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ACTIVELY CONTROLLED SHAFT SEALS FOR AEROSPACE APPLICATIONS

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I. BACKGROUND

The objective of NASA grant NAG 3-974 is to determine the feasibility of utilizing controllable mechanical seals for aerospace applications. Under this grant, a potential application was selected as a demonstration case: the buffer gas seal in a LOX turbopump. Currently, floating ring seals are used in this application. Their replacement with controllable mechanical seals would result in substantially reduced leakage rates. This would reduce the required amount of stored buffer gas, and therefore increase the vehicle payload. For such an application, a suitable controllable mechanical seal was designed and analyzed. By the end of the grant period (December 31,1991), fabrication, assembly, and shakedown testing of the seal will be completed.

The above application is only one of many in the aerospace environment, in which fixed clearance seals (e.g., floating ring seals, labyrinth seals) are currently used. While fixed clearance seals have the important advantage of high reliability, they also have the disadvantage of high leakage rate. Conventional mechanical seals have much lower leakage rates. However, the latter are not extensively used in aerospace applications because of their lower reliability. It is the intent of the controllable mechanical seal to achieve the high reliability of fixed clearance seals and the low leakage rate of conventional mechanical seals.

A typical conventional mechanical seal is shown schematically in figure 1. The nonrotating face is floating, and is free to move in the axial direction (within limits), while the rotating face is fixed axially. The seal is designed such that during normal operation, a thin lubricating film of fluid separates the two faces, preventing excessive wear and face damage. The thickness of this film is determined by the axial location of the floating seal face, which is governed by the forces acting on it. The "closing force," produced by the spring and the sealed pressure acting on the backside of the floating face, forces the floating face toward the fixed face. This force depends on the spring characteristics, the sealed pressure, and the geometry of the floating face cross-section (as measured by the balance ratio). The "opening force," produced by the pressure distribution within the fluid film, opposes the closing force and keeps the film intact. This force depends on the fluid mechanics of the film. Under steady state conditions, the film thickness is fixed by a balance between the closing force and the opening force.

Most conventional mechanical seals are of the hydrostatic type. These seals have flat, axisymmetric faces. The pressure distribution within the film of such a seal is associated with the radial flow field induced by the pressure drop across the seal. Thus, the pressure distribution and the associated opening force are dependent on the radial variation in geometry of the gap between the seal faces (or the radial variation in film thickness), as well as on the sealed pressure. The gap may be uniform, converging (going from the high pressure to the low pressure side), or diverging. However, it is well known that to achieve stable operation, a seal must be designed such that the gap converges.

The amount of convergence, or coning, strongly influences the pressure distribution and opening force. The larger the coning, normalized with respect to the average film thickness, the more convex the pressure distribution and the larger the opening force. Since the closing force is fixed by the spring characteristics, the sealed pressure and the cross-sectional geometry of the floating face (balance ratio), under steady state conditions there is one unique value of the normalized coning for which the opening force equals the closing force and the floating face is in equilibrium. Hence, the steady state average film thickness must be proportional to the coning: the larger the coning, the larger the film thickness.

Normally, a conventional seal is designed and manufactured such that prior to service the coning is effectively zero. However, once it is in service, coning is produced by mechanical and thermal deformation of the faces and their supporting structures. The coning is therefore determined by the fluid mechanic, heat transfer, energy dissipation, and structural deformation processes. Hence the amount of coning, and therefore the film thickness, is strongly dependent on the operating and environmental conditions, as well as on the seal design. Thus, it is a principal task of the designer to configure the seal so that the faces deform just the right amount, at the design point, to produce an optimum film thickness. This film thickness is usually on the order of a few microns, which is much smaller than the clearance in fixed clearance seals (approximately twenty microns). Since the leakage rate is proportional to the cube of the film thickness, it will be much lower for a mechanical seal than for a fixed clearance seal.

