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NONELASTOMERIC ROD SEALS FOR ADVANCED HYDRAULIC SYSTEMS

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ABSTRACT

Advanced high temperature hydraulic system rod sealing requirements can be met by using seals made of nonelastomeric (plastic) materials in applications where elastomers do not have adequate life. Exploratory seal designs were optimized for advanced applications using machinable polyimide materials. These seals demonstrated equivalent flight hour lives of 12 500 at 350° F and 9875 at 400° to 450° F in advanced hydraulic system simulation. Successful operation was also attained under simulated space shuttle applications; 96 reentry thermal cycles and 1438 hours of vacuum storage. Tests of less expensive molded plastic seals indicated a need for improved materials to provide equivalent performance to the machined seals.

INTRODUCTION

The development of hydraulic systems for advanced aircraft and space applications requires use of materials and design concepts that are suitable in more adverse environmental conditions than exist in current applications. The higher fluid temperatures identified with these hydraulic systems preclude the use of many heretofore conventional seal design practices. The universal application of the elastomer to all hydraulic sealing applications is no longer possible and critical dynamic sealing requirements can only be satisfied using materials capable of long life at high fluid temperatures.

The material properties of advanced nonelastomeric (plastics) are acceptable for the entire range of Type III hydraulic system temperatures (-65° to 450° F) and considerably higher temperatures, making these materials prime candidates for experimental seal research for advanced applications involving two-stage linear rod seals for flight control surface actuators. NASA-initiated research was instrumental in the early development of a two-stage seal, using polyimides in exploratory tests to determine sealing characteristics under various operating environments (Refs. 1 and 2).

NASA-sponsored research conducted at Boeing extended the exploratory development of the two-stage seal to that of performance verification

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testing simulating actual high-performance hydraulic system requirements (Refs. 3 to 6). Testing of plastic first-stage seals was discontinued after results showed that the wear properties of machined polyimide and the strength properties of injection molded aromatic copolyester materials limited the life of the first-stage seals below that desired for the applications being investigated. This paper, therefore, describes the design, development and testing conducted on second-stage rod seals, made of both machinable and moldable plastics. The evaluation was particularly important because the material fatigue limit stresses of the plastic seals were in the same order as the imposed stresses that occur during the design life in advanced hydraulic system applications.

Performance measurement under such dynamic stress conditions, simulating requirements for both aircraft and manned space vehicles, was the first step leading to seal development for advanced applications having environments uniquely suitable for plastic materials. The program included analyses of basic seal designs and improvement of those designs based on recommended stress distributions. Seals were fabricated and tested to the fatigue environments of cyclic impulse and fatigue life typical of a supersonic transport or high-performance military aircraft and to similar requirements, plus the thermal cycling and vacuum exposure experienced in space shuttle applications.

DESIGN REQUIREMENTS

Flight control actuator requirements were investigated to establish seal design criteria because such actuation equipment receives the highest degree of time utilization in flight and is subject to the most severe environmental conditions. The general criteria, applicable to the most representative candidate primary flight control actuators having rod sizes nearest those available for the test evaluation, were used to establish the design and test parameters for the rod seals studied. The seal configuration acceptable for application was a continuously pressurized two-stage linear actuator rod seal with bleedoff to return between the first and second stages.

The hydraulic fluids considered during design and used during testing were a high-temperature polyolester, and MIL-H-83282, a synthetic hydrocarbon. The hydraulic test system was of a closed circulation loop design, operating normally over the temperature range of -50° to 450° F and at a nominal working pressure of 4000 psig.

The required flight life of advanced hydraulic components was established at 50 000 hr with flight control actuator overhaul periods at a minimum of 12 500 hr intervals. Seals were to be replaced during each overhaul; thus overhaul life was established as the endurance life for actuator seals. The mean or nominal cycle life for rod seals in flight control actuators during 12 500 hours between overhauls was established at approximately 8×10^{6} cycles.

A relationship was determined for the pressure environment to which the second-stage seal would be exposed during a maximum pressure surge created in an operational maneuver. This relationship was analytically simulated using the Boeing HYTRAN computer program and the mathematical model of an advanced airplane hydraulic system. Pressure conditions observed at the actuator inlet and at the second-stage seal in an actuator representative of the size available for testing are shown in Fig. 1. It was concluded from this analysis that there was no practical way to substantially reduce the second-stage seal pressure requirements below the maximum dynamic return pressure during a surge. It was, therefore, necessary to develop a polyimide seal to satisfy the 1450 psig dynamic requirement. The second-stage seal pressure impulse requirement was thus established as 200 000 cycles of 0 to 1500 to 0 psig with a rise rate between 25 000 to 35 000 psig/sec, based on the HYTRAN simulator runs.

Specific friction criteria for each seal size were based on the mean friction forces obtained during numerous Boeing research tests conducted during the Boeing/DOT SST development. These criteria were 8 lbf maximum friction force at the second-stage seal on a 1.0-inch diameter rod with a 200 psig upstream pressure and 30 lbf maximum friction force at the seal on a 2.5-inch diameter rod, under the same pressure conditions.

The external leakage allowable was 2 drops per 25 cycles at each end of an actuator at any temperature and pressure. A goal of zero leakage was established but was not justification for rejection of the seal if the goal could not be obtained.

SEAL DESIGN

The primary design objective of the rod seal evaluation was to optimize NASA second-stage exploratory seal designs and test the revised designs to realistic and existing requirements for actuators to be used in high-performance aircraft and space vehicles. The nominal geometric seal shape was extremely sensitive to the relationship between material stresses and the imposed hydraulic pressure loading. A stress analysis was therefore performed to determine the specific geometry, adaptable within available test actuators, that would tolerate the highest hydraulic pressure reversals occurring during dynamic operation. There were a number of design iterations leading to the final configuration of a tapered-leg, thickened-apex chevron design as shown on Fig. 2. The seal consists of four elements: two chevrons, the backup, and the strongback.

Material Considerations

Machinable polyimides were the only available plastic materials at the beginning of this program that had acceptable tensile and compressive strength. Selection of a material having a 15% graphite filler, was based on manufacturers published information and supplementary testing performed by NASA and Boeing.

Very little data were found to describe the polyimide material fatigue limit stress. Available data for the selected material were at a specific stress cycle with a zero mean as illustrated in Fig. 3. (A zero-mean stress was defined as equal magnitude tension and compression loading about a zero stress.) The stress cycle needed to be satisfied in the design application was imposed on a tensile-mean stress. The difference between the maximum and minimum values in the design stress cycle from Fig. 3 were significantly less than the material fatigue limit stress, although the maximum design tensile stress at 350° F exceeded the material limit for this temperature. Because the design stress amplitude was only 28% of the material limit and the maximum stress exceeded the material limit in tension by only 9%, it was concluded that a seal made of machinable 15% graphite filled polyimide could meet the high-temperature cyclic stress requirements of the application.

