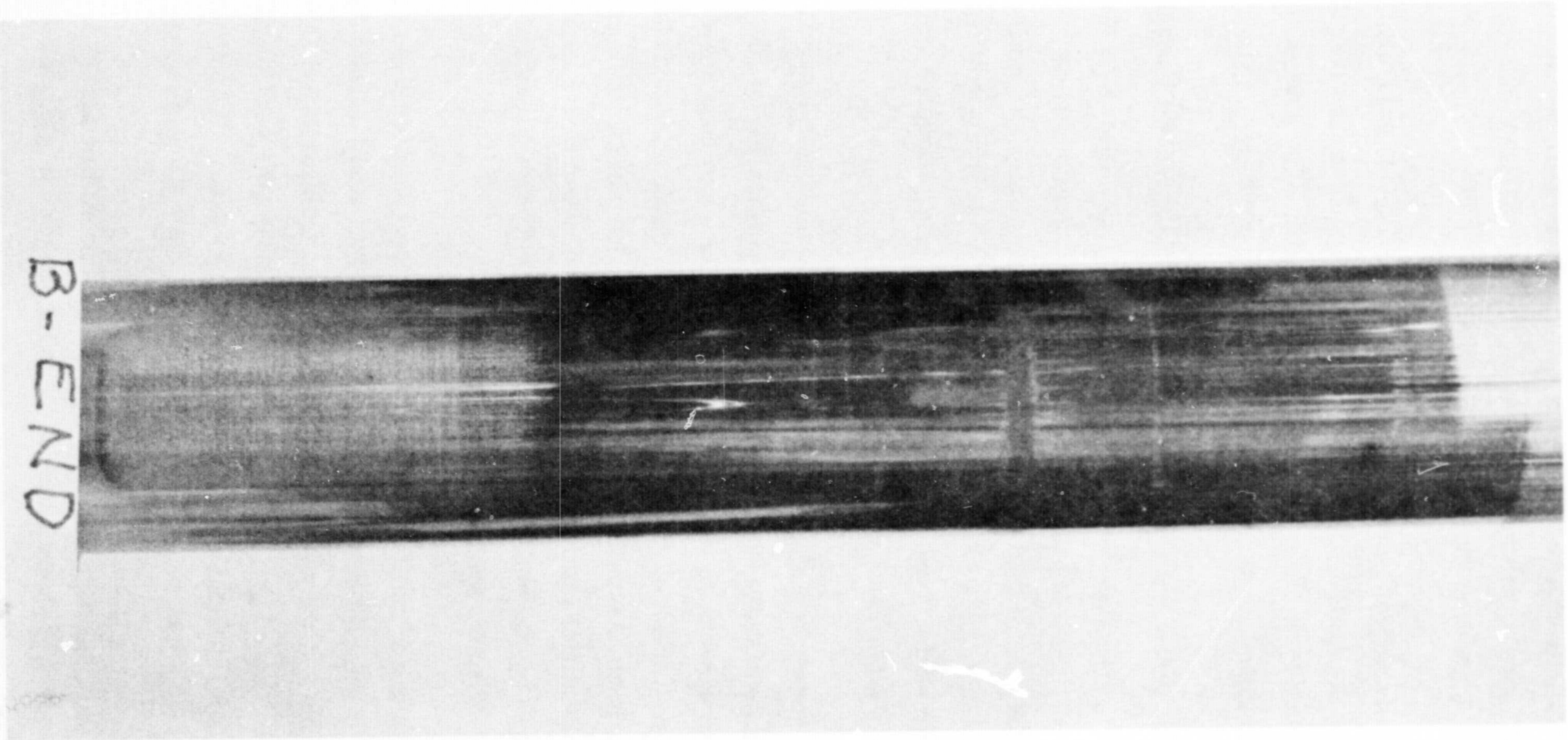


Figure 6-39. Piston Rod, B-End, One-Inch Reed Seal

pattern of longitudinal scratches provided the leakage path for the fluid. The depth of these scratches were determined with a surface analyzer and was found to be approximately 0.0001-inch deep.

The first-stage seals (Figures 6-40 and 6-41) gave satisfactory performance during the entire endurance run. Leakage from the A-end seal, which was the unbalanced type, was 6-to-11 cc per minute. The pressure balanced seal on the B-end exhibited leakage of 10-to-15cc per minute.

6. Three-Inch Vascojet 1000/Silver Alloy Reed Seal (Design AH)

Testing of this configuration was concluded after 428 hours of operation. Leakage at the time testing was terminated ranged between 4-to-5 drops per minute on the A-end seal and 5-to-6 drops per minute on the B-end seal. Accumulated leakage at the conclusion of the test was 217cc and 370cc for the A-end and B-end seals, respectively. Leakage characteristics during the test are depicted in Figure 6-35.

As shown in Figure 6-42 and 6-43, the sealing reeds were in excellent condition. Measurements were taken on the inner diameter of the reeds to determine the amount of wear in this area. These values are compared with the original measurements in Table 6-7. On the basis of this comparison, wear on the sealing reeds was very low. As indicated in Table 6-7, several of the reeds retained more than 50% of their initial interference fit. This indicated that adequate seal contact load was retained by the sealing reeds and that excessive leakage was caused by the multiplicity of score marks on the piston rod.

Both the A and B-end portion of the rod (Figures 6-44 and 6-45), were heavily scored during the long-stroke as well as the short-stroke cycling. The depth of the score marks was determined to be approximately 0.00011-to-0.00017 inch. It was also noted that the scoring extended into the portion of the rod where the first-stage seal normally operates (see Figure 6-46). These score marks were caused by metallic wear particles generated by the reed seals, which were imbedded into the polyimide first-stage seals.

Because of the surface damage on the piston rod, leakage from the first-stage seals was considerably higher than that normally encountered. Leakage