However, when a conventional mechanical seal experiences changes in operating and/or environmental conditions, through short term or long term transients, the amount of deformation and the film thickness change, leading to periods of excessive leakage and periods of face contact. The latter is the more serious consequence, because face contact can produce excessive face temperatures, and thermal and mechanical face damage. It is these effects that have limited the use of conventional mechanical seals in aerospace applications. The face damage produced by prolonged face contact is the primary cause of inadequate mechanical seal reliability. Furthermore, excessive face temperatures produced by prolonged face contact could result in an ignition hazard (i.e., in a LOX environment).

To alleviate the above problems of conventional mechanical seals, the concept of a controllable mechanical seal has been developed. A conventional seal is a passive device. While it can be designed to run at a particular design point in an optimum fashion, it reacts to changes in operating and environmental conditions in an uncontrolled manner, according to the laws of physics. Once the seal has been installed, nothing can be done to improve its performance. In contrast to the conventional seal, the controllable mechanical seal is an active device. It reacts to changes in operating and environmental conditions in a pre-programmed controlled manner, such that optimum operation is achieved at all steady state operating points (within a specified range) as well as during transients. The design of the seal is effectively changed during operation to adapt to local conditions. Such a controllable seal has a variable film thickness, which can be adjusted explicitly. If conditions are such that the film is too thick or too thin, it is readjusted to its optimum thickness while the seal is in service.

The film thickness of a conventional seal is fixed by the location of the floating face, which is governed by the forces acting on it. Hence, the film thickness could be controlled by controlling either the closing force or the opening force. However, the author has found that it is much more effective to control the latter than the former. Since the opening force and the film thickness are strongly dependent on coning, control is achieved by controlling the coning. This is accomplished by incorporating within the seal a piezoelectric actuator to deform mechanically one of the faces and generate coning. The produced coning compensates for the natural mechanical and thermal deformation experienced by the seal faces and supporting structure, and can be adjusted to any desired value by the actuator. It is important to note that the actuator does not directly produce the opening (or closing) force. It is the pressure distribution in the film that produces the opening force, and keeps the two faces separated and the film intact. The actuator regulates the pressure distribution and opening force by controlling the coning. Thus, very precise control of film thickness is achieved.

While the above adjustment of film thickness could be done manually, a much more powerful approach makes use of an automatic control system. Sensors monitor conditions in the film and in the sealed cavity. Based on those conditions, a microprocessor-based control system causes the actuator to adjust the film thickness, according to a preprogrammed strategy. This allows continuous optimization of the film thickness and automatic adaptation to a steady state operating point. Furthermore, it allows response to short term transients, and especially, the minimization of face contact during such events.

Several actively controlled seals, based on the above concept, have been developed for industrial applications [1,2]. These have been successfully tested in the laboratory and in the field. However, the requirements for aerospace applications are very different from those for industrial applications. The biggest difference is in the size requirement. A typical industrial seal, for a 140 mm. shaft, would have an available space of axial length approximately 200 mm. and radial width approximately 50 mm. A typical aerospace turbopump application would have an available space of axial length approximately 40 mm. and radial width approximately 13 mm. Thus, the aerospace seal must be significantly more compact than the industrial seal, and would require a radically new design. In addition, the aerospace seal must operate at higher rotational speeds than the industrial seal (70,000 rpm vs. 3,600 rpm), and at more extreme temperatures (208K-382K vs. 294K).

II. RESULTS OF GRANT TO DATE

A. Seal Design

The controllable aerospace seal developed under this grant [3], is shown in the schematic drawing of figure 2, and in the assembly and detail drawings of figures 3 - 8. This seal is intended to serve as the buffer gas seal in a LOX turbo pump. The design constraints are shown in figure 9.

The tester housing is designed to simulate the seal cavity in the turbopump. The tandem seal configuration is comprised of one rotating face and two nonrotating floating components. This arrangement is well-suited to the proposed application, since one floating face seals the hot turbine gases from the buffer gas (helium), while the other floating face seals the cold gaseous oxygen from the buffer gas. The tandem configuration is also appropriate as a test configuration for other applications (including single seal applications) because it minimizes the axial force transmitted to the shaft, and therefore the thrust load on the bearings.