The large differential between coefficients of thermal expansion for the steel actuator, 7×10^{-6} in./in./$^{\circ}$F, and for the polyimide in the seal, 23×10^{-6} in./in./$^{\circ}$F needed to be considered in the seal design so that leakage paths would not be introduced at the seal ID at high temperature and at the seal OD at low temperature. The magnitudes of these effects are illustrated in Fig. 4, showing the free dimension relationships between the seal and cavity if the full effects of differential expansion and aging or shrinkage were allowed without compensation. Heat treating of the material during fabrication eliminated most of the effect aging during the testing phase.

Stress Analysis

The stress analysis of the chevron seal was treated using a finite-element plate analysis which represented the elemental interactions within the chevron seal. The definition used for each of the finite elements of the chevron seal is a plate as described in Fig. 5. The Boeing SAMECS computer program used the direct stiffness (displacement) method to perform the required analyses. This method of analysis evaluated the response of individual plates, or elements, and subsequently combined those responses to produce compatibility within the structure. Unknown rotations and displacements of the seal were determined using the matrix form of the following equation:

$$[\rho] = [K]^{-1}[F]$$

where [K] was the stiffness matrix, [F] was the force matrix, and [ρ] was the matrix of unknown displacements.

The predominant stresses encountered in the finite-element analysis of the chevron seal were meridional bending stresses. These meridional bending stresses were caused by interaction forces between the chevron legs and boundary reactions at the seal gland OD and ID. Gland depth did not greatly affect the stress.

The freebodies in Fig. 6 illustrate the predominate forces and reactions to which a two-chevron configuration was subjected. Hoop stiffness (K_{hoop}) and beam bending stiffness were interrelated such that, as rod diameter became larger, the load was reacted by increased beam bending. For rod diameters of 1.0 inch and larger, the dominant influence was beam bending. For rod diameters less than 1.0 inch, the hoop stiffness may become dominant due to the lower tensile stresses in the parallel grain of the polyimide material.

Another parameter important to stress analysis was the curved-beam correction factor for bending stresses. Because there was a nonlinear distribution of stresses due to curved-beam bending, correction factors, K_i and K_0, had to be applied as indicated in Fig. 7. The R/c value for the exploratory seals was set at a minimum of 4.0. This value was accepted as a minimum objective for design to keep the inside fiber stresses from exceeding practical limits for the material.

Fig. 8 illustrates, in summary form, the iterations of the stress optimization studies performed for the 1.0 in. rod application. Analysis was initiated with a gland depth assignment, curve 1. Optimizations were then conducted for the effects of leg thickness and leg angle, curves 2 and 3, showing favorable reductions in the tensile stress. This result was not evidenced in the compressive stress analysis. Leg angle optimization, curve 3, showed a large increase in compressive stress. This effect was unavoidable and important information for the analyst to use in designing the backup block, part 1, Fig. 2.

Chevron Final Design

The final design for the chevron seal assembly of Fig. 2 was completed, utilizing the advantages of opposing pressure and preset loads to offset the disadvantages of large presets and provide seal designs that required no loading springs.

The downstream chevron element was rigidly supported by a rounded backup block through contact at the chevron apex. This support produced a less severe stress condition in the downstream chevron element than was present in the upstream element, which was not similarly supported. By placing a "strongback" or rigid body between the downstream and upstream chevron elements of a two-element assembly, the stress distribution in both elements was made similar. Designs for a small, 1.0 in., and a large, 2.5 in., diameter rod assembly, using this rigid supporting technique for both elements, were developed as follows.

Preset for the small-size chevron seal was assigned to provide a minimum of 0.001 in. interference fit at both the inside and outside legs of the two chevron elements at any temperature condition. Maximum preset at the extremes of the temperature range was 0.005 in. in a 0.121 in. gland, consistent with MIL-G-5514.

It was found that no realistic stress solution existed for a chevron with a leg thickness that was capable of resisting 0.005 in. preset and 1000 psi pressure loading without drastically exceeding the ±4.30 ksi allowable stress envelope at 350° F. Acceptable solutions had different leg thicknesses for the deflection and pressure conditions, indicating that a tapered leg chevron was required.

Computer solutions using the finite-element analysis were obtained to evaluate the model shown in Fig. 9 for the region of acceptable configuration design. A limitation of 0.015 in. minimum thickness at the leg tip was established, so that there would be sufficient thickness to support shear forces imposed by the cutting tools during fabrication. The thickness at the apex was required to be 1.2 times the leg thickness at the tangent, based on prior analysis that this ratio would not overstress the apex at maximum preset when the stresses at the tangent were acceptable.

The results of the computer analysis are shown in Fig. 10 for the region of solutions acceptable for the envelope of stress allowables. An evaluation of the results was made to reach a practical solution for the 1.0 in. rod application. A minimum of 20° for the leg angle was established as a compromise to keep the leg length within practical limits. Lower angles required high taper ratios, which increased the chevron contact foot area. A limiting value occurred at 15°, where the entire leg was in contact with the gland.

At a 20° leg angle there was no theoretical solution within the ±4.30 ksi stress envelope. The best practical solution provided stresses within -4.80 to 3.70 ksi, using a chevron tangent point thickness of 0.025 in. This was considered a sound compromise because: (a) the deviation from either the preset or pressure curve was small and judged to be within the conservatism of the material limit, and (b) the accuracy of dimensional control during fabrication could not guarantee the tolerance applied to the angle of crossover between the theoretical preset and pressure curves shown.

An analysis similar to that described above was independently performed to determine the region of acceptable design for the large, 2.5 in., rod application, using a two-element chevron assembly separated by a strongback.

K-Section Seal Design

A design for a K-section seal, similar in concept to the chevron configuration, but incorporating a backup section as an integral part of the sealing element, was initiated independent of the chevron analysis.

Stress considerations were based on the K-section being comprised of a stiff vertical member of constant cross section and two independent, tapered legs. As such, each leg was a flexible cantilever beam, having a specific taper ratio. The loading condition on each leg was composed of a pressure component and a preset component in similar fashion to the chevron leg loads. The pressure-induced load was the same as the load induced in a chevron leg having the same dimensional parameters; however, the K-section leg reacted with greater rigidity to preset loading, because there was no interaction between legs of the K-section.

The reduced flexibility in the K-section legs, due to preset loading, was indicated in the stress sensitivity to the distribution of thickness along the tapered leg. The optimum solution was to have a sealing leg profile with the shape of an ogee curve. The impracticality of such a design required a compromise to make the K-section leg the same as used in the chevron design, with a rigid vertical beam as an integral part of the apex geometry. Fig. 11 is the final K-section design for the small 1.0 in. size seal.