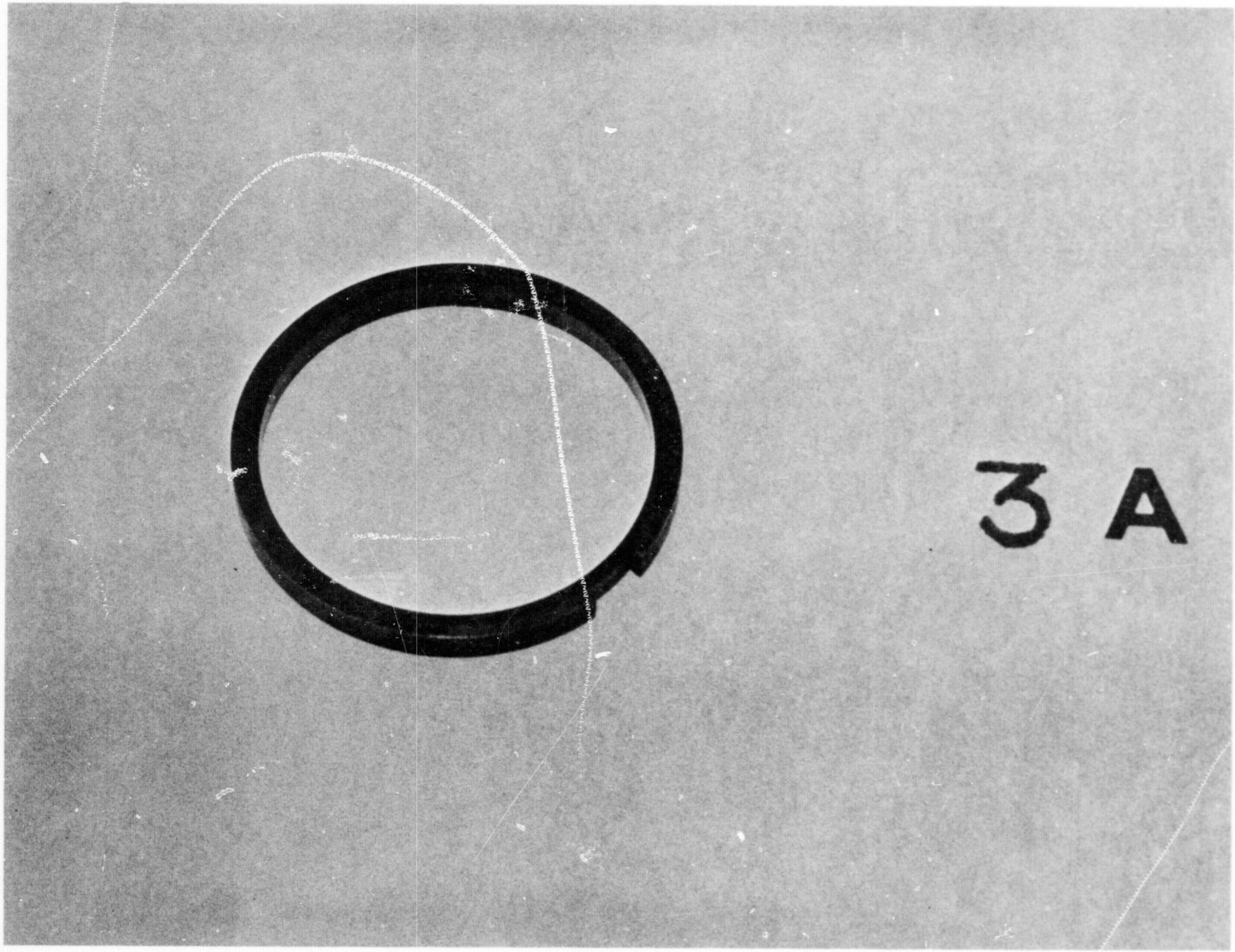


Figure 6-40. First-Stage Seal (A-End), Actuator 1-3

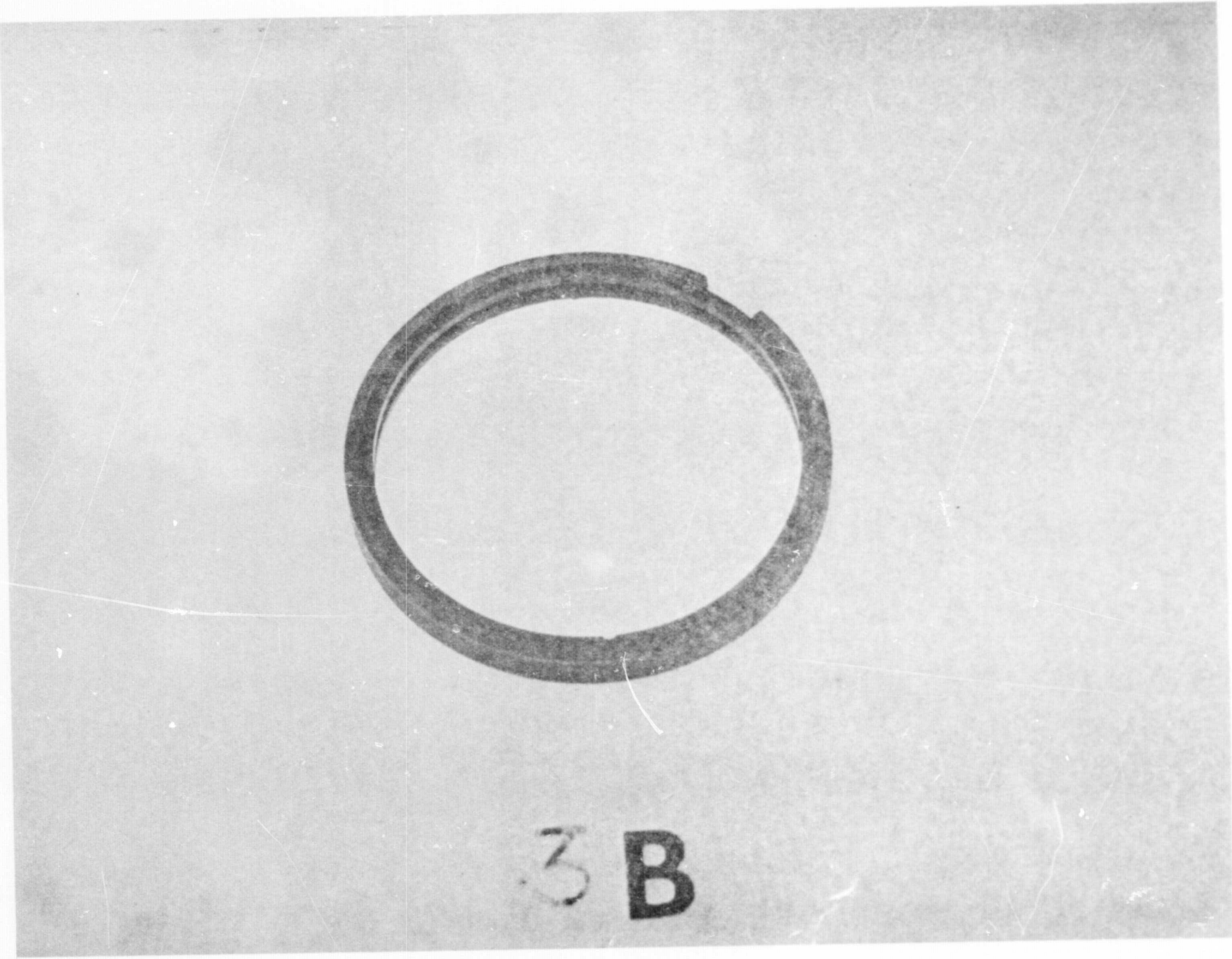


Figure 6-41. First-Stage Seal (B-End), Actuator 1-3

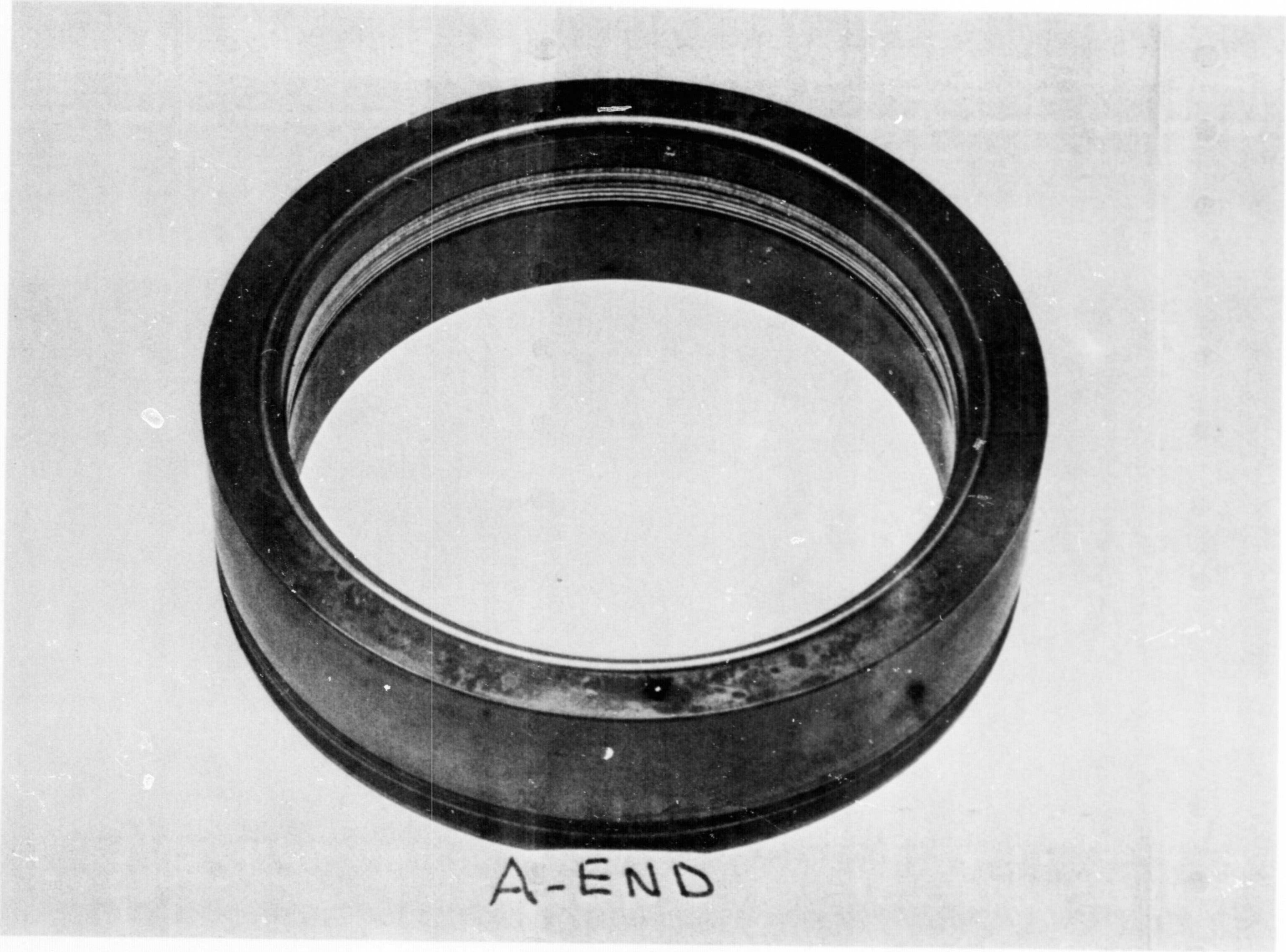


Figure 6-42. Three-Inch Reed Seal, A-End

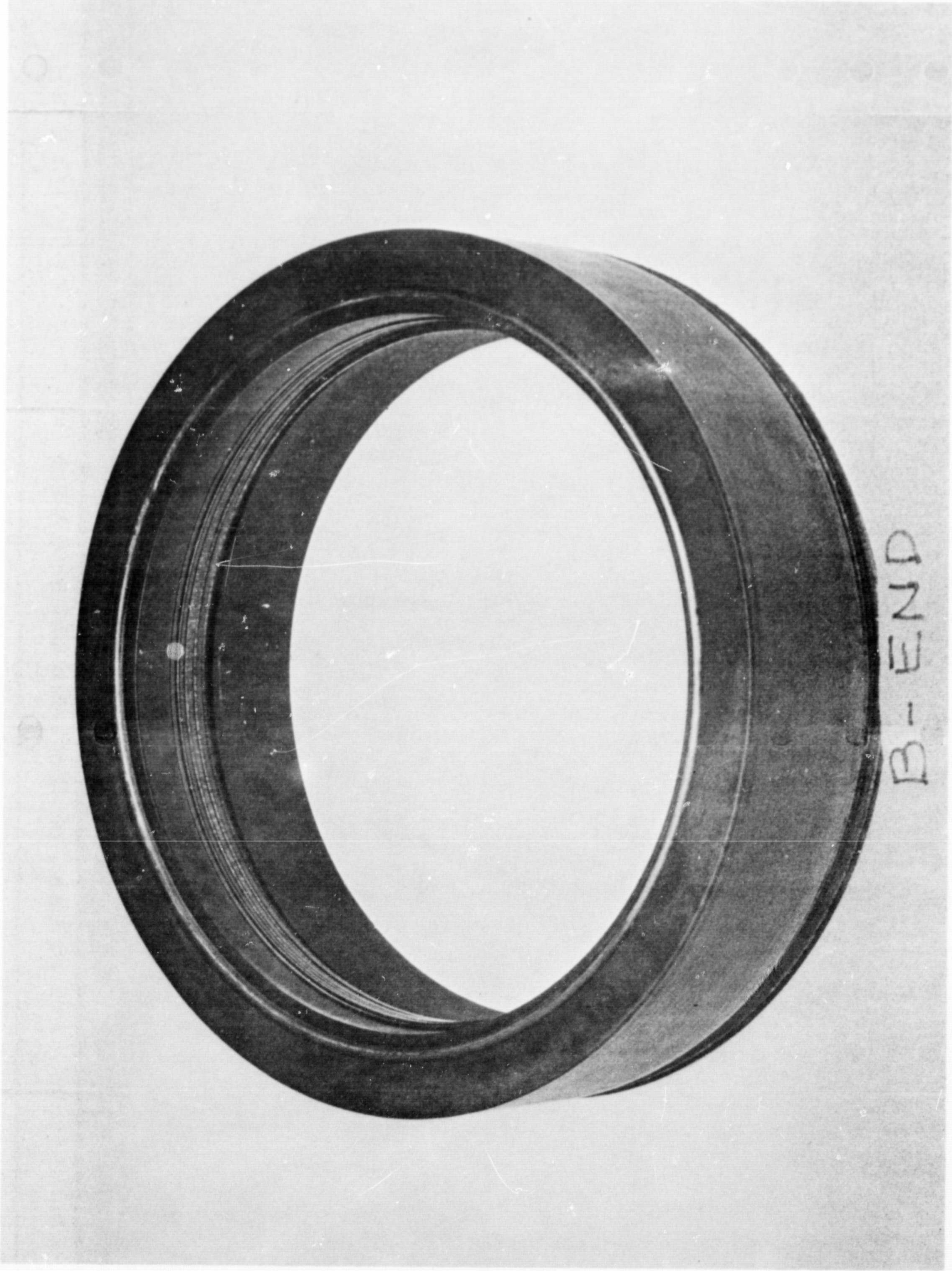


Figure 6-43. Three-Inch Reed Seal, B-End

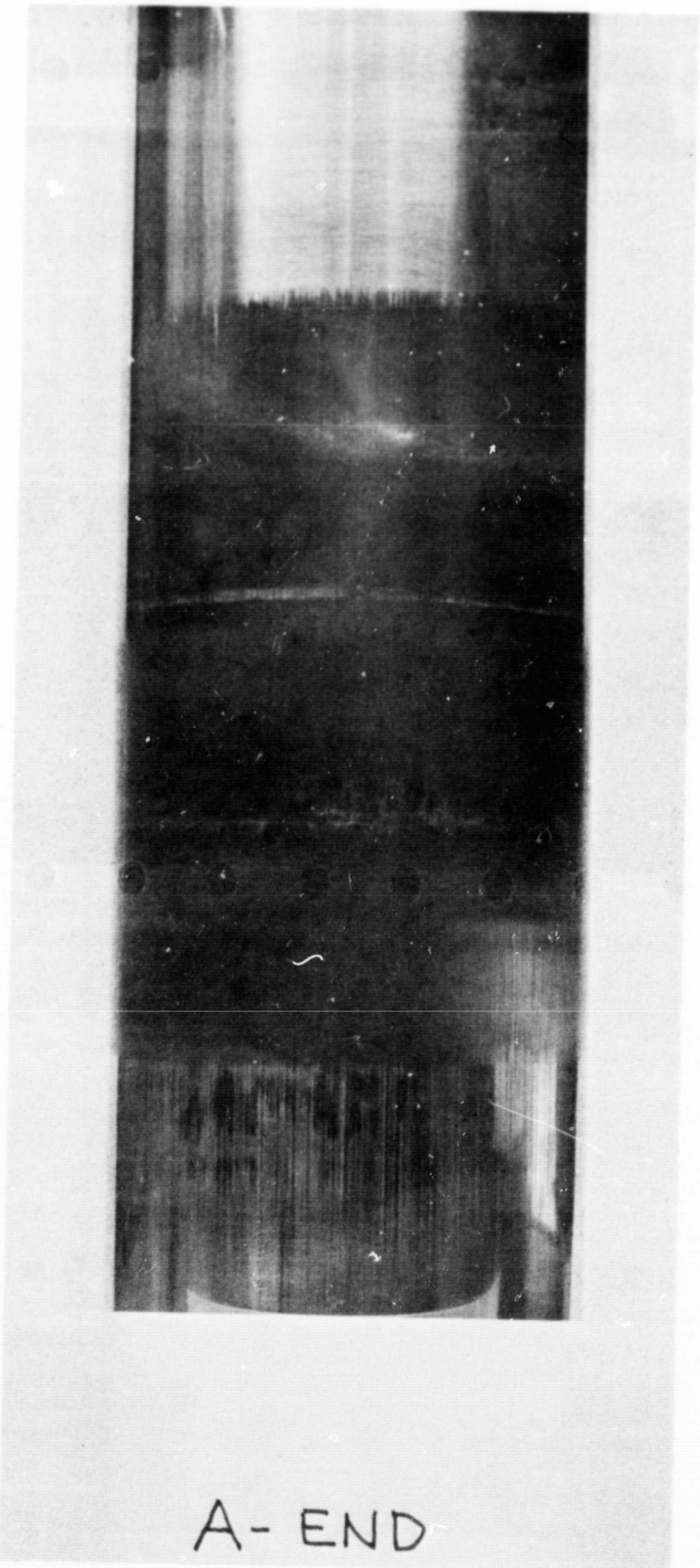


Figure 6-44. Piston Rod, A-End, Three-Inch Reed Seal

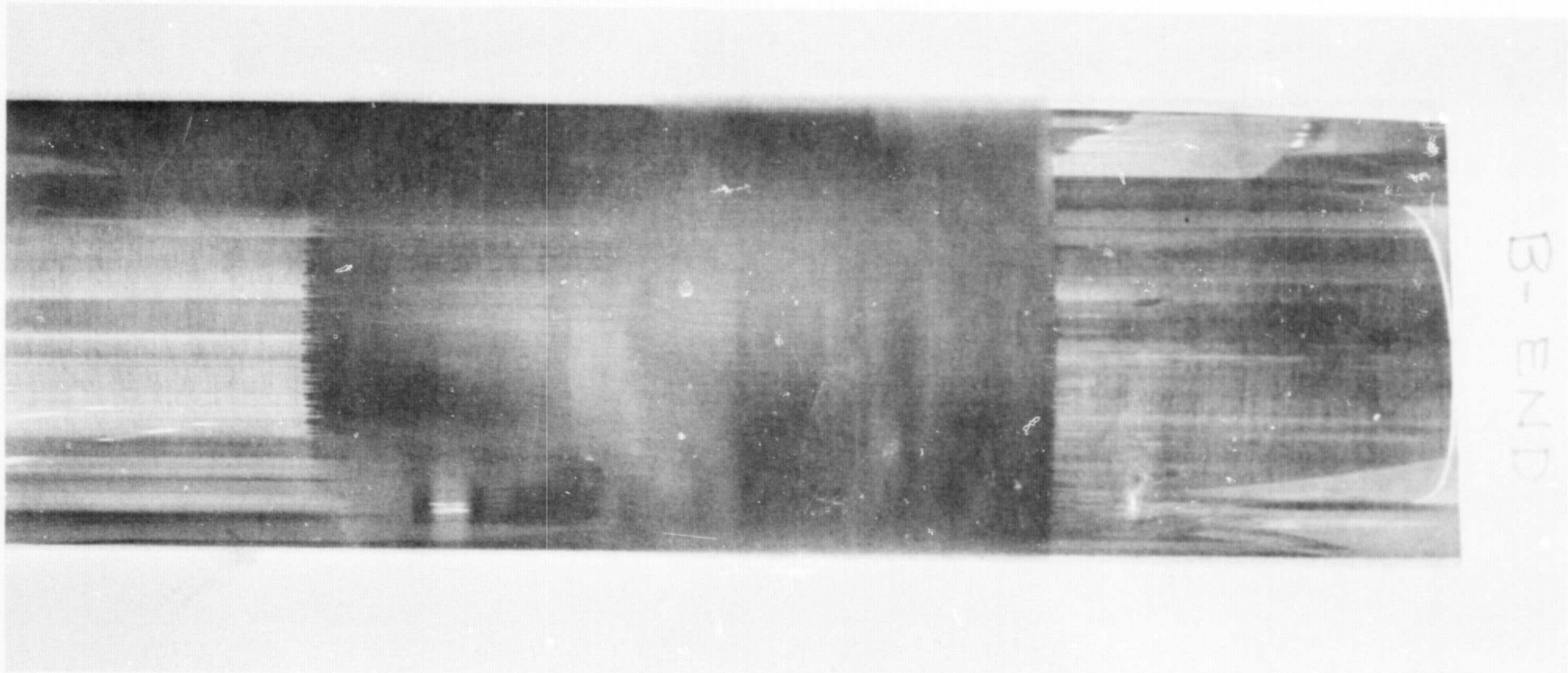


Figure 6-45. Piston Rod, B-End, Three-Inch Reed Seal

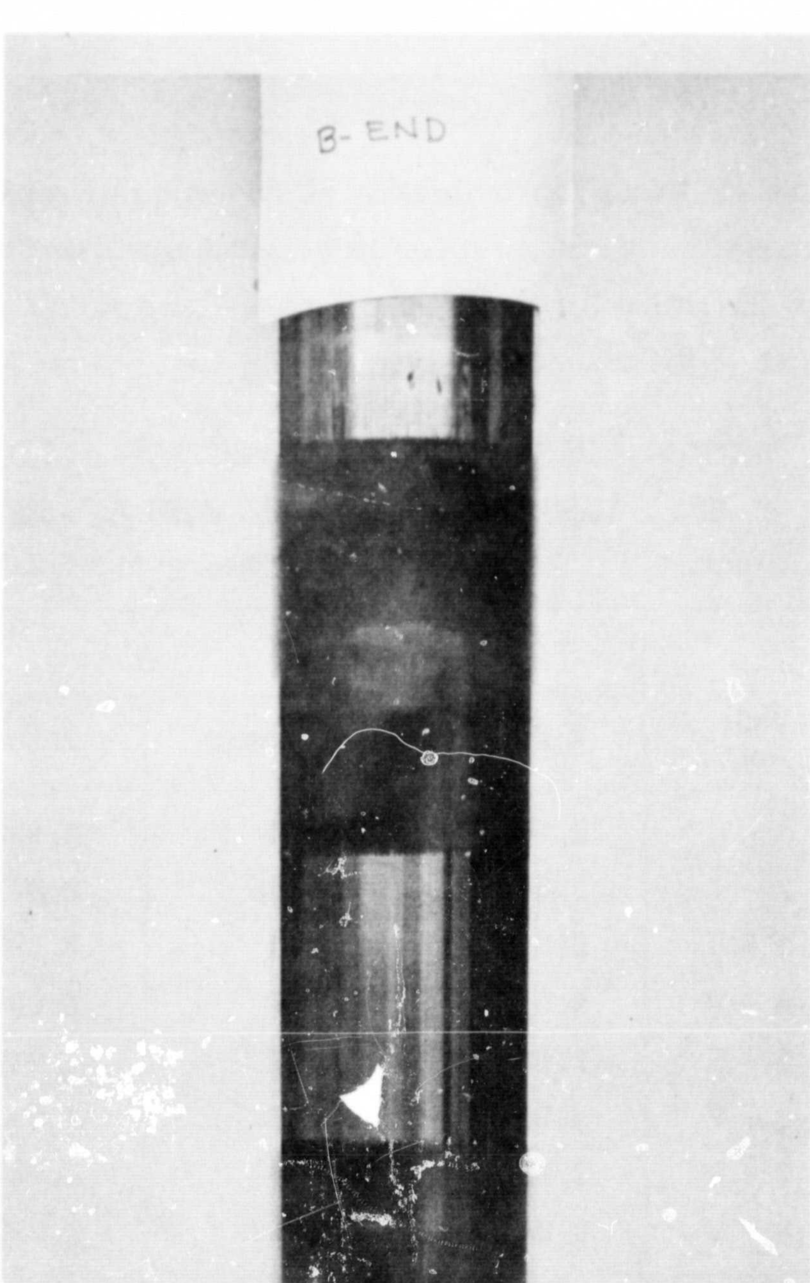


Figure 6-46. Piston Rod, A- and B-Ends

from the A-end seal was approximately 50-to-60cc per minute. The B-end seal experienced leakage in excess of 100cc per minute towards the end of the test. A broken ring joint was the cause for the higher leakage. As shown in Figures 6-47 and 6-48, both seals were of the unbalanced type.

TABLE 6-7. THREE-INCH VASCOJET 1000/SILVER ALLOY REED SEAL (DESIGN AH) SEAL DIMENSIONS BEFORE AND AFTER TEST

ROD DIMENSIONS - 2.9970

End	Seal No.	Before Test		After Test	
		I.D.	Interference	I.D.	Interference
A	1 - S	2.991	0.006	2.9982	-
	2 - V	2.9885	0.0085	2.993	0.004
	3 - S	2.992	0.005	2.9957	0.0013
	4 - V	2.990	0.007	2.9945	0.0025
	5 - S	2.990	0.007	2.9925	0.0045
	6 - V	2.991	0.006	2.9968	0.0012
	7 - S	2.990	0.007	2.994	0.003
	8 - V	2.987	0.0083	2.995	0.002
B	1 - S	2.989	0.008	2.9907	0.0063
	2 - V	2.990	0.007	2.995	0.002
	3 - S	2.992	0.005	2.9962	0.0008
	4 - V	2.991	0.006	2.994	0.003
	5 - S	2.990	0.007	2.9967	0.0003
	6 - V	2.988	0.0082	2.992	0.005
	7 - S	2.990	0.007	2.993	0.004
	8 - V	2.988	0.009	2.992	0.005