The rotating face is constructed of tungsten carbide. An O-ring, set into the shaft, prevents leakage between the face and the shaft. The shaft is partially threaded, and a threaded nut (and a sleeve) holds the face fixed against a shoulder.

Each floating component contains a deformable face assembly, which is comprised of a piezoelectric cylinder bonded to a carbon cylinder. The former contains electrodes along the inner and outer diameters to which a voltage drop is applied. The deformable face assembly floats within a holder, and is supported and sealed by an O-ring. The holder, in turn, floats relative to the housing, and is supported and sealed by a second O-ring. A wave spring washer, between the holder and the housing, supplies a portion of the closing force on the floating component.

The shaft is driven, through a belt drive, by a variable speed motor and power supply. Shaft support is provided by sealed ball bearings, which can function at speeds up to 58,000 rpm. Voltage is supplied to the piezoelectric element by a high voltage power supply. Helium is supplied to the tester from a commercial pressurized cylinder and regulator valve.

The primary variable to be measured and monitored is face temperature. This is done by a thermocouple contained within the floating seal component, and mounted in a groove between the carbon face and the piezoelectric element. The helium temperature within the sealed cavity is also measured with a thermocouple. The leakage rate is measured by two methods. A rotometer type flow meter is inserted in the flow line between the helium supply and the tester, and leakage is collected downstream of the seals, through venting ports in the housing. The rotational speed of the shaft is measured by an optical tachymeter.

B. Analysis

The above seal design was arrived at through an iterative process, involving the evaluation of several potential designs and design variations. The evaluation was performed using a mathematical model that includes four major elements: force balance, fluid mechanics model, structural and heat transfer model, and overall computation algorithm [3].

Under steady state conditions, a balance of the opening and closing forces on the floating seal face determines the axial location of that face, and hence, the film thickness. This algebraic relation contains several design parameters (e.g., balance ratio, spring force) as well as an integral containing the pressure distribution in the fluid film.

The fluid mechanics of the gas film is governed by the compressible Navier Stokes equation. Making the usual lubrication assumptions leads to the compressible Reynolds equation. Assuming steady state and an isothermal film, with no circumferential variations, leads to a closed form expression for the pressure distribution. This expression is then integrated to yield the opening force (on the floating seal face) for insertion into the force balance. It should be noted that the pressure distribution and opening force are strong functions of the (as yet unknown) coning. The solution for the pressure distribution also leads to expressions for the seal stiffness (an indicator of stability) and seal controllability (an indicator of the sensitivity of the seal to the voltage applied to the piezoelectric element). The leakage rate is obtained from the radial gradient in pressure. The viscous heat generation rate, which influences the thermal deformation, is obtained by treating the circumferential flow field as Couette flow.

Finite element structural and heat transfer models of the floating component and rotating faces have been constructed using the commercial program ANSYS. The model of the floating component consists of a piezoelectric element with a carbon face and a holder. The piezoelectric element is modelled by the STIF5 element in the ANSYS library, which is a three-dimensional, eight-noded brick element with translational, voltage, temperature, and magnetic degrees of freedom at each node, for a total of six degrees of freedom per node. The carbon face and holder are modelled with the STIF45 element for structural analysis, which is a three-dimensional, eight-noded brick element with three translational degrees of freedom per node. The rotating face is also modelled with the STIF45 element. For heat transfer analysis, the STIF70 element is used. This element is a three-dimensional, eight-noded brick element with temperature as the only degree of freedom. All models are axisymmetric, with a dimension of 3.5 degrees in the circumferential direction. Appropriate mechanical and thermal boundary conditions are applied.

The overall computation algorithm utilizes the force balance, fluid mechanics model, and structural and heat transfer models, in an iterative procedure to determine the performance of the seal. The seal design, material characteristics, and operating conditions are input. Output includes the film thickness (at ID), the coning, the leakage rate, the temperature profiles for each seal component, and the pressure profile within the gas film.

The above design tools have been used to find the optimum spring force (for selection of wave spring washer) and balance ratio. These have been selected to represent a compromise between a high degree of stability (maximum stiffness) and a high seal controllability.