SCREENING EVALUATION OF MACHINED POLYIMIDE SEALS

Second-stage chevron and K-section seals in both 1.0 and 2.5 in. diameter sizes were manufactured by Boeing from machinable polyimide material.

Screening tests were developed to show the differences between the alternate seal candidates and provide data that would show a quantitative measure of the potential for the seals under stress environments, typical of an advanced hydraulic system application. The screening tests selected were an impulse test to evaluate the structural integrity of the seal cross section, and a friction test to evaluate the forces that contribute to inefficiency of the hydraulic actuator. The results of the screening tests were used to select seal candidates for endurance life cycling.

Impulse Test Results

The impulse test for the second-stage chevron and K-section seals consisted of 200 000 cycles per the profile of Fig. 12. Cycles were accumulated with 40 000 at 100° F, 115 000 at 275° F, 40 000 at 350° F, and 5000 at 400° F. Leakage was measured as cycles-per-drop against an allowable of one drop in 1050 cycles. The most severe leakage observed during testing was one drop in 4368 cycles, obtained at 350° F. This was less than 1/4 of the allowable.

Examination of the seals after impulse testing revealed superficial structural cracks in the legs of some sealing elements. Despite these cracks, all seals retained the ability to provide fluid containment during a test more severe than expected in normal service. There was no evidence of chips or broken pieces from the seal parts.

Friction Test Results

Tests of both static and dynamic friction were conducted to determine the quantitative relationship between the friction forces for both sizes of chevron and K-section configurations of second-stage seals. Data of Fig. 13 shows that the effect of temperature is pronounced with the larger size seal and relates to the increase in normal force produced by the differences in coefficient of thermal expansion between polyimide and steel. The temperature effects on breakout friction were determined to be insignificant resulting in linear breakout friction slopes of 0.5 lbf/psi for the 2.5 in. diameter seals and 0.125 lbf/psi for the 1.0 in. diameter seals. The friction force of the 2.5 inch seal did, however, exceed the desired design friction criteria of 30 lbf at 300° F.

ENDURANCE EVALUATION OF MACHINED POLYIMIDE SEALS

The objective of the endurance test was to provide data on the life of plastic seals in a typical fatigue environment for flight control actuators powered by a high-performance hydraulic system. The endurance tests were conducted with two actuators, simultaneously operated against torque loads of flight control surface magnitude. The K-section seal was tested in the larger actuator and the chevron seal in the smaller actuator. This selection was made, based on the stress analysis that showed the K-section to be more critical than the chevron in the larger size.

A test of four sequences of 770 000 cycles each, at 350° F was conducted. Each sequence consisted of 750 000 cycles at 2% load-and-stroke with 20 000 cycles of a mixture of 25, 50, and 100%, load-and-stroke cycles interspersed.

The average leakages obtained during testing are shown on Table 1 and were all within the allowable of 2 drops/25 cycles, except during 100% stroking with the smaller actuator. The leakage during this 100% stroking condition should not, however, be considered by itself, since a typical flight profile contains only a small portion of 100% strokes. The overall mean for leakage during the entire endurance test approached that of the short-stroke leakage. Post-test inspection of the second-stage seals from both actuators showed no abnormal wear patterns, no evidence of cracking, and the polished contact areas as expected on the inside and outside diameter faces.

Development testing of the 2.5 in. K-section and 1.0 in. chevron second-stage rod seals was continued by extending endurance cycling tests to investigate sealing capability beyond the design temperature of 350° F, and to the upper temperature limit of a Type III hydraulic system, 450° F.

A total of 3.85×10^6 cycles at 400° F with an additional 1.925×10^6 cycles at 450° F were accumulated, representing 9375 equivalent flight hours of an advanced aircraft. Load and stroke conditions that were used in previous testing were repeated in the same proportions.

Leakage measurements are shown on Table 2. There were no unexplainable conditions where leakage was in excess of 2 drops/25 cycles. During one change between short and long stroke cycling at 400° F, residue on one rod, due to thermal decomposition of the fluid film, caused erratic leakage as the rod was retracted through the seal. Subsequently, such residue was manually removed prior to changing stroke conditions. Average leakage data was not truly representative because data taken during many measurement periods showed no evidence of leakage.

Testing at 450° F resulted in a much faster formation of fluid decomposition residue on rods. This residue affected the consistency of leakage measurements and could not be effectively removed by scrapers. As evidenced during testing at 400° F, the average leakage measurements were not representative. Satisfactory seal performance was demonstrated by the ability of the seals to contain fluid within allowable limits under the most severe load/stroke conditions, during both the beginning and ending sequences of cycling. Inspection of the seals following the 400° to 450° F testing showed fluid decomposition residue on the seals. There were no cracks or irregularities and no unusual wear.

The testing completed adequately showed the fatigue life of the second-stage seals to be greater than expected based on the material fatigue limit stress used in the design analysis.

SPACE ENVIRONMENT EVALUATION OF MACHINED POLYIMIDE SEALS

The second-stage K-section seal in the 2.5-in. diameter size was further tested to evaluate its performance in simulated space environments involving thermal cycling and vacuum exposure. The tests consisted of alternating thermal cycling and vacuum tests to simulate repeated reentry heating and space vacuum storage while on station. Fig. 14 shows a typical heating cycle. Thermal cycling was conducted between room temperature and 400° F, with intermediate cooling between cycles to 150° F. The actuator was cycled at 1.0-in. stroke and 0.6 Hz except during the first 1/2 hr of heating in each thermal cycle, during which time the actuator cycling was at 0.25 Hz. All actuator cycling was under simulated load representative of space shuttle actuator reentry conditions.

The test seal completed 96 thermal cycles with no failures or deterioration in seal performance. The maximum leakage observed during any 5-min observation period was one drop. This corresponds to 180 actuation cycles per drop of leakage. The seals thus showed a level of performance 14 times better than the allowable.

The vacuum environment tests were conducted to evaluate the suitability of the polyimide K-section rod seal for space shuttle applications requiring seal reliability during an extended period of continuous vacuum exposure. In a typical space shuttle application, the actuator rod seals would be exposed to periods of cyclic rod actuation alternated with periods when the actuator would be stowed or locked with the seals remaining pressurized. The tests simulated this environment to assess seal leakage variation as a function of changing actuation stroke and fluid base. Tests were performed with MIL-H-83282, a synthetic hydrocarbon hydraulic fluid (7) and with MIL-H-5606, a petroleum-base hydraulic fluid (8).

All vacuum testing was completed with no failures or deterioration of the test seals. The seals were exposed to a total of 2491 hrs under hard vacuum, 583 hrs of these with the actuator rod being stroked for a total of 472 571 extend-retract cycles.

Seal leakage was measured both during rod actuation and while the rod was in the stowed (retracted) position during overnight and weekend periods. Results shown on Table 3 indicate sealing performance during actuation to be better than the allowable of 2 drops per 25 cycles. Leakage during pressurized stowage averaged 2.2 drops/hr for overnight periods and 0.2 drops/hr for weekend periods, indicating that long periods of inactivity assisted in reducing leakage. Adequate sealing was accomplished under all conditions of testing with both of the fluids used.

