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THIRD SEMIANNUAL REPORT

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

by

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ABSTRACT

This report covers the third six-month period of a program to investigate seal materials and to design seals for high temperature hydraulic actuator application. During this work period, assembly and checkout of the second seal test rig has been completed. Work has progressed in the design, fabrication, and test of candidate seal configurations in the one-inch size. Low pressure testing was conducted on the nickel Foametal wedge seal, the Polymer SP V-seal, and the all-metal lip seal fabricated from Vascojet 1000. The latter design met the leakage requirement of less than one drop per minute during 50 hours of testing at 400°F and for 15.5 hours of testing at 500°F. The V-seal exhibited essentially zero leakage during 50 hours of operation at 400°F and 50 hours at 500°F. However, seal leakage exceeded the one drop per minute requirement after 21.75 hours of testing at 600°F. The nickel Foametal wedge seal developed excessive leakage after 17.5 hours of operation at temperatures of +80 to 400°F.

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

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SUMMARY

This report describes activities completed during the third six-month period of NASA Contract NAS 3-7264. The object of this program is to develop hydraulic actuator seals intended to function reliably for 3000 hours in the temperature range of -40 to +600°F. The work being performed to achieve this objective includes the investigation of advanced materials and the development and test of seal concepts.

During this reporting period, progress was made in the following areas:

- Fabrication, assembly, and checkout of the three-inch cycling rig
- Fluid compatibility testing of seal materials
- Design, fabrication, and test of candidate seal designs
- Evaluation of candidate seal designs at low pressure conditions

The three-inch seal cycling rig has been checked out at room temperature, 400°F, 500°F, and 600°F. Operation of the rig with Polymer SP V-seals in the test actuator was satisfactory. Evaluation of seal materials for fluid compatibility is essentially complete. However, additional fluid compatibility testing has been scheduled under a supplemental agreement to the contract.

Low pressure testing of the five approved candidate seal designs in the one-inch size has been initiated. To date, tests have been conducted on the Polymer SP V-seal (Design B), an all-metal lip seal (Design D) fabricated of Vascojet 1000, and the nickel Foametal wedge seal (Design I). Promising results were obtained with the first two configurations. The V-seal was operated for a total of 185 hours. The leakage requirement of less than one drop per

minute was met by this seal during 50 hours of testing at 400°F and 50 hours at 500°F. The seal did not exceed this leakage requirement until after 21.75 hours of operation at 600°F. The Vascojet lip seal also exhibited essentially zero leakage during 50 hours of testing at 400°F and 15.5 hours at 500°F. The wedge seal was operated for four hours at 400°F with zero leakage.

Based on the results obtained thus far, the V-seal and lip seal configurations offer good potential for minimum leakage operation. The major shortcoming of the Polymer SP V-seal is the high rate of thermal expansion and contraction of the material. However, this condition could be compensated for by providing adequate loading to the seal. Adequate lubrication is essential with the Vascojet lip seal to minimize piston rod wear. Use of flame-plated coatings such as tungsten carbide may extend the life of the seal. Increased bearing loads on the wedge seal due to slight lateral motion of the piston rod resulted in excessive wear of the nickel Foametal.

Low pressure testing of the two remaining candidate seal designs is scheduled to be started soon. Testing of the three-inch Polymer SP V-seal at low pressures has been initiated; the seal has completed 50 hours at 400°F with no leakage.

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 is to investigate advanced materials and seal concepts for possible use in fluid power systems of future supersonic aircraft. This investigation is 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.

Emphasis is placed on integrating material properties and seal design to obtain the optimum seal-material combination. The specific tasks to be accomplished are:

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

The progress made on the above tasks is discussed in detail in the following sections.

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TASK I - FACILITIES AND EQUIPMENT

A. GENERAL

Design, fabrication, and installation of the test rigs required for the low pressures test phase (Task IV) have been completed. Work has been initiated to modify and reinstall the fluid compatibility test apparatus for additional testing.

B. SEAL TEST RIGS

The one-inch seal test rig has been operational during this reporting period. Operation of the rig has been satisfactory. A slight lubrication problem, involving the lubrication of the movable joints in the high temperature areas of the rig, was encountered during initial testing. The high temperature anti-galling compound (Fel-Pro C-5) used initially was inadequate under long-term operating conditions; this resulted in seizure of the actuator trunnion pins. The condition was eliminated and has not recurred to date, by using an experimental fluorocarbon grease designated PR 240AC.

Assembly of the three-inch seal test rig (Figure 1) has been completed. However, during assembly of the test actuator, galling of the threads on the end glands was experienced. This condition was due to the sharp edges on the major diameter of the treads breaking off during assembly. These parts were reworked by polishing the sharp peaks off the threads.

The test actuator and cycling rig were checked out at temperatures up to 600°F. During the operation, the Polymer SP V-seals used in the actuators were subjected to 4000 cycles (four-inch stroke) with zero leakage.

C. FLUID COMPATIBILITY TEST APPARATUS

This equipment has been reassembled to facilitate additional fluid-material compatibility testing. The apparatus was also modified to provide a means of agitating the fluid during the degassing period. Details of the modifications are described in the following section. This work was initiated under a supplemental agreement to the present contract.

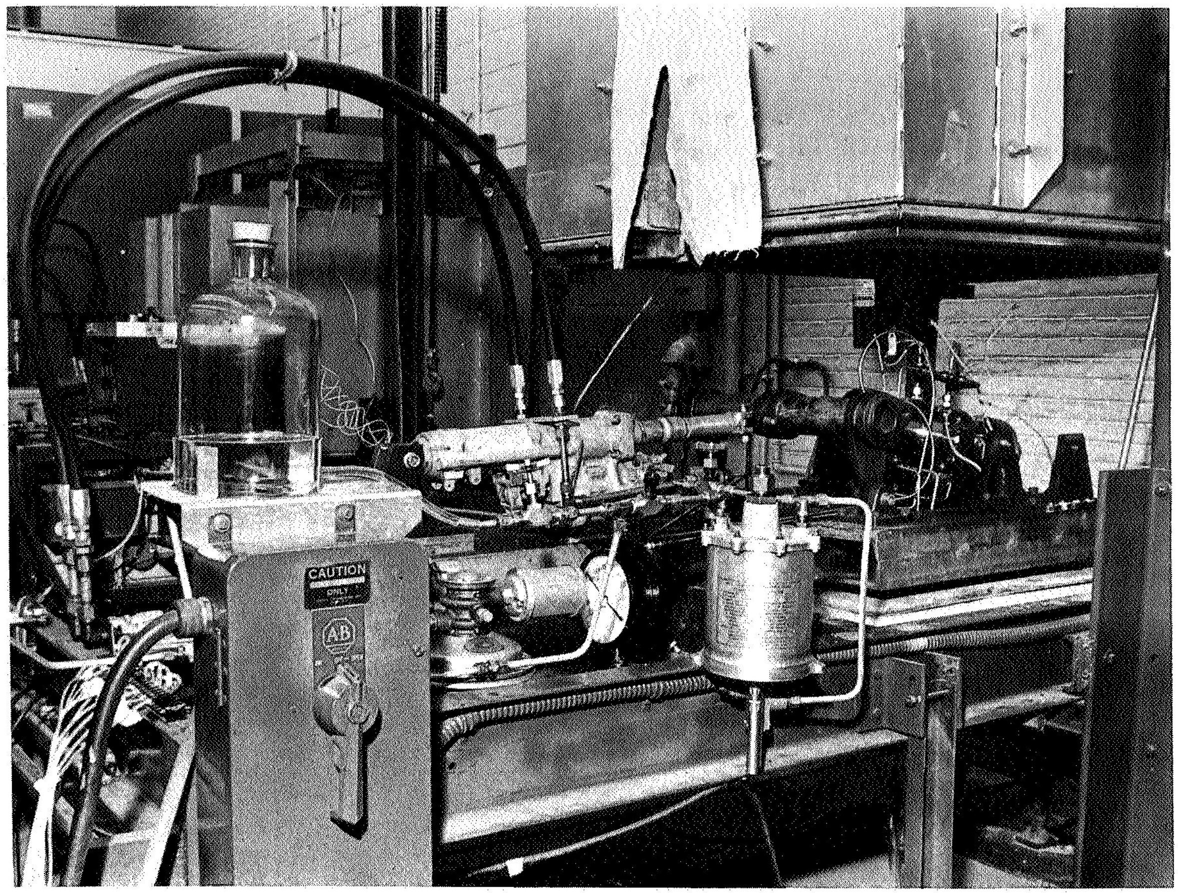


Figure 1. Three-Inch Seal Test Rig

TASK II - MATERIALS EVALUATION

A. GENERAL

Work remaining on this task at the beginning of the present reporting period consisted of a re-run at 400°F of the MCS-3101 halogenated polyaryl fluid and the ten candidate seal materials. This work has been completed. However, Task II has been extended to include compatibility testing of two recently developed silicone base fluids and selected seal and bearing materials.

B. FLUID-MATERIAL COMPATIBILITY TESTING AT 400°F

This test was a repeat of a test conducted earlier in the program for which the data (Reference 1) was inconclusive because of loss of the fluid during degassing. Results of the re-run are summarized in Table 1. The fluid (Figure 2) 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 3) 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.

C. ADDITIONAL FLUID-MATERIAL COMPATIBILITY TESTING

As provided for by supplemental agreement to Contract NAS 3-7264, additional fluid-material compatibility testing will be conducted with Dow Corning XF-1-0291 and XF-1-0294 silicone base fluids. These recently developed fluids will be tested with materials having potential for hydraulic pump application or as seals. The candidate materials are 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% $CaF1_2$ - 62% $BaF1_2$ eutectic
9) Vascojet 1000 (H-11 tool steel)
10) 75% Cobalt - 25% molybdenum alloy
11) Stellite Star J
12) S Monel

The test procedure will be essentially the same as that used in previous compatibility tests (References 1 and 2). The exceptions are that the fluids will be degassed at room temperature and that the fluid will be vibrated during degassing.

Work accomplished to date consists of the fabrication of test specimens from the candidate materials, and fabrication of the chrome-plated test buttons. Reinstallation of the test apparatus is under way. This apparatus is being modified to provide a means of vibrating the fluid during degassing. As shown in Figure 4, the fluid will be vibrated by a linkage system attached to an eccentric plate, which is in turn driven by a variable speed motor. This setup will produce a vibration of 100 to 120 cps at an amplitude of ± 1/8 inch.

TABLE 1

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

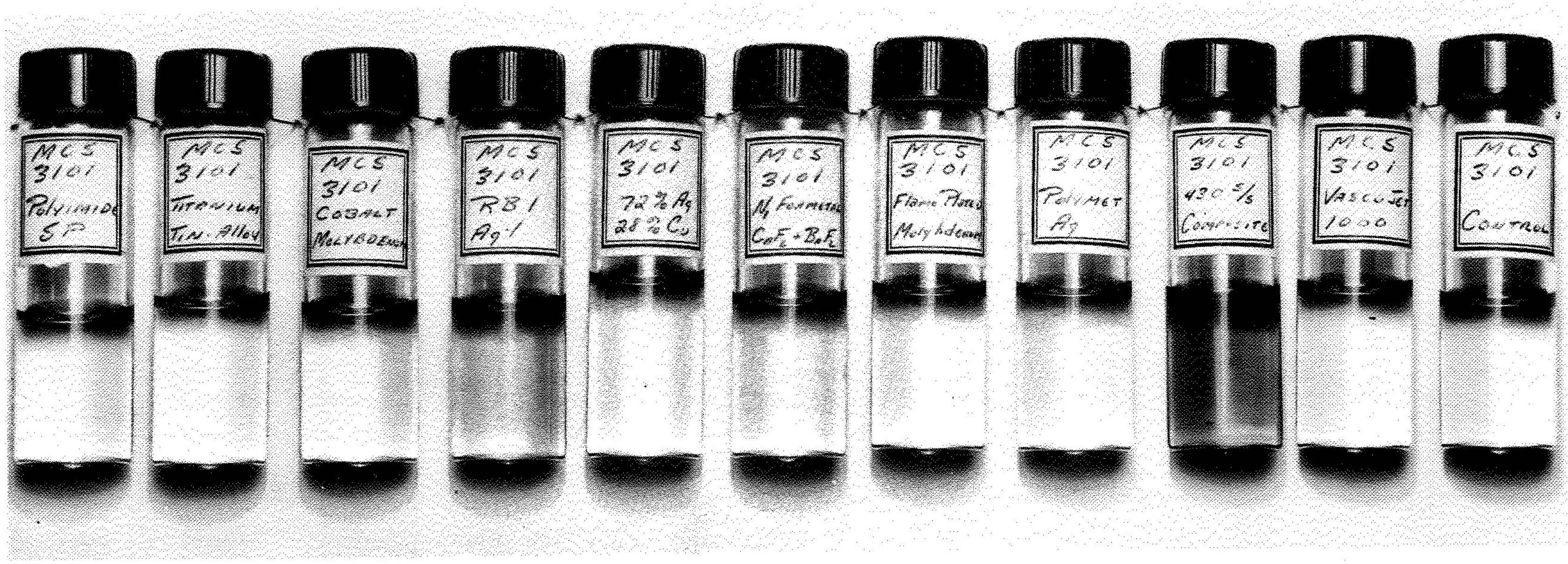


Figure 2. MCS-3101 Fluid After Test At 400°F

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

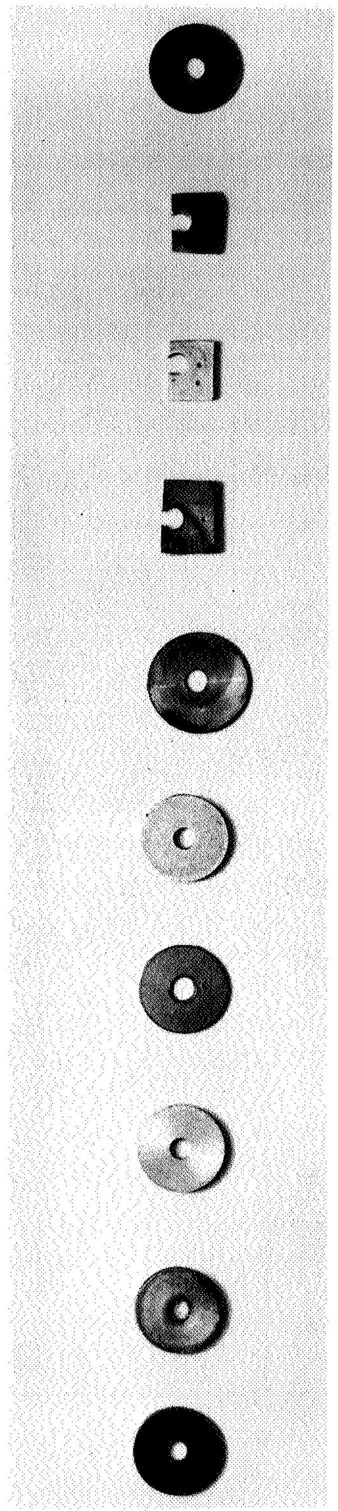


Figure 3. Material Specimens After Test At 400°F

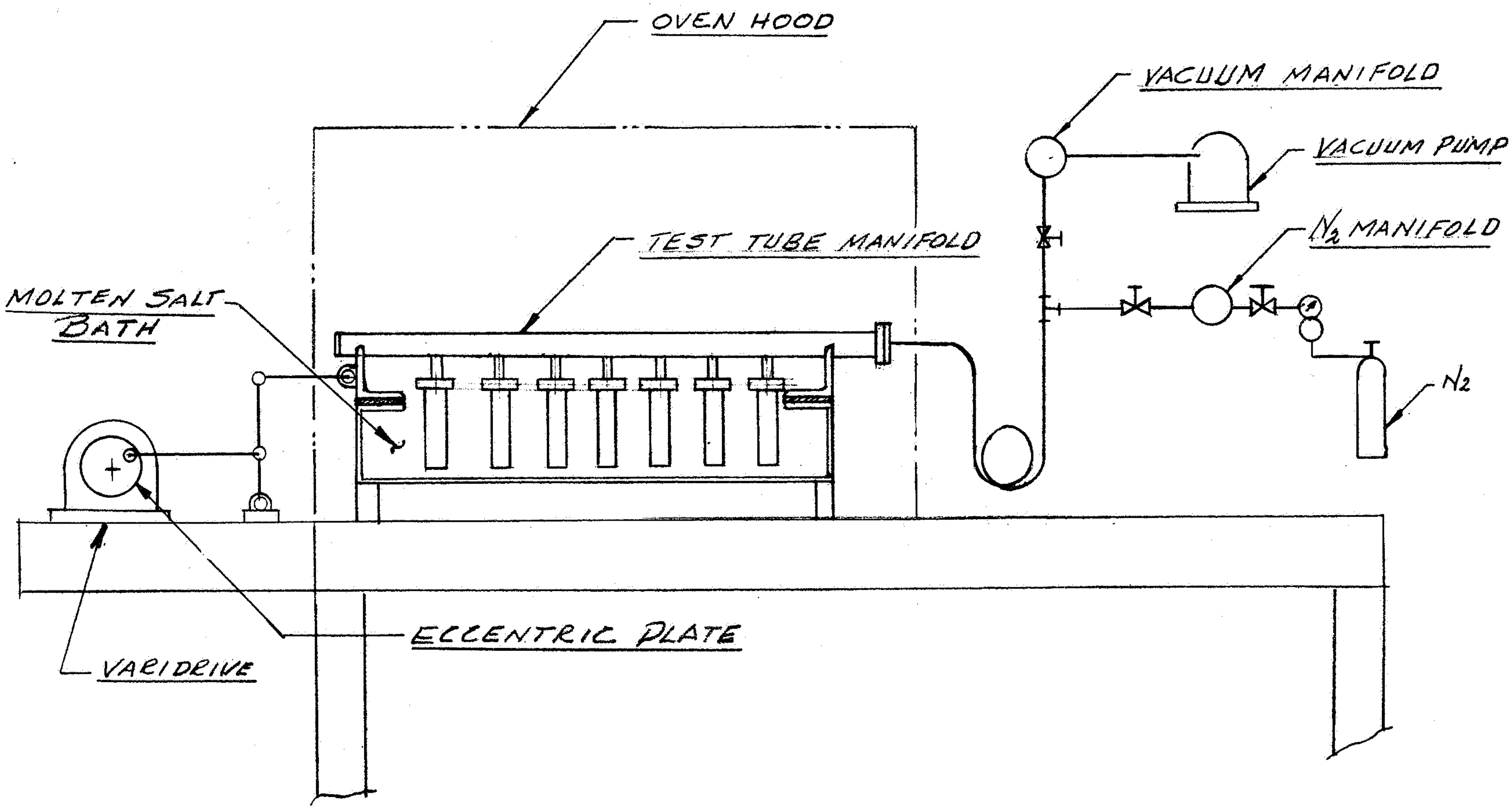


Figure 4. Fluid-Material Compatibility Test Setup

TASK III - SEAL DESIGN AND DEVELOPMENT

A. GENERAL

Detail design and preliminary evaluation of the five candidate seal configuration have been initiated. Preliminary testing was conducted where necessary to substantiate design assumptions. Tests were performed to determine the effects of bearing wear and lateral motion on the various seal configurations. The ability of the seals to operate at over-pressure conditions in the event of first-stage seal failure was also evaluated. To date, work has been completed on Design B (Polymer SP V-seal), Design D (Vascojet lip seal), and Design I (nickel Foametal wedge seal). These configurations have already been evaluated at low pressure conditions (see Task IV). Work has been initiated on the cobalt-molybdenum lip seal and the Vascojet-silver alloy reed seal.

