NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Technical Memorandum 33-587

A Survey of Actuator Shaft Sealing Techniques for Extended Space Missions

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PASADENA, CALIFORNIA

December 15, 1972

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PREFACE

The work described in this report was performed by the Guidance and Control Division of the Jet Propulsion Laboratory.

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ABSTRACT

Actuators for control and articulation aboard Mariner spacecraft have employed output shaft seals to maintain an internal gaseous atmosphere. This, combined with the limitation of temperature extremes through the use of electric heaters or the location of actuators in temperature controlled areas, has resulted in a favorable environment for actuator mechanisms. On future missions, considerably greater expected lifetimes, temperature ranges, and radiation exposures have led to a need to determine the limitations of the present O-ring output shaft seal and to examine other candidate seals. Seals suited both to dynamic and static sealing were examined for potential use in three specific JPL actuator applications and the following candidate seals were selected: (a) O-ring seal, (b) chevron seal, (c) bellows seal (linear actuator only), (d) magnetic fluid seal (rotary actuators only) and as a backup seal to any of the foregoing, (e) the labyrinth seal.

I. INTRODUCTION

Actuators aboard unmanned spacecraft on planetary exploration missions are employed for a variety of functions, including the accurate positioning of scientific instrument scan platforms and high-gain antennas, the precise control of thrust direction during powered flight, and the less precise articulation of solar arrays and solar energy vanes.   Past planetary spacecraft have generally been limited to 1- to 2-year missions and to solar energy fluxes varying by about a factor of 2 throughout their flights.   Future missions are expected to extend into the 5- to 10-year range and solar fluxes received by the spacecraft to vary through the flights by ratios of 100:1 or more.   Such missions intensify the following problems that must be considered in the design of all mechanisms required to survive and operate in space:

(1)   The prolonged effect of space vacuum.

(2)   The cumulative effect of exposure to on-board radiation sources and external space radiation.

(3)   The wide range of temperatures experienced with the actuator either operating or dormant.

These problems affect the choice of actuator component materials that can be used and, as will be shown, they also affect in a major way both lubrication systems and shaft seals, the latter being the subject of this report.

II. ACTUATOR SHAFTS -- TO SEAL OR NOT TO SEAL

The requirement or nonrequirement for actuator output shaft seals depends directly upon the lubrication system employed for the actuator; this, in turn, is determined by the longevity, performance, and environmental parameters set by the mission.   Thus, the lubrication and actuator internal environments are interdependent, the approach used on one directly

influencing the other. For example, if very long life (nonoperational) is required, dry lubricants may be a satisfactory approach since they have quite low vapor pressures even at rather high temperatures, are effective at very low temperatures, and are much less affected by radiation than are wet lubricants. If the integrity of the dry lubricants can be maintained for the required operating life and a means employed to prevent corrosion and contamination of actuator components prior to launch, no seal is required. On the other hand, if long operating life, low friction, and very low backlash are required, wet lubricants are indicated.

In the past, many actuators, including all **JPL** actuators, have operated within a limited temperature range and have employed conventional lubricants in an inert gaseous environment intended to provide a convection heat transfer medium and inhibit degradation of the lubricant. In addition, such an arrangement prevented corrosion within the actuator prior to launch.

The **JPL** actuators which have flown on Mariner spacecraft have employed O-ring dynamic shaft seals to maintain an atmosphere of dry nitrogen at about earth atmospheric pressure. Both the O-rings and working elements have been lubricated with silicone oils and greases since they have low evaporation losses in vacuum which will preserve the life of the lubricants in the event of O-ring failures. Such seals, together with static O-ring seals and hermetic electrical feedthroughs, **limit** gas loss from the actuator to less than 2×10^{-5} standard cm^3/s at 21°C **(70°F).** This leak rate will ensure an internal pressure of greater than 6.89 **X** 10^3 N/m^2 (1 psia) after 30 years of vacuum exposure (see Ref. 1) provided the wetting film of lubricant on the O-rings and the integrity of the O-rings themselves is maintained (i.e. **,** no cracking or hardening due to radiation or vacuum exposure).

The increased possibility of O-ring degradation due to radiation, wider temperature ranges, and extended vacuum exposure has led to a need to determine the limitations of the currently employed shaft seals and to review other techniques in the light of future mission requirements.

111. TYPES **OF** ACTUATORS

Three different types of control and articulation actuators have been employed on JPL spacecraft:

(1)   Electric-motor-driven gear-reducer rotary actuators producing relatively high torques, having high response, high torsional stiffness, very low backlash, and long operating life (Ref. 2).

(2)   Electric-motor-driven ball screw gearless linear engine gimbaling actuators producing relatively high force, having high response and low backlash, and capable of a large number of cycles (Ref. 3).

(3)   Electric-motor direct-drive rotary jet vane actuators producing very low torque outputs and capable of very fast response and a large number of cycles.

General characteristics of these actuators are shown in Table 1. In the past, shaft seal configurations and lubrication techniques for the three types of actuators have been similar. It would be desirable but not essential that this be the case for such actuators in the future.

IV.   ACTUATOR TEMPERATURE ENVIRONMENT

Actuator temperature ranges, both operating and nonoperating, are determined by (1) the mission (i.e., range of solar intensities experienced through the mission), (2) location of actuator (i.e., whether in an actively temperature-controlled or Sun-dependent location), (3) whether heated or not and whether heating is electrical or by radioisotope heating units, (4) whether in the proximity of the auxiliary propulsion engine, and (5) actuator shape factors which influence heat conduction and radiation.

In the past, required test temperatures (which exceed expected operating or survival temperatures) for actuators have not been severe, ranging from about -29°C (-20°F) to about +54°C (+130°F), with a typical range of around 38"C (100°F). Actuators with relatively high energy dissipations, located in temperature-controlled areas, tend to run toward the higher temperatures, and those in Sun-dependent areas tend toward the low-temperature end of the range. The linear engine gimballing and jet vane actuators are subject to heat soakback from the auxiliary propulsion engine and their high temperatures (test) run up to 149°C (300°F). For outer planet missions, the temperature ranges for unheated actuators in Sun-dependent areas will increase to about 149"C (300°F) near Jupiter and 177"C (350°F) near Saturn. Because of very low solar fluxes at these planets, actuator temperatures will tend to reach quite low temperatures. These ranges may

be reduced to about **93°C** (200°F) through the use of commandable (5-10 W) constant-power electric heaters.   However, the use of electric power for heating is costly if required for a nonoperating actuator during a period of peak power usage which can occur, for example, when a spacecraft is occulted from the Sun by the planet which is being explored -- a period when power may be required by science instruments, radio frequency subsystem, data system, scan actuators, gyros, etc.   An additional 10 W for heating, say, an antenna actuator at such times might result in a requirement either for added battery capacity, larger solar arrays, or larger radioisotope thermal generators, with attendant added weight and cost and reduced reliability.

Similarly, the use of a heater to supplement the heat supplied by the motor of an operating actuator with a low duty cycle is undesirable from a power management standpoint at times of peak power.