EVALUATION OF MOLDED PLASTIC SEALS

The high fabrication cost of machining the complex geometry of the chevron and K-section second-stage seals is the major disadvantage that will limit the use of these seals to only specialized applications. A substantial cost reduction was potentially possible if these seals could be molded. The results of NASA evaluations suggested that some new moldable plastics had sufficient tensile strength characteristics comparable to machinable polyimides and that the development of moldable seals was feasible. Because property data on these molding compounds was also limited, impulse and endurance tests of the finished seals were required to demonstrate structural integrity and fatigue life in comparison to the machined seals.

A total of five seals sets of the 2.5 in. chevron configuration were fabricated from five molding materials. The chevron elements of the seal, part 2 on Fig. 2, were the only pieces fabricated by molding, these being the parts that perform the actual sealing function. The remaining parts needed to complete the seal assemblies were machined parts retained for use from previous testing.

The material class, composition and molding process used to fabricate test elements is shown on Table 4.

All of the chevron seal elements made from the aromatic polyimide were compression molded and finished by touchup machining of the sealing surfaces. The compression molded chevron made from the aromatic copolyester was molded in the shape of a rectangular cross-section torroid and then machined to the seal element shape.

Inspection of the finished molded seals showed that the geometric shapes of the chevron elements made from aromatic polyimides all exhibited an obvious shoulder at the sealing surface of the outer leg of each chevron. The presence of this shoulder indicated that molding shrinkage was more than anticipated.

Material nonhomogeneity was very prevalent on the chevron elements using filler materials with most of the MoS_2 and graphite filled elements showing uneven flow of the fillers throughout. The unfilled elements, in contrast, showed excellent material uniformity and the best conformance to drawing diameters, with an average deviation of 0.0014 in. undersize on the ID and 0.0006 in. undersize on the OD dimensions compared to the drawing reference.

Only two chevron elements of the eight machined from blanks made out of the aromatic copolyester compression molding material showed any molding faults which were cause for rejection. All of the chevron elements made by injection molding the copolyester material had major and minor faults with eight of the ten elements immediately rejected because of visible fractures. Fig. 15 is a photograph of a typical crack and shows its proximity to the molding riser position at the inside apex of the chevron cross section. This type of crack and its location indicates that failure was probably caused by stress-relieving during cooling after removing the finished part from the mold and showed that the molding material had marginal strength in the finished cross section.

Dimensional inspections indicated rather poor adherence to drawing dimension specifications for all chevron elements made of the copolyester materials. Compression molded elements showed oversize ID's and OD's implying that overstressing of the outer leg and reduced sealing at the rod would occur at elevated temperature. Injection molded copolyester elements showed oversize OD's and undersize ID's indicating that overstress of both legs would occur at elevated temperature.

Impulse Test Screening

Impulse tests to the same requirements imposed on the machined element seals were conducted on seal assemblies using the molded seal elements.

None of the seals tested demonstrated performance as good as that of the machined seals. The 2.5-in. chevron machined seal assembly had a total leakage of 1.75 cc during the 200 000 cycles of impulse testing.

In comparison, the best assembly using molded elements, made from the aromatic polyimide with a 10% graphite filler, exhibited 22.1 cc total leakage during the 200 000 impulse cycles. The chevrons made of the aromatic copolyester materials structurally failed during impulse cycle calibration, therefore leakage performance could not be measured. These failures were due to the lack of material impact strength as evidenced by the fractures around the entire circumference of the sealing elements, as illustrated in the photograph of Fig. 16.

Dimensional inspections of the chevrons made of the aromatic polyimide series of materials that completed impulse tests all showed evidence of shrinkage averaging 0.0195 in. on the outside diameter. This was evidence that thermal setting had occurred during impulse testing at the higher temperatures.

Endurance Testing

The results of impulse testing showed the unfilled aromatic polyimide to exhibit the least amount of thermal setting, better resistance to surface [illegible], and superior homogeneity compared with either the MoS_2 or graphite filled materials. Seals using elements molded of this material were selected for use during endurance testing.

The requirements imposed during the endurance test were identical to those established for similar testing conducted on machined seals. The test duration was established at 3.85×10^6 cycles of actuation at 350° F with the major portion (3.75×10^6 cycles) conducted under short stroke, 2 percent, operation.

The sealing performance of the molded elements, determined by leakage measurements made during endurance testing, was demonstrated by an average leakage of one drop per 484 actuation cycles for short stroke performance (2 percent) and one drop per 114 actuation cycles during long stroke testing (summation of 25, 50, and 100 percent stroke data). The original chevron seal design criteria for leakage acceptance during endurance testing was a minimum of one drop per 12.5 cycles which was met by the molded chevron elements.

The leakage performance of the molded elements was compared to the performance of machined elements previously evaluated. The results on Table 5 demonstrate that the molded elements exhibit much poorer performance than the machined elements

RESEARCH ACHIEVEMENTS

These development efforts have resulted in a substantial advancement in the state-of-the-art of hydraulic actuator rod seals. It was demonstrated that a two-stage rod seal concept using a machined polyimide second-stage of either a chevron or K-section configuration can adequately contain fluid in advanced systems operating in extreme environments. Specific conclusions reached relative to design considerations are as follows:

On Material Selection

Machinable polyimides with a balanced combination of high ultimate stress allowables and low coefficient of thermal expansion provide the optimum properties for achieving the best performance from the seals. Such performance comes at the expense of complex machining of seal parts. The reduced performance of molded seals is not in itself unsatisfactory for a majority of applications. Moldable materials that have higher impact strength than those tested and more consistent physical properties are desired. With such materials and improved molding techniques that provide better dimensional control, seals of a quality acceptable for industry use are expected to be manufactured at much less expense than machined seals.

On Second-Stage Seals

The life capabilities of the machined second-stage seals were not fully evaluated because fatigue failures were not encountered. Seal size reduction to MIL-G-5514F gland depths precluded the use of springs to provide seal leg expansion to fill the gland at design temperature extremes, necessitating design to higher stress conditions. Chevron seal geometry was optimized for severe fatigue requirements by balancing the design between pressure stresses and preset interference stresses. The K-section seal design was more critical than the chevron due to less flexibility in the sealing element legs.

RECOMMENDATIONS

Machined plastic seals that have passed laboratory tests should be used in flight applications where severe environments require sealing techniques beyond the capabilities of present elastomeric materials. Continued research is recommended to complete the assessment of the practical use of plastic seals in aircraft actuator applications. Evaluations that should be conducted to this end are to continue the endurance validation of the chevron and K-section second-stage seals to their design limitations by testing machined seals to failure. This testing will provide results quantifying the design analysis tool and provide data to update the tool for any design conservatism, making it dependable for future use. This should be accomplished by progressively increasing the impulse test conditions for second-stage seals toward the requirement for first-stage seals until seal fracture is evidenced.