B. DESIGN AND PRELIMINARY TESTING

1. Design B - Polymer SP V-Seal

Detail design of this seal configuration is shown in Figures 5, 6, and 7. Much of the preliminary testing on this seal has been accomplished (Reference 1).

The effects of bearing wear and lateral motion on this seal were simulated with the test setup shown in Figure 8. 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 in-board 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 centistroke at room temperature), which is similar to the viscosity of a hydraulic fluid at 500°F to 600°F. Prior to installation of the

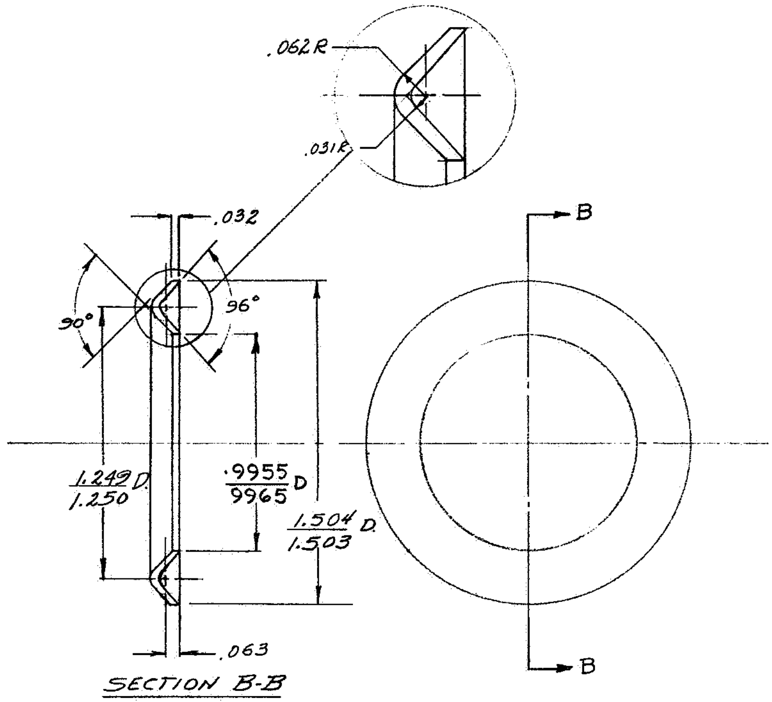


Figure 5. Polymer SP V-Seal

3 3/4"

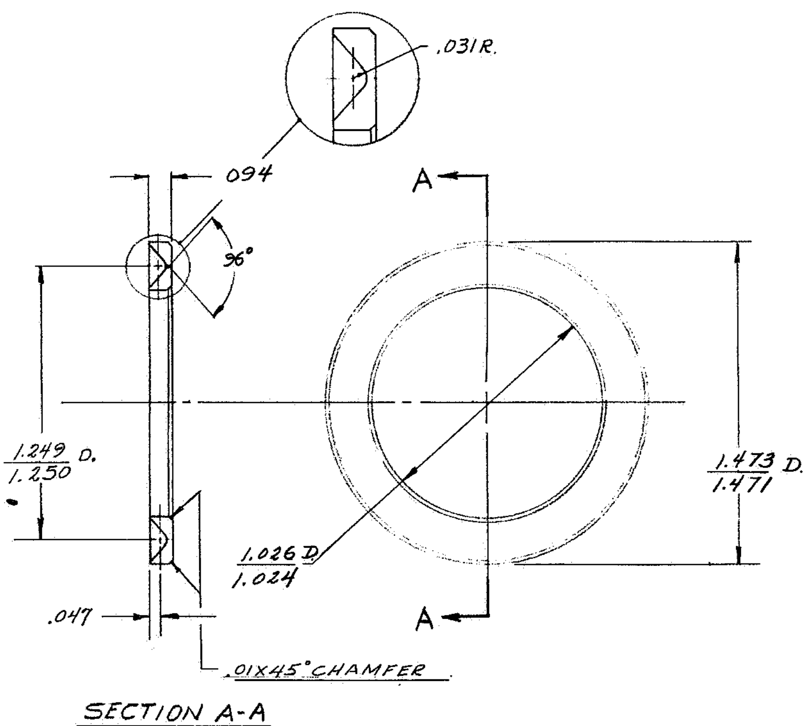


Figure 6. Backup Ring - 321 Stainless Steel

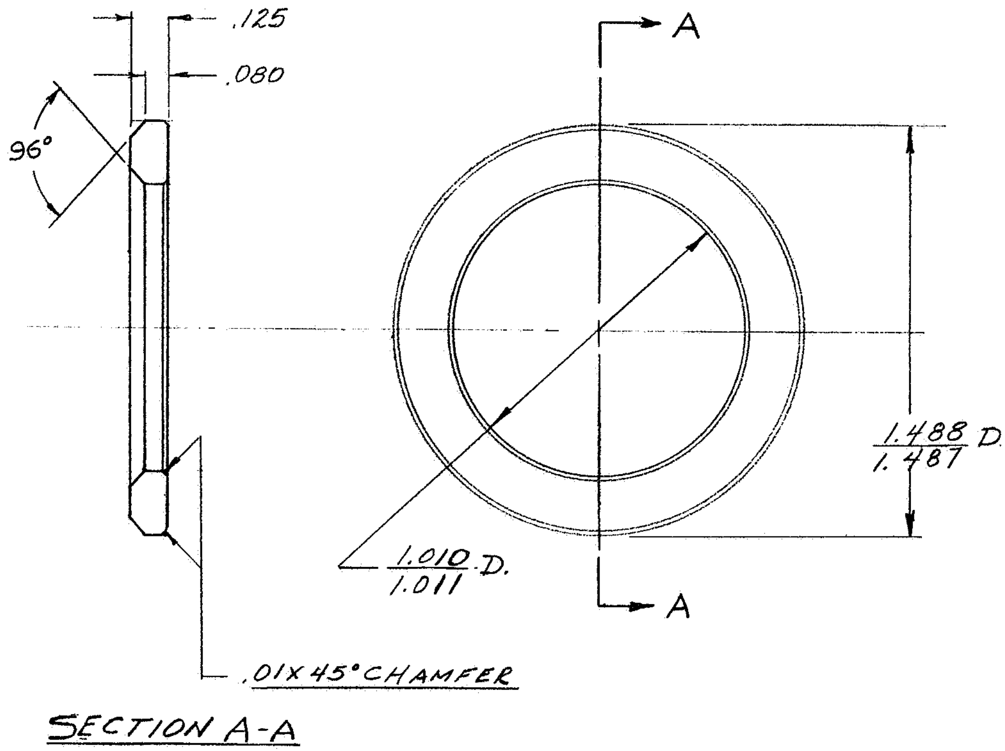


Figure 7. Load Ring - 321 Stainless Steel

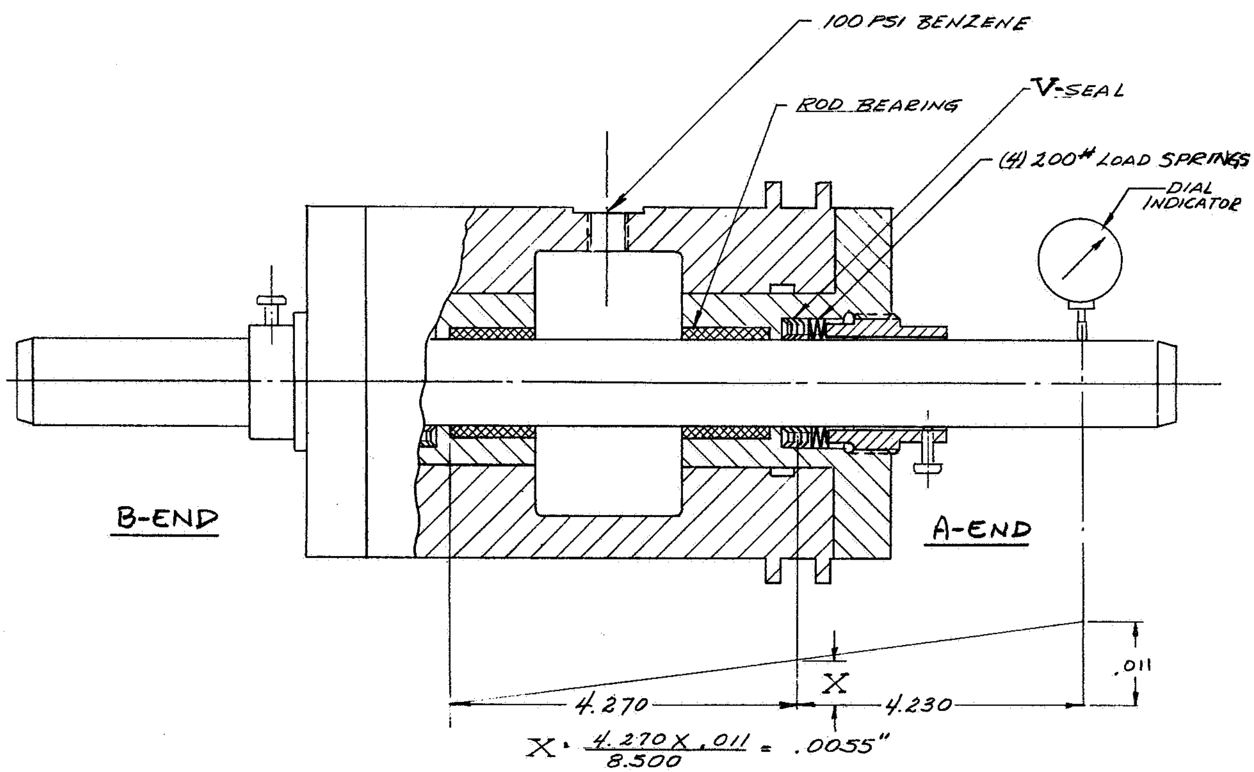


Figure 8. Polymer SP V-Seal - Radial Deflection Versus Leakage Setup

seals in the test fixture (Figure 8) the radial deflection of the seals was determined by loading them in the seal gland shown in Figure 9 and measuring the changes in their inside diameters. The resulting radial deflections, with a 200-pound spring load were 0.0075 and 0.0085 inch, respectively, for the A-end and B-end seals.

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 Polymer SP 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 8, 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 effect on the seal breakout friction. As shown in Figure 10, 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. <u>Design D - Vascojet 1000 Lip Seal</u>

The feasibility of this design was demonstrated in earlier testing reported in Reference 1. The final seal design shown in Figure 11 provides

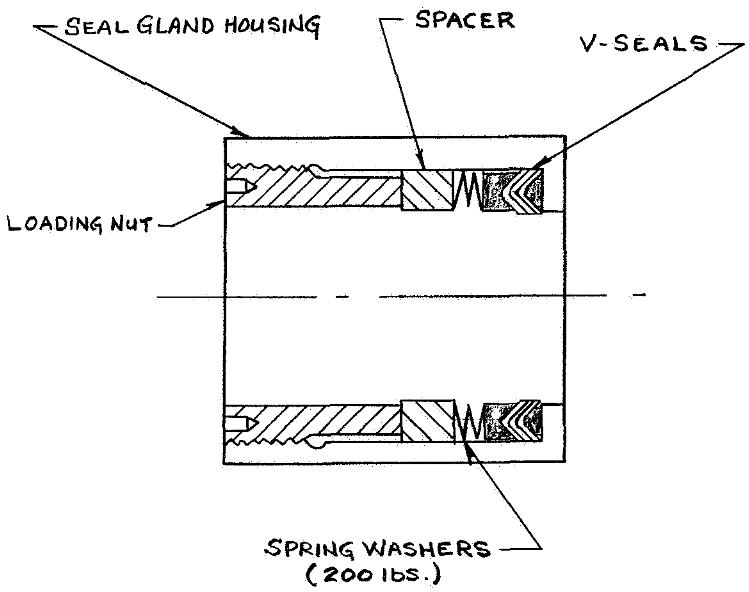


Figure 9. Test Gland - Seal Deflection Test

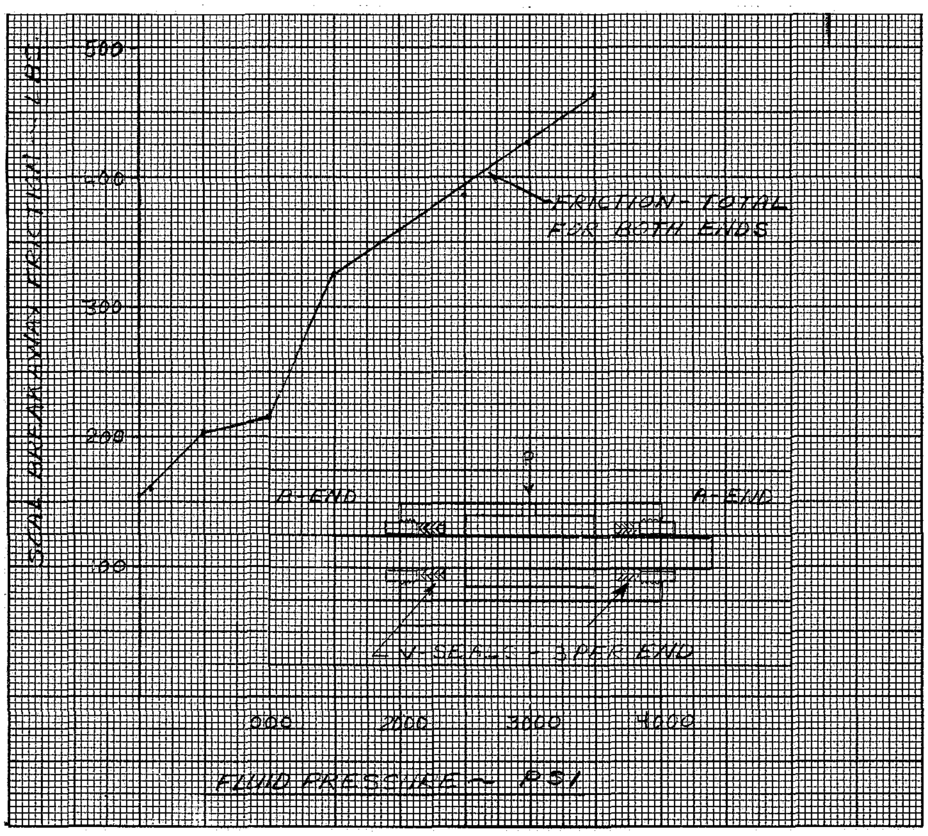


Figure 10. Polymer SP V-Seal - Seal Breakaway Friction Versus Pressure (F-50 Fluid at Room Temperature)