Thus it appears that the elimination of heating of actuators is a desirable goal for reasons of cost, weight, power, and reliability.   To accomplish this goal at least some actuators would require:

(1)   The capability of <u>surviving</u> nonoperating mission temperature excursions of approximately 149°C (300°F) (or at least 93°C (200°F), using radioisotope heating units).

(2)   The ability to <u>operate</u> over a temperature range of 149°C (300°F) (or at least over a 93°C (200°F) range)

Factors which render actuator <u>survival</u> over a 149"C (300°F) temperature range difficult include the following:

(1)   Possible damage to actuator components because of differences in thermal expansion.

(2)   Possible degradation of oil or grease lubricant if the lower temperature limit is below the lubricant's freezing point.

(3)   Maintaining the integrity (i.e., ability to seal) of the shaft O-ring seals and actuator static seals if the lower temperature limit is below the brittle point of the elastomer.

(4)   Excessive loss of oil if labyrinth seal (or no seal) is used and upper temperature limit is set high.

Factors which make actuator <u>operation</u> over a 149°C (300°F) temperature range difficult include those listed above and the following:

(5) Excessive torque requirements due to high lubricant viscosity at the low end of the temperature range, since the maximum temperature is constrained by lubricant film thickness considerations.

(6) Integrity of dynamic shaft O-ring seal if requirement (**3**) above calls for operation below about *-29°C* (-20°F).

Thus it appears that extending actuator operation and survival to wider temperature ranges may be limited by (1) shaft seals or (2) lubricants or (**3**) both.

V.   REQUIRED SHAFT SEAL INVESTIGATIONS

Areas requiring investigation to extend actuator operation and/or survival to wider temperature limits include the following:

(1) Determination of low temperature limits for O-ring shaft seals[1] with shaft rotating (i.e., low-temperature limit for seal operation).

(2) Determination of low-temperature limit for O-ring shaft seals[1] with shaft stationary (i.e., seal low temperature limit for survival).

(3) Determination of low-temperature limit for lubricants without degradation.

(4) Determination of low-temperature lubricant limitations from motor torque standpoint.

Sufficient data are available on high temperature effects both for O-ring material and lubricants, with the possible exception of oils for use with labyrinth seals.   Thus potentially more severe temperature environments, coupled with radiation effects over an extended time period, have led to a need to determine the limitations of the presently used O-ring shaft seals and to survey the current state of the art and applicability of other candidate shaft seals for the actuator applications in question.

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[1] And other candidate seals.

VI.   SHAFT SEAL APPROACHES

There are three potential approaches to actuator output shaft sealing:

(1)   "Hermetic"[2] and "tight"[2] seal (e. g. , O-ring shaft seal, flexing membranes, etc.).   This approach permits the widest selection of lubricants with minimum limitations on additives or on the range of lubricant molecular weights or lubricant vapor pressures.

(2)   Labyrinth seal.   This is a noncontact but close clearance "seal" which permits the slow escape of the lubricating oil vapor.   It maintains a maximum pressure within the actuator enclosure equal to the oil vapor pressure at the prevailing temperature. Thus lower actuator temperatures will result in lower oil losses. Limitations on oil selected for long-term use with this seal include a requirement for a very low vapor pressure and for a highly refined, very narrow molecular weight range and either the use of an anticorrosion additive in the oil or the use of wholly corrosion-resistant materials within the actuator to protect the components prior to launch.

(3)   No seal.   Wet lubricants have been used in space with no seal but this approach is not suited to extended life missions.   Dry lubricants, where applicable, may be employed with no seal. Actuators containing components with no rolling or sliding contact and hence requiring no lubricant (i.e., flexure or magnetic suspensions) may also be feasible for some applications such as the jet vane actuator, which is a gearless, direct-drive unit. Some development has been accomplished on such a suspension for a jet vane actuator.

The last two approaches have an advantage in applications requiring long cycling life (as distinct from survival life), since they involve no seal contact and hence no seal wear, but, as noted, the "no seal" approach

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[2] A "hermetic" seal is defined in Ref. 4 as one designed to permit no detectable gas flow through it (on a very sensitive detector such as a helium mass spectrometer). A "tight" seal is just free of leaks according to a given specification.

imposes unacceptable lifetime limitations on wet lubrication applications,
The labyrinth seal has been used in a number of space applications, generally
on those involving continuous rotation such as on despin assemblies and
where the operating temperature range is limited.

VII.   DYNAMIC VACUUM SHAFT SEAL TYPES

Reference 5 classifies rotating shaft vacuum seals into two general
types: (1) hermetic seals and **(2)** shaft penetrating seals.

Hermetic seals include:

(1)   Electric or magnetic field transmission through a stationary
membrane: permanent magnet coupling, electromagnetic
coupling, electrostatic coupling.

(2)   Mechanical transmission through a flexing metal membrane:
harmonic drive, wobble plate, bellows.

For the high-torque rotary actuator application previously mentioned, the
only seal of the hermetic types noted above which approaches possessing the
required torsional stiffness and torque-carrying capacity is the harmonic
drive, but it has the disadvantages of (1) having sliding and rolling contact in
the space vacuum and (2) low efficiency at low speed.   Harmonic drives are
not applicable to jet vane actuators, since these drives are speed reducing
devices, whereas a direct drive without speed reduction is desired.

Shaft penetrating seals have high torsional stiffness and torque
capacity and contribute no backlash to the actuator but, with one exception,
have the severe disadvantage of not being a hermetic (membrane) seal.   Shaft
penetrating seals include:

(1)   Rubbing

(a)   Solid-to-solid contact: O-ring, chevron, face seal.

(b)   Solid-to-liquid contact: viscoseal, fluid head, slinger,
surface tension.

(2)   Clearance or controlled-pressure differential

(a)   Speed-independent: labyrinth and diffusion pump.

(b)   Speed-dependent: molecular pumping and centrifugal effect.

Many of these classes of seals must be eliminated for the applications in question since they are speed-dependent (i.e., do not seal with the shaft not rotating) or are gravity-dependent. This leaves the O-ring, chevron ring, face seal, surface tension, and labyrinth seals to be considered.

A tentative listing of candidate shaft seals for the three types of actuators, all of which are shaft-penetrating types, is given in Table **2**. Descriptions of these seals are given below.