An investigation of improved molding materials should be conducted to find materials with equivalent impulse and fatigue properties to those of the machinable polyimides. If such molding materials are not available, the design for the molded seal must be altered to accommodate the properties of the molding materials with the most suitable properties. More extensive property testing of materials and further refinement of molding procedures is recommended to establish material/design compatibility before further performance testing of molded seals for advanced airplane and space vehicle hydraulic system applications.

The development testing of plastic first-stage seals, should be resumed to incorporate updated material property information. Additional tests will determine whether seals, designed for a known wear characteristic at high pressure, actually perform in the manner designed.

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TABLE I. - LEAKAGE DURING MACHINED POLYIMIDE SEAL CYCLING @ 350° F

	Seal configuration	
	2.5 in. K-section	1.0 in. chevron
Leakage performance	cycles/drop	cycles/drop
Minimum accepted value	12.5	12.5
Average value 2% load/stroke	75	300
25%	37.5	94
50%	22.5	90
100%	20	8.5
Cycles accumulated @ 2%	3.15×10^6	2.84×10^6
@ 25%, 50%, 100%	0.06×10^6	0.06×10^6

TABLE 2. - LEAKAGE DURING MACHINED POLYIMIDE SEAL

CYCLING @ 400° AND 450° F

Seal configuration	2.5 in. K-section	1.0 in. chevron	2.5 in. K-section	1.0 in. chevron
Test temperature	400° F		450° F	
Leakage performance	cycles/drop		cycles/drop	
Minimum acceptable value	12.5	12.5	12.5	12.5
Average value 2% load/stroke	22 562	1407	17 648	666
25%	2 406	1467	2 988	255
50%	2 154	1232	2 178	715
100%	2 016	1485	1 584	645
Cycles accumulated @ 2%	3.75×10^6	3.75×10^6	1.875×10^6	1.875×10^6
@ 25%, 50%, 100%	0.1×10^6	0.1×10^6	0.05×10^6	0.05×10^6

TABLE 3. - 2.5 IN. K-SECTION SEAL LEAKAGE DURING VACUUM TESTING

Fluid	Average leakage Actuation @ 16 cycles/min		Maximum leakage (single measurement)		
	% stroke (100% = 4 in)	drops/8 hrs	8 hrs actuation cycles/drop	16 hrs storage drops	64 hrs storage drops
MIL-H-83282	100	41	113	8	0
	75	19			
	50	20			
	25	17			
MIL-H-5606	100	124	58	14	22
	75	72			
	50	44			
	25	24			

Average downstream seal pressure = 10^{-6} mm Hg

TABLE 4. - MOLDING MATERIALS FOR SECOND-STAGE SEALS

Material class	Filler	Molding process
Aromatic polyimide	None	Compression
Aromatic polyimide	10% MoS_2	Compression
Aromatic polyimide	10% graphite	Compression
Aromatic copolyester	None	Compression
Aromatic copolyester	None	Injection

TABLE 5. - MOLDED SEAL PERFORMANCE

	Short stroke actuation cycles/drop	Long stroke actuation cycles/drop
Average performance with molded chevron elements	484	114
Average performance with machined K-section elements	224 659	2717

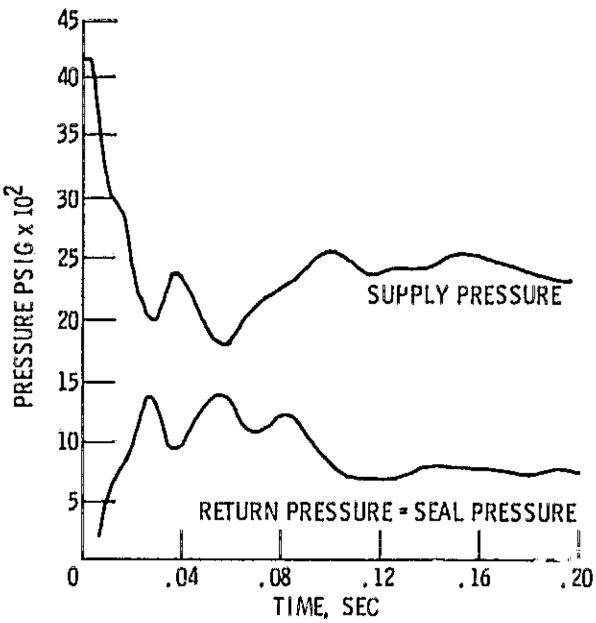

Figure 1. - Computer simulation seal design surge pressure maximum rate, no load.

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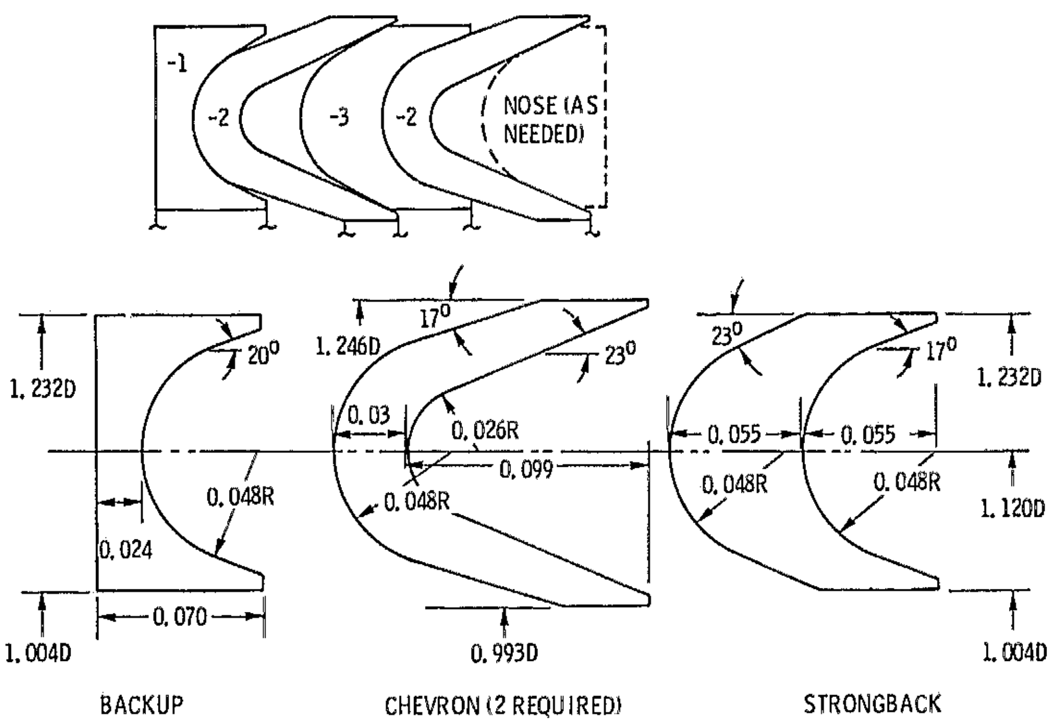

Figure 2. - Chevron second-stage rod seal.