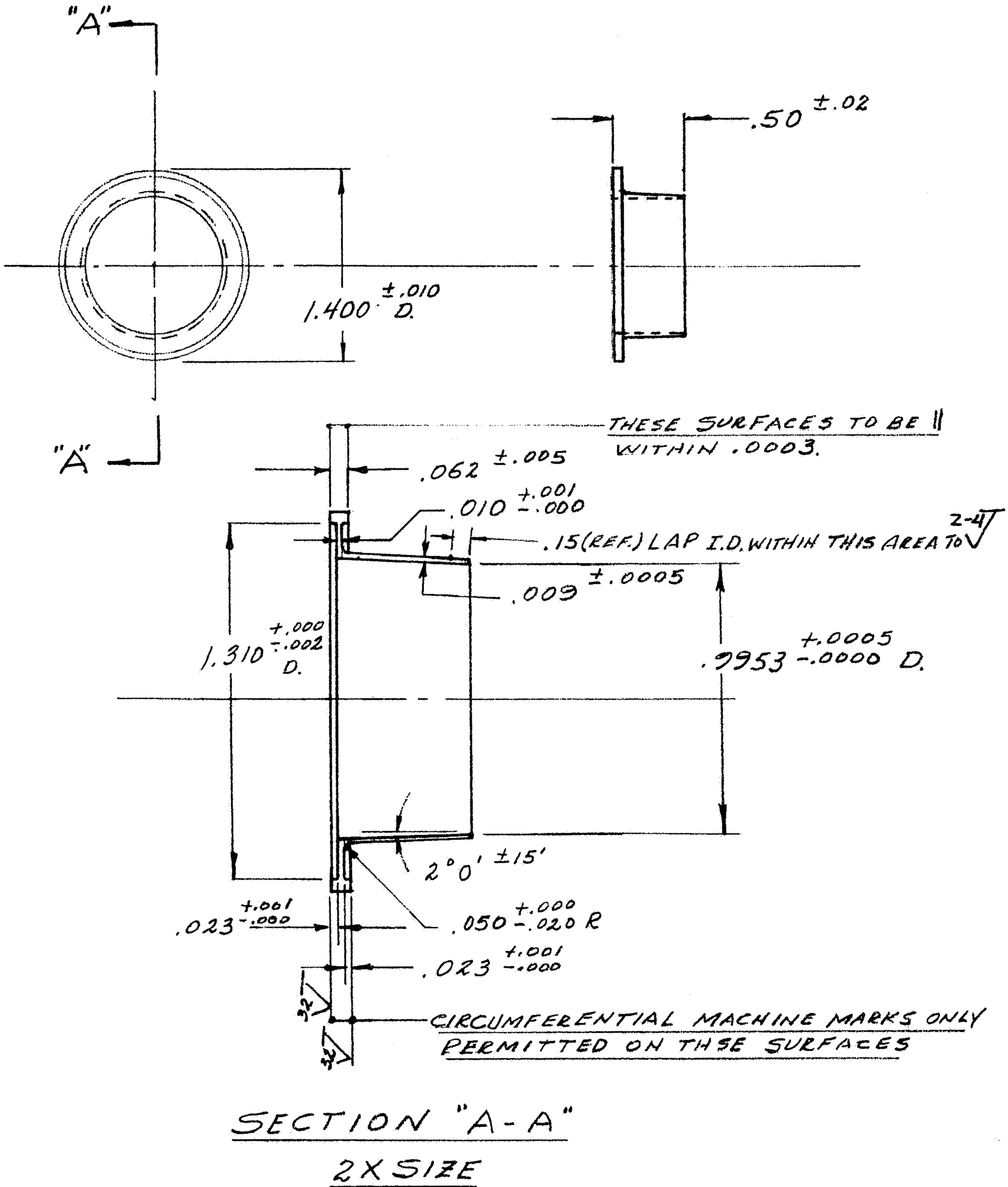


Figure 11. Vascojet 1000 Lip Seal (Modified Flange)

for a seal diametral interference of 0.0038 inch and a seal contact load of 50 pounds per inch of circumference. This configuration was arrived at after additional development and testing to improve the seal's ability to withstand lateral deflections. In these tests the original design of the seal was modified by reducing the thickness of the flange. The original thickness (0.06 inch) of the flange was first reduced to 0.020 inch. However, this reduction did not improve the tracking ability of the seal. Subsequently, the seal flange thickness was decreased to 0.01 inch, which is slightly thicker than the 0.009 inch sealing lip. This configuration enabled the seal to withstand a lateral deflection of 0.0055 inch without leaking. The corresponding rod deflection was 0.010 inch, at which point the rod bottomed on the gland bearings and testing was discontinued. These tests were conducted in the setup shown in Figure 12, Testing was conducted at room temperature, with benzene used as the test fluid.

Of the design modifications investigated, the thinner seal flange appears to be the most promising. Investigation of an earlier design change (Reference 1), which consists of decreasing the sectional thickness of the sealing lip, enabled the seal to withstand lateral deflections up to 0.004 inch. However, the reduced sectional thickness limited the seal to a fluid pressure of approximately 650 psi. In the modified flange configuration (Figure 11), the seal could withstand fluid pressure up to approximately 1200 psi. This configuration was used in the low pressure test phase.

3. Design D - Cobalt-Molybdenum Alloy Lip Seal

As shown in Figure 13, 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, which 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.

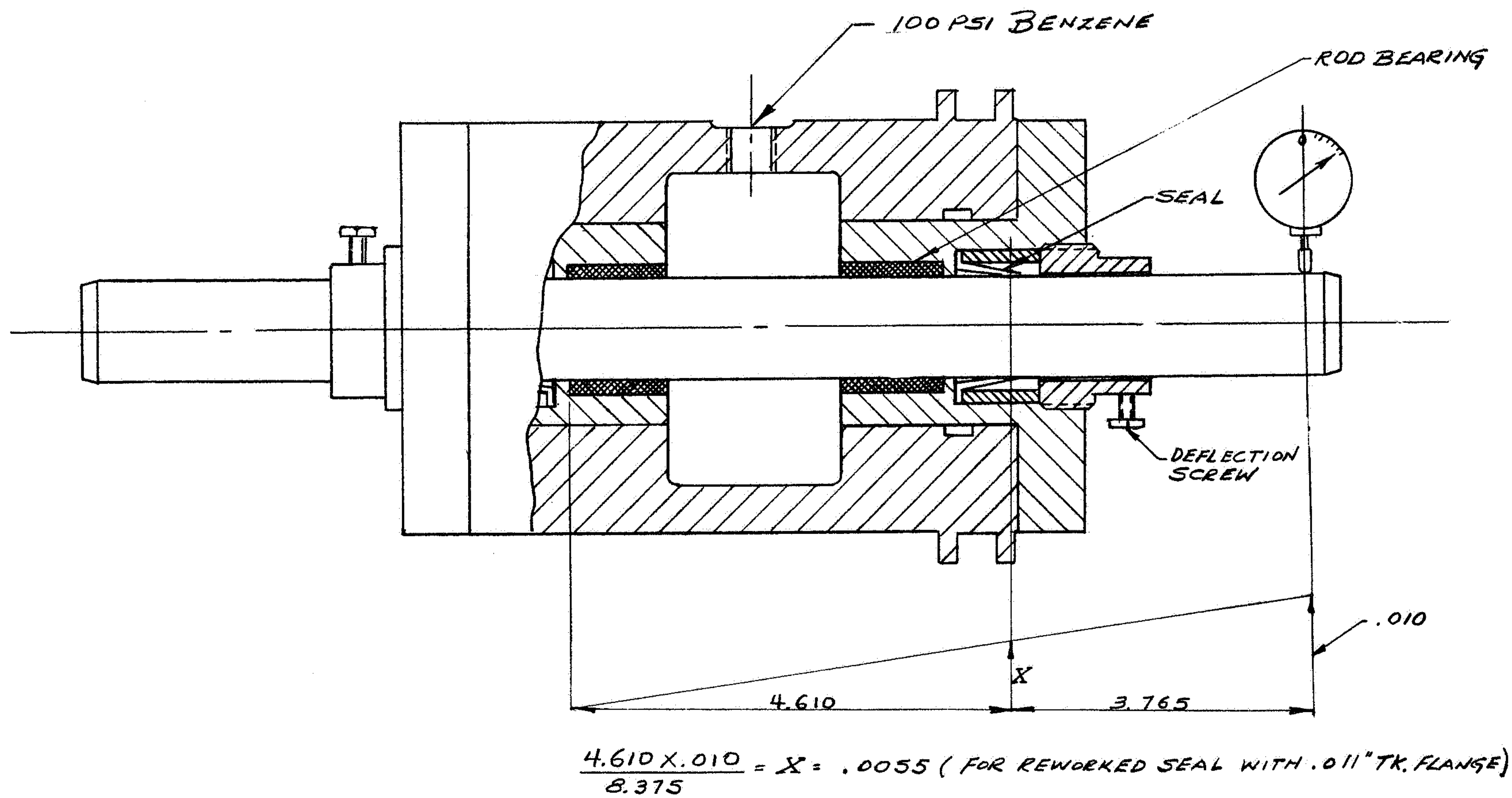


Figure 12. Vascojet 1000 Lip Seal - Radial Deflection Versus Leakage Setup

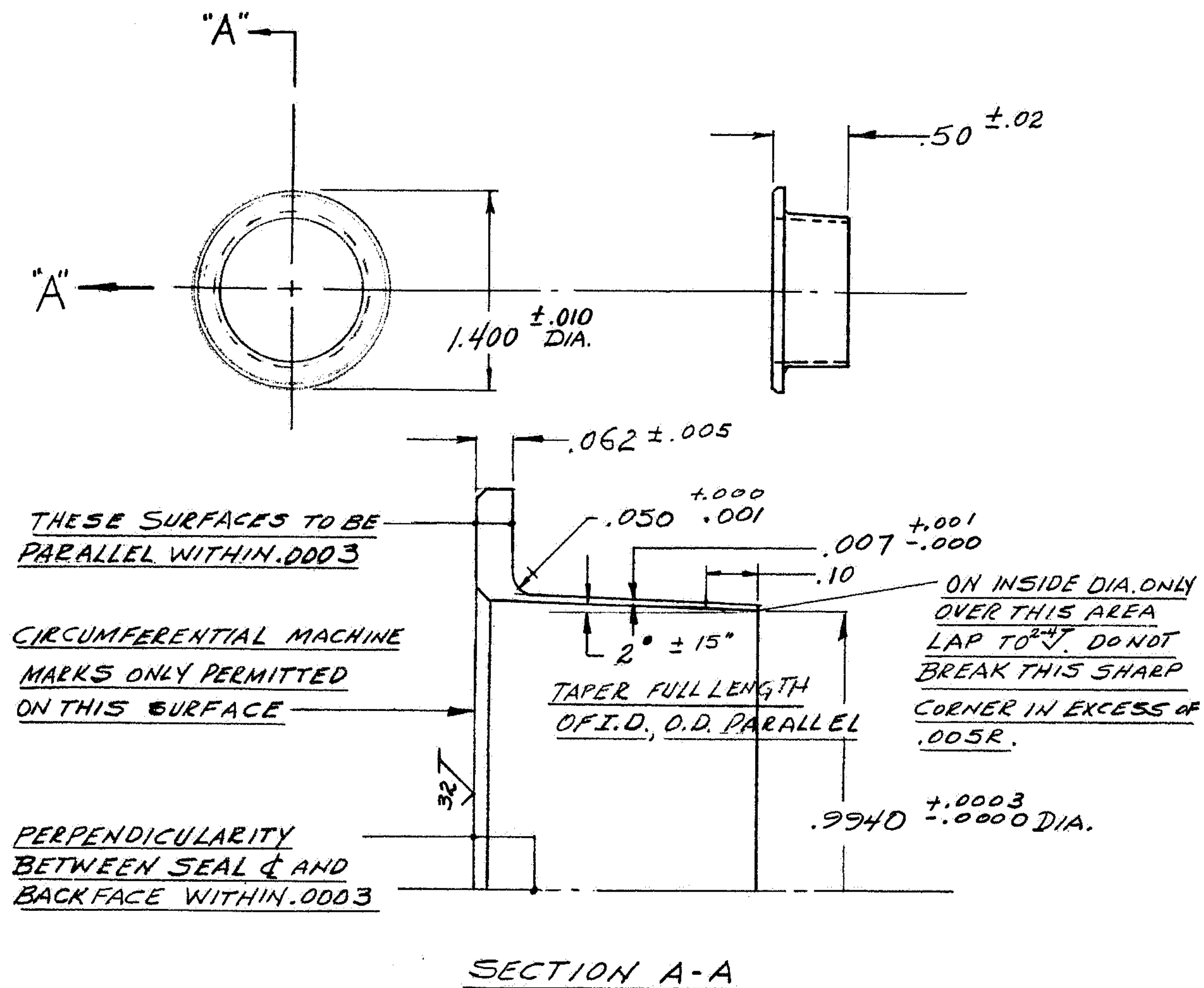


Figure 13. Cobalt-Molybdenum Alloy Lip Seal

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. Since 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 14) appeared to be solid pieces of molybdenum that pulled out during machining.

Additional quantities of this alloy have been received via NASA from the Stellite Division of Union Carbide Corp. Further attempts will be made to fabricate seals from this material.

4. Design I - Nickel Foametal Wedge Seal

Investigations were conducted on wedge seals with various taper angles to determine the most suitable configuration. Since 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.

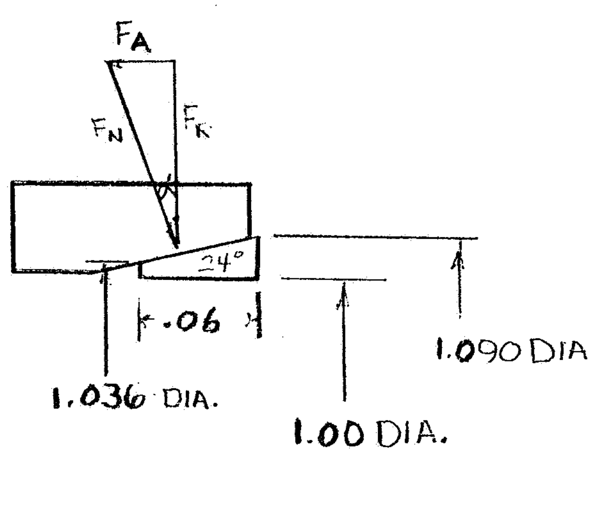


Figure 14. Inclusions in Cobalt-Molybdenum Alloy

Seal wedge angle, $\alpha = 24$ degrees

Coefficient of friction, $\mu = 0.2$ 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, $E = 9 \times 10^6$ for nickel Foametal (Reference 1)

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

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

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

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

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 2.

TABLE 2

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 15. The test apparatus consists of a seal gland, holding fixture, force gage, 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 gage, and the radial deflection of the seal was measured with a dial indicator located at the sealing surface.

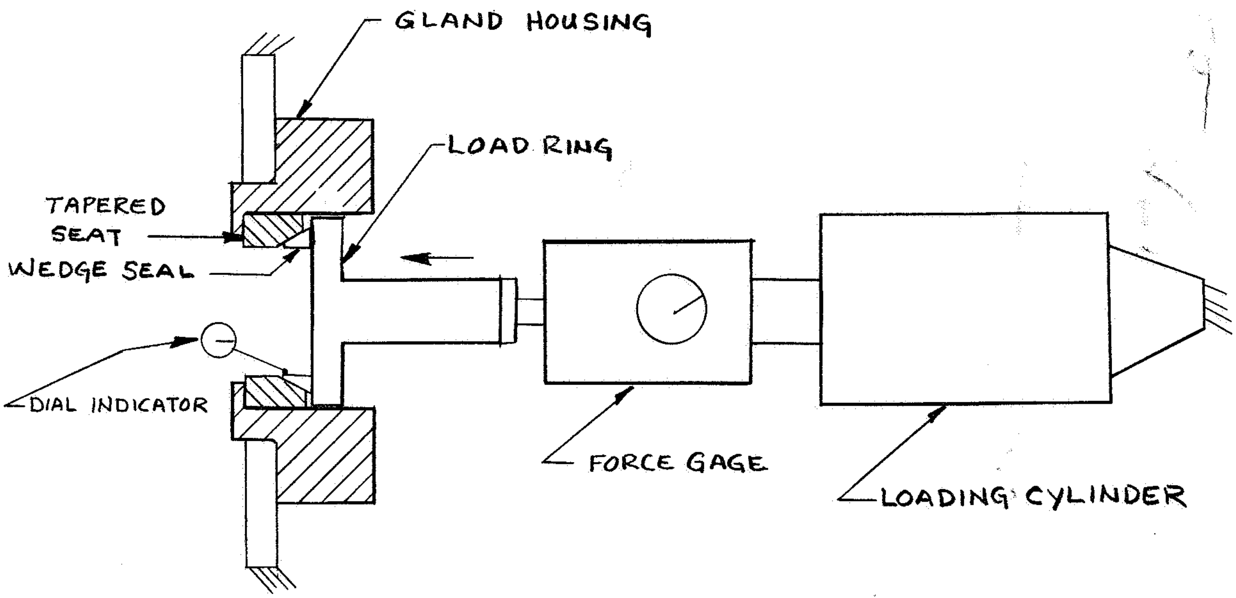


Figure 15. Wedge Seal Deflection Test Setup

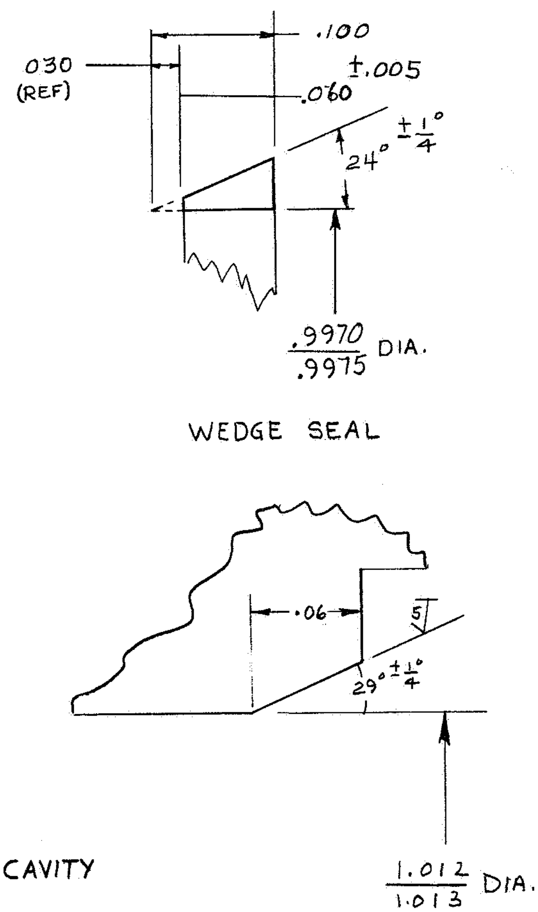


Figure 16. Geometry of Nickel Foametal Wedge Seal

Results obtained with seals having wedge angles of 12, 24, and 30 degrees are shown in Table 2. 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. Since 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 2, 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 16) 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 17, 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 18 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 18. 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 15. As shown in Figure 18, 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