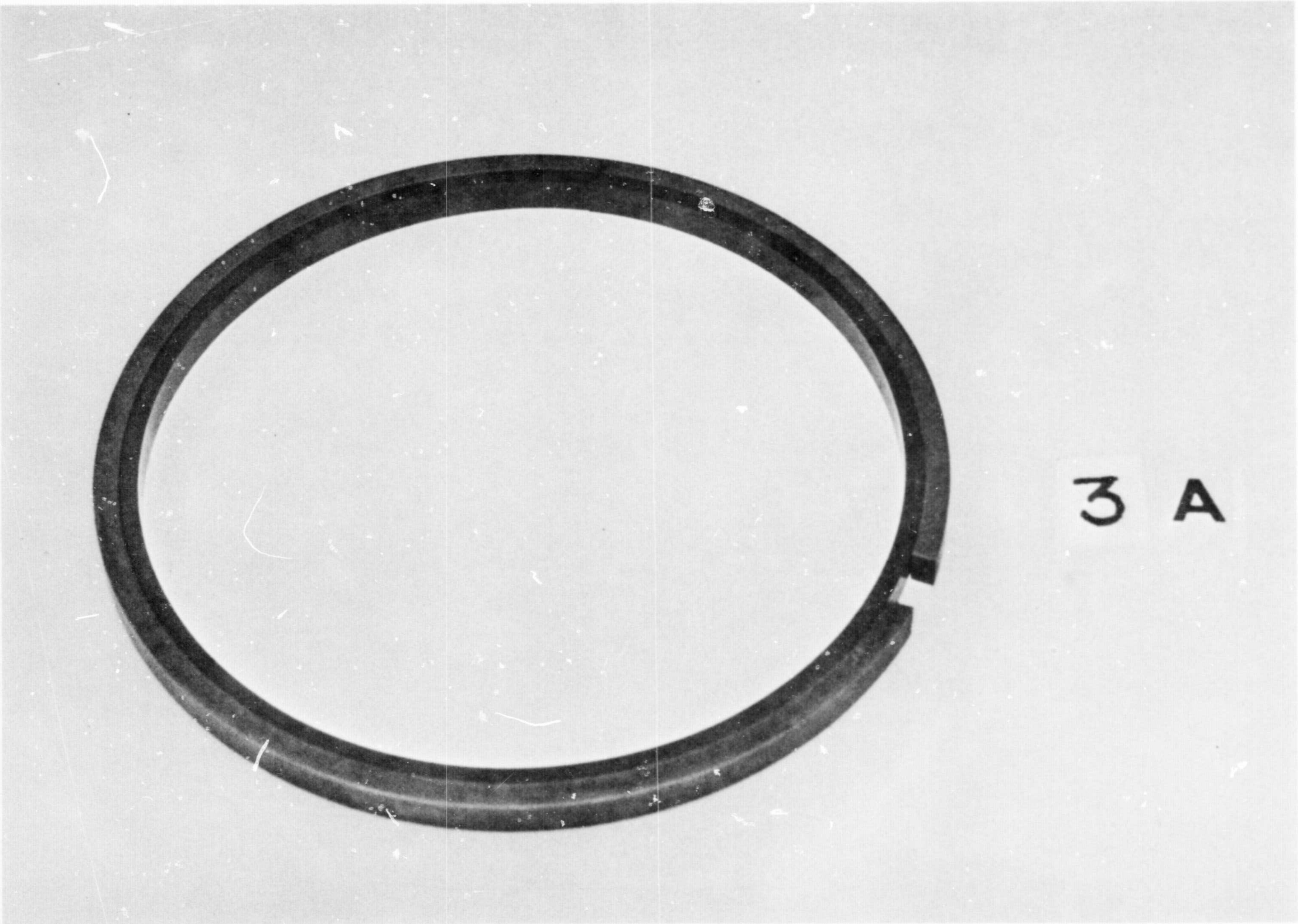


Figure 6-47. First-Stage Seal (A-End), Actuator 3-3

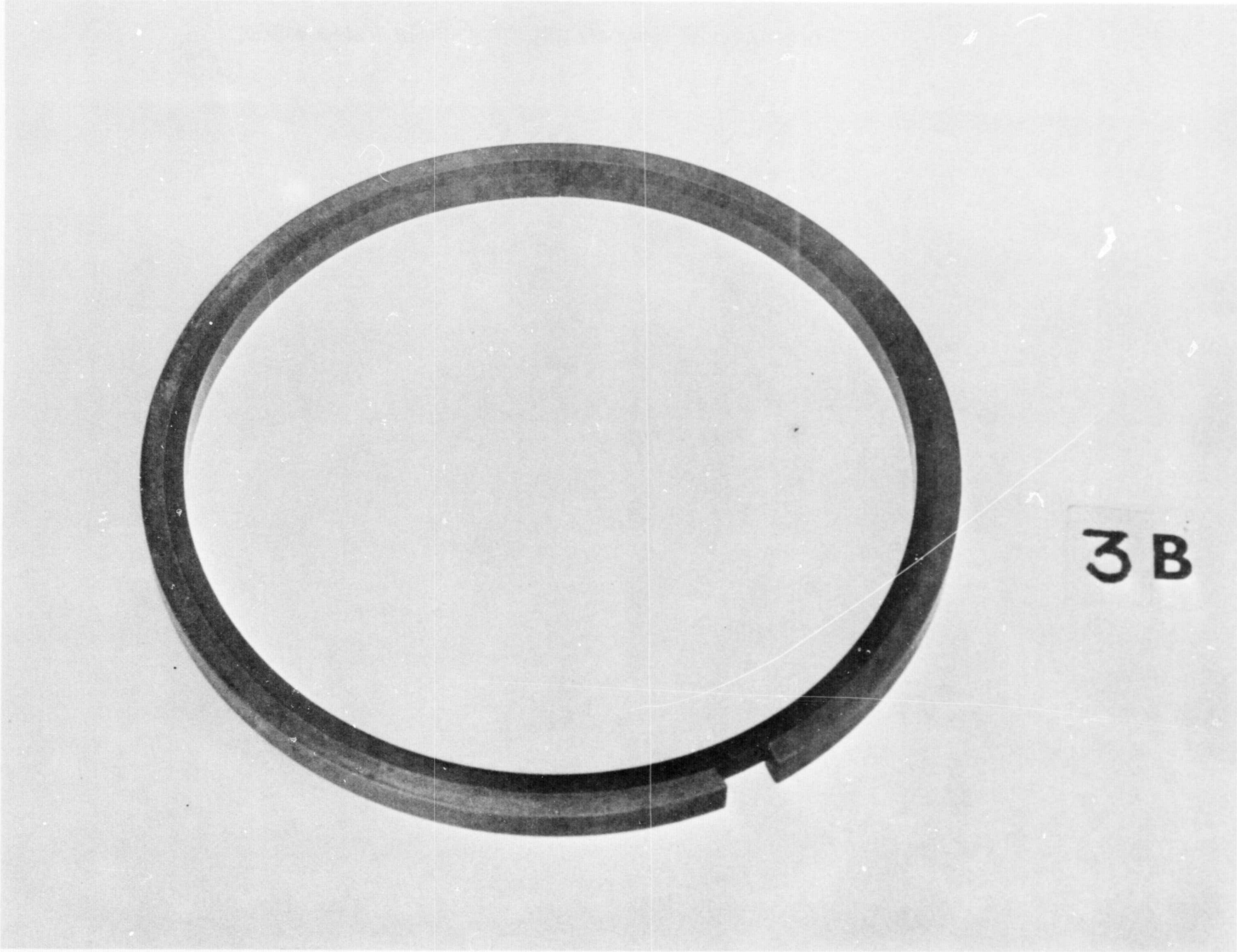


Figure 6-48. First-Stage Seal (B-End), Actuator 3-3

SECTION VII
HIGH-PRESSURE SINGLE-STAGE SEAL

A. GENERAL

Single-stage seals are considered desirable in actuator applications where internal leakage cannot be tolerated. Utility type actuators, which have a much lower cyclic life would also find single-stage seals to be highly desirable because of the possible savings in weight and space. Accordingly, efforts were devoted to the development of such a configuration. A one-inch diameter seal was selected as being a representative size to work with. Performance goals for the seal were: (1) operating life of 100 hours, (2) capable of performing at temperature to 500°F and 3000 psi. The seal was subjected to essentially the same test profile used in the endurance testing.

B. SEAL DESIGN

A review of the seal-material combinations developed during the course of this program was conducted to determine the most suitable configuration for the single-stage seal. Candidate seal designs considered are shown in Table 7-1. These configurations were considered from the standpoint of potential friction and wear when subjected to a working pressure of 3000 psi. As summarized in Table 7-1, performance of the polyimide V-seal (Design B), at low pressures is excellent. However, the growth potential of this material is limited to approximately 600°F. Because of its low mechanical properties, radical design changes would be required to enable the material to withstand an operating pressure of 3000 psi. Although the Vascojet 1000/silver alloy reed seal (Design AH), has also demonstrated good sealing performance at low pressures, application at high pressures would tend to produce excessive pressure energizing of the thin flexible sealing members. The increased contact stresses at the seal interface due to this energizing effect could induce high sliding friction and rapid wear of the seal and/or rod plating.

TABLE 7-1 - CANDIDATE HIGH-PRESSURE SEAL DESIGNS

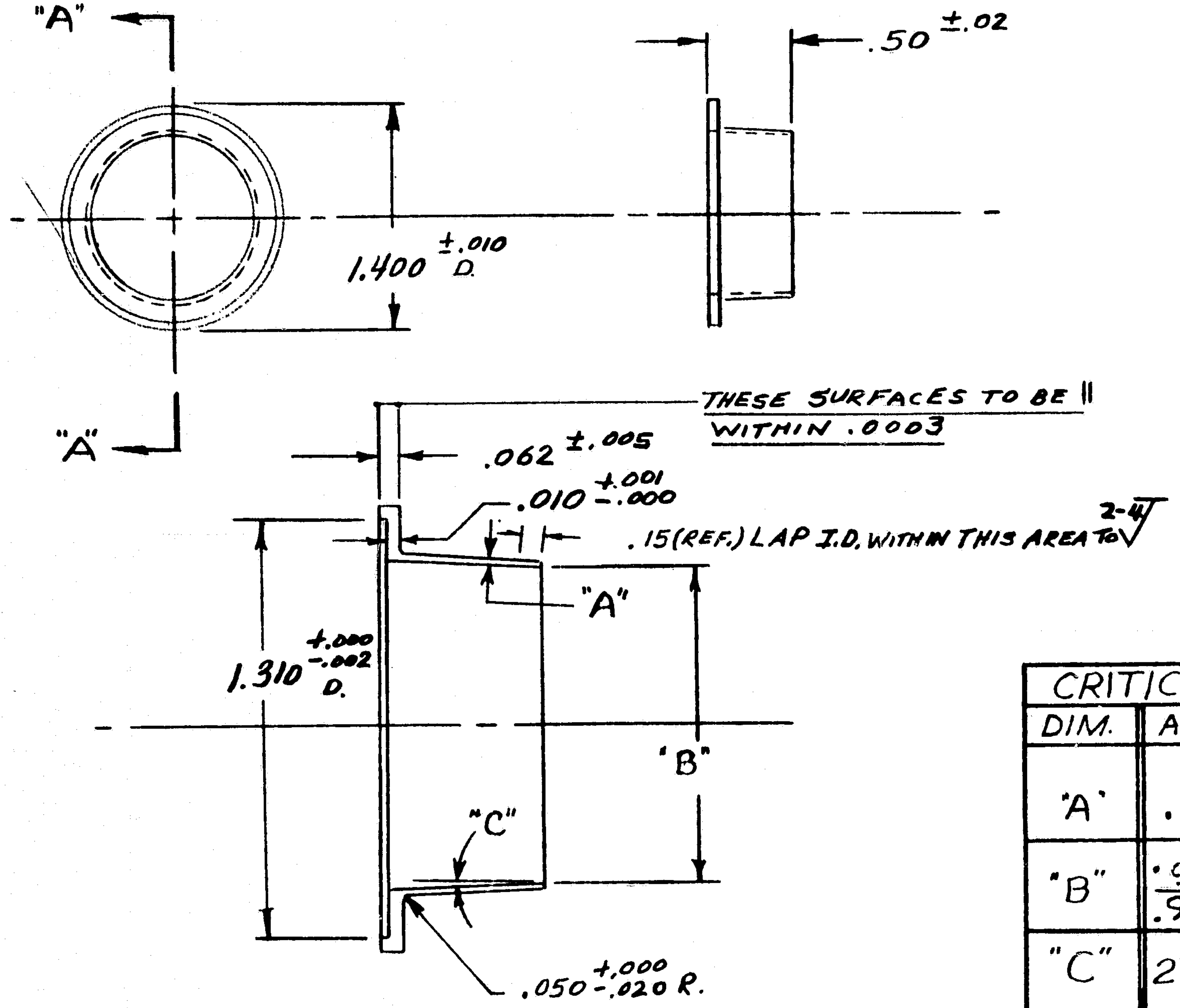
Seal Design	Material	Design Features	Past History	Application at High Pressure
Design B	polyimide	Spring loaded, good wear compensation, fairly low friction	Successfully completed 150-hour low pressure test	Material may not withstand 3000 psi. Requires redesign. Organic material limits growth potential of seal.
Design AH Reed Seal	Vascojet 1000 and silver alloy	Interference fit, pressure energized	Successfully completed 150-hour low pressure test	Potential high friction and rapid wear at 3000 psi. Requires redesign to minimize pressure energizing.
Design D Lip Seal	Vascojet 1000, Cobalt molybdenum alloy	Interference fit, load relieving, fairly low friction	Vascojet lip seal operated for 86 hours (up to 500°F) with very low leakage. Cobalt lip seal operated for 133 hours (up to 600°F) with very low leakage	Good potential for 3000 psi application. Friction and wear should be low because of load relieving feature.

The metallic lip seal (Design D) offers the best potentials of meeting the 100 hour life requirement at 3000 psi. With this particular design, seal contact load is inversely proportional to the working pressure because of the load relieving feature of the seal. Therefore, at 3000 psi the initial sealing load would be reduced to a minimum required for effective sealing. Thus wear and friction would be minimized. Both the cobalt molybdenum alloy and Vascojet 1000 were considered for the lip seal. However, the Vascojet 1000 appears to be a more suitable material for 3000 psi operation because of its higher tensile properties. The high strength obtainable with this material permits the use of a thinner lip for the seal. Greater lip deflection (interference) can also be obtained with the higher strength material, which increases the ability of the seal to compensate for wear.

C. SEAL DEVELOPMENT AND TEST

The configuration of the high pressure lip seal is shown in Figure 7-1. Two versions of the seal were fabricated, each was configured slightly different. The A-seal was designed to obtain maximum seal interference (.0038-inch) for wear compensation. The cross-sectional thickness of the seal was designed as thin as practical to obtain good lateral tracking characteristics when subjected to side loading. To obtain these characteristics the seal was designed to a hoop stress of approximately 110,000 psi, and a seal load of 150 pounds per inch of circumference. This resulted in a seal thickness of 0.019-inch and a seal contact pressure (based on contact width of 0.040-inch) of 3800 psi.

This seal was designed to provide optimum flexibility and wear compensation. However, at 3000 psi the seal would be approaching its tensile yield stress, as the material stresses induced by the 3000 psi fluid pressure becomes additive to the original hoop stress. Consequently, the B-seal was designed to a lower initial hoop stress of approximately 85,000 psi to provide a margin of safety at the 3000 psi operating condition. A seal load of 330 pounds per in of circumference was selected, resulting in a seal thickness of 0.037-inch, which produced a seal interference of 0.003-inch. Although the sealing load was higher than that used in the A-seal, the actual interface contact pressure was maintained at approximately the same level as the A-seal. This condition was possible because the design of



CRITICAL SEAL DIM.		
DIM.	A-SEAL	B-SEAL
"A"	.019 ±.0005	.037 ±.0005
"B"	.9937 / .9940	.9944 / .9947
"C"	2° ±15'	1.45° ±15'

Figure 7-1. High-Pressure Lip Seal (Vascojet-1000)

the B-seal was such that a greater contact area was obtained, which enabled the higher load to be distributed over a wider area.

Seal break out friction was determined on both seals at pressure up to 4000 psi using F-50 silicone fluid. Testing was conducted separately on the A and B seal. In these tests, the seals were checked in pairs of the same configuration. As shown in Figure 7-2, friction for both the A and B seal were approximately the same, which supports the assumption that the contact load experienced by both seals would be approximately the same. Friction for the A and B seal at zero pressure was 340 and 380 pounds, respectively. At 4000 psi, friction was 105 and 130 pounds, respectively. No leakage was experienced from either configuration tested.

Evaluation of these seals were conducted in the cycling rig depicted in Figure 4-2. Arrangement of the seals in the seal tester is typical of that shown in Figure 4-4. The seals were evaluated at pressures to 3000 psi and temperature of 500°F. Long-stroke (2-inch) cycling was performed during the heat up from room temperature to 500°F. The seals were then subjected to short-stroke (± 0.12-inch) cycling for approximately 1.5 hours at 500°F, followed by long-stroke cycling during the cool down from 500°F.

A summary of the test is shown in Table 7-2. The results obtained show that leakage became excessive when piston rod surface damage due to short-stroke cycling occurs. It is apparent that at the higher pressures, surface imperfections becomes critical because of the greater propensity of the fluid to leak. Consequently, the multiplicity of scratches produced by the short-stroke cycling operation became quite detrimental to satisfactory sealing.

TABLE 7-2. RESULTS OF SINGLE-STAGE SEAL

Seal Configuration	Total Time (hr)	Time at 500°F	Total Cycles	Cycles at 500°F	Thermal Cycles	Total Leakage(cc)
A-seal Vascojet 1000 Lip Seal	10.25	0.5	24,130	3,930	2	67
B-seal Vascojet 1000 Lip Seal	25.5	9.5	103,554	72,694	6	453
Polyimide Lip Seal (Replacement for A-seal)	15.25	9.0	79,424	68,764	4	6

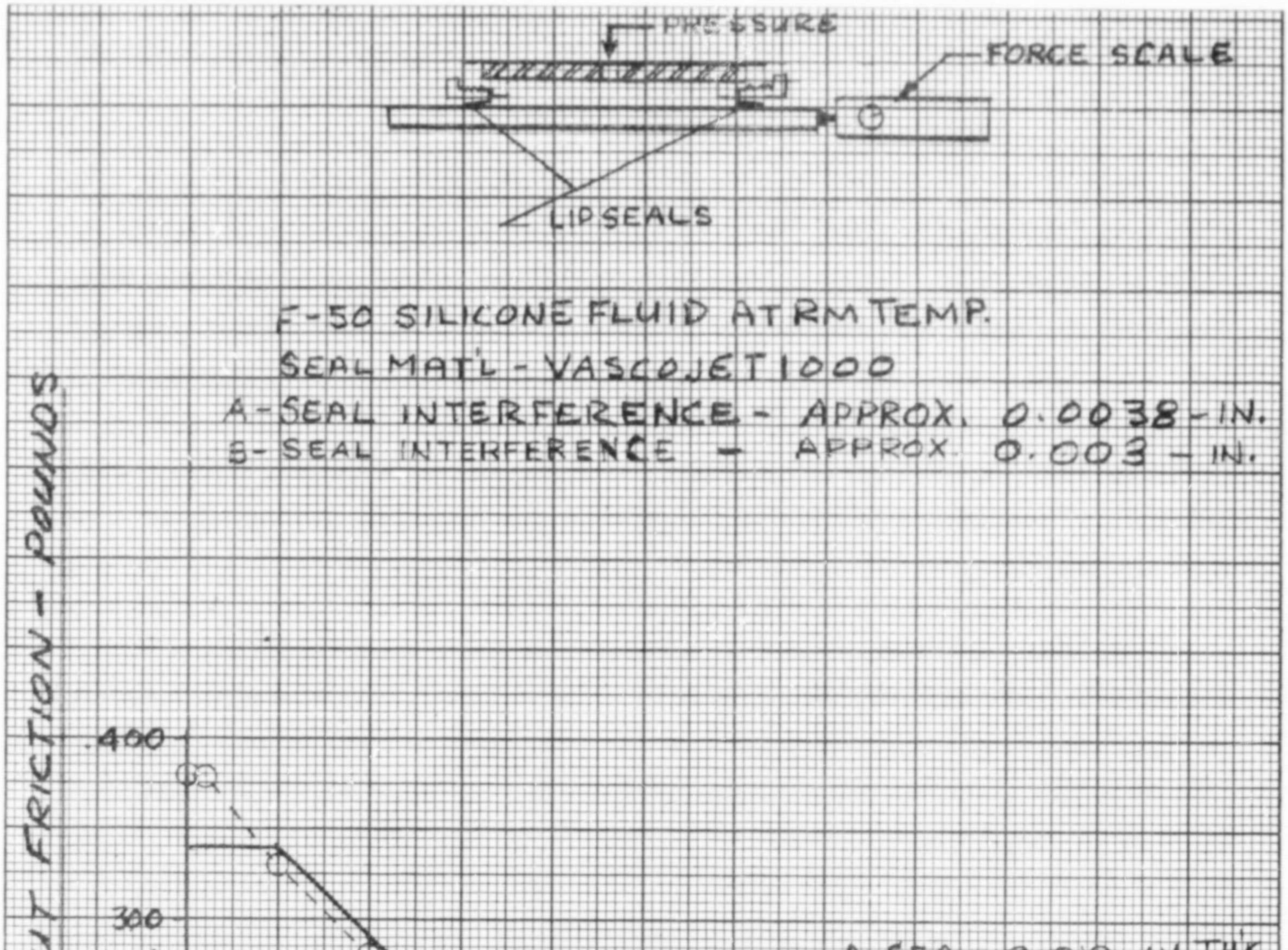


Figure 7-2. Lip Seal Breakout Friction versus Pressure (Total Friction for Two Seals)

The A-seal (Figure 7-3) experienced leakage of 27 drops per minute after 10.25 hours of cycling. Leakage was caused by surface scratches on the chrome plated piston (See Figure 7-4). Measurements taken of the seal indicated that wear of 0.001- 0.0015-inch has taken place at the sealing lip, which resulted in the loss of seal contact load. Leakage from the B-seal was also experienced but to a lesser degree. Total leakage collected on this seal for the same time period was 15cc. As shown in Figure 7-5, piston rod damage due to short-stroke cycling was also experienced. Because of the lower leakage exhibited by this seal, it was decided to continue its evaluation. This was accomplished by replacing the A-seal with a set of polyimide lip seals, which were similar to the configuration shown in Figure 3-1. With this arrangement, the B-seal (Figure 7-6), was operated for a total of 25.5 hours when excessive leakage developed and testing was concluded. Total leakage collected during this time period was 453cc. At the time of failure, the seal could not hold pressure because of the steady stream leakage.