Finite element analyses were performed on the floating component to determine the optimum deformation mode of the piezoelectric element, and the optimum method of support. The results suggest that the piezoelectric element should operate in the transverse mode, and should float within the holder to obtain the maximum amount of coning deformation with a given applied voltage.

Finite element analyses were performed on the rotating face to assure its structural integrity under the action of centrifugal stresses. It was found that the maximum stresses predicted were well within the acceptable limits for the face material (tungsten carbide).

Many performance computations were made to determine the effect on seal behavior of mechanical boundary conditions, thermal boundary conditions, initial coning, sealed pressure, and piezoelectric material. Typical results are shown in figure 10.

C. Bench Tests

A series of bench tests have been performed on sample piezoelectric cylinders to determine the effectiveness of various bonding and insulating agents, and to verify the piezoelectric deformation characteristics. Various voltages were applied across specially prepared samples, and the occurrence or nonoccurrence of electrical breakdown was observed. Based on such tests, a preparation method and insulation material were selected. Suitably prepared samples were then placed on a profilometer, and voltage was applied to produce the transverse mode of deformation.

D. Hardware and Shakedown Tests

The assembly and detail drawings of the controllable mechanical seal are contained in figures 3 to 8. These have been submitted to the machine shop for fabrication and balancing. Fabrication, balancing, assembly, and shakedown tests will be completed by the end of the grant period (December 31, 1991).

III. REFERENCES

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2. Salant, R.F., Giles, O., Key, W.E., "Design of Controllable Mechanical Seals," 15th Leeds-Lyon Symposium on Tribology, Leeds (1988).

3. Wolff, P., "Development of an Actively Controlled Mechanical Seal," MS Thesis, Georgia Institute of Technology, Atlanta (1991).