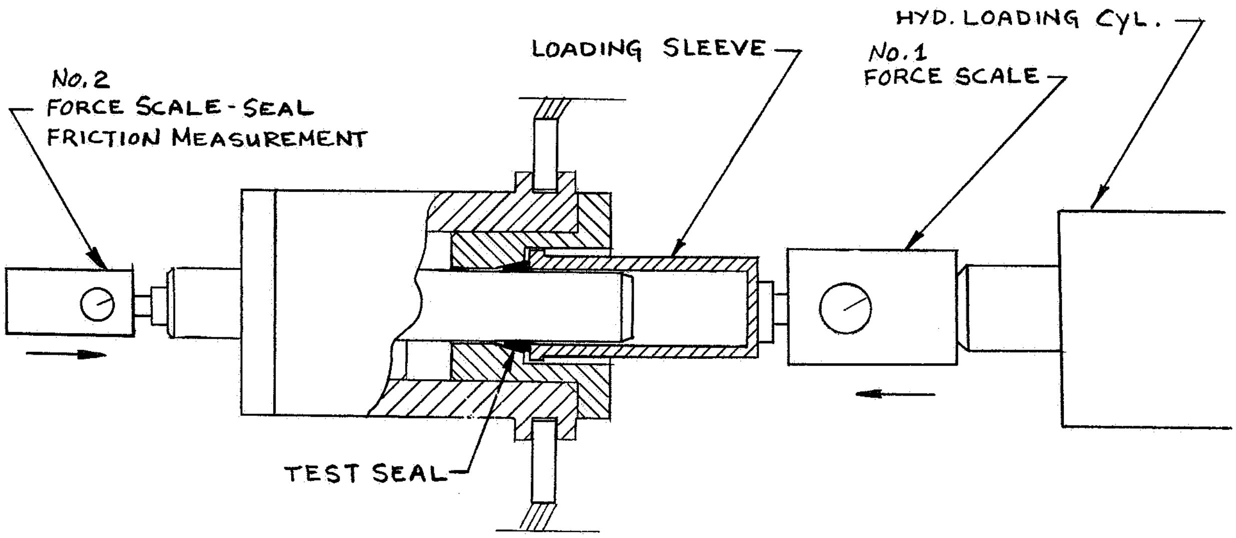


Figure 17. Test Setup for Wedge Seal Friction Measurement

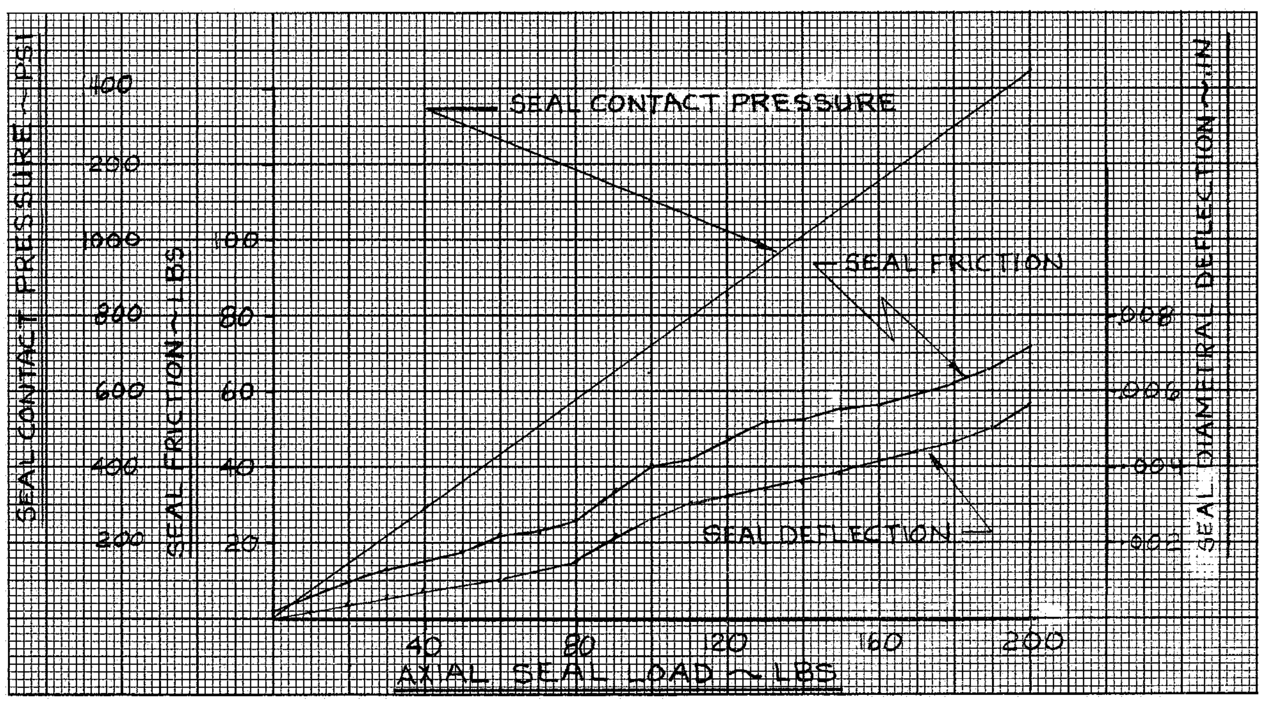


Figure 18. Friction Deflection and Contact Stress of Wedge Seal at Various Axial Seal Loads

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 8) and procedures were the same as those previously described in the Polymer SP V-seal testing. 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 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 17 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 then 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 19) 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 20, a seat angle of 29 degrees was used to ensure proper seating of the seal. The

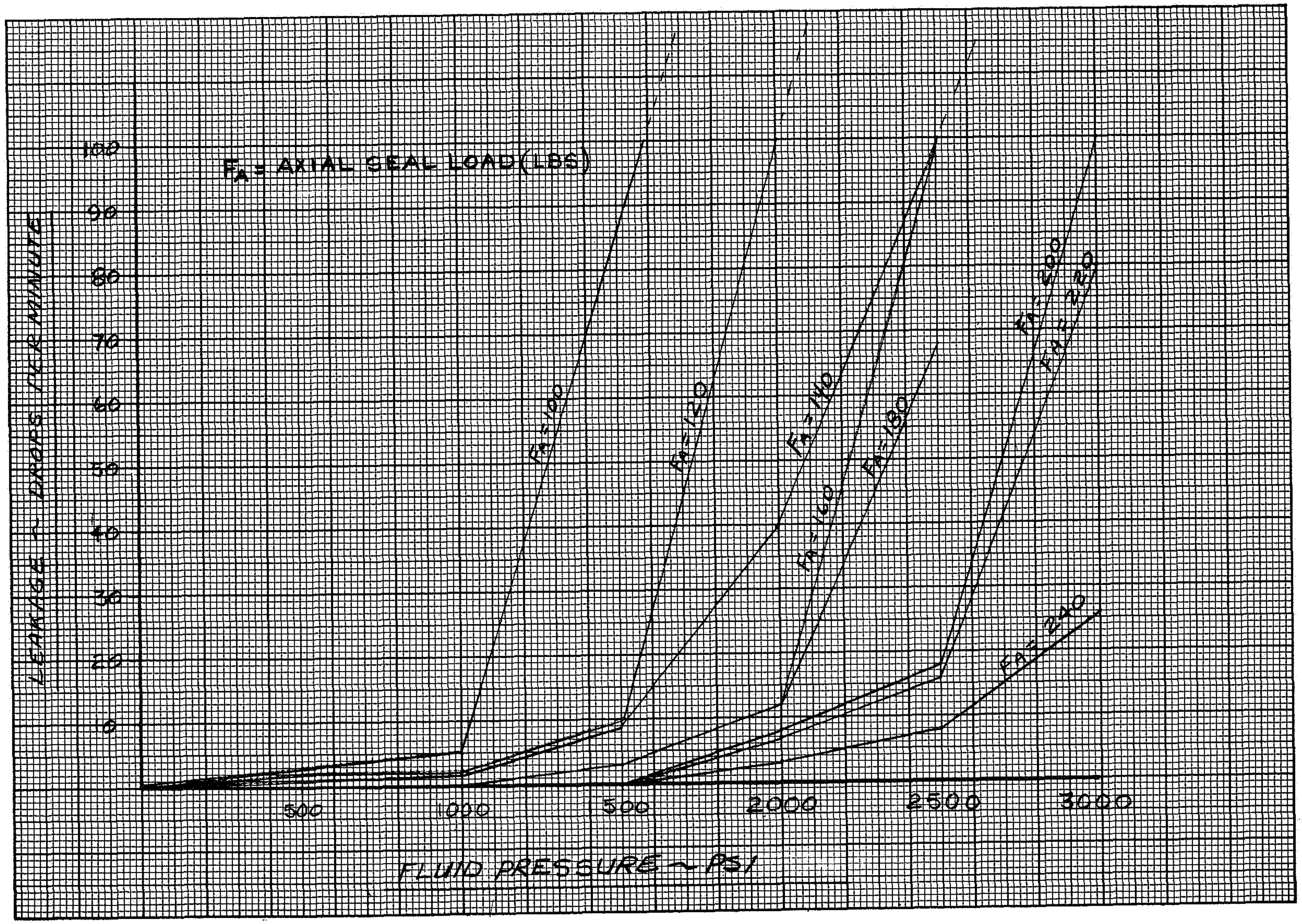


Figure 19. Leakage Versus Pressure - Nickel Foametal Wedge Seal

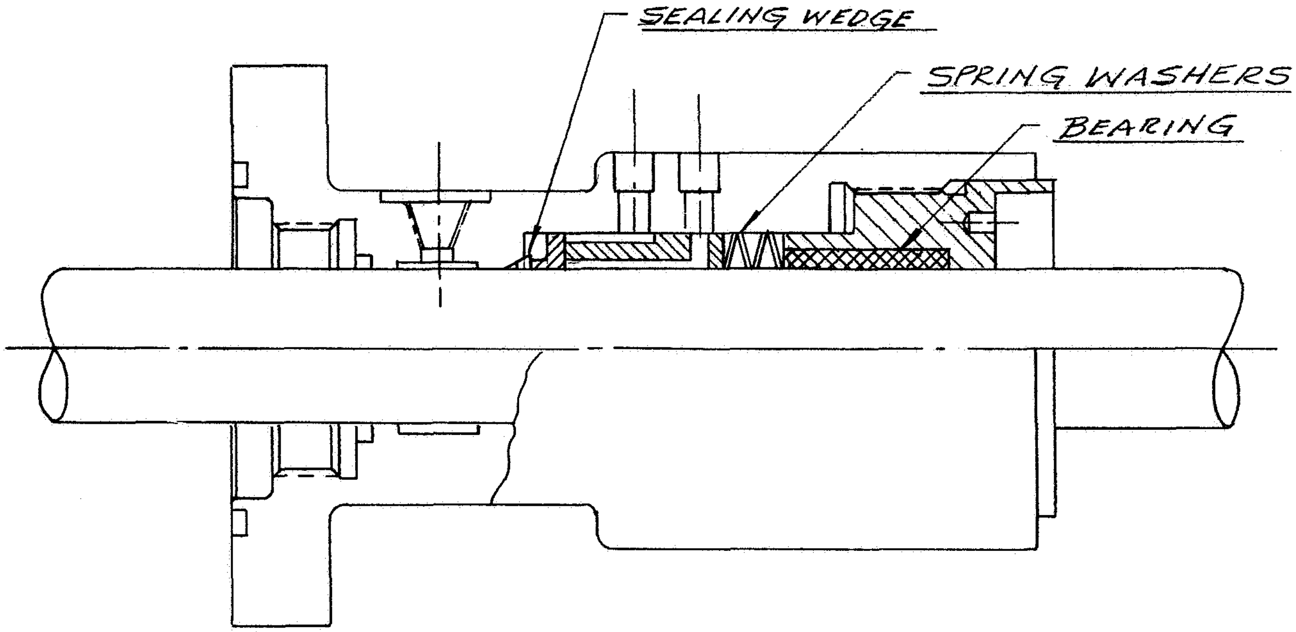


Figure 20. Gland Configuration - Nickel Foametal Wedge Seal

seal was loaded to obtain maximum deflection, which was approximately 0.0055 inch (diametrally). According to the data shown in Figure 18, 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 18, the foregoing conditions would provide a seal that possesses high deflection capability for wear compensation and also minimum leakage at over-pressure conditions. Since only the A-end of the actuator was reworked for the wedge seal, a set of Polymer SP V-seals was installed on the B-end of the unit.

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 21, 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. Since the seal friction load showed only a slight decrease at the termination 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-Copper Alloy Reed Seal

Seals of this design have been fabricated for preliminary testing. No difficulties were encountered in the machining of the silver-copper (72% Ag-28% Cu) alloy. However, the Vascojet 1000 reeds, which were only 0.0055 inch thick, appeared to be considerably distorted after being heat treated. A dimensional check of these reeds is being made to determine the extent of out-of-roundness. Preliminary testing of this configuration will be initiated if the Vascojet 1000 reeds are found acceptable.

Figure 21. Nickel Foametal Wedge Seal After Test

TASK IV - LOW PRESSURE TESTING

A. GENERAL

Low pressure testing was completed on the Polymer SP V-seal (Design B), the Vascojet 1000 lip seal (Design D), and the nickel Foametal wedge seal (Design I) in the one-inch size. Testing of the Polymer SP V-seal in the three-inch size has also been initiated. Testing was conducted with F-50 silicone fluid at 100 psi and at temperatures of 400, 500, and 600°F. Required time at each of these temperature levels was 50 hours. Seal leakage in excess of one drop per minute or a two-fold increase in seal friction was considered failure.

As shown in Figure 22, the test profile consists 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 250 ± 50 cpm for two hours) and long-stroke cycling (± 2 inches at 25 ± 5 cpm for one hour).

3) Long-stroke cycling during the cool-down from the designated temperature level to room temperature.

4) Repeat steps 1, 2 and 3 for the next temperature level.

B. LOW PRESSURE TEST NO. 1 - POLYMER SP V-SEAL

The test configuration as shown in Figure 23 consisted of three V-seals, a loading ring, backup ring, and four spring washers. The spring washers each provides a nominal load of 200 pounds at a deflection of approximately 0.021 inch. The piston rod, which was made from 440C Cres. was chrome-plated,