TABLE 7-3. LIP SEAL DIMENSION

Seal	I.D. (inch)	Lip Thickness (inch)	Interference* (inch)	I.D. (inch)	Lip Thickness (inch)	Interference* (inch)
A-seal	0.9938	0.019	0.0037	0.9960	0.0175 0.018	0.0015
B-seal	0.9943	0.0370	0.0032	0.9970	0.0352 0.0358	0.0005

* Seal interference based on 0.9975-inch rod diameter

During the above testing, a slight variation of the test conditions was introduced. This consisted of shifting the piston rod to a different position at the start of each thermal cycle. Since it was apparent that a certain amount of surface damage usually occurs on the piston rod during short-stroke cycling, this procedure permitted the B-seal to operate on an undamaged portion of the rod during each thermal cycle. The purpose here was to determine what affect

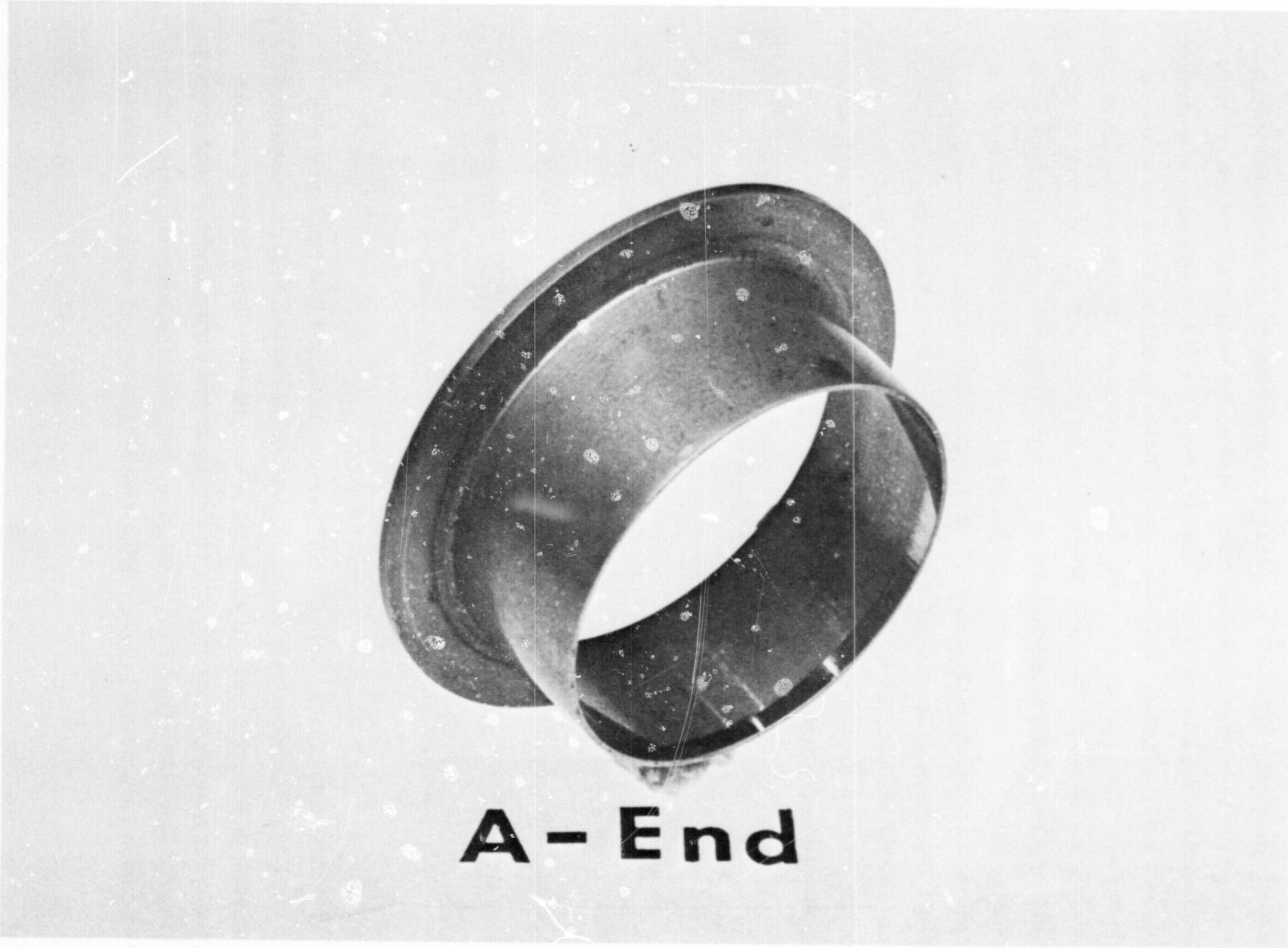


Figure 7-3. High-Pressure Lip Seal, A-End

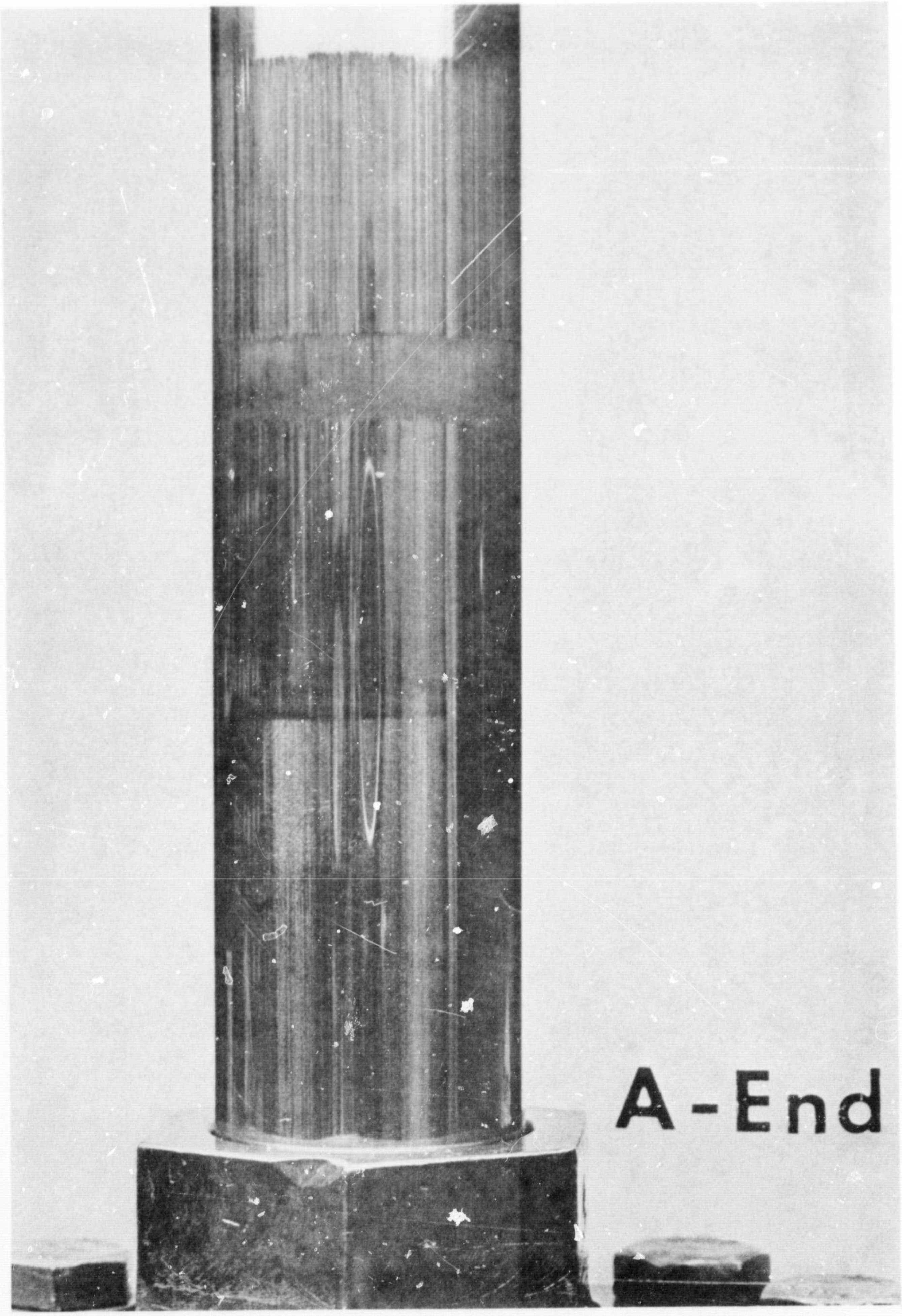


Figure 7-4. Short-Stroke Wear Pattern A-End

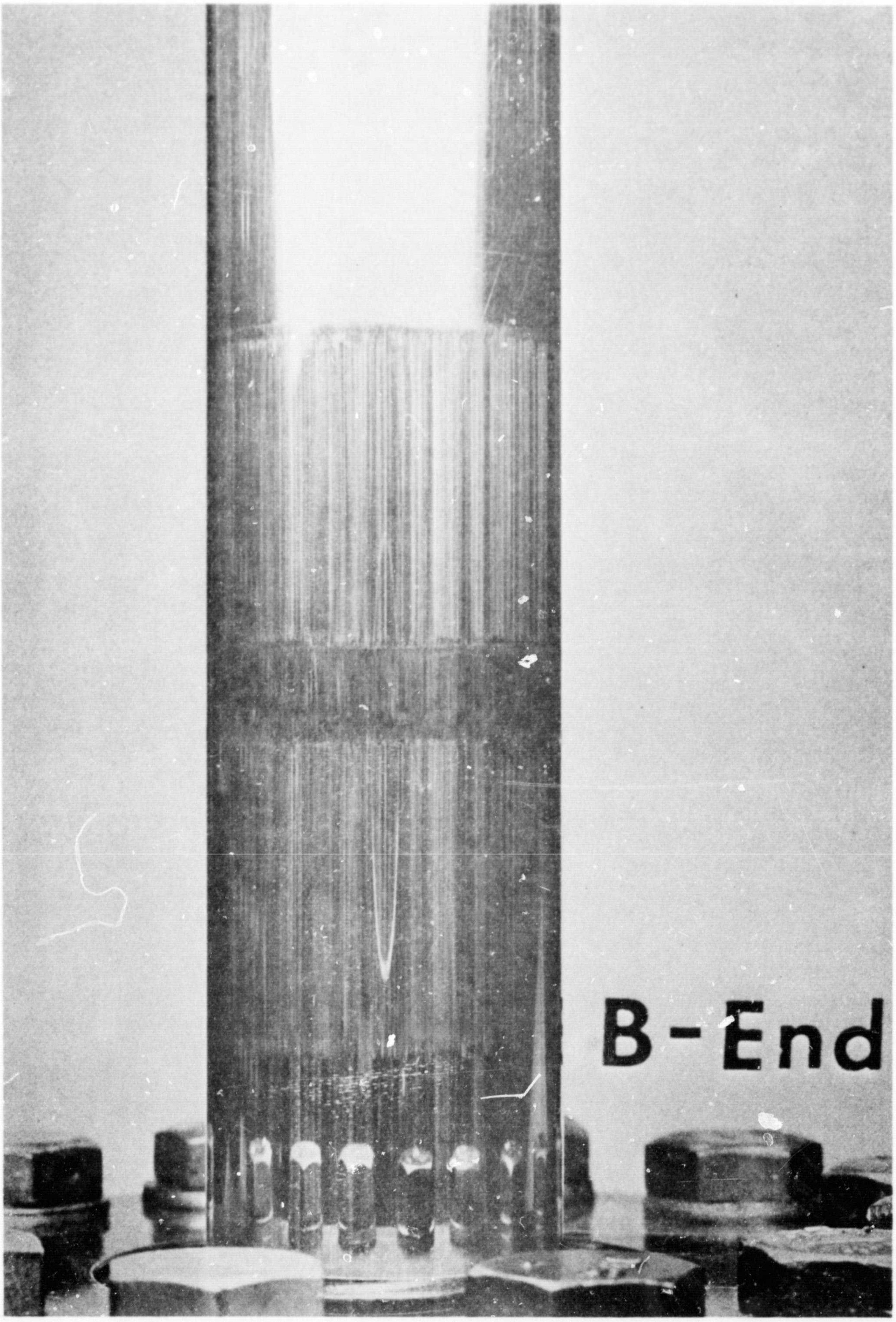


Figure 7-5. Short-Stroke Wear Pattern - B-End

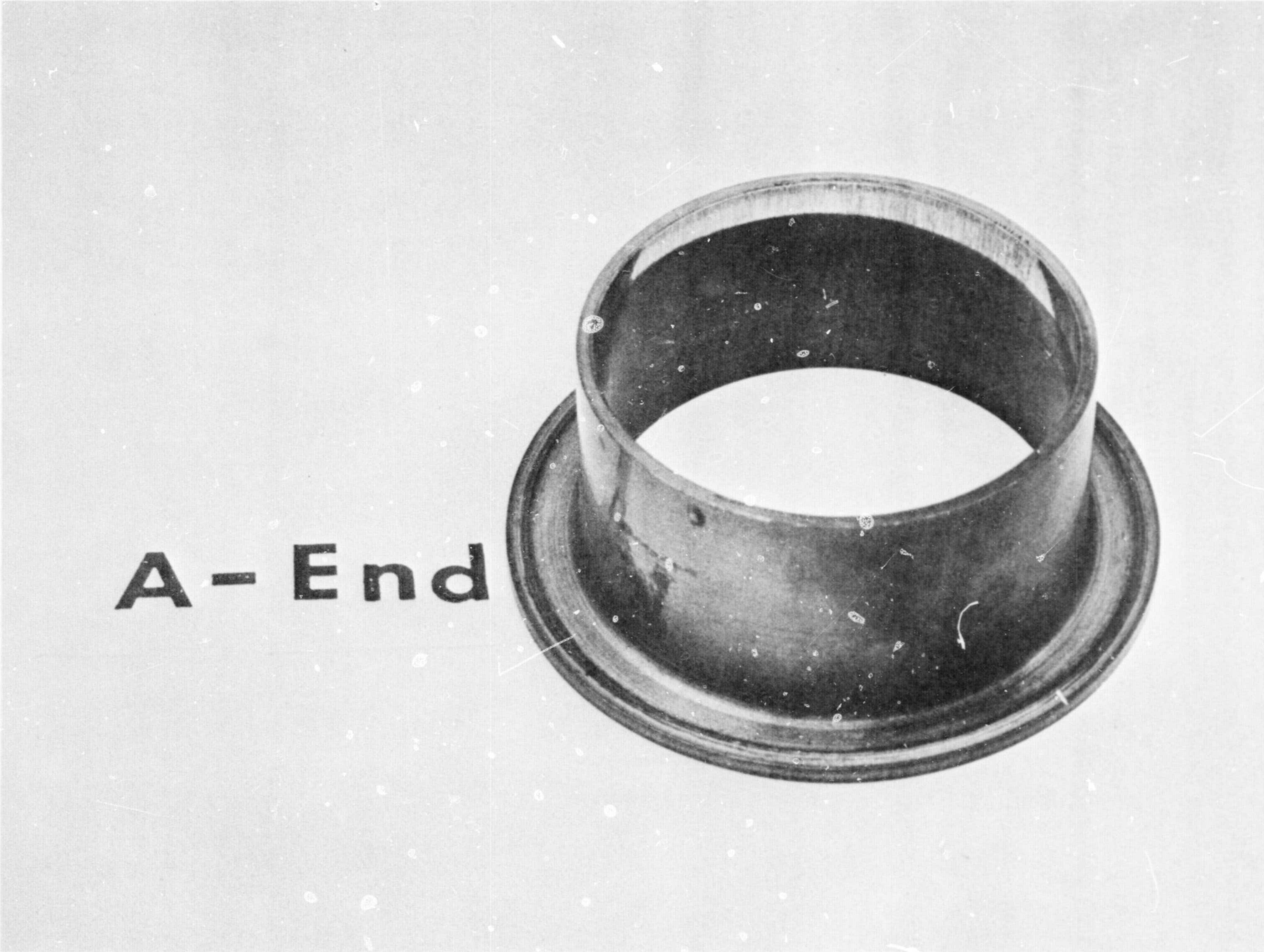


Figure 7-6. High Pressure Lip Seal - B-End

this would have on the leakage characteristics of the seal. It was expected that seal leakage would be low at the start of the short-stroke operation but would then increase as cycles are accumulated. However, a decrease in leakage was not discernable because of the other operating variables. For example, seal wear (Table 7-3) was accumulating which resulted in a decrease in seal contact load and progressive increases in leakage. The decrease in seal contact load was evidenced by the fact that the short-stroke wear pattern became less pronounced with each succeeding thermal cycle (see Figure 7-7).

