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Space Tribology

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INTRODUCTION

Space tribology is a subset of the lubrication field dealing with the reliable performance of satellites and spacecraft (including the space station). It encompasses the entire gamut of tribological regimes, including elastohydrodynamic lubrication (EHL), parched EHL, transient EHL, boundary lubrication, and mixed lubrication.

Historically, the lubricants for space mechanisms have been chosen on the basis of space heritage rather than on the latest technology or the best available materials. Early in the space program, this strategy almost always paid off because mission lifetimes and mechanism duty cycles were minimal. When mission lifetimes were extended, other spacecraft components, such as electronics, batteries, and computers failed before the lubrication systems [Fleishauer and Hilton, 1991]. However, in the last decade these ancillary mechanism components have been greatly improved and the tribological systems have become the limiting factor in spacecraft reliability and performance. Although the tribological components represent only a small fraction of the spacecraft's cost, they are often single point failures that can render an expensive satellite totally useless.

A classic example was the Galileo spacecraft. Galileo was launched from the Space Shuttle Atlantis in 1989. One of the most important components was a high gain, umbrella like antenna that was closed during launch to conserve space. In 1991 this antenna only partially unfurled. It was concluded that four of the 18 ribs were stuck in place. In ground based tests [Miyoshi and Pepper, 1992], it was determined that the Ti-6Al-4V alignment pins, which were lubricated with a bonded dry film lubricant, had galled and seized due to loss of the lubricant. The mission was salvaged only by the use of the low gain antenna and advances in data transmission technology.

LUBRICATION REGIMES

The purpose of lubrication is to separate surfaces in relative motion by interposing a third body that has a low resistance to shear so that the two surfaces do not sustain serious damage or wear. This third body can be a variety of different materials, including adsorbed gases, reaction films, and liquid or solid lubricants.

Depending on the type of third body and its thickness, several different regimes of lubrication can be identified. One way of depicting these regimes is by use of the Stribeck curve shown in Figure 1. Stribeck performed a series of journal bearing experiments in the early 1900's [Stribeck, 1902]. He measured the coefficient of friction as a function of load, speed, and temperature. Later, Hersey performed similar experiments and devised a plotting format based on a dimensionless parameter, ZN/P [Hersey, 1914]. This Stribeck/Hersey curve takes the form of the coefficient of friction as a function of viscosity (Z), velocity (N), and load (P).

At high values of this parameter, which occur at high speeds, high viscosities, and low loads, the surfaces are completely separated by a thick ($> 0.25 \mu\text{m}$) lubricant film. This is the area of hydrodynamic lubrication where friction is determined by the rheology of the lubricant. For nonconformal, concentrated contacts where loads are high enough to cause elastic deformation of the surfaces and pressure-viscosity effects on the lubricant, another regime, elastohydrodynamic lubrication (EHL) occurs. Film thickness in this regime ranges from 0.025 to $1.250 \mu\text{m}$.

As this parameter decreases, film thickness decreases and surface interactions start taking place. In this regime, where both surface interactions and fluid film effects occur, is referred to as the mixed regime. Finally, at low values of ZN/P , the boundary lubrication regime is entered.

The boundary lubrication regime is a highly complex arena involving metallurgy, surface topography, physical and chemical adsorption, corrosion, catalysis, and reaction kinetics [Godfrey, 1980; Jones, 1982]. The most important aspect of this regime is the formation of protective surface films to minimize wear and surface damage. For space mechanisms, AISI 440C stainless steel is the most common bearing material.

The formation of lubricating films is governed by the chemistry of both the film former as well as the bearing surface and other environmental factors. The effectiveness of these films in minimizing wear is determined by their physical properties. These include shear strength, thickness, surface adhesion, film cohesion, melting point or decomposition temperature, and solubility in the bulk lubricant.

Typically, the EHL, mixed, and boundary lubrication regimes occur in lubricated space mechanisms with the boundary lubrication regime being the most stringent. However, a subdivision of EHL, starvation theory, has been described a number of years ago [Wedeven et al., 1971]. It describes the situation occurring in ball bearings having a restricted oil supply, in which pressure build-up in the inlet region of the contact is inhibited, resulting in a film thickness thinner than calculated by classical EHL theory [Dowson and Higginson, 1959; Hamrock and Dowson, 1981].