The elastomeric O-ring seal (Fig. 1) is widely used in industry as a static and dynamic seal. It provides its sealing by pressure created by compression of the elastomer along the O-ring cross section; this pressure is supplemented by the differential pressure of the fluid being sealed. For vacuum or pneumatic applications, O-rings seal better if wetted with grease, with oil, or with water adsorbed from the air. For dynamic vacuum seals, the wetting agent, water or oil[3], is virtually mandatory. For vacuum O-ring dynamic seals, oil[3] has the advantage of substantially reducing the friction, hence the O-ring wear, and of being retained longer because of its lower vapor pressure. For a long mission, it is essential to ensure that the O-rings have a constant supply of wetting fluid -- water or oil[3]. Should this not be provided, the O-ring seal leak rate will increase substantially with the shaft stationary; the O-ring will fail catastrophically rather quickly with the shaft rotating or translating. Sealing at low temperatures may necessitate increased O-ring compression with resultant increased friction and wear, which will shorten O-ring operating life. Leak rates for static O-ring seals range from about 10^{-7} standard cm^3/s to 4×10^{-11} standard cm^3/s per centimeter of seal length (Ref. 6) depending upon the elastomer, the amount of compression used on the O-ring, and the temperature. Leak rates for dynamic operation are perhaps 5 to 10 times as great. For extended missions, the major questions associated with O-ring shaft seals relate to the long-term effects of vacuum and radiation on the properties of the elastomer:

(1) Will the loss of plasticizers and stabilizers in the elastomer take place and cause stiffening and/or cracking?

(2) Will anti-rad components, if used to increase radiation resistance of the elastomer, be lost by long-term vacuum exposure?

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[3]Or grease.

(3)   Will long-term compression set cause the O-ring compression to be relieved after a long period and result in leakage?

The chevron seal (Fig. 2) has been employed in many forms for pneumatic and hydraulic seals but only a few types have been used in vacuum shaft seal applications. The classic Wilson elastomeric vacuum shaft seal uses the pressure differential of the system to provide the sealing force to the lip of the elastomer. For stiffer seal materials such as Teflon, particularly at low temperatures, the pressure forces must be supplemented with other forces, generally supplied by various forms of metal springs.

The C-shaped Teflon spring-backed seal has been employed in some dynamic-shaft vacuum-seal applications without lubricant, but little if any information is available on its leak rate, friction, and wear life. It is claimed to be capable of "sealing" at temperatures ranging from $+204°C$ $(+400°F)$ to $-185°C$ $(-300°F)$. Because sealing forces, hence frictional forces, are high, this seal is not likely to be suited to the jet vane actuator application.

Reference 6 quotes leak rates of 2×10^{-6} standard cm^3/s for a stationary Wilson lip seal 0.96 cm (3/8 in.) in diameter and 5×10^{-6} standard cm^3/s for the same seal with the shaft rotating at 60 revolutions/min. This seal, used for slow linear motion, is quoted as having a leak rate of 10^{-5} standard $cm^3/s/cm$ of travel, independent of speed or time. Since the Wilson seal employs a pressure-loaded elastomeric lip seal, whereas the chevron seal uses Teflon with spring backing, these leak rates are not likely to be applicable and must be determined.

The lapped graphite face seal (Fig. 3) is used extensively for a liquid seal (e.g., automotive water pumps), requires a lubricant and coolant, either water or oil, and is best suited to high-speed rotation. Graphite face seals have been used to seal high-speed rotating shafts in vacuum applications under dynamic pumping conditions and where oil spray lubrication was employed. This seal is not likely to have a gas leak rate acceptable for an actuator application.

The surface tension seal (Fig. 4) first described in Ref. 7, uses a substance to bridge the gap between the shaft and the housing; the substance must be liquid when the shaft is rotated and may be liquid or solid when the shaft is stationary. It is perhaps the only shaft penetration seal which can be truly termed "hermetic" if, as is usually the case, a liquid metal is used as

the sealing material.   For vacuum seals, the sealing substance must have high surface tension, low vapor pressure, a low melting point, and suitable "wetting" characteristics for the shaft and housing metals.   Desirable properties for such a substance might be as follows:

Melting point	**-50°C (-58°F)**
Surface tension, γ	0.500 N/m (500 dynes/cm), minimum
Vapor pressure	10^{-10} N/m^2 ($<10^{-12}$ torr) at 25°C (77°F)

In addition, its viscosity should be low.

No known material will satisfy all these requirements, but the most important properties are (1) low vapor pressure to minimize **loss** of the substance through evaporation and **(2)** high surface tension so that the gap need not be made unacceptably narrow.   These characteristics for some possible materials are given in Table **3.**   From Table **3,** it appears that only metals such as indium, tin, gallium (and perhaps lead and bismuth), and Ga-Sn-In eutectic possess the required surface tensions and vapor pressures; however, high melting points for indium and tin present obvious problems. If the actuator is to be operated at a temperature below the melting point of the metal, as will almost certainly be the case, it will be necessary to supply heat to melt the seal for actuator operation.   Although a very close clearance D is required for this seal, its length is quite short.   It is essential, however, that no rubbing take place between the shaft and housing, or new metal alloys will be formed which may substantially change the melting point of the liquid metal, dewet the sealing surfaces, and cause leaks.

The major advantage of this seal is that it is a true hermetic seal with no leak and with essentially no gas permeation, whether the metal is liquid or solid.   It should also be immune to radiation.   Its major drawback is that the heating power requirement, undesirable from a power management stand-point, adds both complexity and a failure mode.   It would appear that this seal might be best suited to actuators requiring infrequent use.   An attractive variation of this seal involves its use as a one-time hermetic seal for a one-shot actuator required to operate after a very long time in space. In this version, indium --which has very low shear and tensile strength -- would seal the shaft in the form of a thin membrane "soldered" to shaft and housing,   **A** heater would be provided only for operating the actuator during

ground test; for actual use, the seal need not be melted but would be sheared by operation of the actuator.

The major unresolved questions regarding this seal pertain to long-term usage for, say, scan actuators where dewetting or alloying with base metal effects must be determined. Investigations into the power requirements for melting the seal are also required.

The labyrinth seal (Fig. 5) or controlled leakage seal, is described in Refs. 8-11 inclusive. It is a noncontact seal that provides a low conductance molecular flow path from the actuator to the vacuum environment. This path is a very narrow annular clearance between the actuator output shaft and housing, through which molecules of lubricating oil vapor escape. The loss of oil vapor is proportional to the conductance of the leak path. Since the conductance varies directly with path length and inversely with annular clearance to the third power, very narrow and at least moderate length annuli are desirable.

For very long missions, both low conductance through the labyrinth and a low vapor pressure oil with a narrow molecular weight range are required to hold the loss of oil to a small amount during the mission (a few grams or less) and to maintain fixed lubricant characteristics. The loss of oil must be kept very low to prevent its condensation on the spacecraft's optical or thermally controlled surfaces and degrading them. If an oil is made up of constituents with a wide range of molecular weights, the vapor pressures associated with them will also vary widely and the lower molecular weight fractions, having higher vapor pressures, will evaporate first (Refs. 10 and 12), increasing the viscosity, pour point, and other properties of the remaining oil. Such changes could have serious consequences in applications where torque margins are small and operation is required after many years of space vacuum exposure, as may be the case with jet vane actuators.

A further difficulty associated with the selection of lubricating oils for extended missions is that the requirement for minimal oil loss necessitates an oil with very low vapor pressure. This in turn dictates a higher-viscosity oil than that required to obtain the desired lubricant oil film thickness, resulting in higher friction, unless the actuator is heated during operation. This type of seal is poorly suited to exposure to a wide temperature range (unless of course the wide range exists only for relatively short periods of

time) because of the drastic change in vapor pressure -- hence oil evaporation rate -- with temperature. This change is of the order of 10^4 for a temperature range of **93°C** (200°F) and results in increased oil losses by a factor of 10^4.