The polyimide lip seal, which replaced the A-seal, exhibited exceptionally low leakage. Leakage accumulated during 15.25 hours of testing was 6cc.

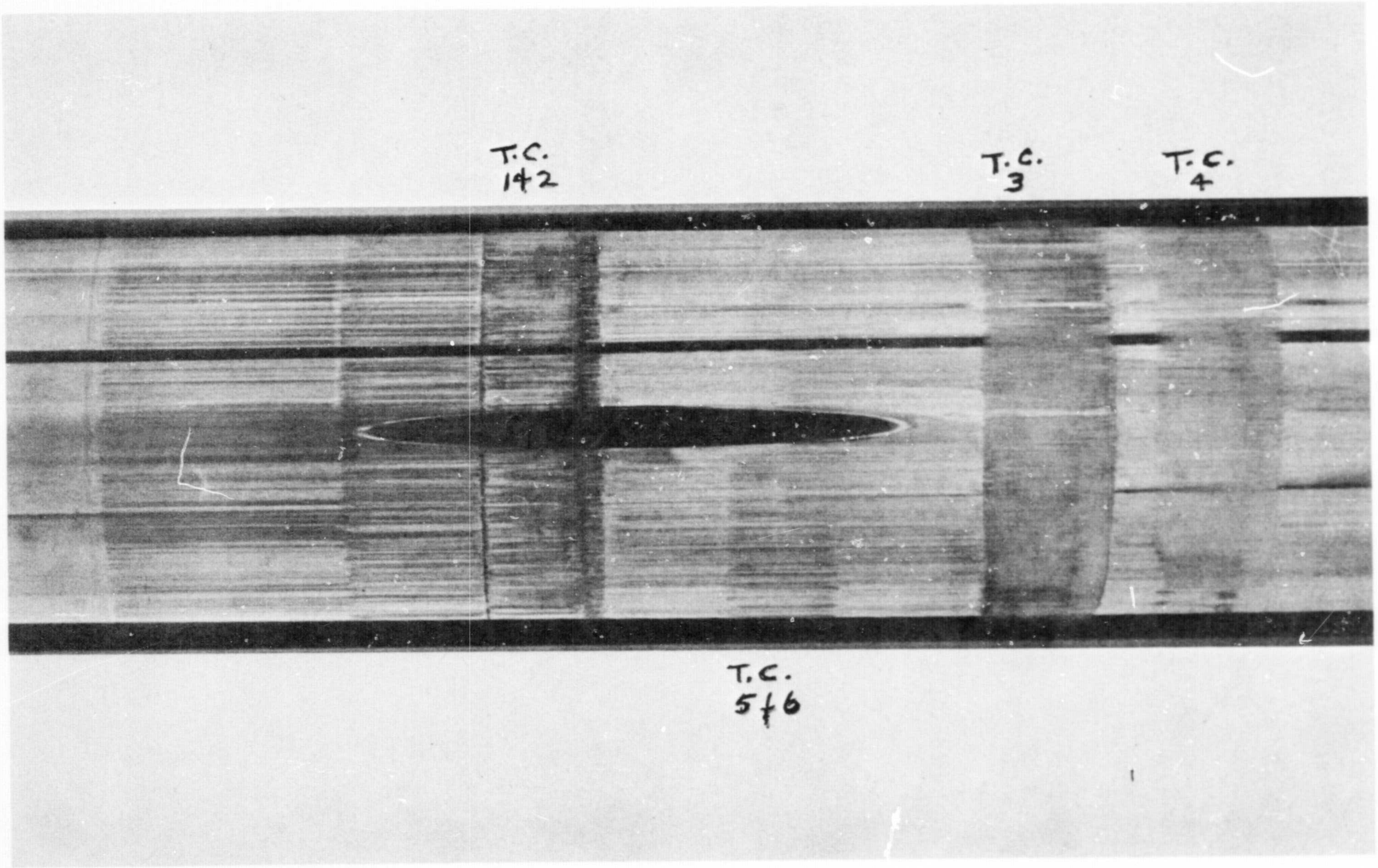


Figure 7-7. Short-Stroke Wear Pattern at Each Thermal Cycle

SECTION VIII
CONCLUSIONS

The areas of endeavor specified in the contract work statement (Appendix B), were accomplished. Substantial achievements were attained in the design and development of actuator rod seals for high temperature hydraulic system applications. Conclusions that can be drawn from these efforts are summarized in the following paragraphs.

1. The soundness of the two-stage seal concept was adequately demonstrated in the endurance testing. This approach appears to be the most practical for flight control actuators, which accumulate many millions of cycles and which can tolerate limited internal leakage. The leakage experienced by the polyimide first stage seals appears acceptable for most two-stage applications.

2. The use of polyimide split contracting rings for the high pressure first stage is completely feasible. However, careful design of the ring configuration is essential to prevent ring joint breakage due to the high notch sensitivity of the material.

3. Of the three low pressure seals developed and evaluated in the endurance test, the polyimide V-seal demonstrated the highest potential for meeting 3000 hours of operation. The material failure experienced in the testing is attributable to temperature-fluid effects and also to possible deficiencies in the seal design.

4. The Vascojet 1000/silver alloy reed seal exhibited extremely low wear. However, the Vascojet 1000 reeds tend to score the piston rod during rapid short-stroke cycling. The wear pattern generated formed a multiplicity of leakage paths for the fluid.

5. The cobalt molybdenum alloy lip seal also produced scoring of the chrome plated piston rod but to a lesser degree. Failure of the material during the seal fabrication process occurred quite frequently due possibly to defects and consequently its full potential was not adequately explored.

6. The feasibility of designing a seal with load relieving or pressure balancing features was demonstrated with the lip seal (Design D). The ability of a seal to operate effectively with minimum seal interface loading is highly desirable as it decreases the amount of heat generated at the interface and thus reduces wear.

7. The life of hard metal seals is dependent upon its duty cycle and stroke length. Long-stroke cycling appears to have little effect on the seal and its mating chrome-plated surface. Rapid short-stroke cycling is more detrimental to hard metal seals since they operate under more severe lubricating conditions. Consequently, metal to metal contact occurs with resulting high temperature and accelerated wear of the seal and rod surface.

8. Failure of the single-stage seal to attain 100 hours of operating life was due to failure of the chrome plating. The tendency for the fluid to leak at high pressures was difficult to overcome with imperfect contact surfaces.

SECTION IX

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APPENDIX A

EVALUATION OF SECOND-STAGE SEAL DESIGNS

A. INTRODUCTION

The basic criteria, method, and results of the evaluation of second-stage seal designs are presented in the following sections. The designs that were considered represent a spectrum of approaches. These approaches were found to fall into four basic configuration categories: wedges, lips, reeds, and specially shaped elements such as a V, C or X.

The state of the art of high temperature rod seal design is still primarily empirical. Until recently, little has been done to generate rational methods of analyses for the many conceivable combinations of shapes and materials. In addition, published data are limited largely to test results in terms of friction, leakage, and life. Therefore, the analysis of seal designs relies heavily on experience, simple approximations in calculations, and intuition.

An effort was made to optimize the various approaches, considering the materials available as well as the specific application in mind. However, the number of combinations and possible variations precluded extensive study of any one design. Rather, efforts were concentrated on the designs that reached the greatest degree of development and consequently offer the greatest potential for meeting the goals of this program. By applying various practical criteria, a manageable number of likely candidates have been selected that still cover a broad variety of approaches.

B. BASIC APPROACH

A positive-contact type of second-stage rod seal is required if leakage is to be kept to very small values (one drop per minute at failure). Experience shows that this goal is quite difficult to attain, particularly in the larger rod sizes. Furthermore, it must be met over a broad range of pressures, from 0 (during system shutdown) through nominal operating (return line) pressure, to surges of several times nominal pressure; and over a fluid temperature range of -40°F to +600°F.

An overall seal configuration is selected primarily to attain the desired loading of the sealing surface at the rod surface, and to accomplish other functions such as static sealing and alignment. The seal material's mechanical properties govern its ability to transmit an adequate sealing stress to the rod over the entire pressure and temperature ranges, and to continue to deliver this sealing stress after a reasonable amount of wear occurs. The seal's wear rate will then determine its useful life. The configuration must also overcome practical limitations such as adverse tolerance buildups, eccentricities and out-of-roundness of the rod. These practical problems in effect reduce the seal's theoretical ability to accommodate wear.

Allowable stress levels that are appropriate to the particular seal material provide certain design constraints on the configuration. These stresses will generally fall into one of the following categories:

1) Hoop stresses that limit the radial deflection available for wear compensation.

2) Combined stresses that limit the transfer of loading energy to the sealing surface.

3) Sealing (bearing) stresses that produce acceptable friction and wear rates.

The principal tradeoffs to be considered in selecting seals are described below:

1. Wear Rate Versus Wear Compensation. The need for the sealing surface to deflect radially is proportional to its wear rate. A fairly high wear rate, coupled with a considerable ability to deflect, might be preferable to a low wear rate and little ability to deflect. Compensation for wear is obtained if the material is elastic and if the loading method does not induce excessive local stresses in some other part of the seal element.

2. Leakage Versus Friction and Wear. Bearing stresses at the seal-rod interface that will produce a low-leakage seal will vary with different materials. The configuration must attain a satisfactory stress level throughout wide ranges of temperature and pressure without causing excessive friction and wear. Any seal design is a compromise among many possible values of stress that can be experienced over the attainable environmental range, and both before and after a change in cross-section due to wear occurs.

3. Reliability Versus Performance. Redundant elements, conservative allowable stresses, and special provisions to support the seal during pressure surges are desirable features from the standpoint of reliability. The effect of such features must be balanced against any adverse effect they may have on friction, wear or compensating ability. Furthermore, care must be taken that an inherent weakness that can result in abrupt failure is not built into the design. Any penalty resulting from a necessary compromise in design will be made in such a manner so as to preclude any built in catastrophic failure potentials. However, by compromising one feature to improve another, it is possible that disadvantages such as reduced service life, or increased allowable leakage or friction may occur.

4. Performance at Nominal Versus Design Conditions. The seal's specific duty cycle may dictate additional compromises in its design. For example, in flight control actuators, friction is frequently a dominant factor. This may lead to a deliberate reduction in initial loading and, consequently, a foreshortened life due to the early occurrence of high leakage rates. Utility actuators can tolerate higher friction (assuming this does not result in excessive wear), but usually experience a relatively low number of cycles during the vehicle life. The seals for these two applications may thus differ considerably from one another in design.

In summary, the seal configuration must exploit the material's strong points while offsetting its weak points to an acceptable degree. Any gross mismatch among the various design factors will render the seal ineffective.

C. SEAL RATING SYSTEM

The foregoing considerations were used as the basis for setting up the seal rating system. This system assists in making a preliminary selection of designs that warrant further development effort. Candidates are rated on certain factors deemed essential to attaining program objectives. Several candidates are thereby eliminated from further consideration because of an unacceptable quality or because of low overall score. The remaining designs are then rated on the basis of other factors, considered desirable but not absolutely essential, to aid in the final selection.

This system does not assure the identification of a distinct order of merit among the various candidates. Assigned values for many of the factors are estimated and, consequently, are subject to argument. However, the rating does identify the least promising candidates and reduces the final group to a manageable size. Final selection must to some extent depend on intuition for the most practical approaches.

Scoring of factors is limited to a three-digit spread (0-1-2). The lack of quantitative data precludes precise scoring; thus, more refined gradations are considered impractical at this time. The standard of grading is relative to the group average. The evaluation covers all elements of the installation (such as static and dynamic seals, bearings and loading devices) that affect actuator design.

D. ESSENTIAL FACTORS

1. Sealing Mechanism. The seal conforms to the rod by means of a feasible loading technique. Appropriate stress levels and patterns are generated in the seal element throughout the temperature and pressure range. Geometry and behavior of elements can be predicted and controlled within workable limits. Effective static sealing is available. Pressure does not increase leakage.

2. Wear and Compensation. The seal exhibits either a low wear rate or the ability to compensate for wear, and wears rod surface at a low rate. The wear process is not detrimental to overall performance of seal installation (i.e. no adverse effect on static sealing). Pressure does not increase wear.

3. Reliability. Seal failure is slow and detectable. Seal construction is rugged and inherently resistant to abuse in use. Redundant sealing elements available.

Precise definition of operating conditions and behavior is not critical; possesses inherent latitude to withstand pressure surges.

4. Short-term Potential. Knowledge of concept is available or can be acquired in the near future. Configuration can be optimized and evaluated within time period of program.

E. DESIRABLE FACTORS

1. Design Features. Materials exhibit compatible coefficients of expansion. Rod does not require exotic plating. Friction is relatively low (for good servo performance). The seal has a relatively high degree of self-compensation for wear. Geometry lends to pressure balancing or pressure relief.

2. Cost. Relatively easy to manufacture (reasonable tolerance requirements, accessible geometry for machining). Materials are available and machineable.

3. Serviceability. Easy to install and remove. Does not require custom tailoring; no special installation tools required.

4. Past experience with similar approach.

F. SEAL RATINGS

Rating work sheets for each seal design are attached (pages A-10 to A-28). Final overall scores are summarized in Table A-1. In making the final selection, a minimum overall score of 10 was established for acceptance. Five seal configurations and one alternate were selected on this basis. The seal-material combinations recommended for further development are shown in Table A-2. Since certain seal designs were applicable to more than one material, alternate combinations are also included.

The final recommendations represent a broad coverage of the various available seal concepts and materials. It is possible that other seal-material combinations may also offer considerable potential. However, it would not be feasible to evaluate all possible combinations within the scope of the program.

The final selection of configurations was influenced by past and recent experience, and are considered to be representative of the best features of the many designs that were reviewed.

The recommended seal designs and their appropriate materials are provided below.

1. Design B: V-seal/Polymer SP

This design consists of three V-shaped sealing elements, a load ring, back-up ring, and loading springs. The latter provides the loading force to effect a seal and also acts as a wear compensating device. Polymer SP V-seals have undergone substantial development and evaluation in previous seal programs. Results have been quite satisfactory at temperatures up to 600°F. Low friction and low wear characteristics of the SP material offer good potential for long-life operation. Nickel Foametal impregnated with CaF_2 + BaF_2 is suggested as an alternate material with this design.

2. Design D: Lip Seal/Vascojet 1000

3. Design D: Lip Seal/Cobalt-Molybdenum Alloy

This seal (2 & 3 above) utilizes an interference fit over the rod to effect a seal. The stresses induced by the interference provide compensation for wear. This design also provides a contact width of approximately 0.030 to 0.040 inch on assembly. This enables the sealing force to be distributed over a relatively large area, which results in lower contact stresses. By having the sealing lip facing away from the fluid, excessive buildup of contact stresses due to fluid pressure is avoided. This arrangement also permits relieving of the contact load at the seal interface since fluid pressure will tend to open the inner diameter of the seal. Such a feature is advantageous in reducing friction and wear. Recent test results (Ref. Progress Reports No. 7 and 8 for the present contract) indicate the feasibility of this design.

4. Design F: Spring-Loaded Lip Seal/Silver Alloy Material

This design consists of a truncated-cone. Independent loading of the static and dynamic portions of the seal is accomplished by using separate adjusting nuts. Finer load adjustments are thus maintained. Spring loading of the sealing lip provides compensation for wear. Preliminary testing of this design using silver alloy as the seal material has indicated the feasibility of this loading arrangement. Relatively light loading was required to effect sealing with a .025-inch thick seal. Seal breakout friction at 0 psi was approximately 40 pounds. Alternate materials recommended for this configuration are nickel Foametal impregnated with $CaF_2 + BaF_2$, and Polymer SP.

5. <u>Design I: Wedge Seal/Nickel Foametal Impregnated with $CaF_2 + BaF_2$</u>

This design consists of a wedge shaped (45°) sealing element and loading springs. The spring load provides the axial force to effect a static seal at the tapered seat. The radial force component provides the load at the seal-rod interface. The spring washers also provides the means for wear compensation. Testing of this configuration was conducted in previous high temperature seal programs (References 9 and 10) at temperatures of 800°F and 4000 psi and achieved promising results. Alternate material recommended for this design is the silver alloy (72%Ag+ 28% Cu).

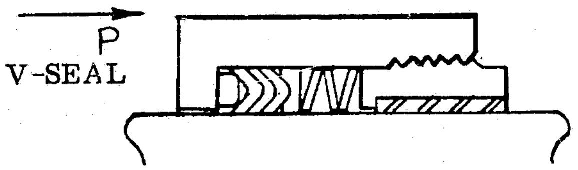
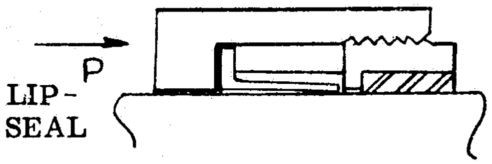
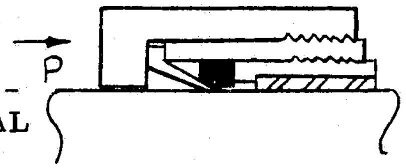
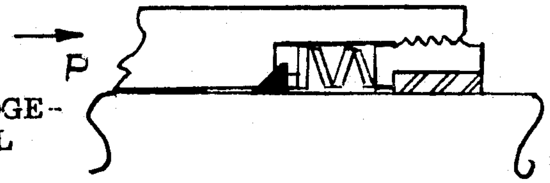
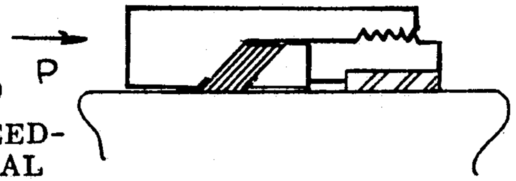
6. <u>(Alternate Design), Design AH: Reed Seal/Vascojet 1000 and Silver Alloy (72%Ag + 28% Cu)</u>

This configuration is recommended as an alternate to be used with a combination hard and soft metal. The seal consists of alternate elements of hard and soft metal. The purpose of the soft metal is to provide added conformability. The sealing elements are approximately .005 to .007 inch thick. Wear compensation is inherent due to the interference fit and the action of fluid pressure. An alternate combination recommended for this design consists of cobalt-molybdenum alloy and silver alloy (providing the cobalt material can be obtained in sheet form). Polymer SP is also recommended. However, the latter would be formed with a steeper angle, because of the limited ability of the SP sheet material to be formed to a low angle.