But starvation theory fails to adequately describe instrument bearing behavior since there is no oil meniscus. Another subdivision of EHL, parched elastohydrodynamics, describes this behavior where there is no free bulk oil in the system [Kingsbury, 1985; Schrittz et al., 1994]. The lubricant films in this regime are so thin that they are immobile outside the Hertzian contact zone. This regime is of particular importance to space mechanisms since parched bearings require the least driving torque and have the most precisely defined spin axis.

Finally, another area of EHL that is of importance to space mechanisms involves transient or non-steady state behavior. Unlike steady state EHL behavior, non-steady state behavior is not well understood. However, many practical machine elements such as rolling element bearings, gears, cams, and traction drives operate under non-steady state conditions. This is where load, speed, and contact geometry are not constant over time. In particular, stepper motors, which are commonly used in many space mechanisms, operate in this regime. This regime has been studied theoretically for line contacts [Wu and Yan, 1986; Ai and Yu, 1988; Hooke, 1994] and experimentally for point contacts [Sugimura et al., 1998].

MECHANISM COMPONENTS

Spacecraft contain a variety of instruments and mechanisms that require lubrication. Devices include solar array drives, momentum, reaction, and filter wheels, tracking antennas, scanning devices, and sensors. Each of these devices has unique hardware, and therefore lubrication requirements.

Gyroscopes, which are used to measure changes in orientation, operate at high speeds, typically between 8,000-20,000 RPM with high accuracy. This makes the bearings the most important element of a gyroscope. Fluctuations in the bearing reaction torque, noise, and excess heat generation can cause a loss of null position in the gyroscope. The ideal lubricant for a gyroscope provides a high level of wear protection, produces minimal friction, and has a low evaporation rate [Kalogeris et al., 1993]. Also, a fixed, small (3 mg) amount of lubricant is used and must provide lubrication throughout the life of the gyroscope. Gyroscope gimbal supports are low speed applications and the bearings operate in the boundary regime only.

Momentum wheels, which typically operate between 3000-10,000 RPM, pose their own lubricant selection criteria. Currently, the majority of problems experienced by momentum wheels are related to the lubricant. Inadequate lubrication, loss of lubricant, or lubricant degradation are the reason for the majority of wheel failures [Kalogeris et al., 1993]. As higher speed wheels are designed, lubricants will be subjected to higher operating temperatures, which can increase creep or degradation rates. Current design practices to ease lubricant problems include use of improved

synthetic lubricants, labyrinth seals and barrier coatings, lubricant impregnated retainers, or a lubricant resupply system.

Reaction wheels have similar design concepts as momentum wheels, but operate at lower speeds. The support bearings spend more time in the mixed lubrication regime. Therefore, lubricants chosen for reaction wheel use must also have good boundary lubrication characteristics.

Control momentum gyroscopes (CMG) combine the aspects of the gyroscope and the momentum wheel to provide attitude control of a spacecraft. Therefore, considerations of both groups must be weighed when selecting a lubricant for use in a CMG [Kalogeras et al., 1993].

Devices that utilize scanning or rotating sensors represent another space mechanism that requires lubrication. An example would be a scanning horizon sensor. This device detects the earth's horizon, which allows spacecraft to orient themselves. Moderate operational speeds (400-1,600 RPM) and low loads in the bearings make lubricant selection easy. On the other hand, sensors that use oscillatory motion place a high demand on the lubricant. Typically, the angle of oscillation is slight and the bearing operates in the boundary lubrication regime only. With the small oscillatory angle, no new lubricant is brought in the contact zones [Postma, 1999].

Slip rings are another example of a common mechanism used in space applications that require lubricants. Low speed operation and electrical conductivity are two important factors that affect lubricant selection. Excessive electrical noise is the most common failure mechanism in slip rings [Kalogeras et al., 1993]. This is usually due to surface contamination, which can be reduced by proper lubricant selection.

Many other mechanisms that require lubrication are used in space applications. Some examples include solar array drives (SAD), which rotate a spacecraft's solar arrays, ball, roller, and acme screws, and many types of gears and transmission assemblies [Sarafin and Larson, 1995].

LIQUID AND SOLID LUBRICANTS

Both liquid and solid lubricants are used for space applications. The choice is often left to the designer. However, each class has merits and deficiencies. The relative merits have been tabulated by Roberts and Todd (1990) and appear in Table 1.