FIGURES

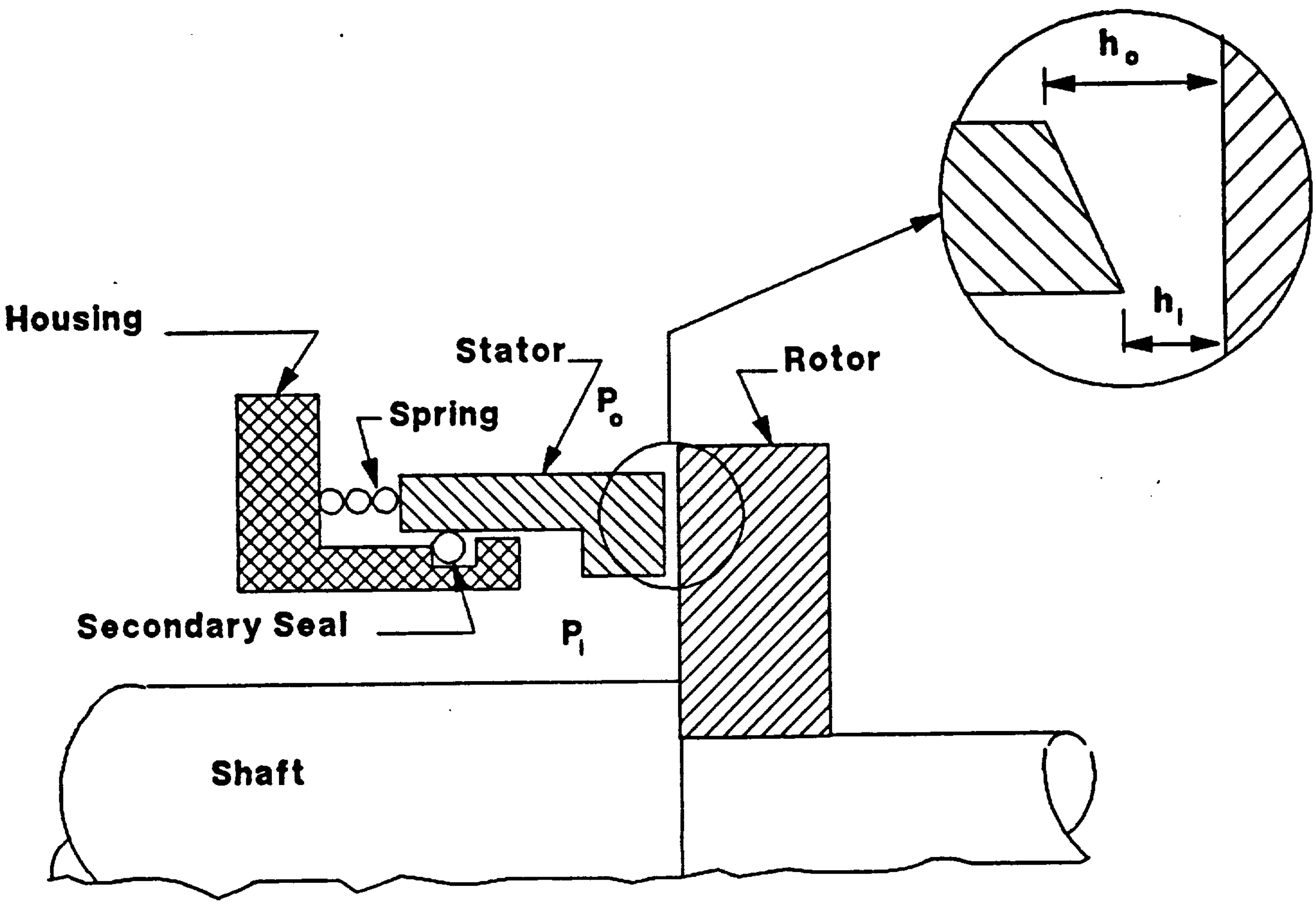

Figure 1. Diagram of Mechanical Seal