TABLE A-1. SUMMARY OF OVERALL SCORE

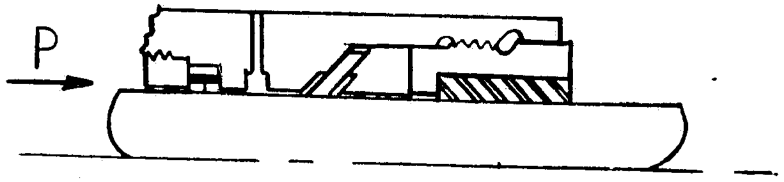
Materials	Plastics			Soft Metal							Hard Metal								
Seal Config.	A	B	AH	B	C	F	G	I	IR	L	B	D	E,H	F	J	K	M	N	O
Essential Factors																			
1	2	2	2	2	1	2	1	2	1	2	2	2	2	2	1	1	2	1	1
2	2	2	2	2	1	1	1	2	2	1	1	2	1	1	1	1	1	2	2
3	1	2	2	2	1	1	1	1	2	1	1	1	2	1	2	1	1	1	1
4	2	2	2	2	2	2	1	1	1	2	2	2	2	1	1	1	1	1	1
Subtotal	7	8	8	8	5	6	4	6	6	6	6	7	7	5	5	4	5	5	5
Desirable Factors																			
1	1	1	1	1	1	1	-	1	1	1	1	2	1	1	1	-	1	2	2
2	2	1	2	1	1	1	-	1	1	1	0	1	2	1	1	-	0	1	1
3	1	1	1	1	1	1	-	2	2	1	0	2	1	1	1	-	1	1	1
4	1	2	1	2	0	1	-	2	2	1	2	2	2	1	1	-	2	1	1
Subtotal	5	5	5	5	3	4	-	6	6	4	3	7	6	4	4	-	4	5	5
Total Score	12	13	13	13	8	10	4	12	12	10	9	14	13	9	9	4	9	10	10

TABLE A-2. RECOMMENDED SEAL DESIGNS AND MATERIALS

SEAL DESIGN	SEAL TYPE	RECOMMENDED MATERIALS	ALTERNATE MATERIALS
B	V-SEAL 	Polymer SP	Nickel oametal with Ca F_2 + Ba F_2
D	LIP-SEAL 	Vascojet-1000	
D	SAME	Cobalt-molybdenum alloy (75% Co + 25% Mo)	
F	LIP-SEAL 	Silver alloy (72% Ag + 28% Cu)	Nickel Foametal with Ca F_2 + Ba F_2, Polymer SP
I	WEDGE-SEAL 	Nickel Foametal with Ca F_2 + Ba F_2	Silver alloy (72% Ag + 28% Cu)
(ALTERNATE) H or AH	REED-SEAL 	Vascojet - 1000 and silver alloy (72% Ag + 28% Cu) combination	Cobalt-molybdenum alloy and silver alloy (72% Ag + 28% Cu) combination, Polymer SP

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **A** - **REED SEAL - PLASTIC (POLYMER SP)**



DATE 12/65

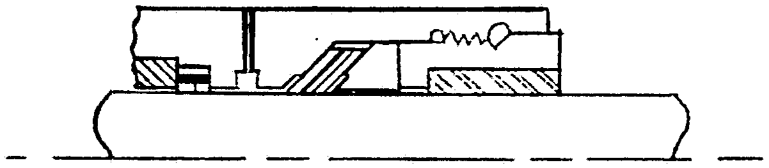
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Pressure energizing reduces leakage. Static seal adjustment independent of dynamic seal. Behavior of Polymer SP predictable from experience in other configurations.	No dynamic load adjustment.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	Use of SP sheet material permits "stretch" compensation. Compensationability recently demonstrated.	Pressure energizing may increase wear rate.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Capable of withstanding pressure surges. Material relatively resistant to foreign objects. Short term integrity of material well known.	No redundant elements. Long-term aging effects uncertain. High shear stresses may occur at edge of clamped portion.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Previous experience with material. Variables relatively easy to adjust and to evaluate	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Self-compensating. Hard chrome plate rod.	Coefficients of expansion mis-matched. Friction increases at higher pressure.
2. COST Easy to manufacture, machinable.	2	Simple - manufacture from sheet stock. No double concentricity problem - self centering.	Polymer SP somewhat difficult to hold to close tolerances.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Omission of spacer will show up in pressure check. Interchangeable. Easy to install.	May require torque wrench to avoid overloading static lips. Requires care in "stretching" over rod and presetting of lips.
4. EXPERIENCE Past exp. with sim. approach.	1	Experience with SP V-seal)Design B) somewhat applicable. Limited experience with configuration A.	

COMMENTS: Rising friction vs. pressure characteristic - best suited for 2-stage flight control services or utility services. Consider multiple elements (see AH). Sharp incidence angles unobtainable with sheet material - use of solid stock will sacrifice some flexibility.

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **AH - REED SEAL - PLASTIC (POLYMER SP)**



DATE 12/65

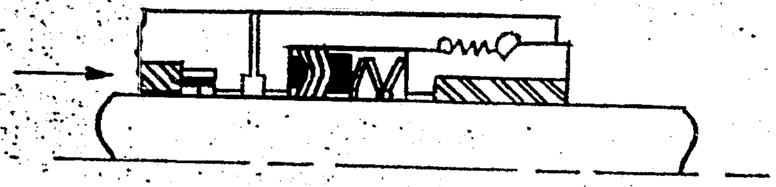
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	2	Pressure energizing reduces leakage. Static seal adjustment independent of dynamic seal. Behavior of Polymer SP predictable from experience in other configurations.	No dynamic load adjustment. Several possible static leakage paths, but material tends to conform well.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	2	Use of SP sheet material permits "stretch" compensation. Elements may transfer sealing function downstream as wear occurs, providing some increase in compensation life.	Pressure energizing may increase wear rate of upstream lips.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	2	Capable of withstanding pressure surges. Redundant elements. Material relatively resistant to foreign objects. Short term integrity of material is well known.	Long term aging effects uncertain.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Previous experience with material. Variables relatively easy to adjust and to evaluate.	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Self compensating. Hard chrome plate rod.	Coefficients of expansion mismatched. Friction increases somewhat at higher pressure.
2. COST Easy to manufacture, machinable.	2	Simple-manufacture from sheet stock. No double concentricity problem-self centering.	Polymer SP somewhat difficult to hold to close tolerances.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Omission of spacer will show up in pressure checks. Interchangeable. Easy to install.	May require torque wrench to avoid overloading static lips. Some seal elements may be omitted without showing up in pressure checks. Requires presetting of lips.
4. EXPERIENCE Past exp. with sim. approach.	1	Experience with SP V-seal (Design B) somewhat applicable. Limited experience with configuration A.	

COMMENTS: **Sharp incidence angles unobtainable with sheet stock - use of solid stock will sacrifice some flexibility. If friction vs. pressure characteristic can be controlled, this configuration is suited for single stage service.**

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **B - V-SEAL - PLASTIC (POLYMER SP)**



DATE 12/65

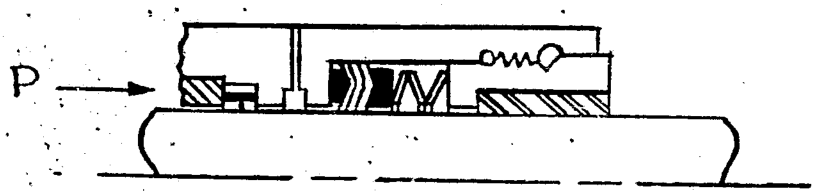
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	2	Proven loading technique and static sealing. Behavior predictable from experience. Pressure energizing reduces high pressure leakage. Loading adjustable.	Static and dynamic loading are inter-related.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	2	Demonstrated ability to compensate. Slight "stretch" fit can be employed. Elements transfer sealing function downstream as wear occurs, providing some increase in compensation life.	Pressure energizing may increase wear rate of upstream lips.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	2	Redundant elements. Material relatively resistant to foreign objects. Short term behavior of material is fairly well known. Considerable cyclic life has been obtained with this configuration. Capable of withstanding surges.	Long term aging effects uncertain.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Much data available - Configuration has been refined. Design and manufacturing techniques have been established.	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	May be capable of some self-compensation if "stretch" is employed. Hard chrome plate rod.	Coefficients of expansion mismatched. Friction increases somewhat at higher pressures.
2. COST Easy to manufacture, machinable.	1	Manufacturing techniques well established	Complex. Polymer SP somewhat difficult to hold to close tolerances. Must be made from solid stock. Close control of concentricities required.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Interchangeable. Reverse installation of seal elements not likely, and will show up in pressure check.	Omission of spacer or spring may not show up in pressure check. Can be installed in wrong sequence.
4. EXPERIENCE Past exp. with sim. approach.	2	Extensive experience.	

COMMENTS: **If friction vs. pressure characteristic can be controlled, this configuration is suited for single stage service.**

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **B - V-SEAL - SOFT METAL**



DATE 12/65

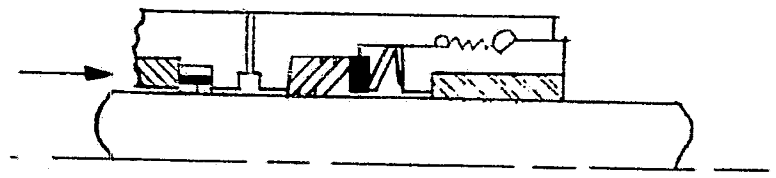
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Proven loading technique and static sealing. Behavior predictable from experience. Pressure energizing reduces high pressure leakage. Loading adjustable.	Static and dynamic loading are interrelated. Taper fairly critical in providing proper seat.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	Compensation demonstrated. Limited downstream	Pressure energizing increases wear rate of upstream lips.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	Redundant elements. Short term behavior well known. Capable of withstanding pressure surges.	
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Much data available - configuration has been refined. Design and manufacturing techniques have been established.	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion. Hard chrome plate rod.	Friction increases at higher pressure. Limited "stretch" of soft metals precludes any degree of self compensation.
2. COST Easy to manufacture, machinable.	1	Manufacturing techniques well established.	Complex. Close control of concentricities required.
3 SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Easy to install. Reverse installation of seal elements not likely, and will show up in pressure check.	Omission of spacer or spring may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2	Extensive experience.	

COMMENTS: **Rising friction vs. pressure characteristic - best suited for 2-stage seal applications or utility services.**

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **C** - **BEVELLED RING SEAL - SOFT METAL**



DATE 12/65

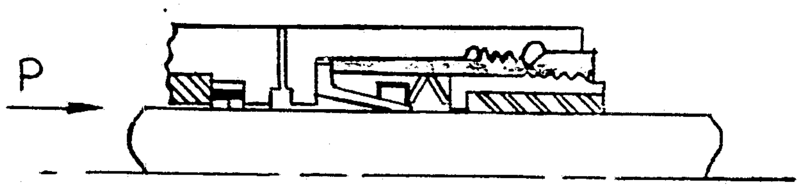
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	1	See comment at bottom.	Behavior uncertain and difficult to predict. Effectiveness of static seal and load adjustment questionable.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	If inner edge seals statically, very small area is exposed to pressure.	Limited compensation because of geometric stability of design.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements: resists pressure surges.	1	Multiple seal elements.	Principal is obscure and redundancy uncertain.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Relatively simple to evaluate	No data available.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion. Low friction - small pressure - energizing effect. Probably can use hard chrome plate.	Limited "stretch" in soft metals precludes any degree of self-compensation.
2. COST Easy to manufacture, machinable.	1	Relatively easy to manufacture	Close concentricities required. Requires lapping in.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Easy to install. No special tools required.	Backwards installation may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	0	None	

COMMENTS: A similar approach was once patented in England for application to high pressure air compressors, with good results claimed.

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN F - LIP SEAL - SOFT METAL



DATE 12/65

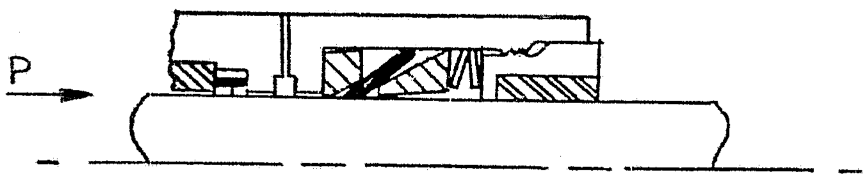
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	2	Static seal independent of dynamic seal. Dynamic load adjustable. Loading demonstrated in limited testing.	Stress levels at sealing lip somewhat difficult to predict.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	Wear appears to be low in limited testing (35000 cycles). Some compensation available.	Relationship of variables difficult to assess.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	1	Load ring may protect against catastrophic failure under pressure surges.	No redundant elements. May not be capable of withstanding pressure surges without permanent set and increase in leakage.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Manufacturing techniques established. Limited testing has indicated feasibility of concept.	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion.	Limited "stretch" in soft metals preclude any degree of self compensation.
2. COST Easy to manufacture, machinable.	1	All parts readily machinable.	Close tolerance requirement for proper load ring action.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Unlikely to be installed backwards - will show up in pressure check.	Missing loading elements may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1	Experience with V-seals somewhat applicable. Limited testing conducted on lip seal.	

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN G - DOUBLE WEDGE SEAL - SOFT METAL



DATE 12/65

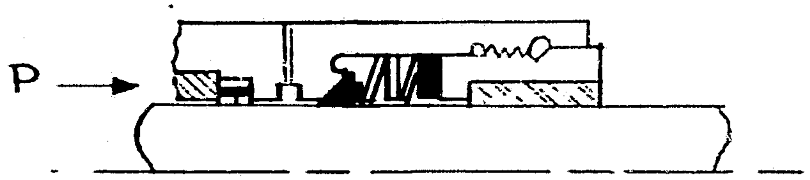
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	1	Adjustable.	Behavior difficult to predict.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1		Ability to compensate not apparent.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements: resists pressure surges.	1	Rugged seal element - not sensitive to damage by foreign object.	No redundant elements.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1		Extensive development required. No data available.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.			
2. COST Easy to manufacture, machinable.			
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.			
4. EXPERIENCE Past exp. with sim. approach.			

COMMENTS

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **I - WEDGE SEAL - SOFT METAL OR GRAPHITE**



DATE 12/65

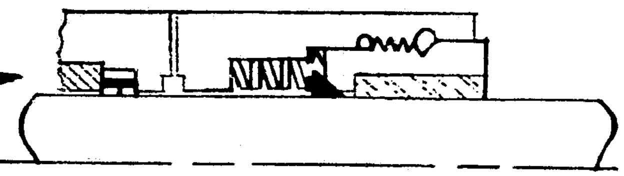
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded: predictable: controllable: good static sealing: pressure decreases leakage.	2	Demonstrated loading technique with good stress levels. Behavior can be predicted with reasonable accuracy. Load adjustable.	Static and dynamic loading interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate: wear not detrimental to performance: pressure decreases wear.	2	Demonstrated degree of compensation. Very small area exposed to pressure. May be capable of pressure balancing at some increase in exposed area.	
3. RELIABILITY Fails slowly and observably: rugged, resists abuse; redundant elements: resists pressure surges.	1	Seal element is fairly rugged and well supported. Not likely to fail catastrophically.	No redundant element. Preload necessary to withstand unseating by pressure surges plus rod friction may not be compatible with desired low-pressure friction characteristic.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	...ne test data available. Relatively easy to evaluate.	Experience shows that larger rod sizes (above 1") have excessive friction.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available: self compensating.	1	·ly well-matched coefficients of expansion. ·tion level fairly constant due to small effect of ·sure. Hard chrome plate rod.	Not self-compensating due to low "stretch" available.
2. COST Easy to manufacture, machinable.	1	Fairly easy to manufacture. Self-centering.	Diametral tolerances fairly critical. Excessive bearing clearances may cause poor seating.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	2	Easy to install. Improper installation will probably show up in pressure check.	Requires run-in, particularly with soft metal.
4. EXPERIENCE Past exp. with sim. approach.	2	E· rience documented.	

COMMENTS: Graphite exhibits greater compressibility (higher $\frac{S}{E}$) than silver.