LIQUID LUBRICANTS

Many different chemical classes of liquid lubricants have been used for space applications in the last three decades. These include mineral oils, silicones, polyphenyl ethers, esters, and perfluoropolyethers. Recently, a synthetic hydrocarbon (Pennzane™) has been replacing many of these older lubricant classes. Each of these types will be discussed briefly. However, since the great majority of current spacecraft use either a formulated Pennzane™ or one of the PFPE materials, these two classes will be discussed in much greater detail.

Mineral Oils.—This class of lubricants consists of a complex mixture of naturally occurring hydrocarbons with a fairly wide range of molecular weights. Examples include V-78, BP 110, Apiezon C, Andok C (Coray 100) [Bertrand, 1991], and the SRG series of super refined mineral oils, which includes KG-80 [Dromgold and Klaus, 1968]. These latter fluids have been highly refined either by hydrogenation or percolation through bauxite to remove polar impurities. This makes them poorer neat lubricants, but greatly improves their response to additives. Apiezon C is still available commercially, but production of all others was discontinued many years ago. Nevertheless, the SRG oils have been stockpiled by some companies and are still used to lubricate bearings for momentum and reaction wheels. Their estimated shelf life is in excess of 20 years [Dromgold and Klaus, 1968].

Esters.—British Petroleum developed a triester base lubricant in the 70's designated BP 135. This material was laboratory tested but production was stopped and it was never flown. Another ester that was used in the past is designated as NPT-4 (neopentylpolyol ester), but is no longer marketed. Nye Lubricants markets another series of low volatility neopentylpolyol esters (UC₄, UC₇ and UC₉). Esters are inherently good boundary lubricants and are available in a wide viscosity range.

Silicones.—This fluid class was used early in the space program. They are poor boundary lubricants for steel on steel systems. Versilube F-50, a chloroarylsiloxane, was an early example. Comparisons of this fluid in boundary lubrication tests with a PFPE and a PAO have been reported [Kalogeris et al., 1993]. Relative life is shown in Figure 2. The silicone performed poorly by degrading into an abrasive polymerized product.

Synthetic Hydrocarbons.—There are two groups of synthetic hydrocarbons available today: poly-alphaolefins (PAO) and multiply alkylated cyclopentanes (MACs). The first class is made by the oligomerization of linear α -olefins having six or more carbon atoms [Shubkin, 1993]. Nye Lubricants markets a number of PAOs for space applications. Properties for three commercial PAOs appear in Table 2.

The other class of hydrocarbons is known as MACs. These materials are synthesized by reacting cyclopentadiene with various alcohols in the presence of a strong base [Venier and Casserly, 1993]. The products are hydrogenated to produce the final product, which is a mixture of di-, tri-, tetra- or penta-alkylated cyclopentanes. Varying reaction conditions controls the distribution. For the last several years, only one product has been available for space use which is primarily the tri-2-octyldodecyl substituted cyclopentane [Venier and Casserly, 1991]. This product is known as Pennzane™ SHF-X2000, marketed as Nye Synthetic Oil 2001A. Various formulated versions are also available. A primarily di-substituted (lower viscosity, but higher volatility) version is also now available. Properties of the 2001A product appear in Table 3. Recent experience with this fluid appears in Carré et al., 1995. A six year life-test of a CERES elevation bearing assembly using a Pennzane/lead naphthenate formulation yielded excellent results [Brown et al., 1999].

Perfluoropolyethers.—These fluids which are designated as either PFPE or PFPPE have been commercially available since the 1960s and 70s in the form of a branched fluid (Krytox™) manufactured by DuPont [Gumprecht, 1966], a linear fluid (Fomblin™ Z) [Sianesi et al., 1973], and a branched fluid (Fomblin™ Y) [Sianesi et al., 1971], both manufactured by Montefluos. Another linear fluid (Demnum™) was developed in Japan by Daikin [Ohsaka, 1985]. Preparation and properties of these fluids appear in Synthetic Lubricants and High-Performance Functional Fluids, R. L. Shubkin, editor, Perfluoroalkylpolyethers, Del Pesco, 1993. Some typical properties of these fluids appear in Table 4.

Silahydrocarbons.—A new type of space lubricant has been developed by the Air Force Materials Laboratory [Snyder et al., 1992]. These materials contain only silicon, carbon, and hydrogen and therefore do not exhibit the poor boundary lubricating ability observed with silicones. In addition, these unimolecular materials have exceptionally low volatility and are available in a wide range of viscosities. There are three types based on the number of silicon atoms present in the molecule (i.e. tri, tetra or penta) [Paciorek et al., 1990 and 1991].