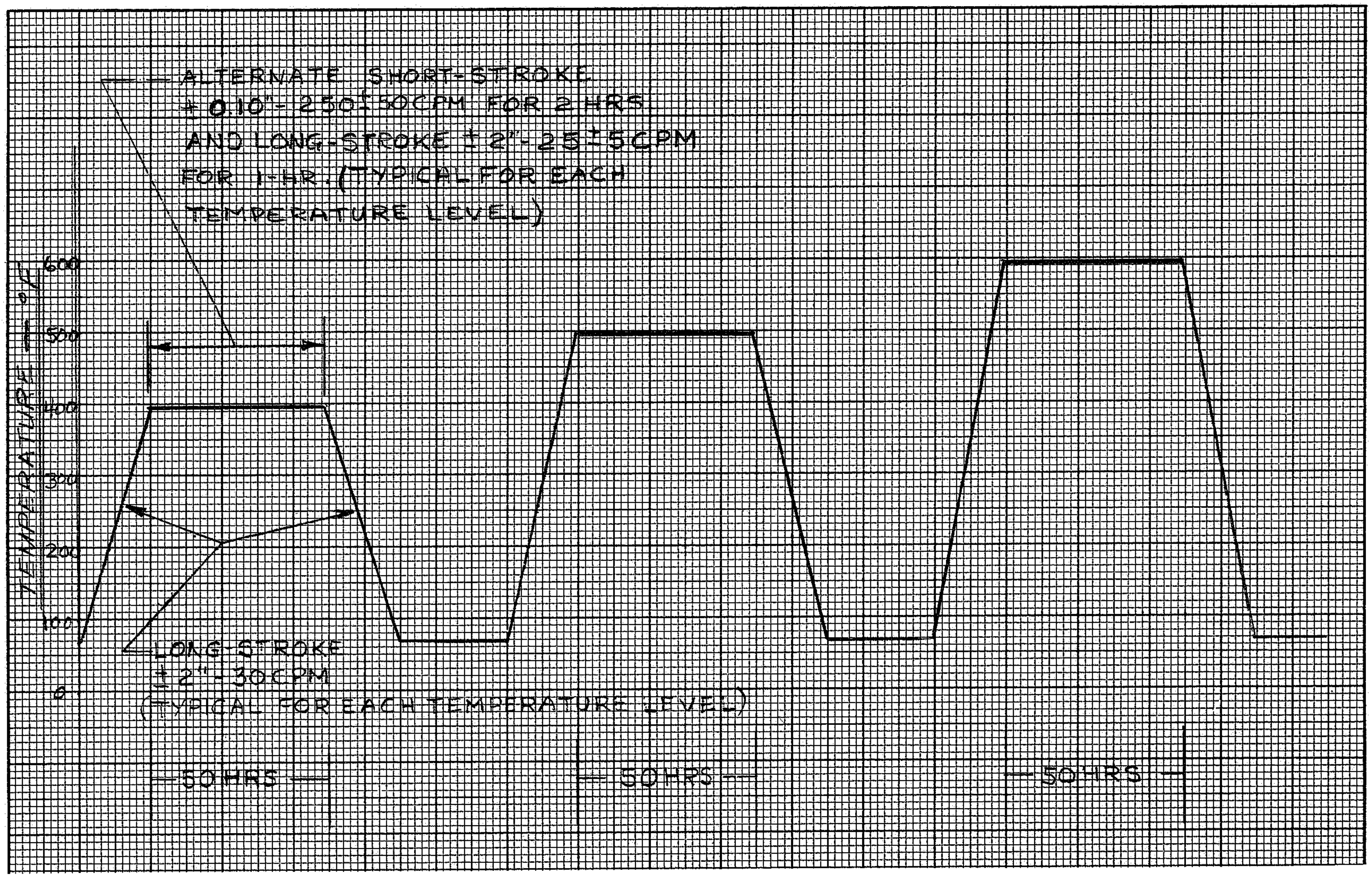


Figure 22. Test Profile - Low Pressure Seal Test

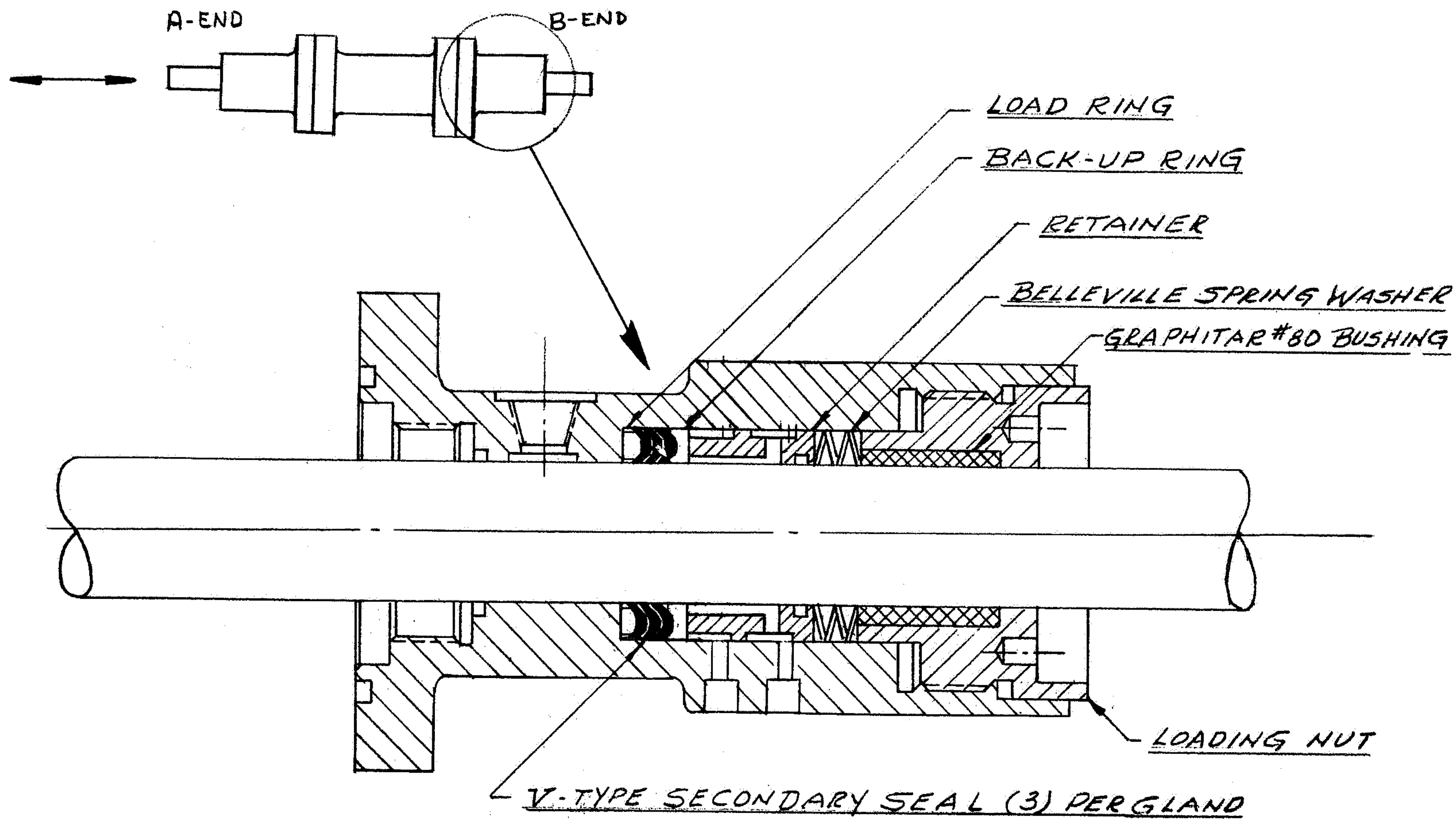


Figure 23. Polymer SP V-Seal Gland Configuration

ground, and lapped to a finish of 4 RMS. Hardness of the base material was Rc 52-53. The seal gland was fabricated from 17-4PH Cres. In the test configuration, 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.

Leakages from the static portion (outer diameter) and dynamic portion (inner diameter) of the seal were monitored separately to identify the source of leakage.

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. This was reflected in the total cycles obtained at 600°F. A summary of the results is tabulated below.

Low Pressure Test No. 1 - Polymer SP V-Seal
100 psi - F-50 Silicone Fluid

Temp.	Total Hours	Hours @ Temp.	Total Cycles	Cycles @ Temp.	Long Stroke Cycles @ Temp.	Short Stroke Cycles @ Temp.
400°F	61.50	50	641,563	621,960	29,686	592,274
500°F	67.50	50	657,292	629,988	25,348	604,640
600°F	56.75	50.2	559,487	446,094	27,953	519,591
Totals	185.75	150.2	1,858,342	1,798,042	62,987	1,716,505

The only measurable leakage obtained during the 400°F testing (61.5 hours) 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 portion of the B-end seal, respectively. This leakage was collected over a period of 67.5 hours. For the same time period, 2 cc of leakage were 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 .14 cc from the dynamic part 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. A review of the temperature records showed that fluid temperatures were between 610 and 620°F. It was then suspected that the increased leakage was caused by excessive differential expansion between the Polymer SP material and the 440C piston rod. Consequently, the test was continued to ascertain if this condition had occurred. During the subsequent operation, the fluid and ambient temperatures were gradually decreased to determine the effect this would have on leakage. At a fluid temperature of 440°F, seal leakage decreased to zero. The fluid and ambient temperatures were again brought back to 600°F, where zero-leakage operation was maintained on the B-seal for the remainder of the 50-hour test. However, leakage from the A-seal varied from 2 to 38 drops per minute.

Following a final check of the seal friction, the test actuator was disassembled (Figure 24) 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. Since 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 25).

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 26, 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 Company for analysis. Analysis of the coating by X-ray emission was inconclusive. However, the analysis did show that the scrapings

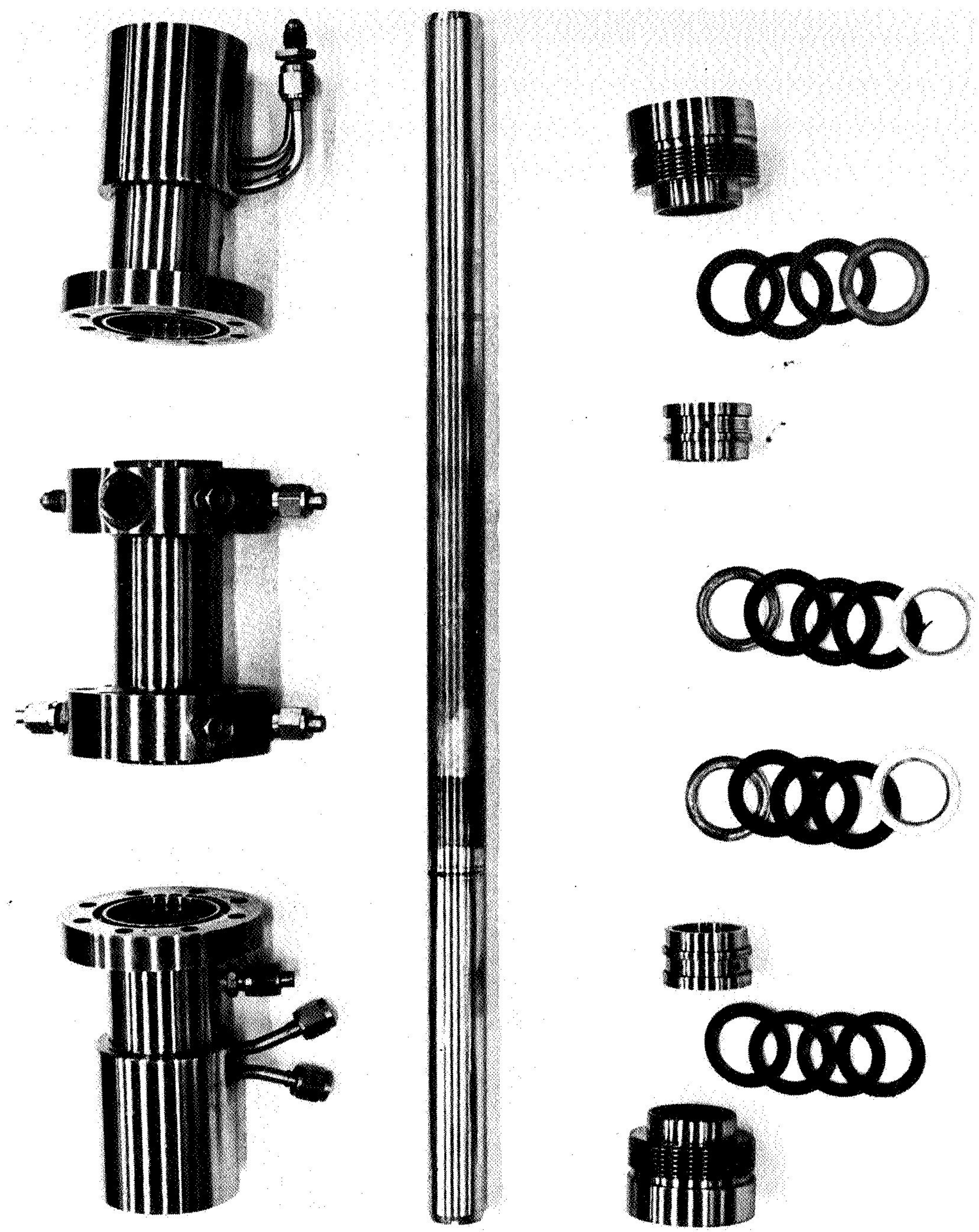


Figure 24. One-Inch Test Actuator - Polymer SP V-Seal Test

Figure 25. Rod End-Gland Bearing

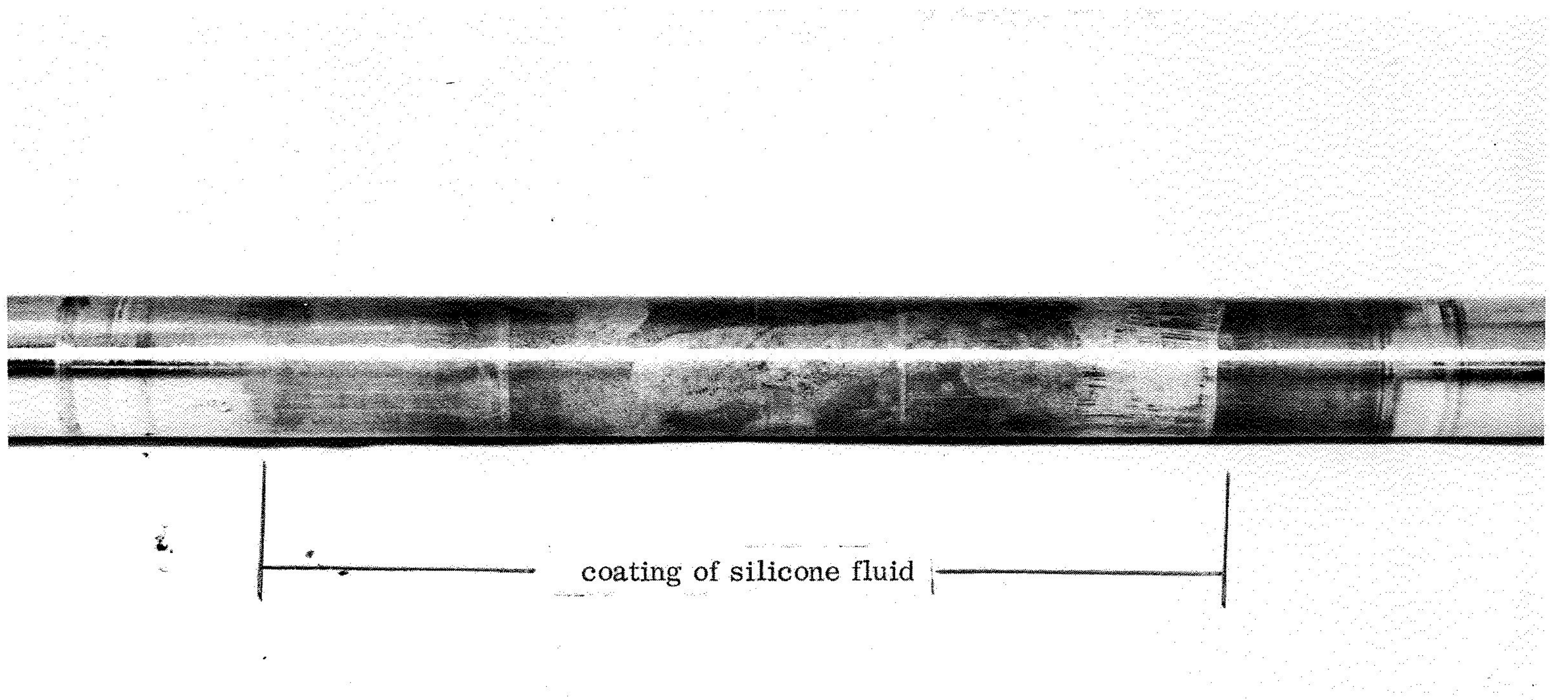


Figure 26. Piston Rod - Polymer SP V-Seal Test

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 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 3

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

	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

Inspection of the Polymer SP 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 27). 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 28) 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 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.

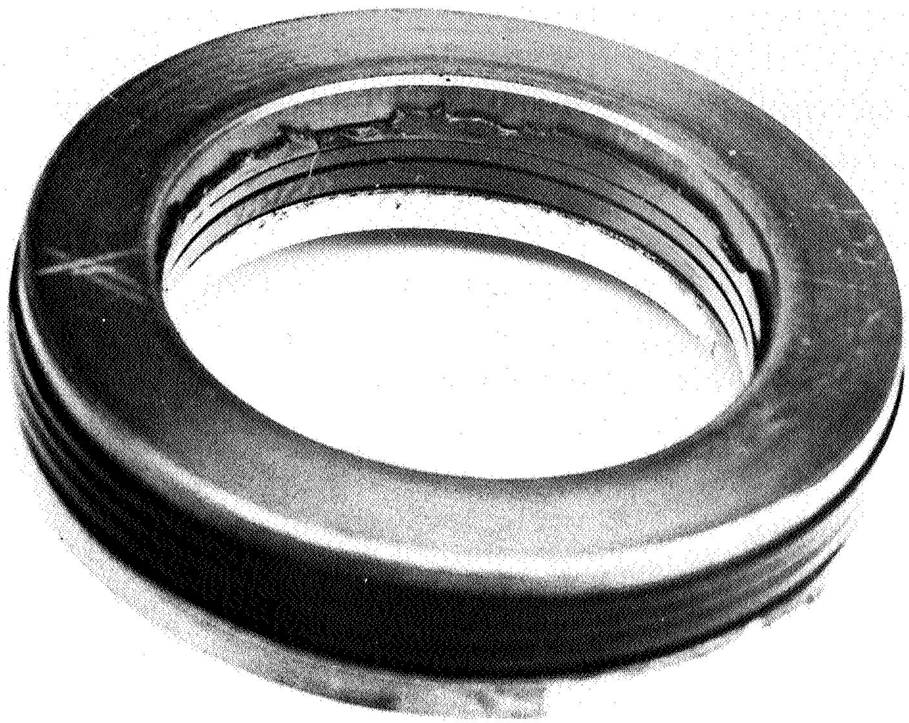


Figure 27. SP V-Seal - B-End

Figure 28. Polymer SP 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, since 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 29), 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 Polymer SP material. The shrinkage experienced at the inside diameter of the seal could be beneficial at room temperature since 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 differential expansion between the Polymer SP seals and the 440 C piston rod.

Figure 29. Extrusion of Polymer SP V-Seal - B-End

Thermal coefficients of expansion (in./in./°F) of Polymer SP 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.

C. LOW PRESSURE TEST NO. 2 - VASCOJET 1000 LIP SEAL

The test configuration and details of the seal are shown in Figures 30 and 11. The seals were fabricated from Vascojet 1000 and heat treated to obtain a hardness of Rc 50. The chrome-plated piston rod was fabricated of 440C Cres. and heat treated to hardness of Rc 52-53.