Because of the aforementioned difficulties, this type of seal has limitations for some future missions. However, the labyrinth seal may be employed as a backup seal to the "hermetic" seal in many actuators with little or no increase in weight **or** size of the actuator. In some cases the output shaft journal bearing can serve as the labyrinth seal, such bearings typically have a clearance of a few micrometers, or ten-thousandths of an inch. Reservoirs of lubricant in close proximity to bearings and gears would also be required.

The tubular elastomer seal (Fig. 6) has the advantage of being a no-friction, no-wear seal, but, like the O-ring seal, involves an organic material whose properties may degrade after long vacuum and radiation exposure. In order to permit the required angular rotation or elongation, a considerable length is required which will adversely affect the size of the actuator. This type of seal is probably not suited to the jet vane actuator (because of the varying torsional spring rate of the seal as a function of its temperature) nor to the linear actuator (because of the large extension required). Its major advantage is that it is a "hermetic" no-wear seal, but it is best adapted to applications, unlike those under discussion, where motion is limited to very small angles or extensions.

The magnetic fluid seal (Fig. **7** and Ref. 13) is a relatively new development, employing.a magnetic fluid which is retained by the magnetic field in the annulus between the shaft to be sealed and permanent magnets located in the housing. The base fluid may be water, kerosene, or a variety of oils including some, but not all, with fairly low vapor pressures. Colloid-sized 10-nm (100-$\overset{o}{A}$)magnetic particles (magnetite, $FeO \cdot Fe_2O_3$), coated with a dispersant, are suspended in the base fluid.

These seals are usually employed as "hermetic" feedthroughs for vacuum chambers and are said to be leak tight to beyond 10^{-11} standard cm^3/s of helium. The seals exhibit a viscous torque drag due to shearing of the oil in the close clearance magnetic gaps and also exhibit a starting torque drag several times larger than the running torque if the seal has been

at rest for a few minutes. This behavior may be attributable to alignment of the magnetic particles in the magnetic field. The seal is said to maintain its integrity with the magnetic fluid frozen (to cryogenic temperatures) and when thawed to operate with no degradation.

The major disadvantages of this seal for the applications in question appear to be the viscous drag which increases with decrease in temperature and the limitations of the base fluids used. The viscous drag (breakaway) torque for a 1.27-cm (1/2-in.) diameter seal is about 0.37 N-m (3.3 in. -lb) at -7"C (20°F); for a 0.63-cm (1/4-in.) diameter seal the breakaway torque is about 0.015 N-m (2.2 in. -oz). On the 0.63-cm (1/4-in.) diameter seal, running torque increases with speed as would be expected, but it decreases with increased speed at low temperatures (but not at higher temperatures) on the 1.27-cm (1/2-in.) diameter seal. This suggests that the fluid is thixotropic in behavior (i.e., non-Newtonian), a property attributable to many colloids, according to Ref. 14. The second disadvantage is that certain fluids such as silicone oils which have desirable **viscosity-vs-temperature** characteristics and low vapor pressures (see Ref. 15) have not been found to be suitable for producing magnetic fluids. Two low vapor pressure base fluids are available; one, a silicate ester liquid, is said to have a vapor pressure of about 6×10^{-3} N/m^2 (5×10^{-5} torr) at 27°C (80°F);the other is a fluoroalkylpolyether which is stated to have about a decade lower vapor pressure than the ester. Such vapor pressures might result in the loss of excessive amounts of base fluids, with unknown effects on the seal, in the applications in question.

Because of these many uncertainties, this seal requires investigations along the following lines:

(1)   Possible development of more suitable lower-vapor-pressure magnetic fluids.

(2)   Compatibility (i.e., immiscibility) of magnetic fluid with actuator lubricant.

(3)   Stray magnetic field produced by the seal.

(4)   Radiation effects on the magnetic fluid.

From the foregoing discussion, it would appear that the magnetic fluid seal is a candidate for the high-torque rotary seal but not for the jet vane actuator.

The metal bellows seal appears to be a leading candidate for sealing the linear actuator shaft, since it is a true metal membrane hermetic seal which should be immune to radiation, vacuum, and time. The effective sealing area of the bellows will be larger than the seal area of the present O-ring, and this will result in greater unbalanced forces which will vary the motor current required for fixed speed of the actuator in the two directions of travel. The variation in the two directions caused by the unbalanced pressure force may be reduced by (1) reducing the effective area of the bellows to a minimum and (2) reducing the internal pressure of the actuator to the minimum acceptable.

VIII.   RADIATION SUSCEPTIBILITIES OF CANDIDATE SEALS

Many of the shaft seals contain polymeric materials which are more susceptible to radiation degradation from on-board radioisotope thermoelectric generators and planetary radiation belts than most other materials used on spacecraft. Such polymeric seal materials and total acceptable gamma doses for the seals under discussion are given in Table **4.**

As an example, the total gamma dose at a distance of 1. 52 m (5 ft) from the radioisotope thermoelectric generators on Mariner Jupiter-Saturn 1977 spacecraft over a 10-year period is approximately 10^2 J/kg (10^4 rad).

The degradation threshold fluences noted in Table **4** for seal materials are comparable to those for many other polymeric sealants, insulating materials, and electronic parts (diodes, field effect transistors, capacitors, etc.) used on spacecraft and will have to be taken into account in the specific mission design with additional shielding or alternative material considered where required.

IX.   DISCUSSION

From the list of candidate seals in Table 2, the following three seals should be deleted for the actuator applications in question **for** the reasons noted:

(1)   Face seal -- high gaseous leak rate, which renders it unsuitable for actuators; best suited to sealing liquids on high-speed rotating shafts.

(2)   Tubular elastomer seal -- limited angular rotation; best suited to sealing shafts with very small angular rotation or extension.

(3)   Surface tension seal -- high power requirement to melt seal. Best adapted to actuators requiring single operation, or multiple operation on mission where actuation will not be required at times of peak power demand.

The labyrinth seal, although not eliminated from consideration, is best adapted to missions involving limited actuator temperature ranges, and hence is not a leading candidate for likely future missions. This seal can probably best serve as a backup to the main seal.

The metal bellows seal appears to be a leading candidate for the linear actuator. If a labyrinth backup seal is employed, the use of the shaft bearing as the labyrinth may not be feasible unless the bellows is placed on the high-pressure side (away from the vacuum) of the bearing.

The remaining seals, i. e., the elastomeric O-ring, chevron ring, and magnetic fluid seals, require further testing and evaluation to determine their low-temperature limitations, both operating and survival.

X.   CONCLUSIONS

On the basis of the foregoing discussion on candidate shaft seals and consideration of Appendix **A,** which lists their advantages and disadvantages, it is proposed that the specific actuators under discussion be limited to the following seals:

Rotary actuator, high torque	Linear actuator	Rotary actuator, jet vane
O-rings	Metal bellows	O-rings
Chevron rings	Labyrinth[4]	Labyrinth[4]
Labyrinth[4]		
Magnetic fluid		

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[4] Backup seal for use in combination with a hermetic seal.