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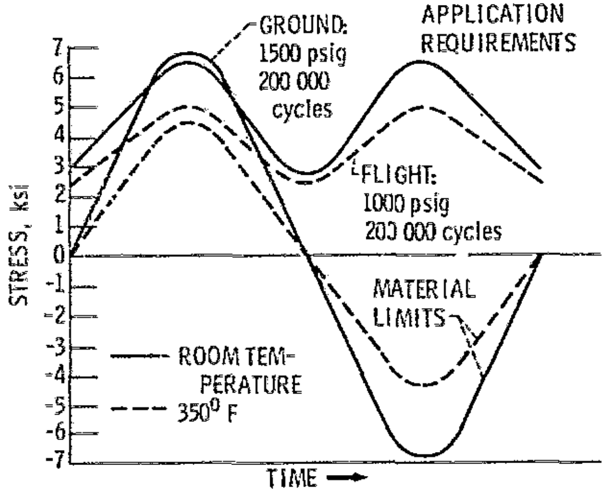

Figure 3. - Seal impulse stress cycle.

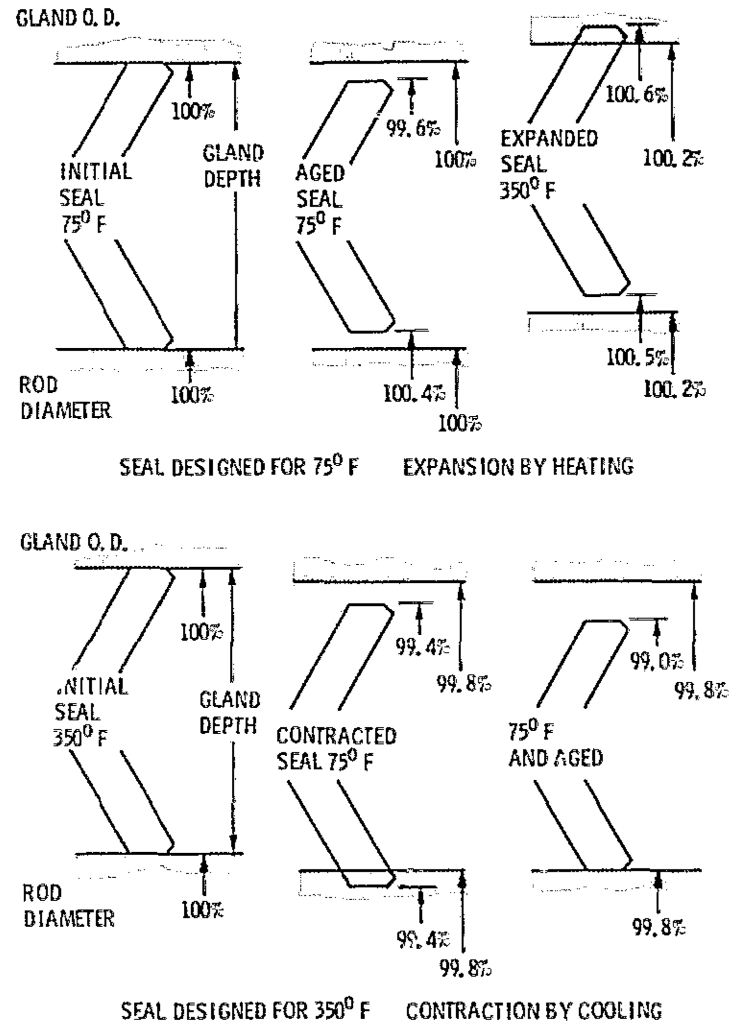

Figure 4. - Effects of temperature and aging.

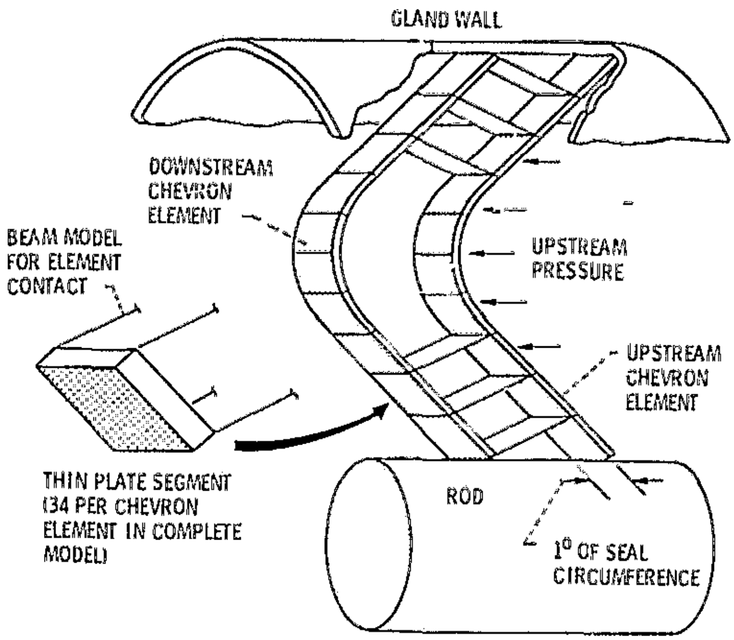

Figure 5. - Typical stress model.

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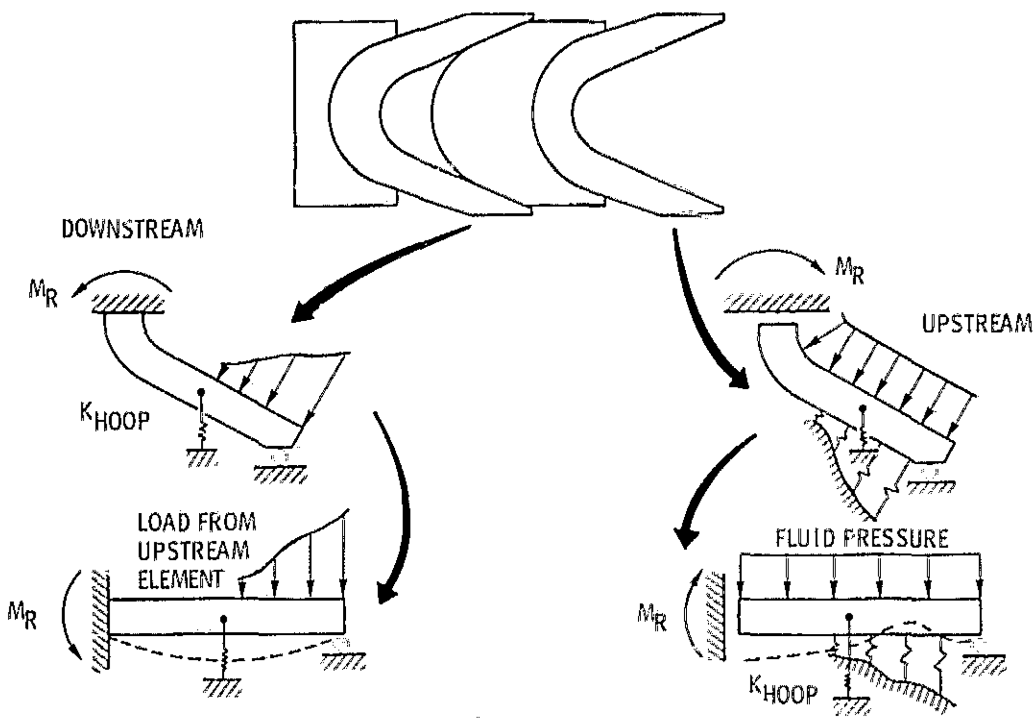