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. A summary of the test is shown below:

Low Pressure Test No. 2 - Vascojet 1000 Lip Seal
100 psi - F-50 Silicone Fluid

Temp.	Total Hours	Hours @ Temp.	Total Cycles	Cycles @ Temp.	Long-Stroke Cycles @ Temp.	Short-Stroke Cycles @ Temp.
400°F	65.5	50	497,305	468,231	24,949	443,282
500°F	20.75	15.5	150,885	144,085	7,891	137,594
Totals	86.25	65.5	648,190	612,316	32,840	580,876

As shown in Figures 31 and 32, the seals completed the 400°F operation with exceptionally low leakage. Total leakage during 65.5 hours of 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

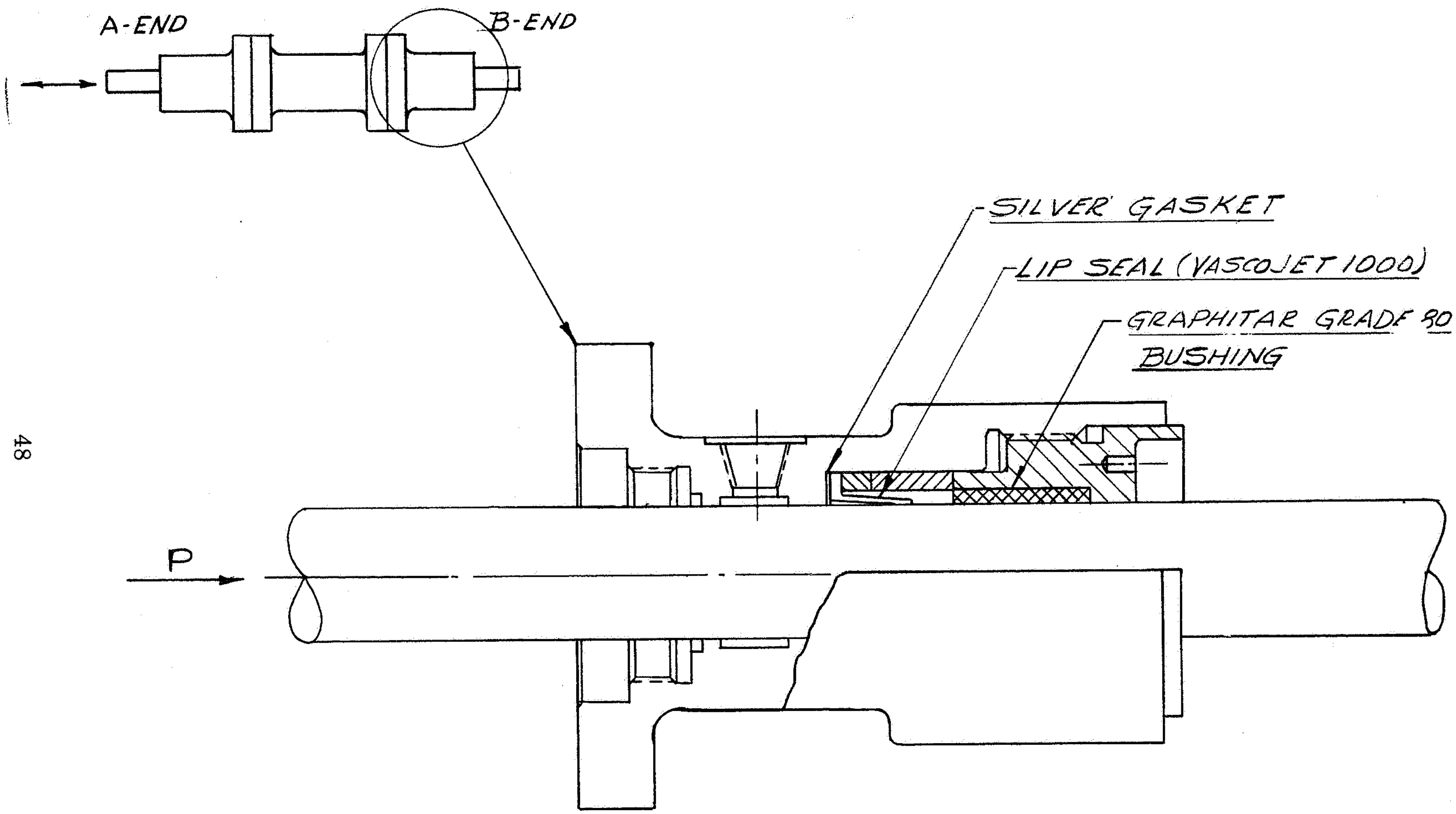


Figure 30. Vascojet 1000 Lip Seal Gland Configuration

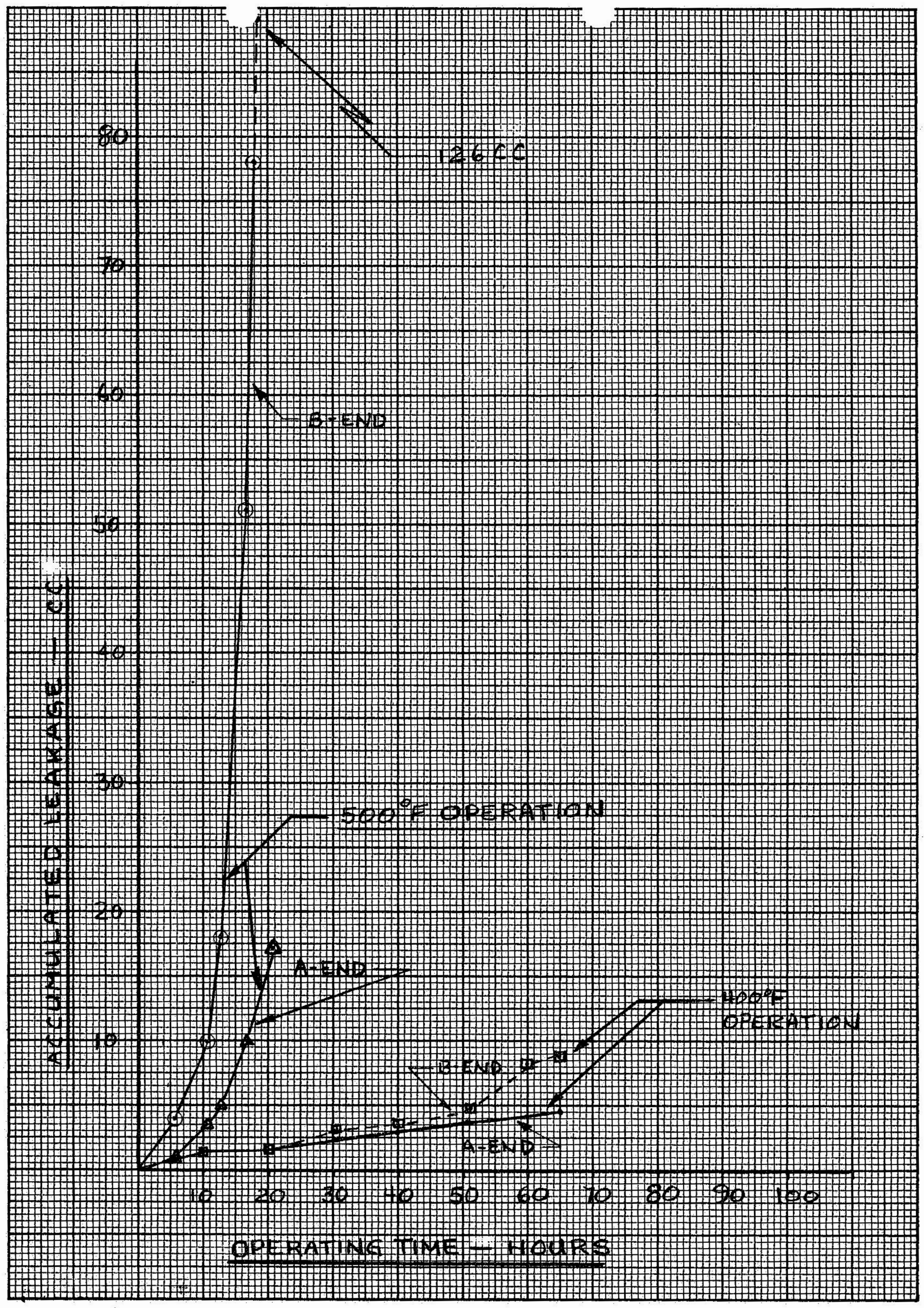


Figure 31. Accumulated Leakage Versus Time - Low Pressure Test No. 2 - Vascojet 1000 Lip Seal

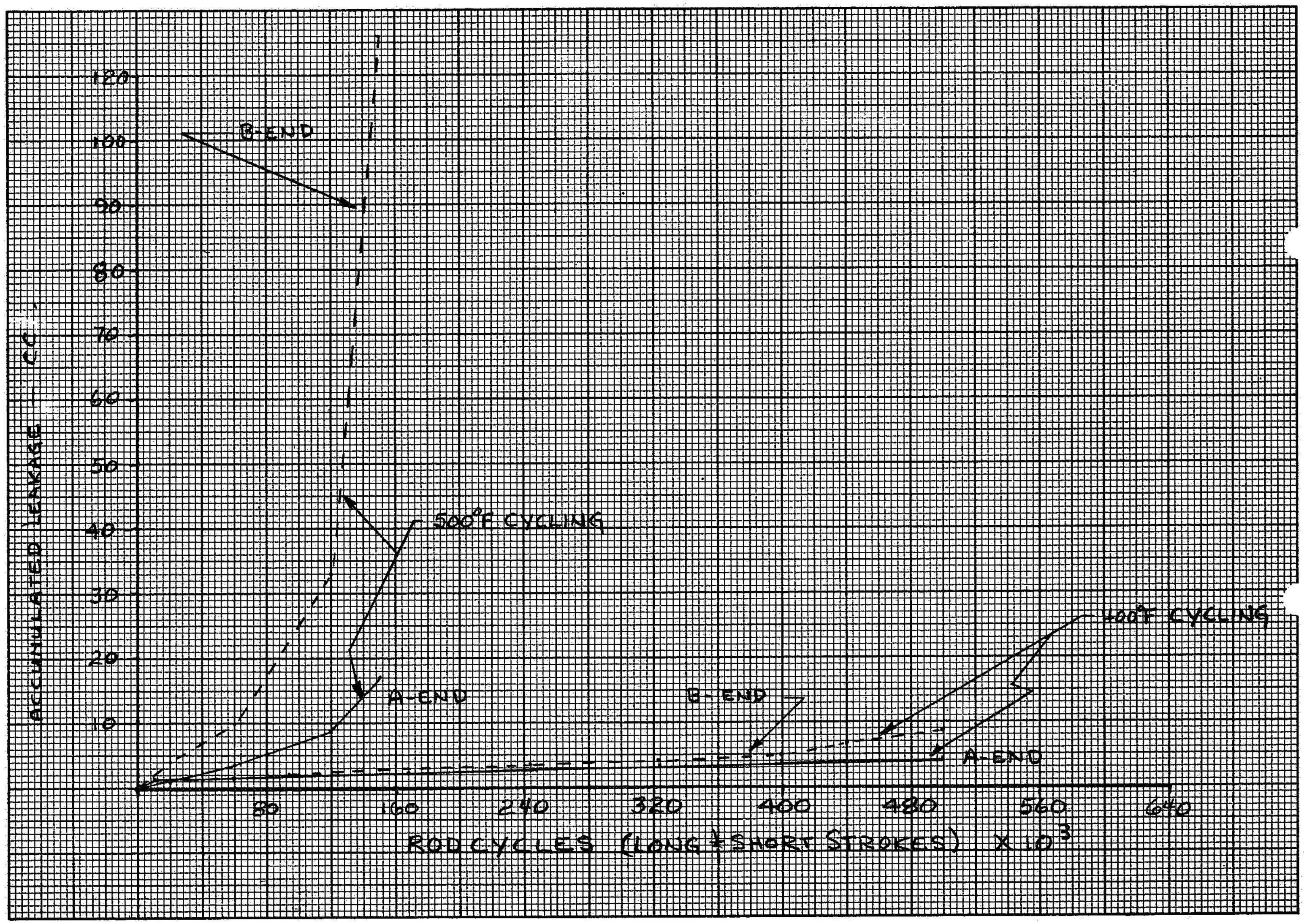


Figure 32. Accumulated Leakage Versus Rod Cycles - Low Pressure Test No. 2 - Vascojet 1000 Lip Seal

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 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 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 33) indicated that no catastrophic failure of the chrome plating had occurred. The chrome-plated rod (Figures 34 and 35) 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 36, the wear patterns generated