Seal tests are required better to define limitations of the O-ring, chevron ring, and magnetic fluid types of seals and to aid in the selection of the dynamic/static shaft seal best suited to a specific mission.

Factors that will influence seal selection for a specific mission will include the following:

(1)　Required <u>operating</u> life of the actuator.

(2)　Required <u>survival</u> life of the actuator.

(3)　Actuator operating profile throughout mission.

(4)　Actuator temperature range during:

(a)　Operating periods.

(b)　Dormant periods.

(5)　Total radiation exposure fluences.

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Table 1. Characteristics of typical Mariner actuators

Item	Rotary actuator (high torque)	Linear actuator	Rotary actuator, jet vane (low torque)
Output torque or force	5.7 N-m (50 in.-lb)	445-890 N (100-200 lb)	0.014-0.021 N-m (2-3 in.-oz)
Angle of rotation or stroke	200-300 deg	1.91 cm (3/4 in.)	±30 deg
Rotational or linear speed	1 deg/s	1.27 cm/s (1/2 in./s)	10^3 deg/s
Backlash, maximum	0.05-0.10 deg	0.010 cm (0.004 in.)	None[a]
Output shaft diameter, cm (in.)	1.27-2.54 (1/2-1)	1.27 (1/2)	0.32 (1/8)
Required life, cycles	10^5	10^5	10^5
Maximum permissible seal friction	Not critical, approx. 0.11 N-m (1 in.-lb)	Not critical, approx. 8.9 N (2 lb)	Critical, 0.002 N-m (0.25 in.-oz)
Permissible seal leak, standard cm^3/s	10^{-5}	10^{-5}	10^{-5}
Envelope size	10.2 cm (4 in.) diameter × 12.7 cm (5 in.) length	8.9 cm (3-1/2 in.) diameter × 15.2 cm (6 in.) length	5.1 cm (2 in.) diameter × 5.1 cm (2 in.) length
Shaft position indicator	Film rotary potentiometer	Linear variable reluctance transducer	Conductive. plastic rotary potentiometer
Weight, kg (lb)	1.71 (3-3/4)	1.25 (2-3/4)	0.45 (1)
Motor type	Permanent magnet stepper	dc	Brushless dc
Power, maximum, W[b]	10	45	4

[a]Direct drive.

[b]Does not include heater power, if any.

Table 2. Preliminary candidate shaft seals[a]

Rotary actuator (high torque)	Linear actuator	Rotary actuator, jet vane (low torque)
O-rings	O-rings	O-rings
Chevron rings	Chevron rings	Chevron rings
Face seal	Labyrinth	Face seal
Surface tension	Metal bellows	Surface tension
Labyrinth		Labyrinth
Tubular elas-tomer seal[b]		Tubular elas-tomer seal
Magnetic fluid seal[b]		Magnetic fluid seal

[a]See Section X, p. 15, for final selection of candidate seals.
[b]Not included in Ref. 5.

Table 3. Characteristics of some possible materials for surface tension seals

Item	Water	Mercury (Hg)	Indium (In)	Tin (Sn)	Gallium (Ga)	Ga-Sn-In eutectic	Organic[a] liquids
Melting point, °C (°F)	0 (32)	-39 (-38)	156 (313)	231 (448)	30 (86)	5 (41)	-38 (-36)[b]
Surface tension γ, N/m (dynes/cm)	0.073 (73)	0.487 (487)	0.559 (559)	0.520 (520)	0.735 (735)	0.60 (~600)	0.02 (~20)
Gap D for 1.01×10^4 N/m² (14.7 psi) pressure differential, cm (in.)	1.5×10^{-4} (6×10^{-5})	7.6×10^{-4} (3×10^{-4})	1×10^{-3} (4×10^{-4})	1×10^{-3} (4×10^{-4})	1.5×10^{-3} (6×10^{-4})	1.3×10^{-3} (5×10^{-4})	5×10^{-5} (2×10^{-5})
Vapor pressure at 25°C, (77°F) N/m² (torr)	2.8×10^3 (21)	$\sim 10^{-1}$ ($\sim 10^{-3}$)	$<10^{-10}$ ($<10^{-12}$)	$<10^{-9}$ ($<10^{-11}$)	$<10^{-10}$ ($<10^{-12}$)	$\sim 10^{-9}$ ($\sim 10^{-11}$)	$\sim 10^{-5}$ ($\sim 10^{-7}$)

[a]DC 704 silicone vacuum pump fluid.
[b]Pour point.

Table **4.** Radiation resistance of polymeric seal materials[a]

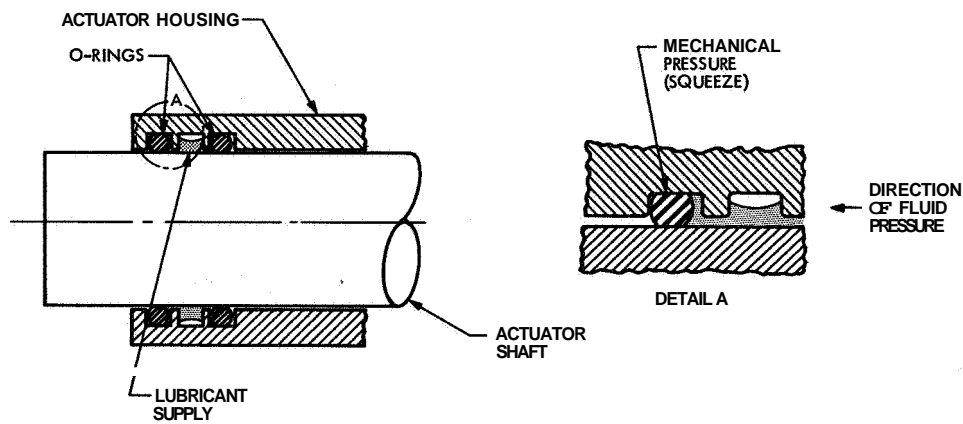| Material	Seal	Acceptable Gamma Dose J/kg (rad)
Viton A	O-ring	10^4 (10^6)
Teflon (TFE)	Chevron	10^3 (10^5)
Chlorophenyl methyl polysiloxane (GE F-50)	Labyrinth	$<10^6$ ($<10^8$)
Silicate esters	Magnetic fluid	$<10^6$ ($<10^8$)
Perfluoroalkyl polyether (Krytox 143AC)	Magnetic fluid	$<10^6$ ($<10^8$)