Figure 6. - Chevron structural freebodies.

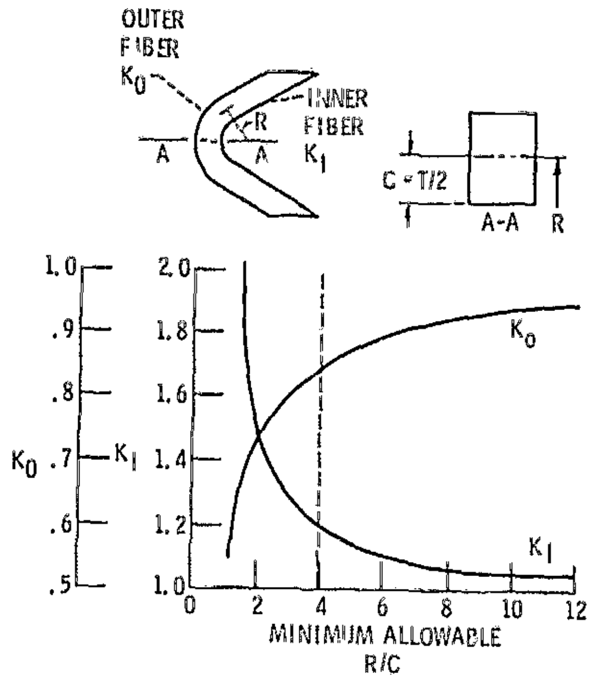

Figure 7. - Curved beam correction factors.

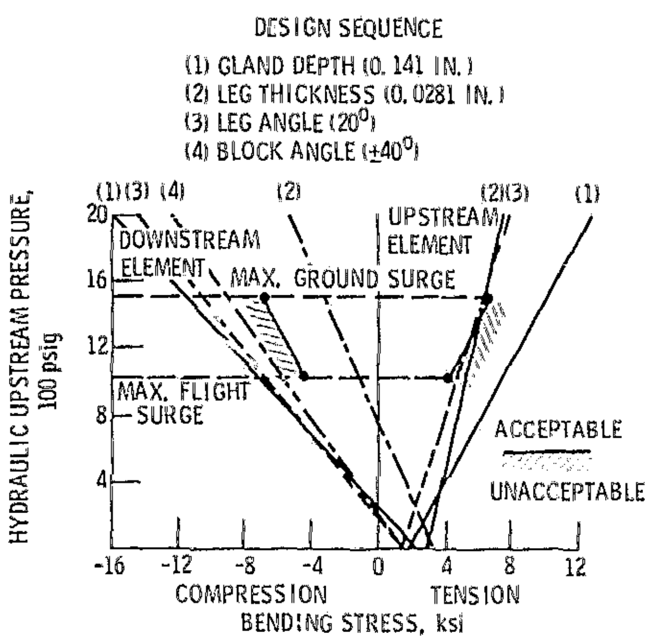

Figure 8. - Stress optimization, 1.0 inch chevron.

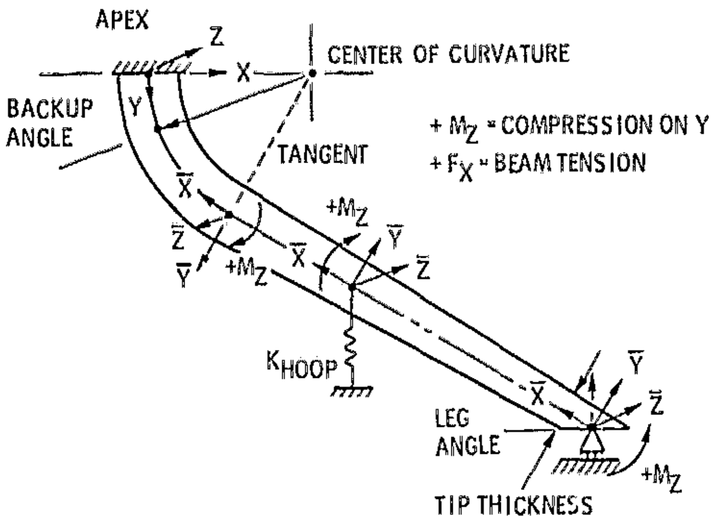

Figure 9. - Tapered leg analysis model.

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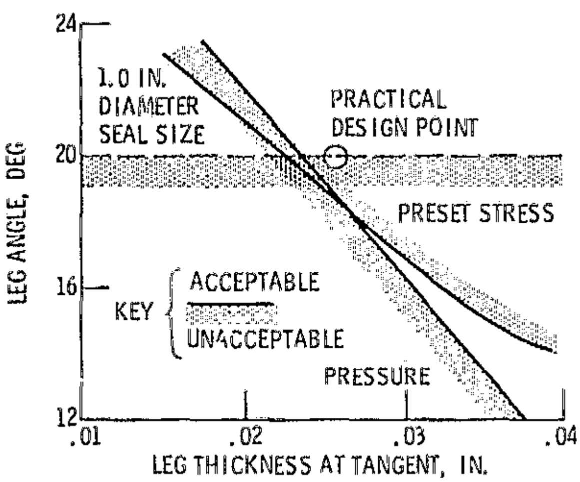

Figure 10. - Finite element stress envelope.