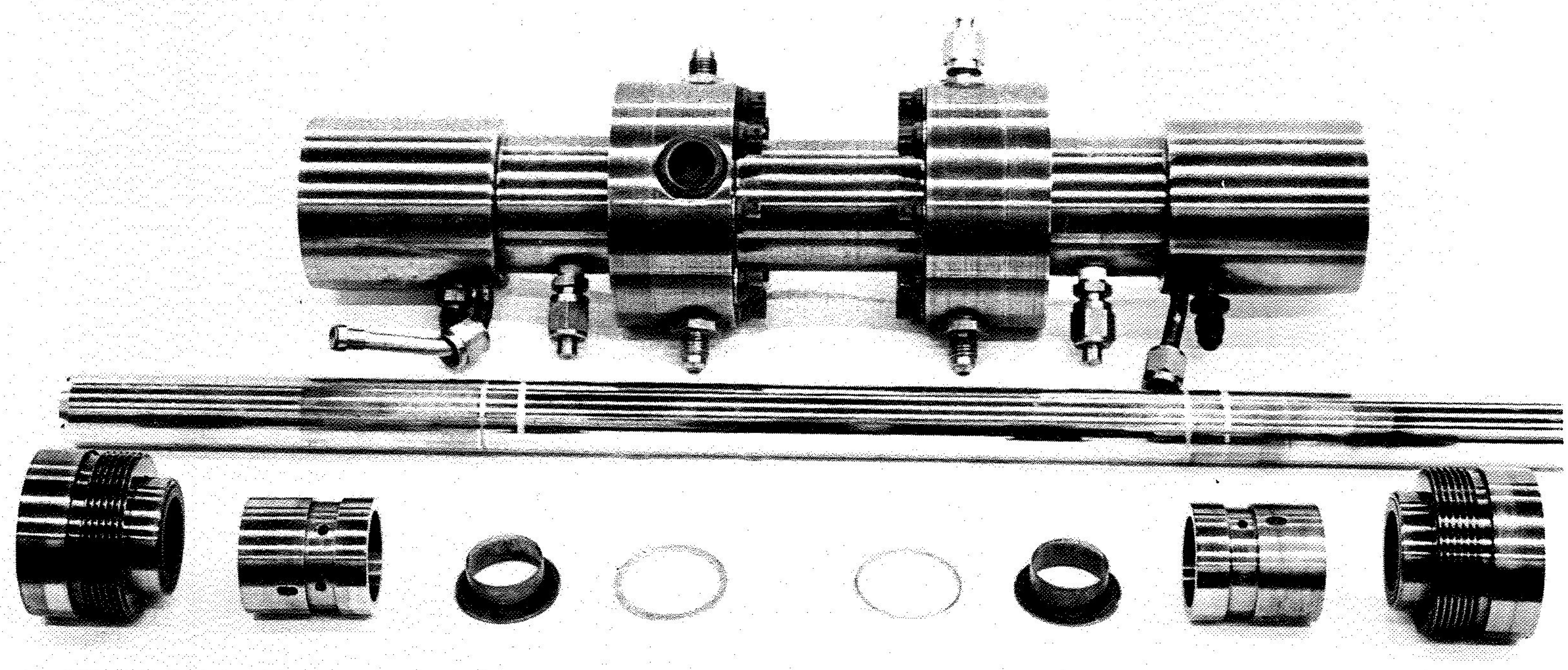


Figure 33. One-Inch Test Actuator - Vascojet 1000 Lip Seal Test

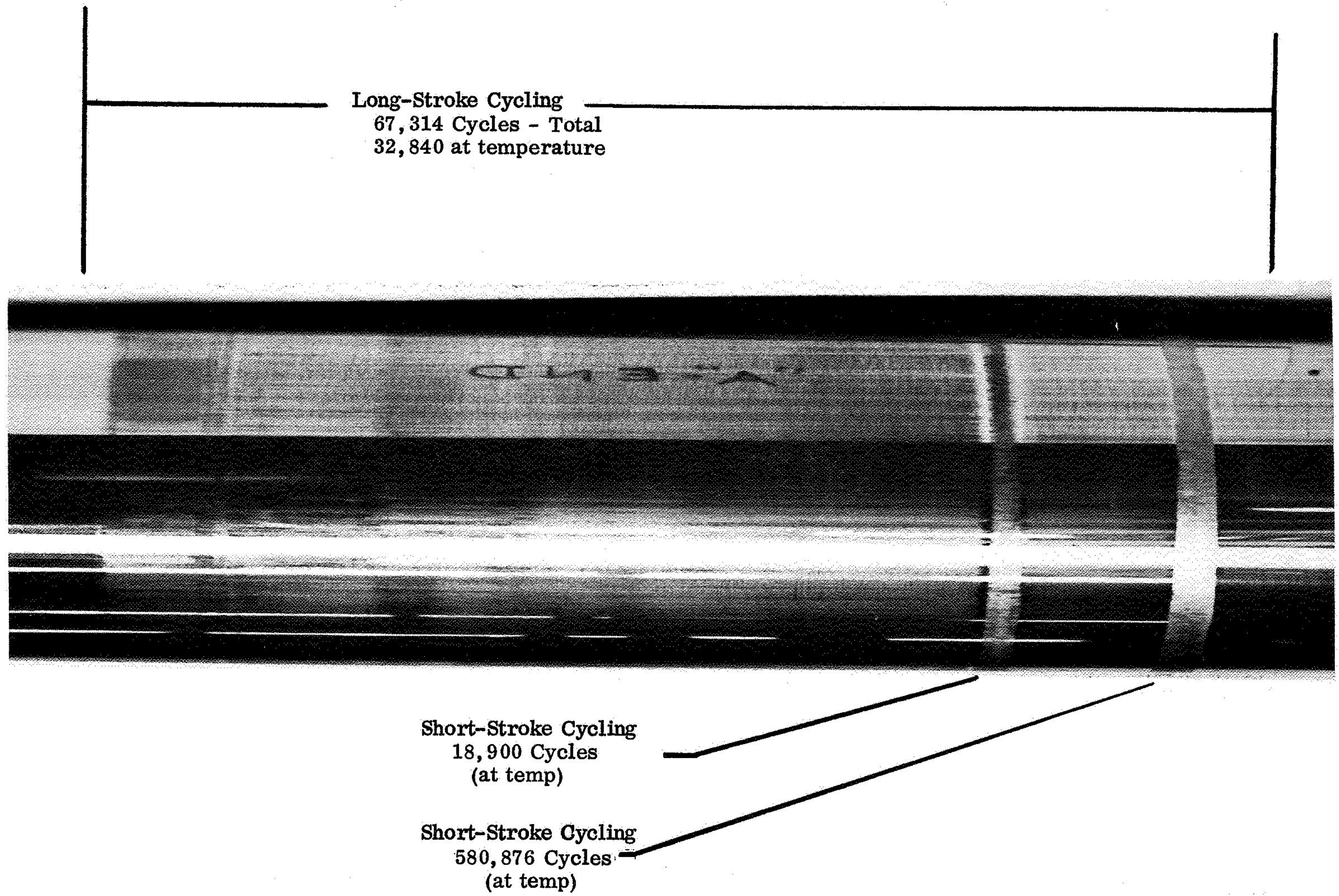


Figure 34. Piston Rod A-End - Vascojet 1000 Lip Seal Test

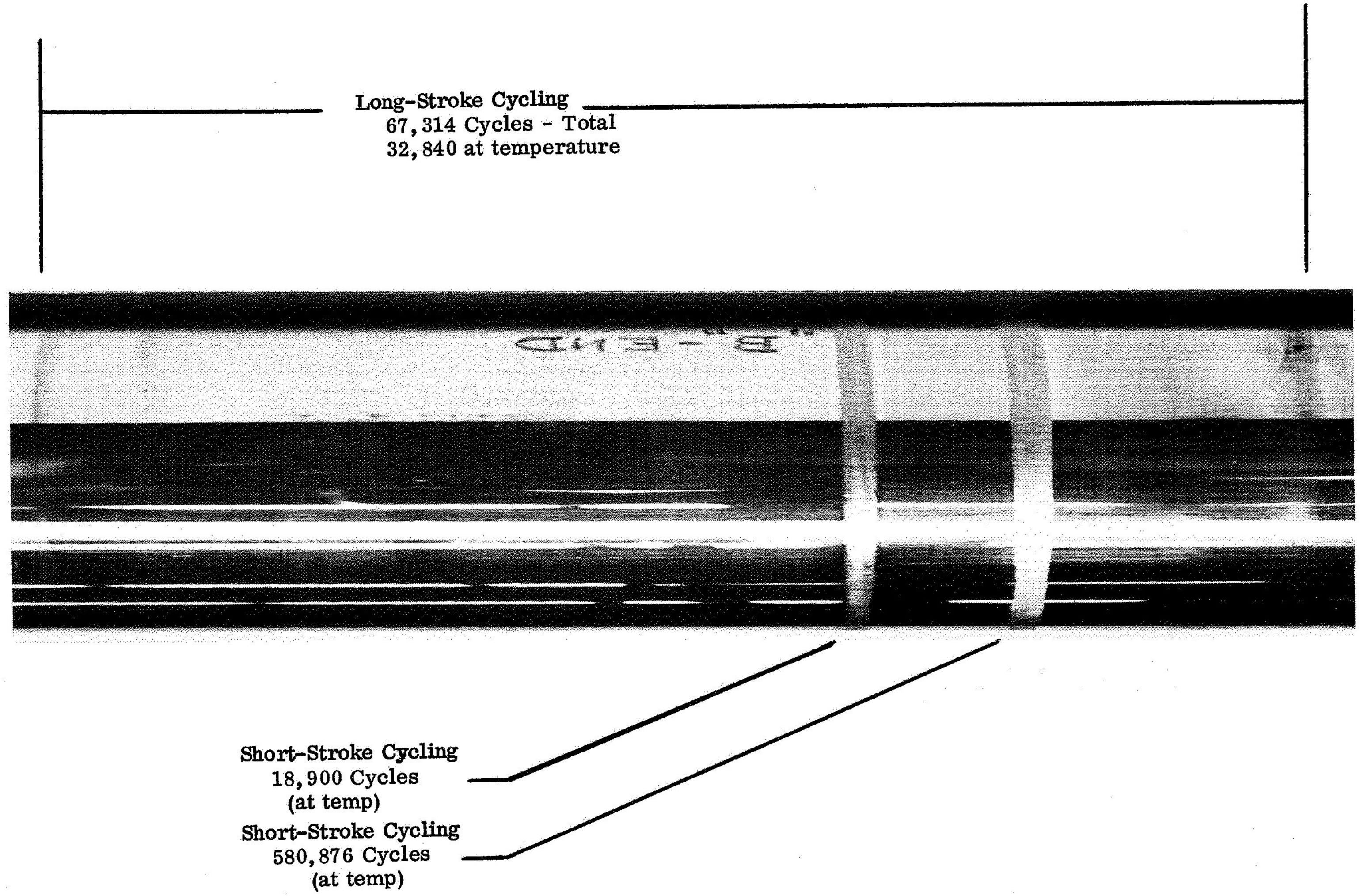
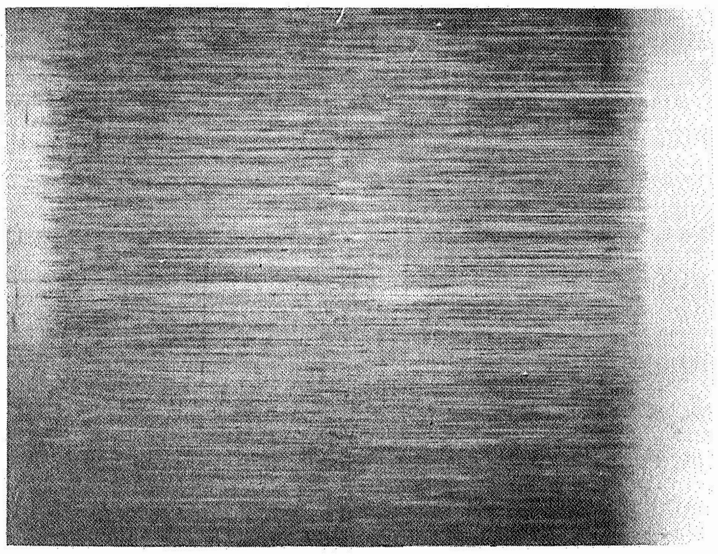


Figure 35. Piston Rod B-End - Vascojet 1000 Lip Seal Test



A-End 25X

580,876 Short-Stroke Cycles

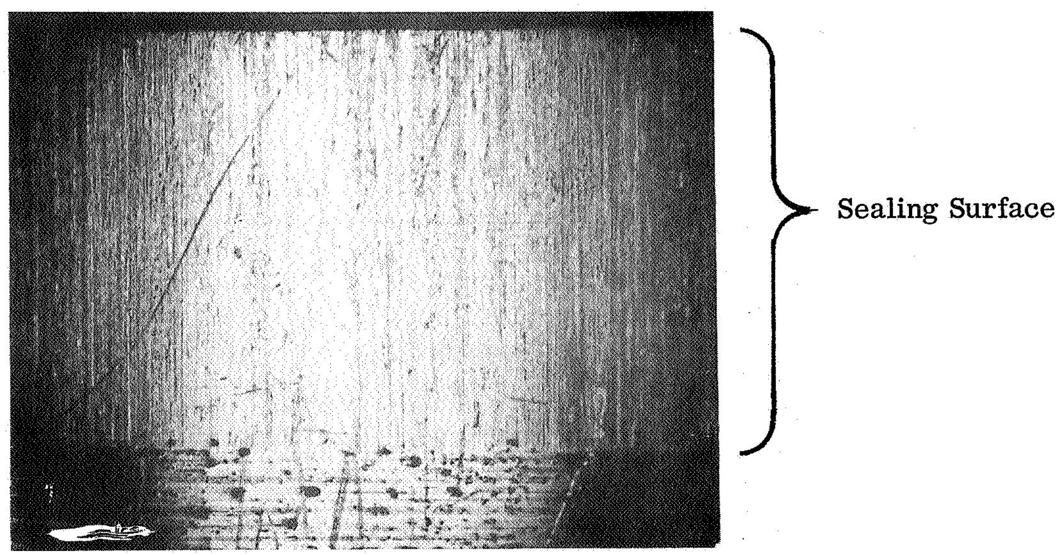
B-End 25X

580,876 Short-Stroke Cycles

Figure 36. Short-Stroke Cycling Wear Pattern on Piston Rod

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 37. Although good contact was achieved over the entire seal circumference (Figures 38 and 39), 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. A slight out-of-roundness of the seals due to heat treating may have produced this effect.

The effective contact pressure remaining on the seals was determined 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.



25X

Figure 37. Wear Pattern on Vascojet 1000 Lip Seal - B-End

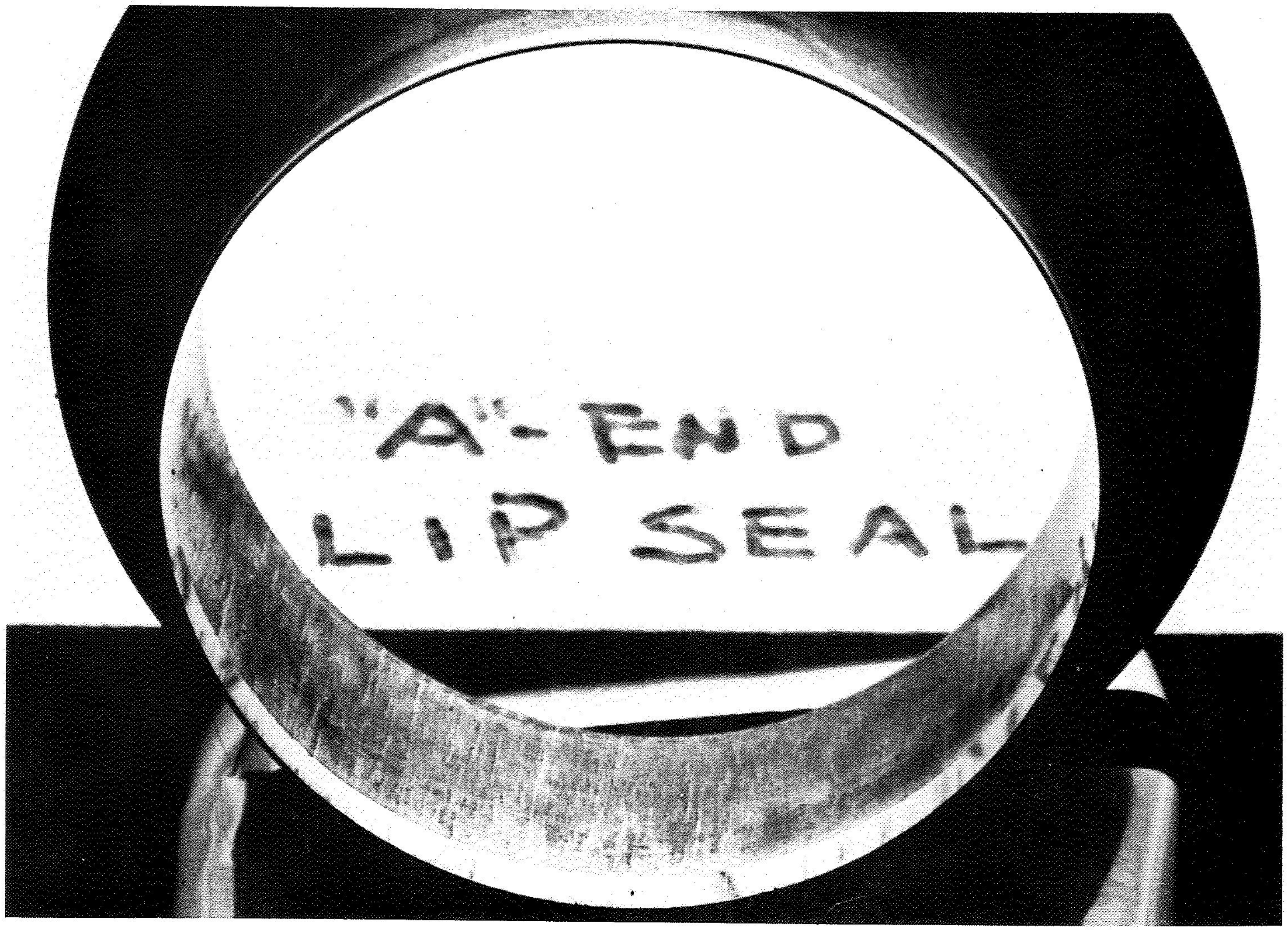


Figure 38. Vascojet 1000 Lip Seal - A-End

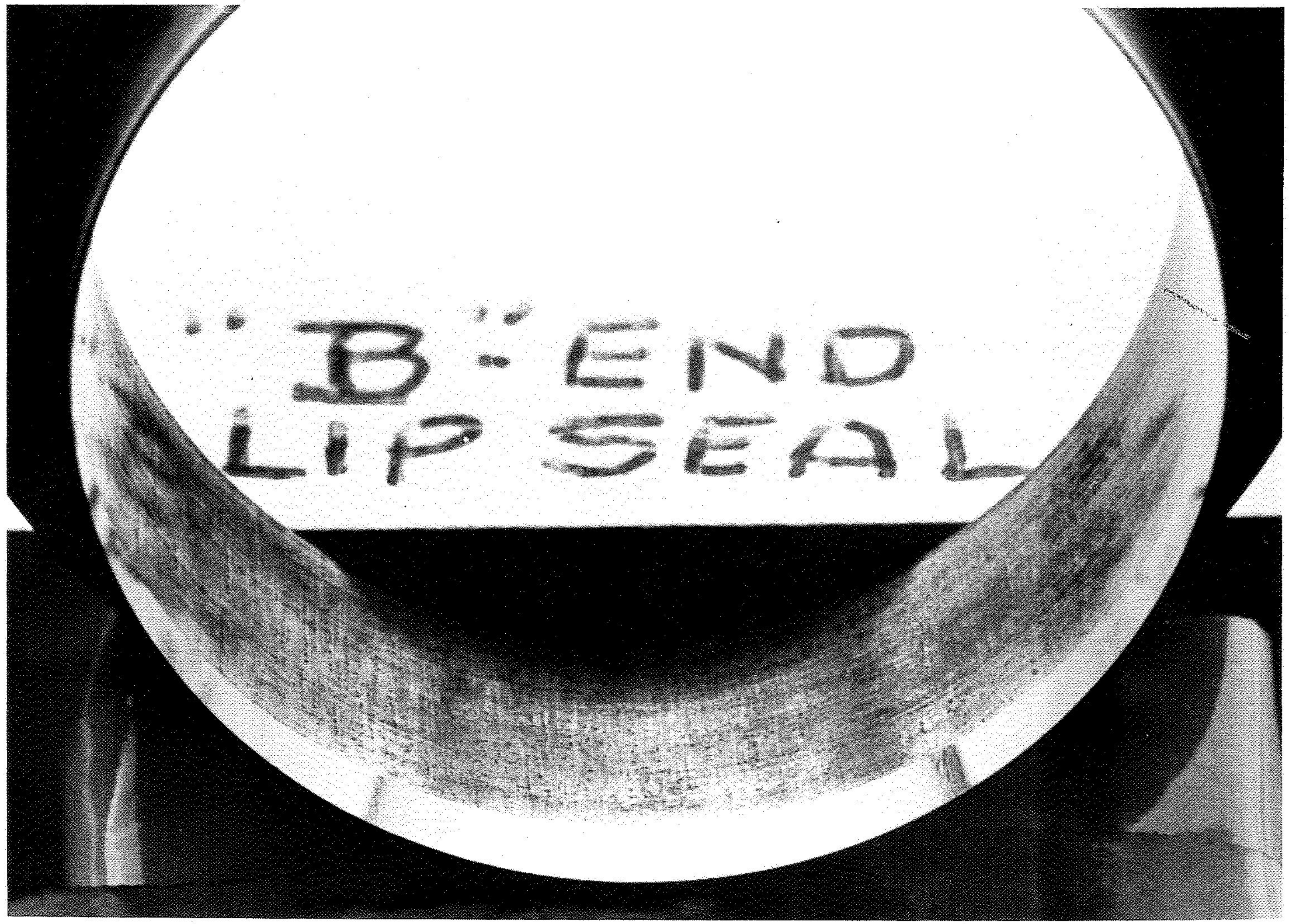


Figure 39. Vascojet 1000 Lip Seal - B-End

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 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.