A series of silahydrocarbons have been synthesized and their kinematic viscosities as a function of temperature have been measured (Figure 3) [Jones et al., 1998]. For comparison, a Pennzane™ plot has been included. As can be seen, the viscosity properties of the silahydrocarbons bracket the Pennzane data.

EHL properties of two members of this class have been measured [Spikes, 1996]. A trisilahydrocarbon had a α value of $16 \text{ GPa}^{-1} (\pm 0.3)$ at 21°C , while a pentasilahydrocarbon had a value of $17 \text{ GPa}^{-1} (\pm 0.3)$. At 40°C , the trisilahydrocarbon had a α value of $11 \text{ GPa}^{-1} (\pm 1)$ and the penta, $13.5 \text{ GPa}^{-1} (\pm 1)$. For comparison, the α value for Pennzane™ at 30°C is $9.8 \text{ GPa}^{-1} (\pm 0.3)$, estimated by the same method. Therefore, these silahydrocarbons will generate thicker EHL films than Pennzane™ under the same conditions.

LIQUID LUBRICANT PROPERTIES

Numerous reviews of liquid lubricants for space applications have been published [Fusaro and Khonsari, 1991; Stone and Bessette, 1998; Zaretsky, 1990]. Liquid lubricant data also appears in some handbooks [Fusaro et al., 1999; Roberts, 1999; McMurtrey, 1985]. Since most applications today use either PFPEs or Pennzane™ (MAC) formulations, these two classes will be presented in more detail.

Perfluoropolyethers and MACs

A liquid lubricant has to possess certain physical and chemical properties to function properly in a lubricated contact. In order to be considered for space applications, these lubricants must have vacuum stability (i.e. low vapor pressure), low tendency to creep, high viscosity index (i.e. wide liquid range), good elastohydrodynamic and boundary lubrication properties, and resistance to radiation and atomic oxygen. Optical or infrared transparency is important in some applications.

Volatility

Although labyrinth seals are extensively used in space mechanisms, lubricant loss can still be a problem for long term applications [Hilton and Fleischauer, 1990]. For a fixed temperature and outlet area, lubricant loss is directly proportional to vapor pressure. For a similar viscosity range, the PFPE fluids are particularly good candidates compared to conventional lubricants, as shown in Figure 4 [Conley and Bohner, 1990]. Vapor pressure data for four commercial PFPE fluids and Pennzane™ appear in Table 4.

Creep

The tendency of a liquid lubricant to creep or migrate over bearing surfaces is inversely related to its surface tension. Therefore, PFPE fluids, which have unusually low surface tensions (γ_{LV} , 17 to 25 dynes/cm at 20°C), are more prone to creep than conventional fluids, such as hydrocarbons, esters and silicones. However, these fluids may be contained in bearing raceways by using low surface energy fluorocarbon barrier films on bearing lands [Kinzig and Ravner, 1978]. But, there is a tendency for PFPE fluids to dissolve these barrier films with prolonged contact [Hilton and Fleischauer, 1993]. Therefore, they are not effective in preventing the migration of PFPEs. Pennzane™ based lubricants have higher surface tensions and are thus less prone to creep.

Viscosity-Temperature Properties

Although liquid lubricated space applications do not involve wide temperature ranges, low temperatures (i.e., -10° to -20°C) are sometimes encountered. Therefore, low pour point fluids that retain low vapor pressure and reasonable viscosities at temperatures to 75°C are desirable. The viscosity-temperature slope of PFPE unbranched fluids is directly related to the carbon to oxygen ratio (C/O) in the polymer repeating unit as shown in Figure 5 [Jones, 1995]. Here, the ASTM slope is used for the correlation. High values of the ASTM slope indicate large changes of viscosity with temperature.

In addition, branching, such as the trifluoromethyl pendant group in the Krytox fluids causes deterioration in viscometric properties. Comparison of ASTM slopes for three commercial fluids appears in Figure 6. Here the low C/O ratio fluid Fomblin Z has the best viscometric properties. The Demnum™ fluid, with a C/O ratio of 3, has intermediate properties, while the branched Krytox™ fluid, has the highest slope.