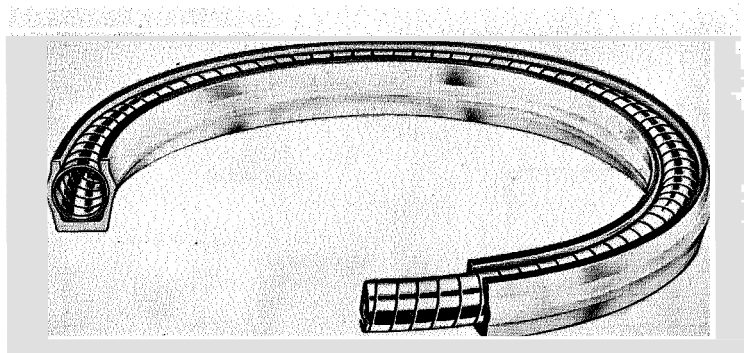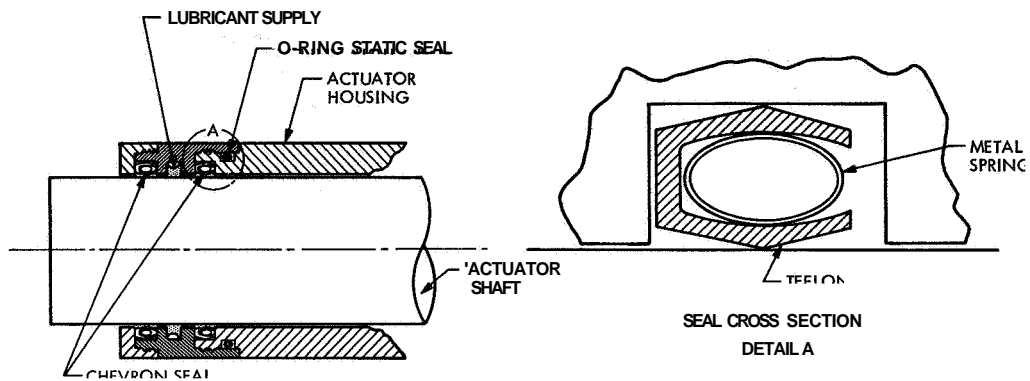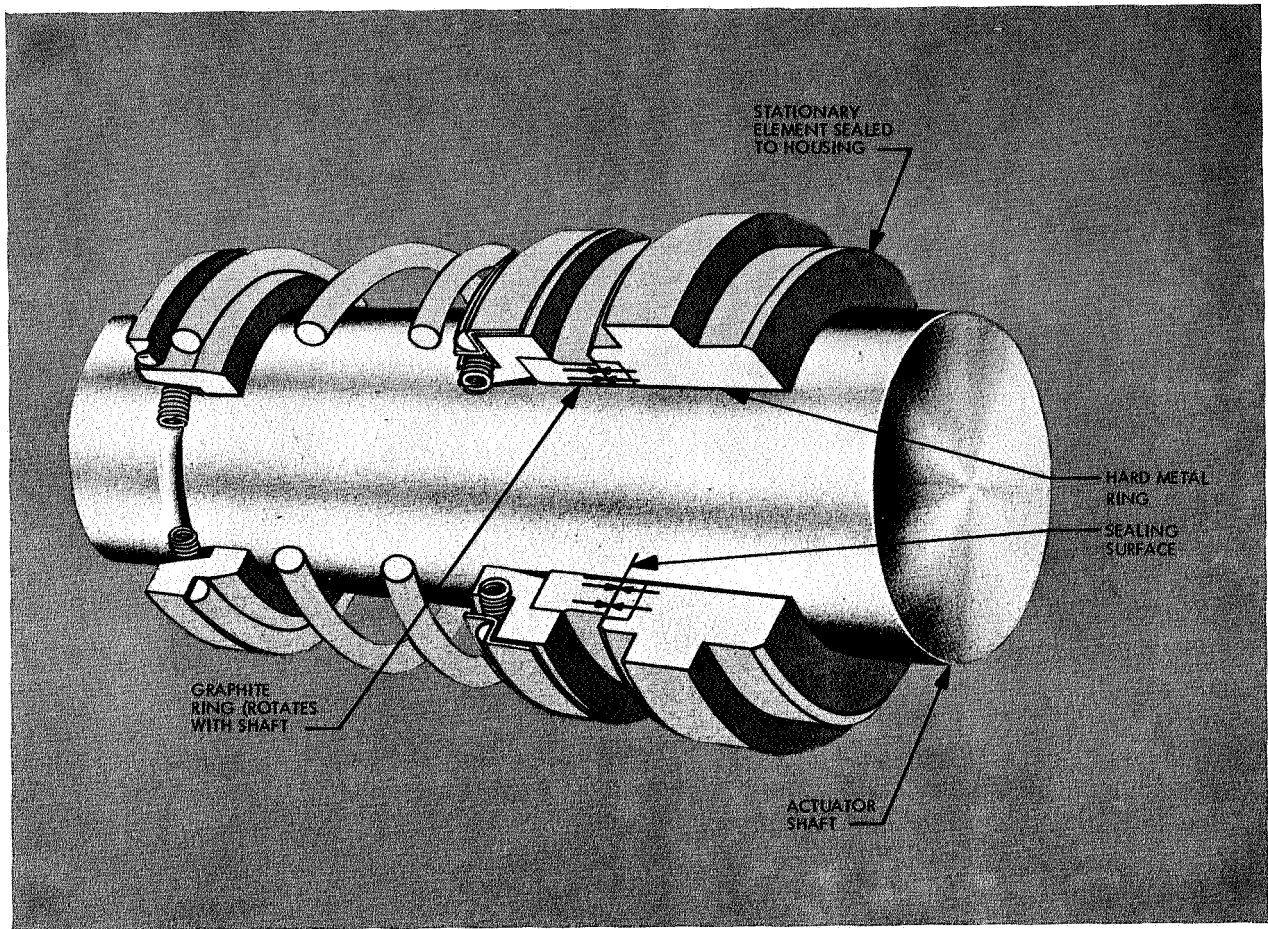[a]Refs. 14 and 16.

Fig. 1. O-ring seal

Fig. 2. Chevron ring seal

Fig. 3. Face seal
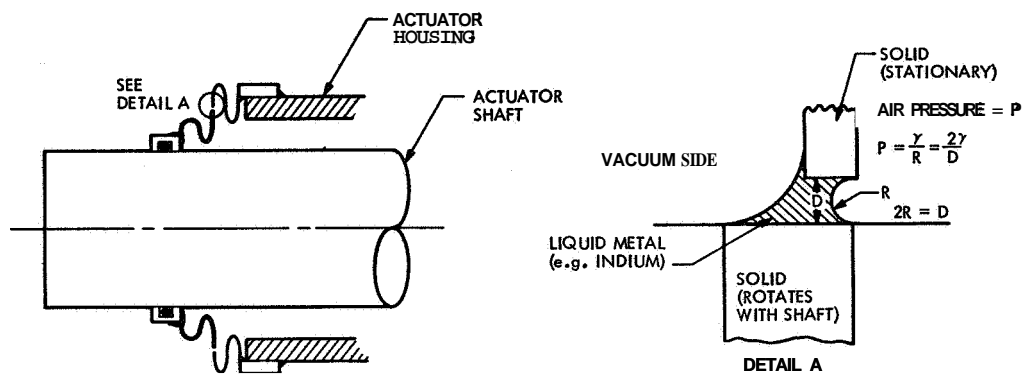
Fig. **4.** Surface tension seal

LABYRINTH
CLEARANCE,
0.005-0.008 mm
(0.0002-0.0003 in.)