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **IR - REVERSE WEDGE SEAL - SOFT METAL OR GRAPHITE**



DATE 12/65

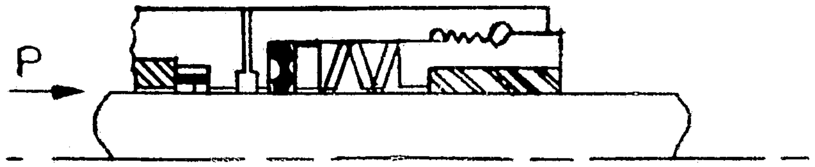
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	1	**Loading technique demonstrated to some extent in Design I. Behavior reasonably predictable. Pressure has slight tendency to decrease leakage.**	**Requires shimming for load adjustment. Static and dynamic loading interrelated. Requires extra static seal. Taper fairly critical in providing proper seat.**
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	**Design I has demonstrated a degree of compensation. Relatively small unbalanced area acted on by pressure. Capable of being balanced to considerable degree.**	**Pressure has slight tendency to increase wear.**
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	2	**Seal element fairly rugged and well supported. Not likely to fail catastrophically. Capable of withstanding pressure surges.**	**No redundant element.**
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	**Limited data available from Design I. Relatively easy to evaluate.**	**May exhibit same size limitation as Design I.**
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	**Fairly well-matched coefficients of expansion. Friction level fairly constant due to small unbalanced area. Hard chrome plate rod.**	**Not self-compensating due to low "stretch" available.**
2. COST Easy to manufacture, machinable.	1	**Fairly easy to manufacture. Self-centering**	**Diametral tolerances fairly critical. Excessive bearing clearances may cause poor seating, but bearing is piloted on same piece as seat.**
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	2	**Easy to install. Improper installation will probably show up in pressure check.**	**Requires run-in, particularly with soft metal.**
4. EXPERIENCE Past exp. with sim. approach.	2	**Some experience with Design I applicable.**	

COMMENTS: **Graphite exhibits greater compressibility (higher S/E) than silver.**

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN L - "C" SEAL - SOFT METAL



DATE 12/65

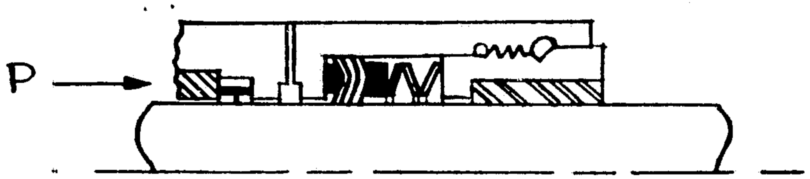
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded: predictable: controllable: good static sealing: pressure decreases leakage.	2	Reasonably predictable behavior. Pressure energizing tends to reduce leakage at high pressure. Load adjustable.	Geometry limits loading ability; static and dynamic loading interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	Some degree of compensation demonstrated in high pressure application.	Pressure energizing tends to increase wear at high pressure. Stable geometry may limit compensation.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	1	Capable of withstanding pressure surges.	No redundant element.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Some test data available. Relatively easy to evaluate.	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Fairly well-matched coefficients of expansion.	Friction increases at higher pressure. Not self-compensating because of limited "stretch" of soft metal.
2. COST Easy to manufacture, machinable.	1	Fairly simple configuration.	Double concentricities must be held closely.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Easy to install. Omission of spring or spacer will probably show up in pressure check.	Reverse installation may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1	Some experience.	

COMMENTS: Optimum low pressure characteristic may not be suitable for high pressure application.

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN B - **V-SEAL - HARD METAL**



DATE 12/65

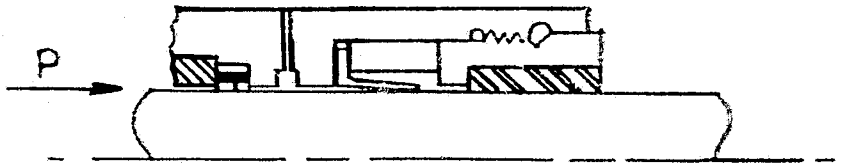
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	**Loading technique and static sealing proven with other materials. Behavior reasonably predictable. Pressure energizing reduces high pressure leakage. Load adjustable.**	**Static and dynamic loading are interrelated.**
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1	Compensation with "stretch" fit. Downstream transfer of sealing function may occur as wear progresses.	**Wear rate may be high for thin lip and high bearing load. Pressure energizing may increase wear rate.**
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Redundant elements. Capable of withstanding pressure surges.	**Sealing lips susceptible to damage by foreign objects.**
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	**Previous experience with configuration generally applicable.**	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	**Well matched coefficient of expansion. Self-compensating due to "stretch".**	**May require flame plated rod. Friction may be high because of high bearing loads and pressure energizing.**
2. COST Easy to manufacture, machinable.	0		**V-shape fairly expensive to machine in thin sections. Complex. Close control of concentricities required.**
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	0	**Reverse installation of seal elements not likely and will show up in pressure check.**	**Tends to jam in cavity due to high bearing loads and permanent set. Requires care in "stretching" over rod. Omission of element may not show up in pressure check. Requires run-in.**
4. EXPERIENCE Past exp. with sim. approach.	2	**Extensive experience with V-seals.**	

COMMENTS: **Rising friction vs. pressure characteristic.**

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **D - LIP SEAL - HARD METAL**



DATE 12/65

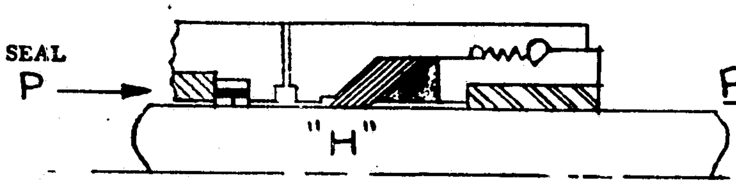
REV

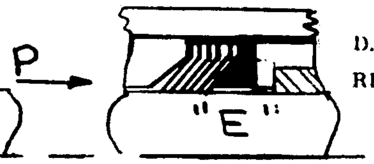
ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Demonstrated loading technique and static sealing. Reasonably predictable behavior. Static seal independent of dynamic seal.	Slight leakage inherent in pressure relief behavior. Load not adjustable.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	2	Low wear rate - no thin wear section and pressure relief Wear probably gradual. Compensation inherent due to "stretch".	Increased leakage with wear due to lower contact pressure (increased contact area).
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	See above. Not overly sensitive to damage by foreign objects.	No redundant elements. May not be capable of withstanding pressure surges without backup.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	Configuration previously tested for short time. Relatively easy to evaluate.	
DESIRABLE FACTORS			
1. DESIGN FEATURES -Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	2	Well-matched coefficient of expansion. Self-compensating due to "stretch". Hard chrome plate rod. Low friction due to pressure relief.	May require flame plated rod for best wear life.
2. COST Easy to manufacture, machinable.	1	Simple, fairly easy to manufacture. No double concentricity problem - self-centering. Rough drawing may be feasible in production.	Close tolerance on wall. Taper may be critical.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	2	Difficult to install improperly. Taper fit eases problem of "stretching" over rod. Improper installation will probably show up in pressure check.	Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2	See (4) above.	

COMMENTS: **May be suitable for first stage seal, or single stage seal if friction is not critical.**

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **E & H - REED SEAL - HARD METAL**





DATE 12/65

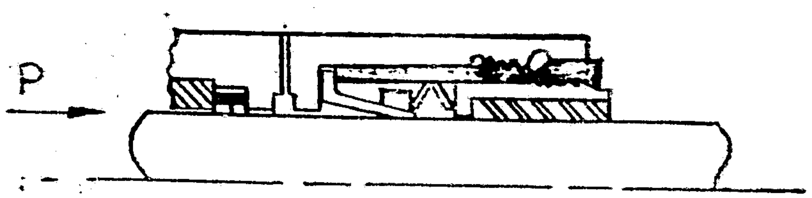
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded: predictable: controllable: good static sealing: pressure decreases leakage.	2	**Pressure energizing reduces high pressure leakage. Approach has been demonstrated in other programs. Static and dynamic adjustments independent.**	**Several possible static leakage paths, but high clamping forces are available. No dynamic load adjustment.**
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	**Compensation fair with "stretch" fit. Elements transfer sealing function downstream as wear occurs, providing some increase in compensation life.**	**Wear rate of individual lips may be high for thin lip and high bearing loads. Pressure energizing may increase wear rate of upstream lips.**
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	2	**Capable of withstanding pressure surges. Redundant elements.**	**Sealing lips may be susceptible to damage by foreign objects.**
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	2	**Previous experience with configuration documented.**	
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	**Well-matched coefficient of expansion. Self-compensating due to "stretch".**	**May require flame plated rod. Friction may be high because of high bearing loads and pressure energizing.**
2. COST Easy to manufacture, machinable.	2	**Fairly simple and easy to manufacture. No double concentricity problem - self centering. Rough drawing may be feasible in production.**	
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	**Fairly easy to install thin lips relatively eady to "stretch" over rod.**	**Some seal elements may be omitted without showing up in pressure check. Requires run-in. May require torque wrench to avoid overloading static lips.**
4. EXPERIENCE Past exp. with sim. approach.	2	**Previous experience with configuration.**	

COMMENTS: **Rising friction vs. pressure characteristic - best suited for 2-stage flight control services or utility services.**

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN F - LIP SEAL - HARD METAL



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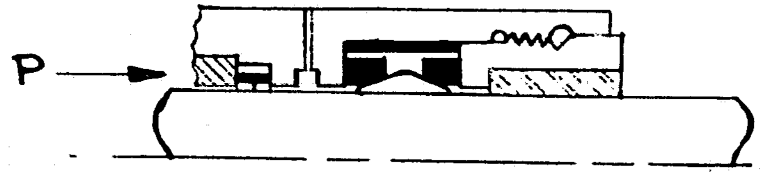
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	2	Static sealing inherent. Individual adjustment of static and dynamic seals. Load adjustable.	Stress levels at sealing lip somewhat difficult to predict.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	Compensation may be inherent due to "stretch".	Relationship of bearing loads to wear rate, friction and leakage uncertain.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse; redundant elements: resists pressure surges.	1	Load ring may protect against catastrophic failure under pressure surges.	No redundant elements. Lip somewhat sensitive to damage by foreign objects.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	Previous experience with design "D".	Fairly difficult to control parameters, particularly thin lip of seal.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Well-matched coefficient of expansion. Self-compensating due to "stretch". Some pressure relief may be available to reduce friction.	May require flame plated rod.
2. COST Easy to manufacture, machinable.	1	All parts readily machineable.	Close tolerance-requirements on thin lips.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Unlikely to be installed backwards - likely to show up in pressure check.	Missing loading elements may not show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	1	Previous experience with somewhat similar configuration.	

COMMENTS

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **J** - **BOEING RING SPRING SEAL - HARD METAL**



DATE 12/65

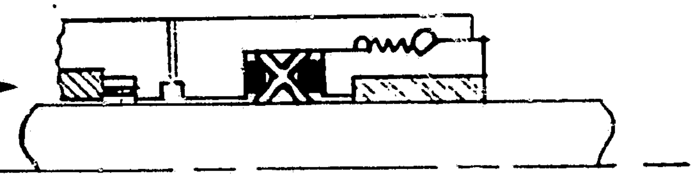
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	1	Demonstrated initial loading ability.	Load difficult to maintain. No positive static seal - pressure may relax spring load, permitting increased leakage.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	Wear likely to be uniform.	High forces required for compensation, and limited by geometry or ring spring.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse; redundant elements: resists pressure surges.	2	Resistant to pressure surges. Rugged element. Seal element resists damage by foreign objects.	No redundant elements.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1		Extensive development required. Proprietary to Boeing - may be impractical to develop within program scope.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Well matched coefficient of expansion	Requires flame plated rod. Friction may be high because of high bearing load.
2. COST Easy to manufacture, machinable.	1	Cannot be installed backwards. Missing elements will show	Fairly expensive because of close tolerance requirements on all three members.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Cannot be installed backwards. Missing elements will show up in pressure check.	Requires run-in
4. EXPERIENCE Past exp. with sim. approach.	1	Previous experience with seal in 1/2-in. rod size.	

COMMENTS

SEAL RATING - LOW PRESSURE SECOND-STAGE SEAL

DESIGN K - X-Seal - Hard Metal



DATE 12/65

REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable; good static sealing: pressure decreases leakage.	1	Static sealing good. Some load adjustment inherent.	Experience has shown this design to be very difficult to control as a dynamic seal. Static and dynamic loading interrelated.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	1	"Stretch" fit may augment loading for compensation.	See above - high wear rate can occur if precise control of variables is not attained.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	1	Resists pressure surges.	See above. Foreign objects may damage rod or lips. Difficult to obtain uniforms loading necessary for redundancy.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1		Proprietary item requiring extensive development in conjunction with vendor.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.			
2. COST Easy to manufacture, machinable.			
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.			
4. EXPERIENCE Past exp. with sim. approach.			

COMMENTS

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

DESIGN **M - BFG SEAL - HARD METAL**

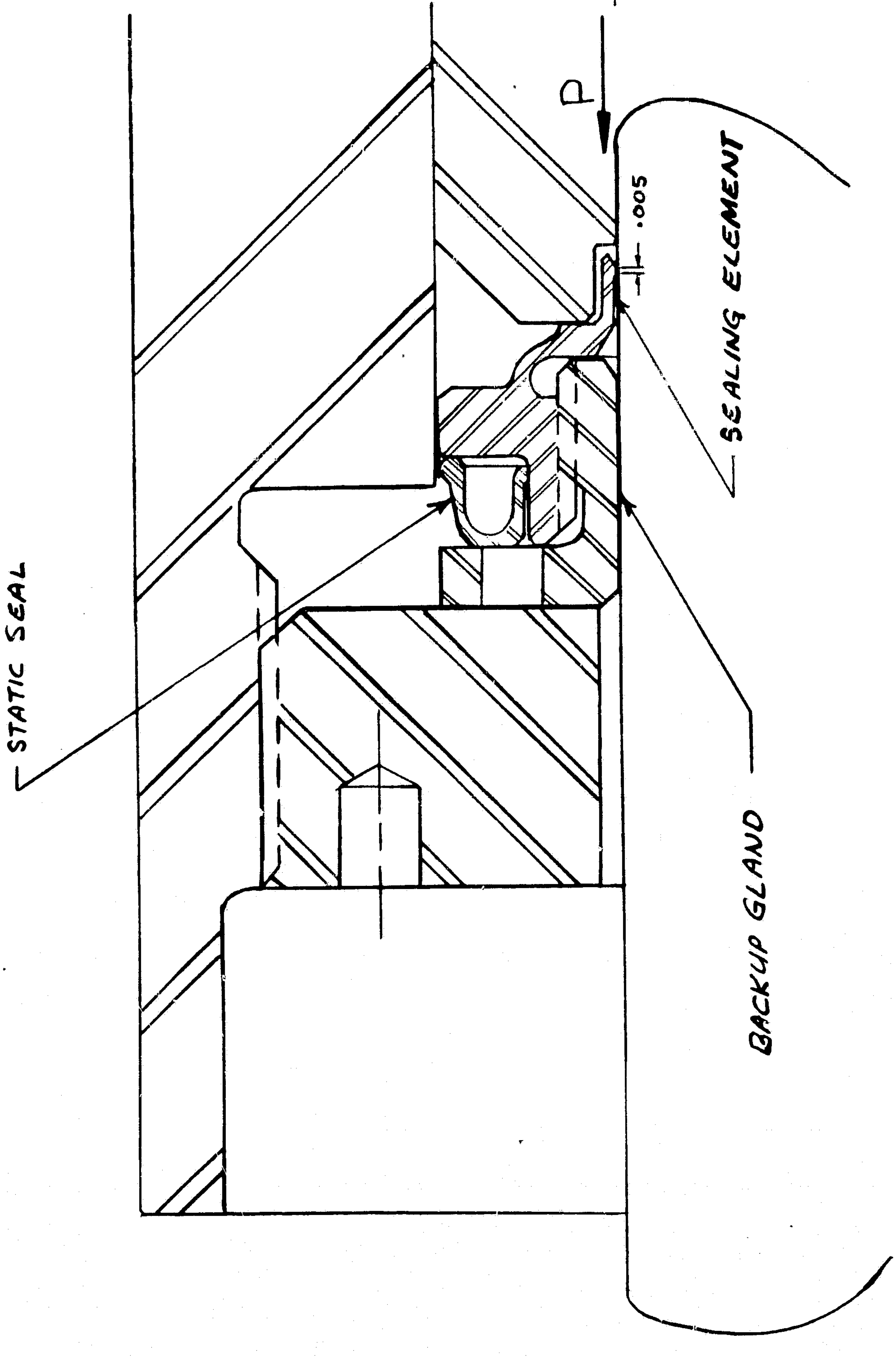
(see page A-27)

DATE 12/65

REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable; controllable; good static sealing; pressure decreases leakage.	2	Demonstrated loading ability. Behavior predictable to some extent through experience. Pressure energizing decreases leakage.	Static & dynamic seal loaded together. Loading difficult to control and adjust.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance; pressure decreases wear.	1	Compensation inherent due to "stretch".	Wear rate may be high for small sealing lip and hard material. Pressure surges can cause high contact stresses at sealing surface.
3. RELIABILITY Fails slowly and observably; rugged, resists abuse; redundant elements; resists pressure surges.	1	Not likely to fail catastrophically because of backup gland.	No redundant element. Foreign objects may damage lip or rod.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	Some experience.	Complex proprietary item requiring development in conjunction with vendor.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	1	Coefficients of expansion well matched. Some self-compensation if "stretched".	May require flame plated rod, according to vendor. Friction increases at higher pressure.
2. COST Easy to manufacture, machinable.	0		Complex and expensive. Multiple concentricities and concentric surfaces for static seal are critical.
3. SERVICEABILITY Easy to install, no spec. tool required. foolproof.	1	Not likely to be installed backwards, and would show up in pressure check.	Static seal can be installed backwards, but may show up in pressure check. Requires run-in.
4. EXPERIENCE Past exp. with sim. approach.	2	Tested at lower (300°F) temp. by vendor.	