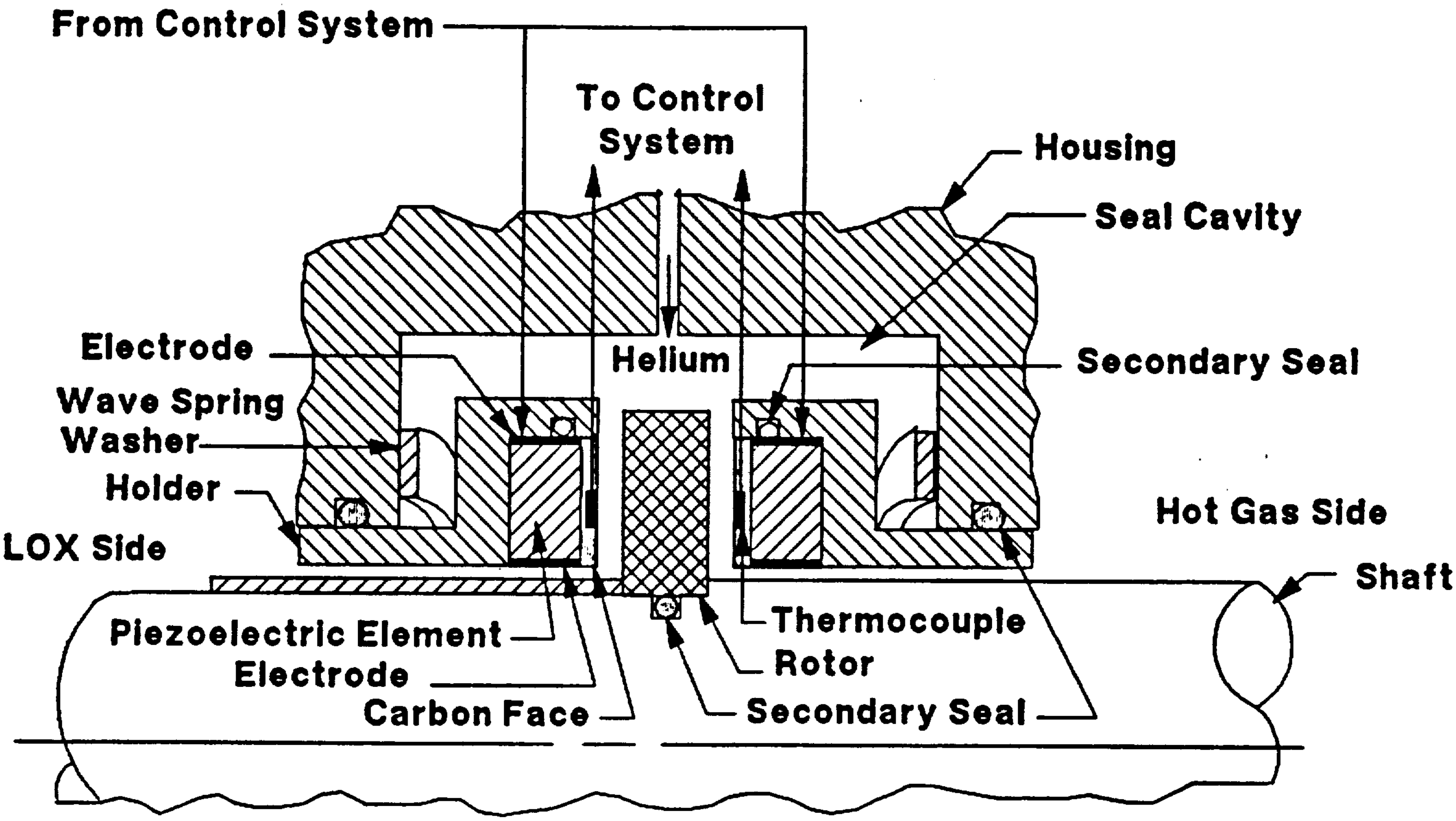

Figure 2. Proposed Seal Design