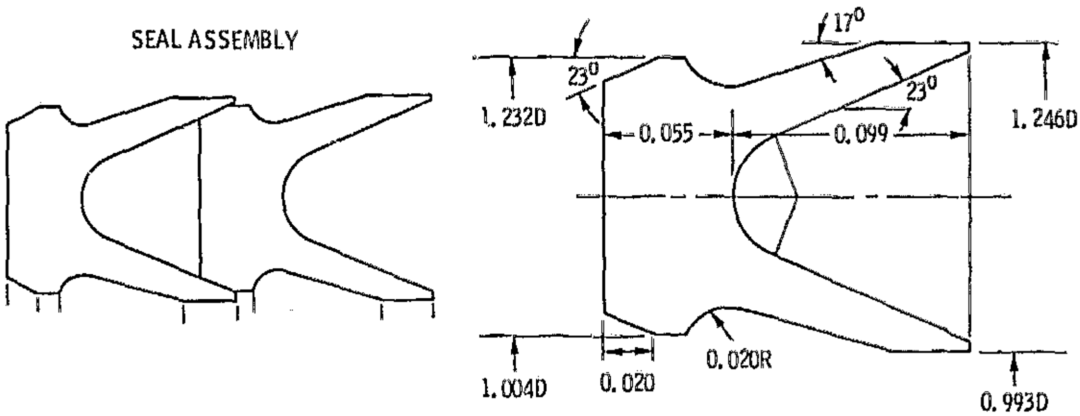

Figure 11. - K-section second-stage rod seal.

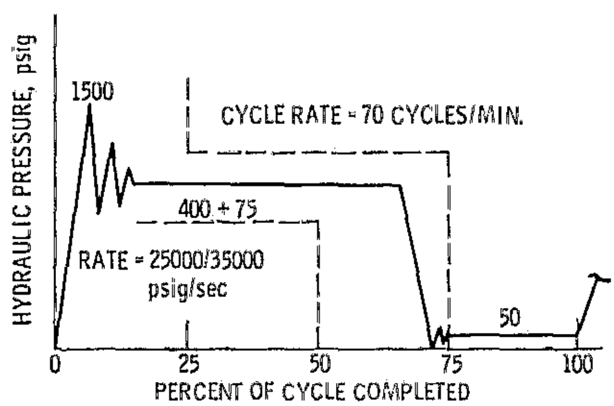

Figure 12. - Second stage impulse cycle.

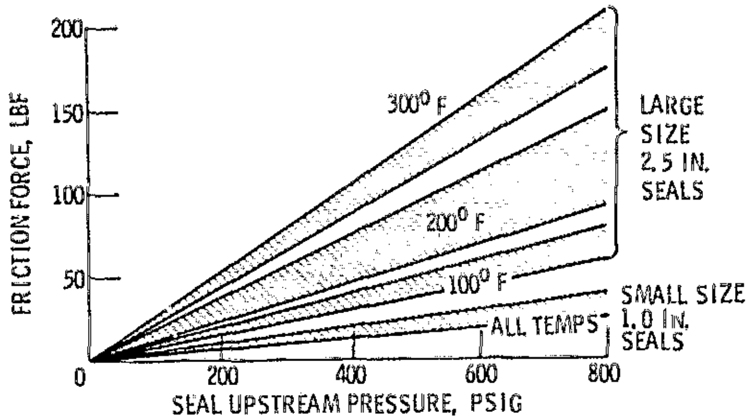

Figure 13. - Second-stage s-al friction.

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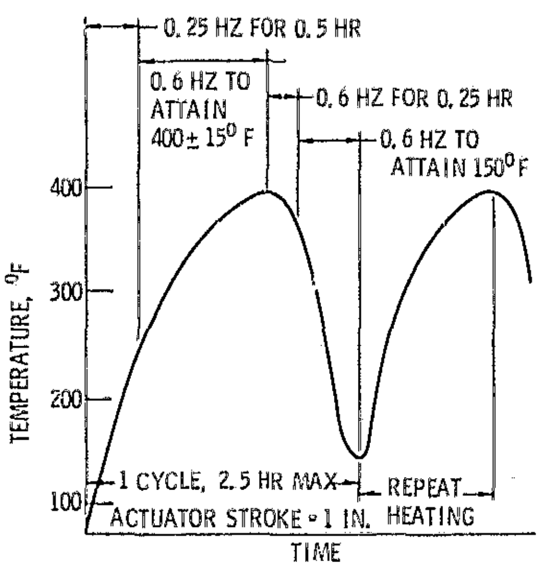

Figure 14. - Typical thermal test cycle.

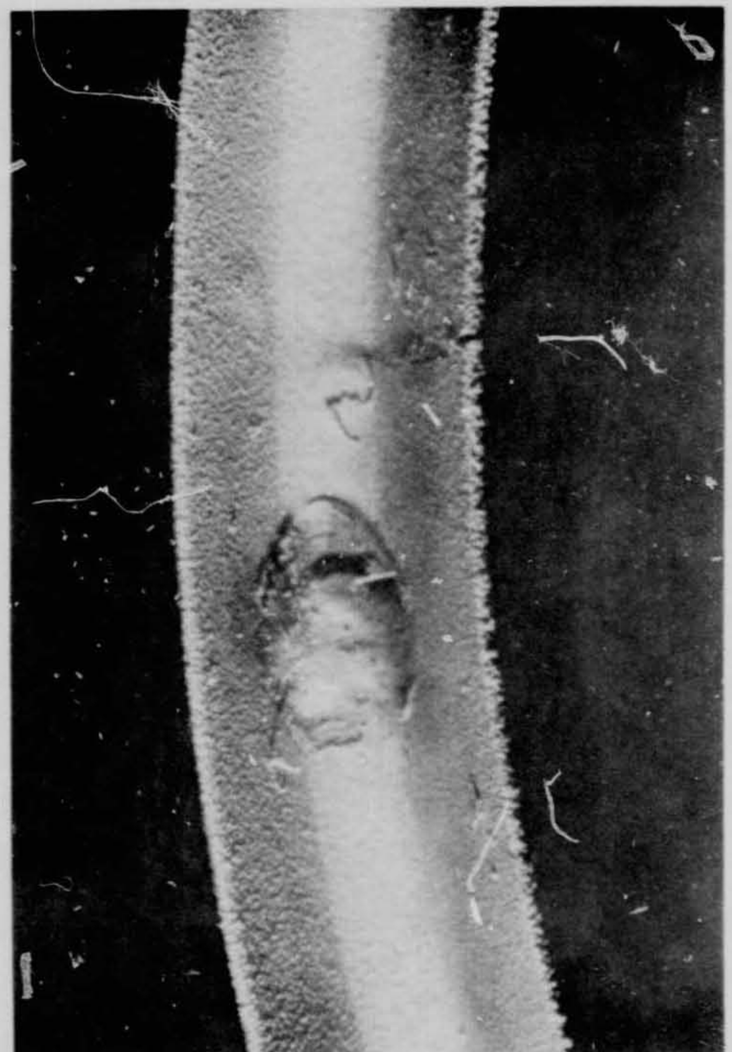

Figure 15. - Fabrication fracture, molded chevron.

Figure 16. - Impulse test fracture of molded chevron.