COMMENTS



DESIGN-M BFG HARDMETAL SEAL

SEAL RATING - **LOW PRESSURE SECOND-STAGE SEAL**

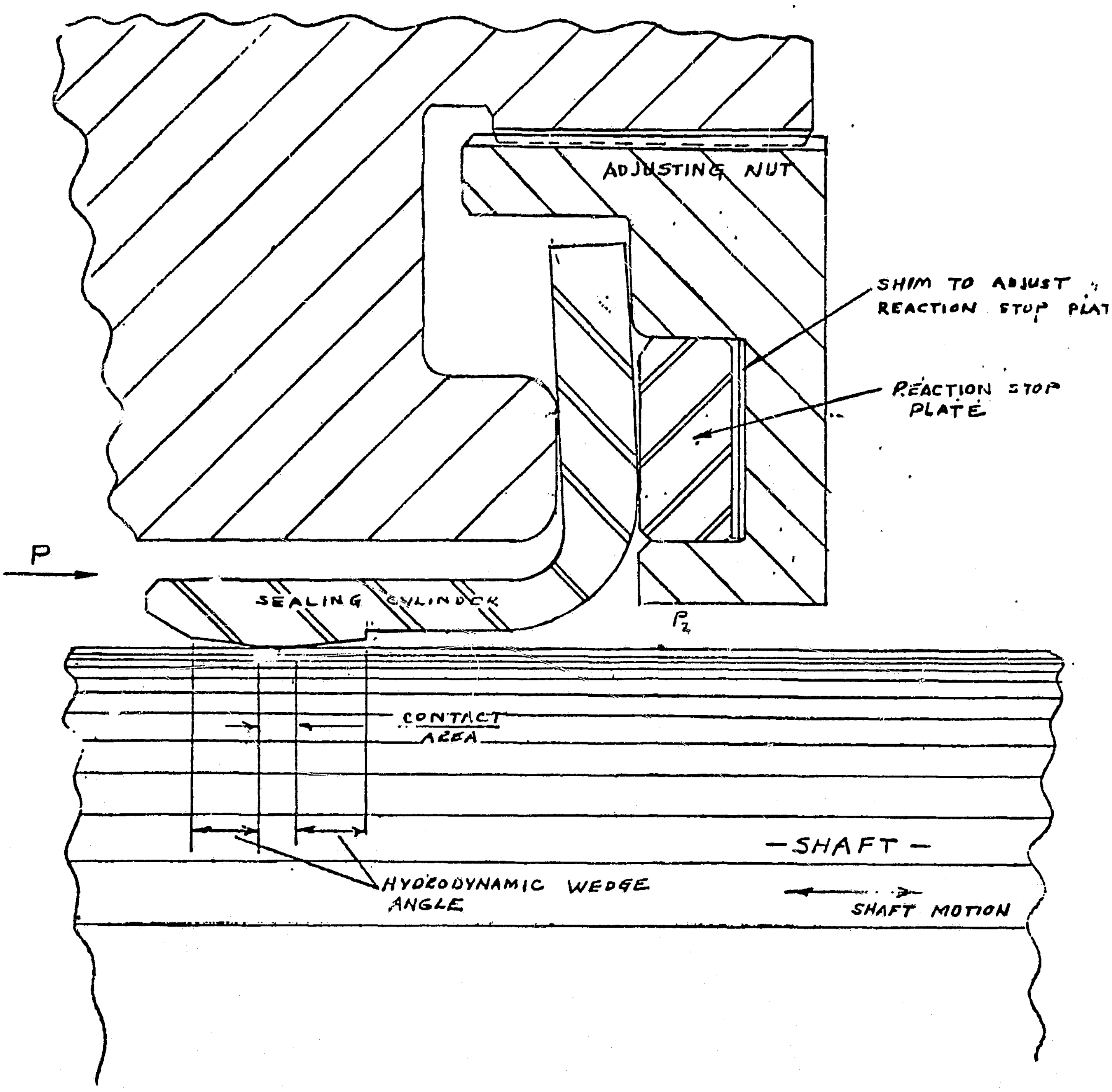
DESIGN **N - (NASA DESIGN 2) - HARD METAL**
O - (NASA DESIGN 5)

(see pages A-29 and A-30)

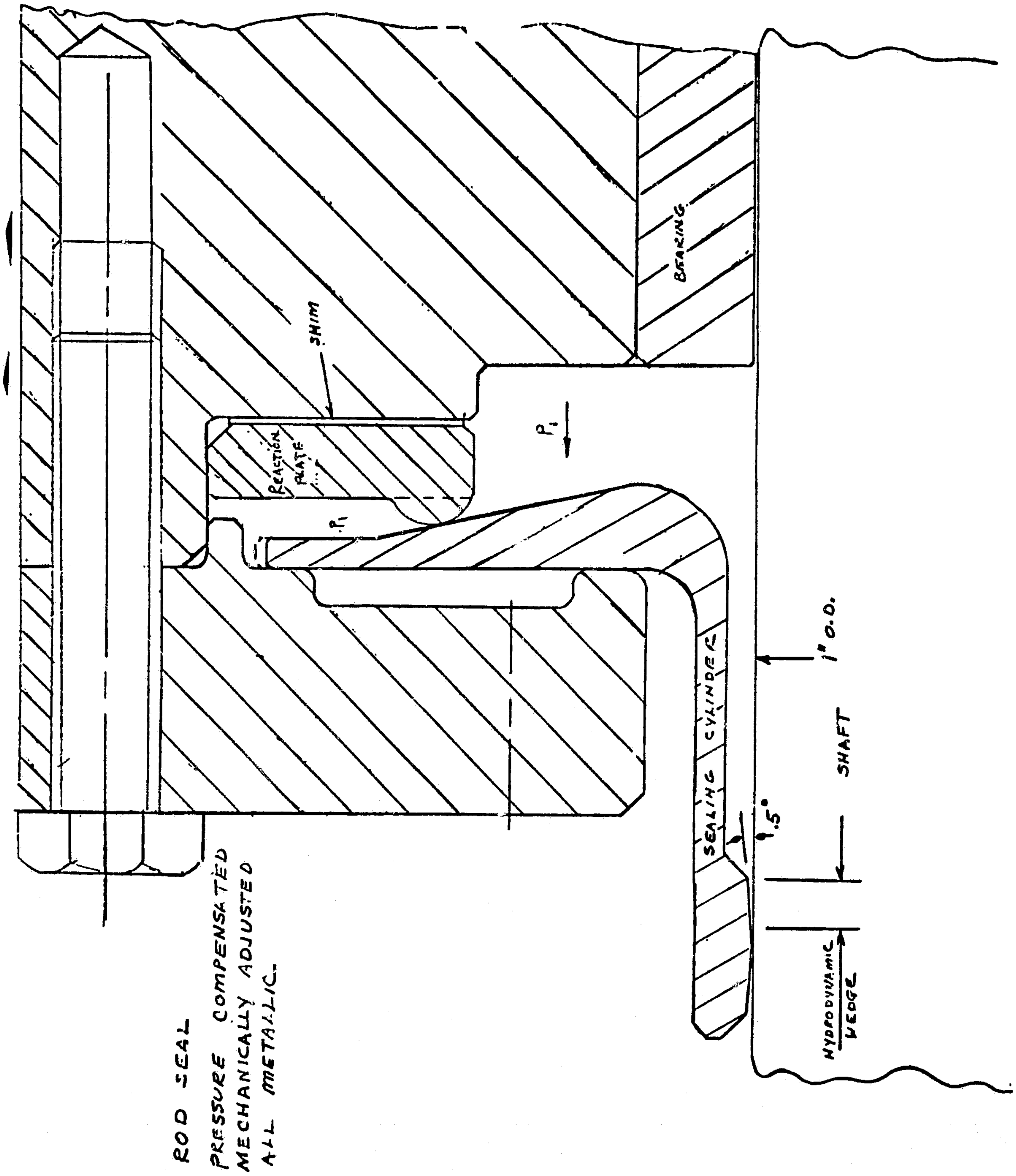
DATE 12/65
REV

ESSENTIAL FACTORS	RATING	ADVANTAGES	DISADVANTAGES
1. SEALING MECHANISM Conformable; readily loaded; predictable: controllable: good static sealing: pressure decreases leakage.	1	Loading similar to Design M. Pressure has little effect on leakage for wall thickness shown.	Static and dynamic seal loading interrelated. Loading difficult to control and adjust - requires shimming to final adjustment.
2. WEAR & COMPENSATION Low wear or able to compensate; wear not detrimental to performance: pressure decreases wear.	2	Compensation inherent in loading. Pressure has little effect on wear.	Wear rate may be high initially for line contact area and hard material. Hydrodynamic effect on outstroke questionable for low rod speeds. No effective hydrodynamic action on instroke.
3. RELIABILITY Fails slowly and observably: rugged, resists abuse: redundant elements: resists pressure surges.	1	Not likely to fail catastrophically because of reaction stop-plate.	No redundant element. Foreign objects may damage rod.
4. SHORT TERM POTENTIAL Knowledge available or can be acquired in reasonable time.	1	Similar to Design M.	Complex and sensitive geometry for loading.
DESIRABLE FACTORS			
1. DESIGN FEATURES Compatible coef. of exp.; No exotic plating on rod; low friction-pressure balancing or pressure relief available; self compensating.	2	Coefficients of expansion well matched. Self compensating. Pressure has little effect on friction.	May require flame plated rod (similar to Design M).
2. COST Easy to manufacture, machinable.	1	Fairly simple construction.	Squareness of mating parts critical.
3. SERVICEABILITY Easy to install, no spec. tool required, foolproof.	1	Not likely to be installed backwards, and would show up in pressure check.	Requires trial and error shimming. Probably requires run-in and readjustment.
4. EXPERIENCE Past exp. with sim. approach.	1	Some experience with similar design.	

COMMENTS **May be suited to non-control system high pressure applications.**


ADJUSTING NUT
SHIM TO ADJUST REACTION STOP PLAT
REACTION STOP PLATE
P
SEALING CYLINDER
CONTACT AREA
-SHAFT-
HYDRODYNAMIC WEDGE ANGLE
SHAFT MOTION

DESIGN-N


ROD SEAL
PRESSURE COMPENSATED
MECHANICALLY ADJUSTED
ALL METALLIC.
SHIM
REACTION PLATE
BEARING
P_1
P_1
SEALING CYLINDER
SHAFT
1" O.D.
1.5°
HYDRODYNAMIC WEDGE

DESIGN-O

NAS3-7264

APPENDIX B

EXHIBIT "A"

High Temperature Hydraulic System Actuator Seals

The Contractor shall furnish the necessary personnel, facilities, services and materials and otherwise do all things necessary for, or incident to, the work described below:

The work to be performed shall provide for the investigation of materials and designs of seals for potential use with hydraulic fluids in advanced supersonic aircraft. This investigation shall be directed to seals intended to function efficiently and reliably for 3000 hours in the temperature range -40°F to 600°F.

TASK I - Apparatus for Evaluation of Seal Materials and Designs

A. Facilities shall be provided for the measurement of hardness, elasticity, mechanical strength and other mechanical properties important to hydraulic system seals.

B. Facilities shall be provided in which seal materials can be evaluated at temperatures of 400°, 500°, and 600°F for chemical compatibility with five fluids to be selected by the Contractor with the approval of the NASA Project Manager. Periods of 150 hours shall be used. The fluids shall be degassed and compatibility shall be established in an inerted atmosphere system. Inerting shall be accomplished with 99.99 percent by volume nitrogen having an oxygen content of not more than 50 ppm, a hydrocarbon content (as methane) of not more than 50 ppm, and a dew point of -90°F or lower.

C. Facilities shall be provided for a rod seal test unit using one-inch diameter seals at 100 psi. Operation shall alternate between operation at 30-40 CPM with ± 2 to ± 4 inch stroke and 100-300 CPM with ± .05 to ± .10 inch stroke to simulate maneuvering and auto-pilot inputs to the actuator. The fluid temperature levels shall be 400°F, 500°F, and 600°F with the temperature for the seal in the actuator unit no less than the fluid temperature. Leakage and actuator forces shall be measured.

D. Facilities shall be provided for a rod seal test unit using seals of 1 inch and 3 inch diameters at pressures from 0 to 4000 psi. Operation shall include a cycling rate of 15-20 CPM with stroke length alternating from strokes (± ½ to ± 1 inch) to strokes (± 2 to ± 4 inch). Operation shall also include a cycling rate of 100-300 CPM with a constant stroke length (± .05 to ± .10 inch). The fluid temperature level shall be 500°F and the temperature of the seal in the actuator unit shall be no less than the fluid temperature. Leakage and actuator forces shall be measured.

E. Each component of the complete fluid systems test apparatus shall be identified by a code number which shall be scribed on the component. A complete log shall be maintained on each component to include the following: Manufacturer's designation and specifications; materials certification report; inspector's report; record of all tests (time and conditions): record of all post-test inspection reports, including photographs and failure analysis where applicable: record of all repairs and substitution of new components. These logs shall be updated at weekly intervals and maintained in a file which is available for inspection by the NASA Project Manager.

TASK II - Materials Selection, Procurement and Testing

A. The following classes of materials shall be considered for seals and/or gland bearing materials. Ten materials shall be selected by the Contractor with the approval of the NASA Project Manager. The selected materials shall be obtained and formed into appropriate test specimens for a one inch rod seal. In all cases, unless specifically approved by the NASA Project Manager polished hard chromium plating shall be used for the mating surfaces.

1. Polymide high temperature polymer (unfilled and metal filled)

2. Silver-metal composites developed by Illinois Institute of Technology under Air Force Contract No. AF33(616)-7310

3. Silver-polymer composite (Polymet)

4. Silver-base alloys or other soft phase duplex structures

5. Other metallic matrix materials

6. High strength metals (steel, titanium, cobalt, etc.)

7. New types of high temperature elastomeric materials

8. High temperature carbon graphite

B. Tests measuring bearing characteristics, hardness, elasticity, mechanical strength and other mechanical properties important to hydraulic system seals shall be made on the selected materials. All properties shall be determined at the projected maximum operating temperature, except hardness. Chemical compatibility tests with five fluids selected by the Contractor and approved by the NASA Project Manager shall be made at temperatures of 400°, 500° and 600°F. It is anticipated that these five fluids will be of the following types:

1. Chlorinated phenyl methyl silicone, General Electric Co. - F50

2. Super-refined mineral oils, MLO 60-294.

3. Monsanto Co. MCS 293 modified polyphenylether.

4. Monsanto Co. MCS 310 Halogenated polyaryl fluid.

5. DuPont fluid PR-143-AB, fluorocarbon.

Using the results of these tests the Contractor shall select five materials from the ten materials tested for further investigation under the following TASKS. The five materials selected shall be subject to the approval of the NASA Project Manager.

TASK III - Seal Design Development

A. Seals shall be designed which most effectively use the mechanical properties of the individual materials selected in TASK II for further investigation. Design studies shall provide for such consideration of the selected materials as to give optimum rod-end seal designs for each material. Such designs may logically provide for spring mounting to compensate for reduced elasticity, pressure balancing to improve endurance and the use of coatings or films as needed because of varied conformability. The seal designs shall be subject to the approval of the NASA Project Manager.

TASK IV - Low Pressure Tests

A. The five seal materials and designs shall be tested in the one-inch diameter 100 psi pressure rod and seal test facility described in TASK I. The best seal material and designs from one (1) inch diameter test shall be evaluated in three (3) inch diameter seals under otherwise identical conditions. The test fluid shall be chlorinated phenyl methyl silicone unless the NASA Project Manager directs that another fluid shall be used instead. Operation shall be for 50 hours (or until seal failure, if less than 50 hours) at each of the fluid temperature levels 400°, 500°, and 600°F. Operation shall alternate between operation at 30-40 CPM with ± 2 to ± 4 inch stroke and operation at 100-300 CPM with ± .05 to ± .10 inch stroke. Seal leakage and actuator forces shall be measured. Seal leakage in excess of one drop per minute or a two-fold increase in required operating force shall be criteria for seal failure. These criteria may be modified with the approval of the NASA Project Manager.

TASK V - High Pressure Tests

A. Three materials selected by the Contractor from the results of the low pressure tests and approved by the NASA Project Manager shall be tested in the 0 to 4000 psi rod seal test unit described in TASK I. These tests shall run for a total of 3000 hours or until seal failure occurs. A single test apparatus capable of both the low pressure tests described in TASK IV and the high pressure tests to be described in this TASK V may be used. Rod end seals of 1 inch diameter and rod end seals of 3 inch diameter shall be tested concurrently in the same unit at a pressure of 3000 to 4000 psi with the following operational cycle:

1. Operation for 35 minutes at 15-20 CPM alternately using ± 1/2 to ± 1 inch stroke and ± 2 to ± 4 inch stroke. Fluid Temperature shall increase approximately linearly from 100°F to 500°F during this period. Ambient temperature shall be increased in a 150 minute period.

2. Operation for 125 minutes at 100-300 CPM using ± .05 to ± .10 inch stroke. Fluid temperature shall be at 500°F.

3. Operation for 20 minutes at 15-20 CPM alternately using ± 1/2 to ± 1 inch stroke and ± 2 to ± 4 inch stroke. Fluid temperature shall decrease approximately linearly from 500°F to 100°F during this period.

B. In addition, a single-stage high pressure one (1) inch diameter seal, fabricated from the best material under Para. "A", TASK V, shall be developed and evaluated in the endurance rig described in TASK I, at:

1. Temperature, 500°F.

2. Pressure, 3000 psi, maximum.

3. Test profile described in TASK V, Para. A.

4. The seal shall be designed and tested to failure or 100 hours with subsequent redesign and testing subject to approval of the NASA Project Manager.

C. For all operations seal temperatures shall be no less than the fluid temperatures. Following every 20 such cycles the seal assembly shall be subjected to a 4 hour cold-soak at -40°F ambient. The leakage in the cold system shall be checked at the end of the soak period and continuously during warm-up prior to the subsequent operational cycle.

D. The operational cycle described above shall be considered typical of test requirements but shall be subject to redirection by the NASA Project Manager. The test fluid shall be chlorinated phenyl methyl silicone unless the NASA Project Manager directs that another fluid shall be used instead.

E. Seal failure criteria sufficient for the termination of a run are leakage in excess of one drop per minute or a two-fold increase in operating force.

APPENDIX C
SOURCE OF MATERIALS

Polyimide (Polymer SP)	E.I. DuPont de Nemours and Co., Inc.
Polymet	The Polymer Corporation
Silver Alloy	Handy and Harmon Co.
Nickel Foametal	Metallurgical Department General Electric Company
Westinghouse composite	Westinghouse Electric Corp.
Metco flame-plate	Metco Company
Vascojet 1000	Vanadium - Alloys Steel Company
Silver impregnated fiber composite	Republic Aviation Division Fairchild Hiller
Titanium - Tin alloy (90% Ti, 10% Sn)	NASA-Lewis Research Center
Cobalt-Molybdenum alloy (75% Co, 25% Mo)	NASA-Lewis Research Center
G.E. F-50 silicone fluid	General Electric Company
MCS-3101	Monsanto Chemical Co.
PR 240 AC high temperature grease	E.I. DuPont de Nemours and Co., Inc.
Graphitar	United States Graphite Co.
XF-1-0291 Silicone fluid	Dow Corning Chemical Co.
XF-1-2094 Silicone fluid	Dow Corning Chemical Co.
Polyimide split ring seals	Koppers Co.
Impregnated nickel Foametal (CaF_2, BaF_2 entectic)	NASA-Lewis Research Center