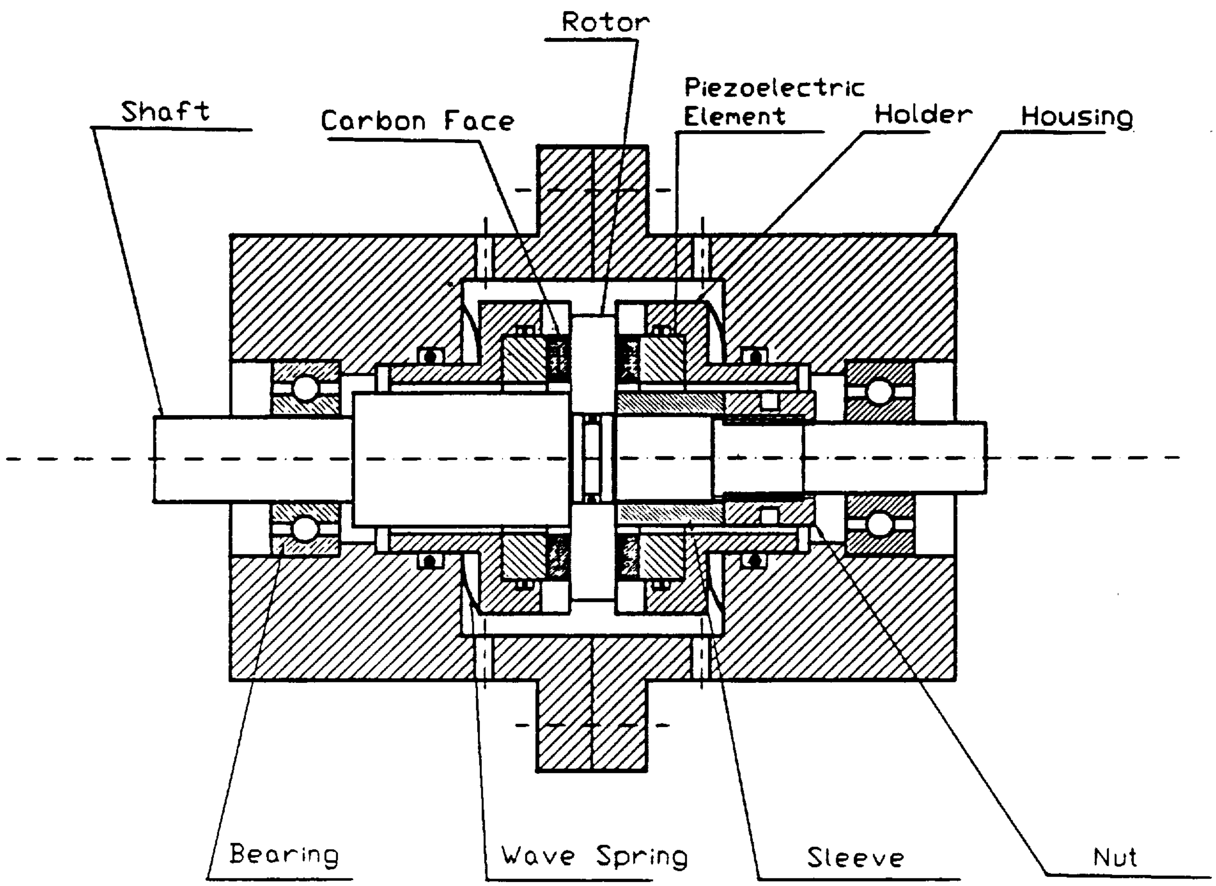

Figure 3. Assembly Drawing

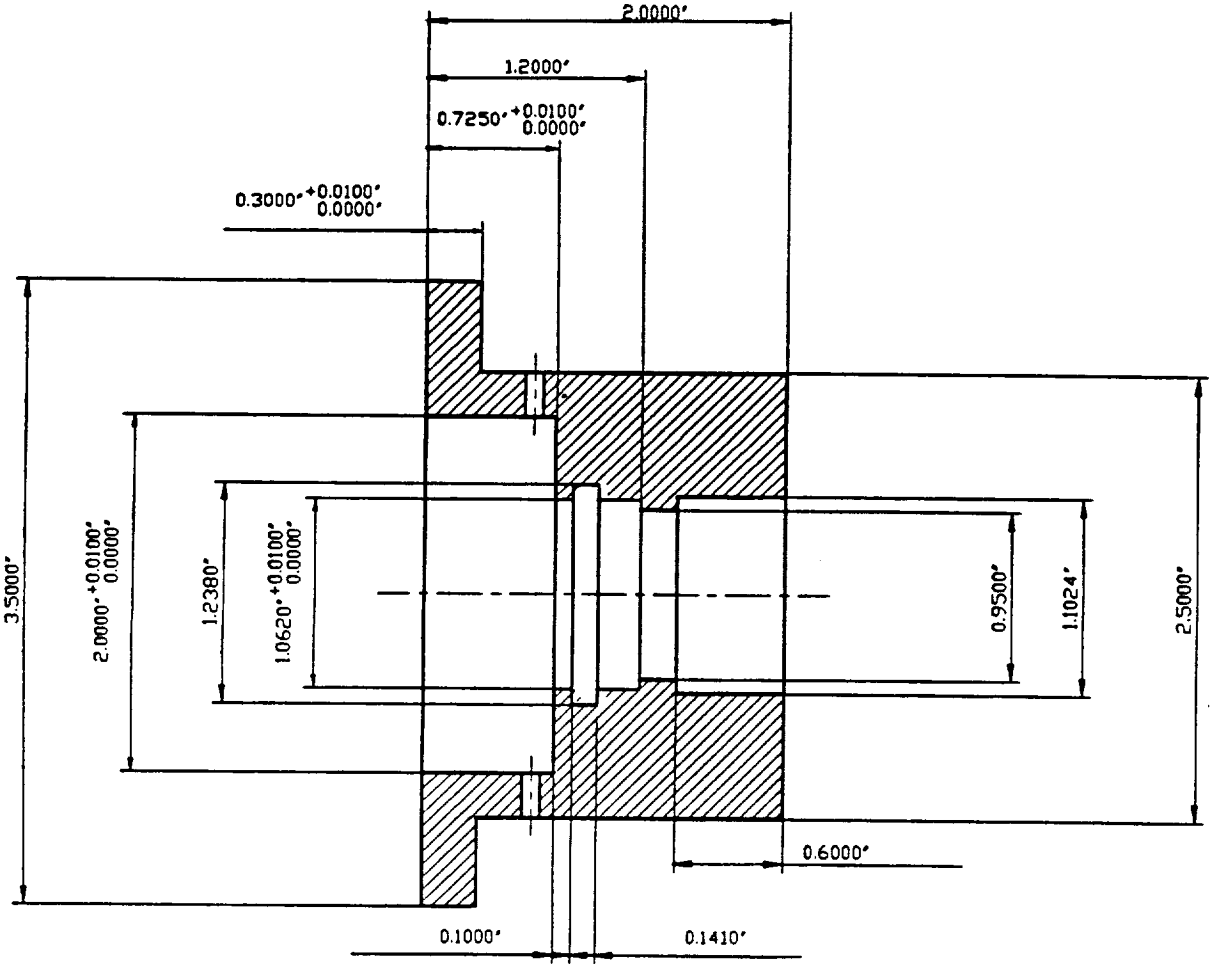