HOUSING

BEARING
CLEARANCE
(ALSO SERVES
AS LABYRINTH)
0.0025-0.0050 mm
(0.0001-0.0002 in.)

ACTUATOR
SHAFT

Fig. 5.   Labyrinth seal

ACTUATOR
SHAFT

CLAMP (TYP

ACTUATOR
HOUSING

ELASTOMER
TUBE

DETAIL A

REENFORCING
METAL RINGS

Fig. 6.   Tubular elastomer seal

INSERT SHOWS MAGNETIC FLUID (BLACK) TRAPPED BETWEEN SHAFT POLE TIPS AND STATIONARY POLE BLOCK

PERMANENT MAGNET

POLE PIECES
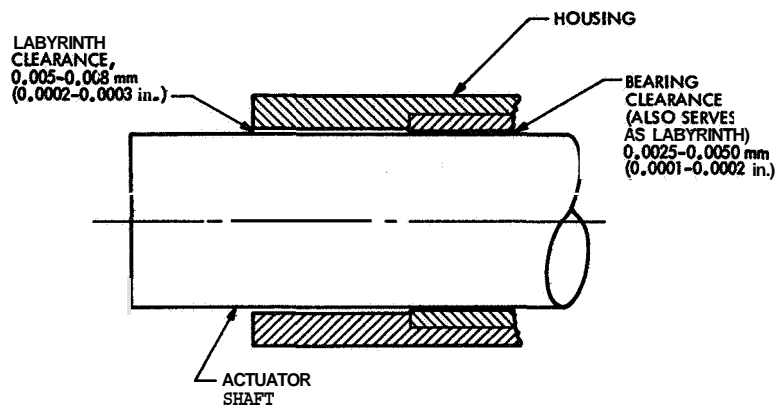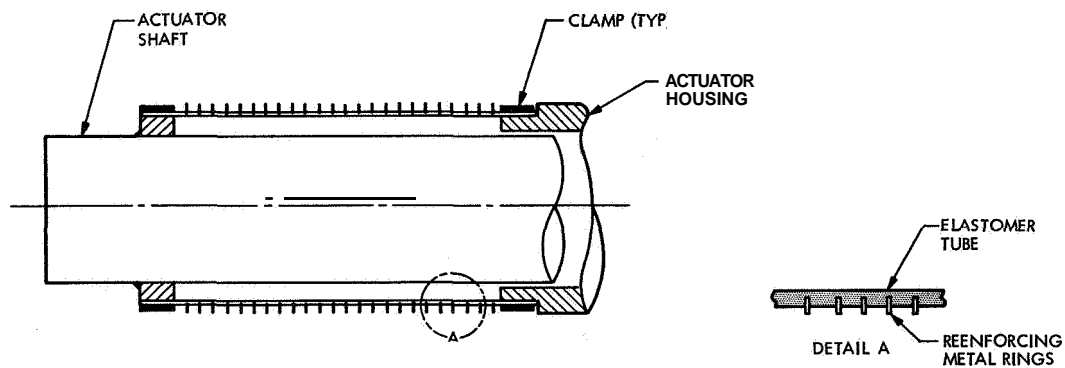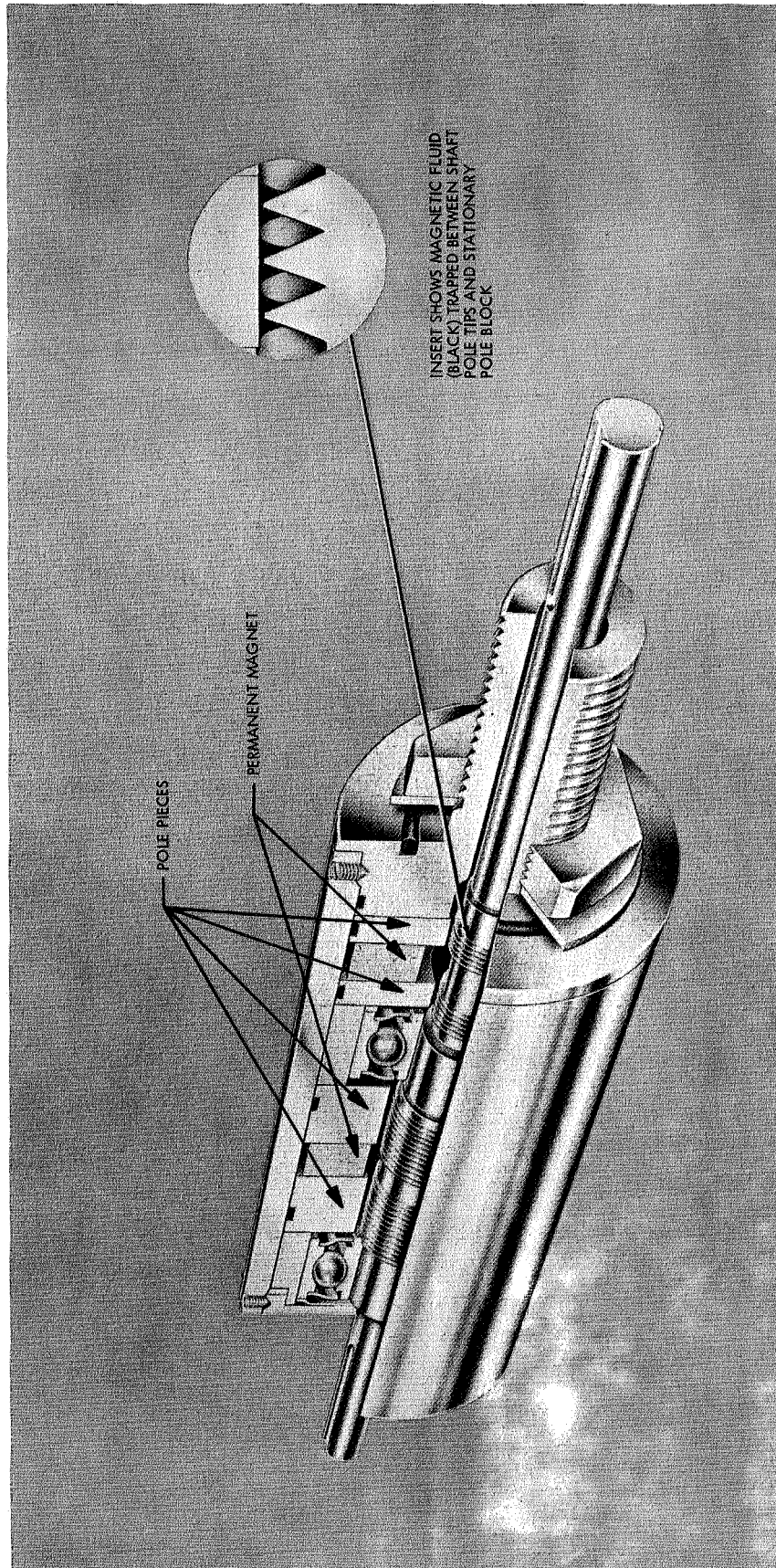
Fig. 7. Magnetic fluid seal. (The seal depicted is a magnetic fluid, hermetic rotary feedthrough for vacuum chamber use.)