TABLE 4

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

	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 Polymer SP 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 40, 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 41) 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

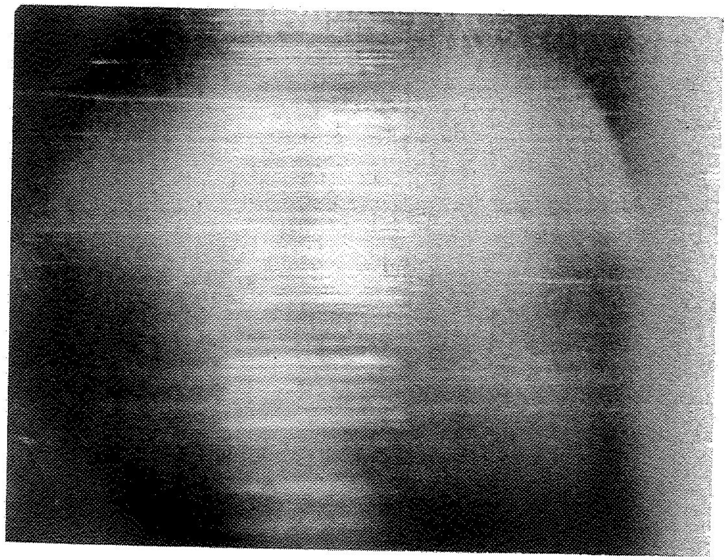


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Figure 40. Run No. 1 - Steady Cycling (300 cpm), 100 psi Fluid Pressure

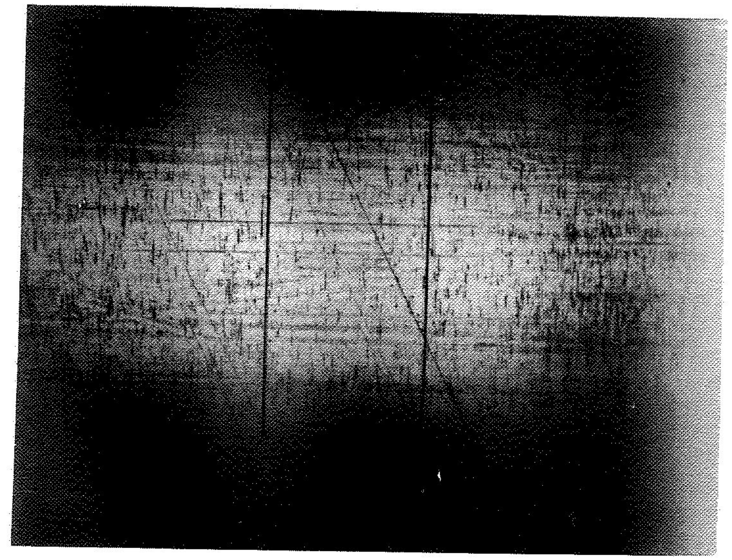

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Figure 41. Run No. 2 - Random Cycling (0 to 300 cpm), 100 psi Fluid Pressure

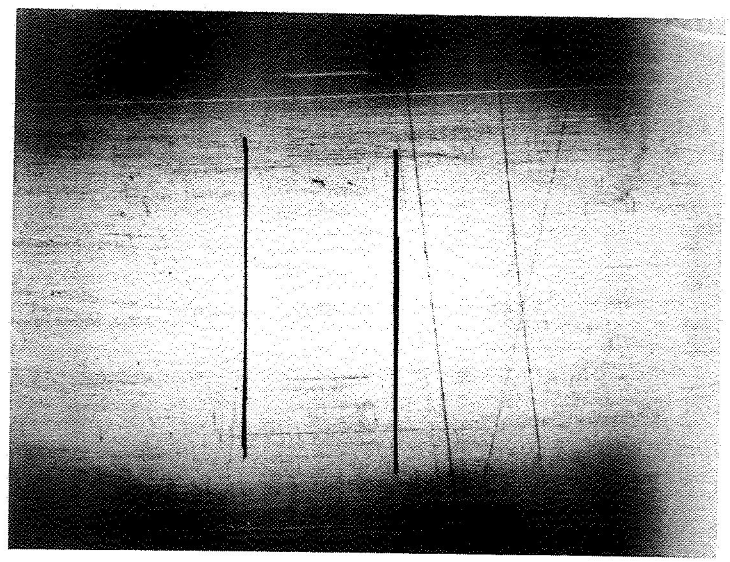

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Figure 42. Run No. 3 - Steady Cycling (300 cpm), 300 psi Fluid Pressure

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 42) 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 steady cycling at constant rate and amplitude. 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, which in turn provides a better lubricating film during subsequent operation.

D. LOW PRESSURE TEST NO. 3 - NICKEL FOAMETAL WEDGE SEAL

The gland configuration for this test is shown in Figure 20. Figure 16 depicts the design of the wedge seal. In the fabrication of the seal, the surface pores of the nickel Foametal (see Figure 43) 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

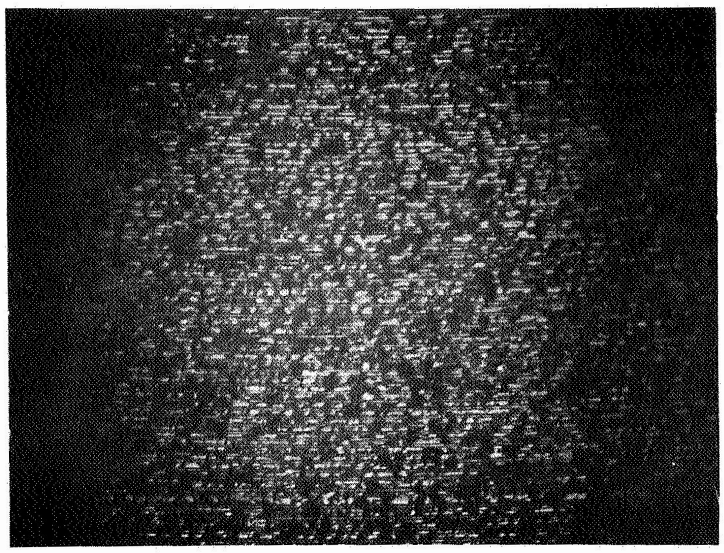

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Figure 43. Finished Surface on Nickel Foametal Wedge Seal

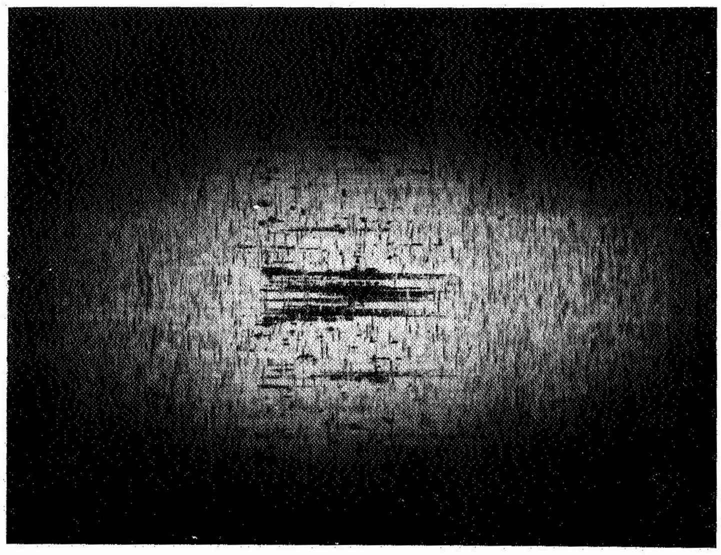

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Figure 44. Transfer of Nickel Foametal onto Piston Rod

pressure at the seal interface was approximately 830 psi. The seal friction resulting from these loading conditions was 50 pounds. Since only the A-end of the test actuator was reworked to accept the wedge seal, a set of Polymer SP V-seals was installed on the B-end.

Testing of the wedge seal was discontinued after 17.5 hours of operation when excessive leakage developed. A summary of the test is given below.

Low Pressure Test No. 3 - Nickel Foametal Wedge Seal
100 psi - F-50 Silicone Fluid

Temp.	Total Hours	Hours @ Temp.	Total Cycles	Cycles @ Temp.	Long-Stroke Cycles @ Temp.	Short-Stroke Cycles @ Temp.
400°F	17.5	7.5	100,248	79,648	4225	75,423

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.

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 44, some transfer of the nickel Foametal onto the piston rod was experienced. Since 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 that might minimize the effects of side loading would be to make the seal seat separate from the gland. This would enable 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 45).

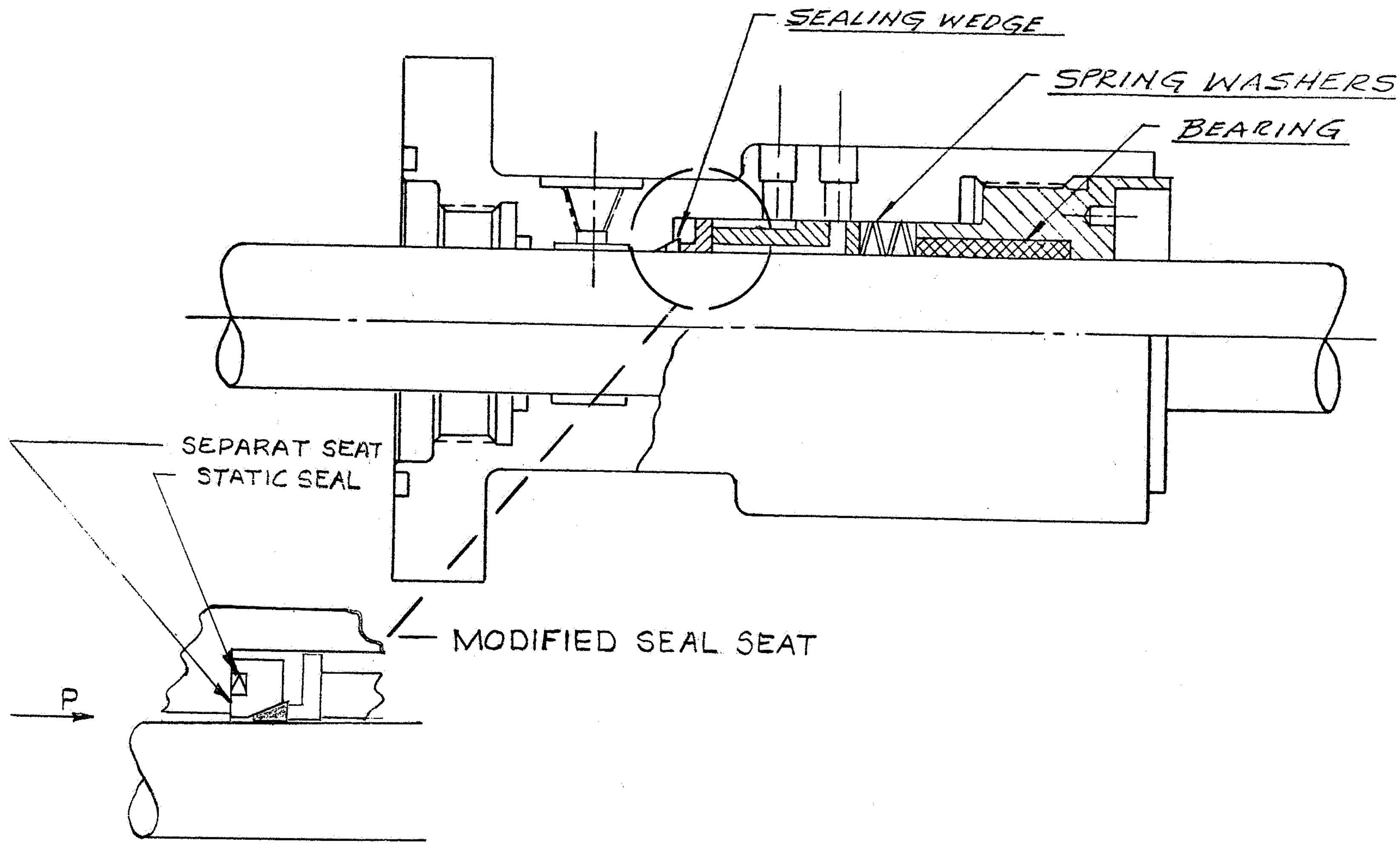

Figure 45. Modified Gland Configuration - Nickel Foametal Wedge Seal

E. LOW PRESSURE TEST NO. 4 - POLYMER SP V-SEAL, THREE-INCH SIZE

Testing of the Polymer SP V-seal in the three-inch size is in progress. The test actuator is shown in Figure 46. The gland configuration is similar to that shown in Figure 23 for the one-inch V-seal. Results obtained thus far are very encouraging. The seals have completed 50 hours at 400°F with practically zero leakage. A total of 469,900 cycles has been completed during this time period.

Further details of this test will be included in the report for the next reporting period.

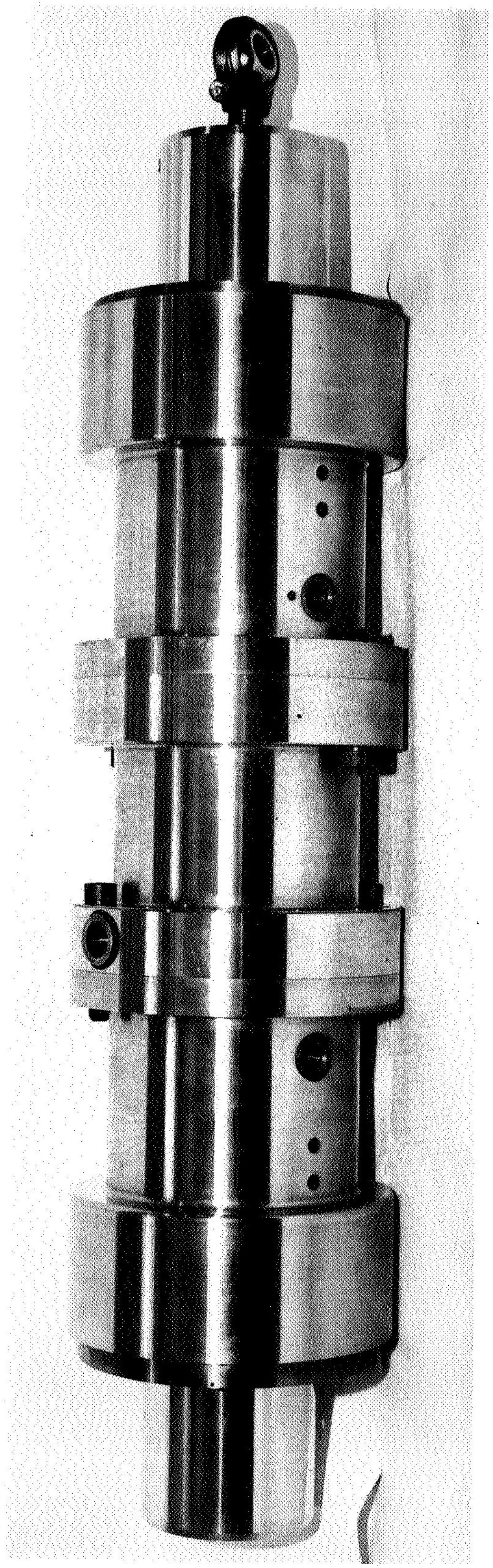

Figure 46. Test Actuator - Three-Inch Polymer SP V-Seal Test

FUTURE EFFORT

The schedule for the next six-month period of the program is shown in Figure 47.

<u>Program Activities</u>

1. Fluid-material compatibility testing
2. Low pressure testing of one-inch seals
3. Low pressure testing of three-inch seals
4. Selection of three materials for endurance test
5. Fabrication of seals for endurance test
6. Modification of rig for high pressure test
7. Endurance test - Task V
8. Design and test single-stage seal
9. Final Report

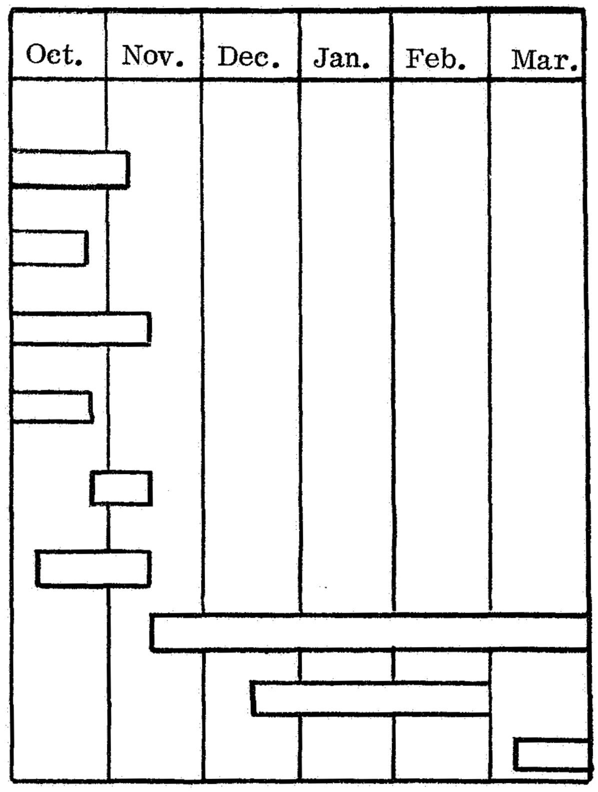

Figure 47. Schedule of Activities for Next Six-Month Period

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REFERENCES

1. Lee, J., "High Temperature Hydraulic System Actuator Seals For Use in Advanced Supersonic Aircraft," Fairchild Hiller (Republic Aviation Division) Report No. FHR 2702-2, 1966.

2. Lee, J., "High Temperature Hydraulic System Actuator Seals For Use in Advanced Supersonic Aircraft," NASA CR-54496, 1965.

APPENDIX A

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^{o}, 500^{o} and $600^{o}F$. 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^o, 500^o, and 600^oF. Operation shall alternate between operation at 30-40 CPM with $\pm$ 2 to $\pm$ 4 inch stroke and operation at 100-300 CPM with $\pm$ .05 to $\pm$ .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 B
SOURCE OF MATERIALS

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-1-0294 Silicone fluid	Dow Corning Chemical Co.