Figure 4. Housing

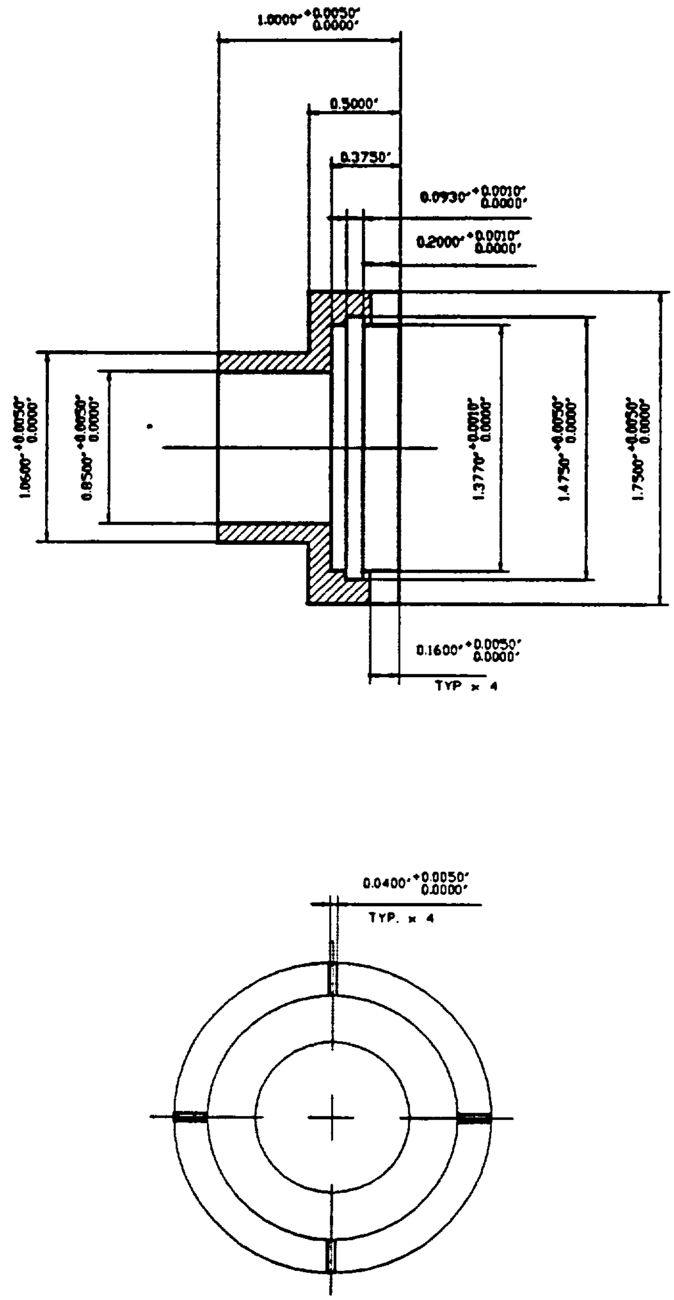

Figure 5. Holder