APPENDIX A

ACTUATOR SHAFT SEALS -- ADVANTAGES, DISADVANTAGES, AND UNKNOWNS OF VARIOUS TYPES

I.   ELASTOMERIC O-RING SEAL

A.   Advantages

(1)   Extensive experience with this type seal over nominal temperature range and up to 2-year lifetime.

B.   Disadvantages

(1)   Experiences (minimal) loss of grease or oil, at least over nominal temperature range; i. e. , seal is not "hermetic."

(2)   Has appreciable friction -- disadvantage for some applications.

(3)   0-seal requires "wetting" by some liquid to provide low leak rate; if this liquid is lost, actuator will be subject to a higher leak rate and much greater wear.

(4)   Not suitable for high-speed shaft rotation. [5]

C.   Unknowns

(1)   Lower temperature limit (both static and dynamic) unknown -- must be tested.   Extension of low-temperature limit may require greater O-ring compression and/or different elastomer with resulting greater friction torque.

(2)   Long-term life of suitable elastomers in combined vacuum, radiation and oil environments unknown.

(3)   Sealing capability and break-out friction after very long storage periods (in space) unknown; compression set of elastomer and possible evaporation of lubricant would affect these.

(4)   Break-out torque at temperatures below lubricant freezing point and elastomer brittle point unknown.   Possibility of damage to O-rings must be investigated, as must be possible gross leakage.

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[5] Not a disadvantage for the application in question.

II.	CHEVRON SEAL (TEFLON C-SHAPED SPRING-BACKED SEAL, e. g. , OMNISEAL, BAL SEAL, TEC-RING, ETC.)

A.	Advantages

(1)	Should have lower temperature limit than O-rings.

(2)	Teflon does not require "wetting" like elastomers to seal. May be more immune to lubricant loss than O-ring.

(3)	Suitable for higher shaft speeds than O-rings. [6]

B.	Neutral Factors

(1)	Teflon is more immune to long-term vacuum exposure, but not to radiation, than are elastomers.

C.	Unknowns

(1)	Friction torque, leak rate, required compression, and lubrication requirements, all unknown, must be tested.

III.	LAPPED-GRAPHITE FACE SEAL

A.	Advantages

(1)	Suitable for very high shaft speeds. [6]

B.	Disadvantages

(1)	Requires ample oil lubrication for sealing at all speeds and for cooling at high speeds. Will fail rapidly without lubrication.

(2)	Leak rate unknown but higher than O-ring seals. Not likely to be suited to unpumped (i. e., static) vacuum sealing.

IV.	LIQUID-METAL SURFACE TENSION SEAL

A.	Advantages

(1)	True hermetic seal when metal is either frozen or liquid.

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[6] Not an advantage for the application in question.

(2) **No** oil or grease required on seal. Low-viscosity, `high-vapor-pressure`, low-pour-point oils may be used in actuator.

(3) Unaffected by radiation and low temperature.

(4) In normal mode, frozen and stationary or liquid, revolving or stationary, it has no leak and essentially no permeation.

B. Disadvantages

(1) Power may be required to melt metal seal to operate actuator-- this may be of the order of 5 W (continuous during actuator operation).

(2) Requires development.

(3) Requires very close clearance but for very short length, i.e., 0.005-0.008 mm (0.0002-0.0003 in.) radial clearance by 0.25 mm (0.01 in.) long; shaft must not touch housing.

C. Unknowns

(1) It should be possible to make this seal fail-safe, if heater malfunctions, by shearing the seal -- estimated to require a torque of 5.7 N-m (50 in.-lb). This requires testing to determine the torque required and to determine whether shearing the seal opens up a large leak.

V. LABYRINTH SEAL (WITH LOW VAPOR PRESSURE LUBRICANT)

A. Advantages

(1) Fail-safe. Cannot lock shaft and prevent actuator from operating.

(2) Has flown successfully on several satellites.

(3) Frictionless.

(4) Should be unaffected by radiation, if oil is unaffected by total fluence.

(5) Suitable for high speeds. [6]

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[6]Not an advantage for the application in question.

B.   Disadvantages

   (1)   Best suited to low-temperature operation, as the lower the
          temperature, the lower the rate of loss of oil, due primarily to
          lower vapor pressure. However, with very-low-vapor-pressure
          oils, the pressure at low temperature might be inadequate to
          protect the actuator.

   (2)   Escaping oil vapor may cause problems with optical and thermal
          control surfaces.

   (3)   Requires a fairly long close clearance, like 0.0127 mm (0.0005
          in.) radial clearance.

   (4)   Low pressure in actuator (molecular flow regime) reduces heat
          transfer.

   (5)   Long-term operation would necessitate use of a highly refined
          oil of narrow vapor pressure range (i.e., range of molecular
          weights) to prevent a large change in viscosity and vapor pressure
          toward the end of the mission.

   (6)   Presents possibility of corrosion and/or contamination prior to
          launch, as actuator is open to atmosphere.

   VI.   TUBULAR ELASTOMER SEAL

A.   Advantages

   (1)   No sliding sealing surface as is the case with O-rings, and
          hence no friction and no leak; leakage is limited to permeation.

   (2)   Requires no grease or oil, yet provides "hermetic" seal to
          permanent gases in actuator housing.

   (3)   No speed limit.

B.   Disadvantages

   (1)   Requires considerable torque to rotate through 200-300 deg;
          stiffness increases with decreasing temperature.

   (2)   May be better adapted to small-diameter, small-angle-of-
          rotation applications, such as jet vane actuator.

(3)   Seal will not work below brittle point of rubber, -54 to *-73°C*
      (-65 to -100°F), unless seal is heated.

(4)   Requires development.

VII.   MAGNETIC FLUID SEAL

A.   Advantages

(1)   Frictionless; viscous drag only.

(2)   Suitable for high-speed rotation.[6]

(3)   Essentially hermetic seal.

B.   Disadvantages

(1)   Maximum temperature approximately 149"C (300°F).[5]

(2)   Base fluids with very low vapor pressure are not presently
      available.

(3)   Seal is heavier and longer than other seals.

(4)   Exhibits viscous drag at room temperature, increasing con-
      siderably at lower temperatures.

(5)   Breakaway torque is two to three times running torque.

C.   Unknowns

(1)   Magnetic field requires investigation to determine acceptability
      for missions where only very low on-board magnetic fields are
      permissible.

(2)   Effect of low temperatures on magnetic fluid must be determined.

(3)   Effect of radiation on magnetic fluid must be determined.

(4)   Long-term effects of evaporation of magnetic base fluid unknown.

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[5]Not a disadvantage for the application in question.

[6]Not an advantage for the application in question.