Elastohydrodynamic Properties

The operation of continuously rotating, medium to high-speed bearings relies on the formation of an elastohydrodynamic (EHL) film. This regime was briefly discussed in the introduction. A more detailed discussion appears in [Wedeven, 1975]. The two physical properties of the lubricant that influence EHL film formation are absolute viscosity (μ) and the pressure-viscosity coefficient (α) [Hamrock and Dowson, 1981].

Both molecular weight and chemical structure influence viscosity. Except for low molecular weight fluids, α values are only related to structure [Spikes et al., 1984]. Pressure-viscosity coefficients can be measured directly with conventional high-pressure viscometers [Jones et al., 1975; Vergne and Reynaud, 1992] or indirectly from optical EHL experiments [Cantow et al., 1987; Aderin et al., 1992]. Conventional viscometry normally uses the Barus equation [Barus, 1893] for correlations.

$$\mu_p = \mu_o e^{\alpha p}$$

where:

- μ_p = absolute viscosity at pressure, p and
- μ_o = absolute viscosity at atmospheric pressure
- α = a temperature dependent but pressure independent constant

This implies that a plot of $\log \mu_p$ versus p should yield a straight line of slope α . Unfortunately, this simple relationship is seldom obeyed. The pressure-viscosity properties that are important in determining EHL film thickness occur in the contact inlet where pressures are much lower than in the Hertzian region. Therefore, the slope of a secant drawn between atmospheric pressure and 0.07 GPa is typically used for film thickness calculations.

Some researchers [Jones et al., 1975] favor the use of another pressure-viscosity parameter, the reciprocal asymptotic isoviscous pressure (α') based on work by Roelands [Roelands, 1966]. Pressure-viscosity coefficients (α') for several lubricants at three temperatures (38°, 99°, and 149°C) are given in Table 5.

Figure 7 contains α values for the branched PFPE, Krytox™ 143AB. Data obtained by conventional (low shear) pressure-viscosity measurements are denoted with open symbols. Indirect measurements from EHL experiments (effective α values) are shown with solid symbols. There is good agreement comparing the different sources as well as different measurement techniques. Figure 8 contains similar data for the unbranched PFPE Fomblin™ Z-25 as a function of temperature. Here, there is a definite grouping of the data with effective α values being substantially lower than values from conventional measurements.

Two possibilities exist for this discrepancy. First, inlet heating can occur during the EHL measurements, thus leading to lower viscosities, lower film thicknesses and resulting in lower calculated α values. The second possibility is a non-Newtonian shear thinning effect, which can occur with polymeric fluids. Shear rates in EHL inlets can range from 10^5 to 10^7 sec^{-1} [Foord et al., 1968]. However, the EHL measurements do represent actual film thicknesses that may be expected in practice. Effective α values for several non-PFPE space lubricants including Pennzane™ base fluid and some Pennzane™ formulations appear in Table 6 [Spikes, 1997].

From EHL theory, the greatest film thickness at room temperature should be obtained with a lubricant having the largest α value, assuming approximately equal inlet viscosities. However, for many applications, lubricants must perform over a wide temperature range. In this case, the EHL inlet viscosity can be the overriding factor if the temperature coefficient of viscosity is high. This can cause a crossover in film thickness as a function of temperature for some PFPE fluids as shown by Spikes [Spikes et al., 1984], in Figure 9.

Boundary Lubrication

As described in the introduction, boundary lubrication is the regime where surfaces are not completely separated, which results in surface asperity interactions. The most important aspect of this regime is the formation of protective surface films to minimize wear and surface damage. The formation of these films is governed by the chemistry of both the film former as well as the contacting surfaces. Non-additive hydrocarbons, mineral oils, and esters will react in a boundary contact to produce "friction polymer" [Lauer and Jones, 1987]. Except for electrical contacts, this material is usually beneficial, but does represent a lubricant loss mechanism. But, these conventional lubricants are never required to act alone and almost all are formulated with antiwear, anti-corrosion, extreme pressure, or anti-oxidant additives to enhance their performance.

Contrast this with a PFPE boundary lubricant, a relatively inert, very pure fluid, which in past years contained no additives. If these fluids were really inert, they would not provide any surface protection except for some local fluid film effects (micro-EHL) and some removal of wear debris. However,

these fluids do react with bearing surfaces producing a series of corrosive gases and friction polymer which, in turn, react with existing surface oxides to produce metallic fluorides [Mori and Morales, 1989; Carré, 1986; Herrera-Fierro et al., 1993]. These fluorides are effective in-situ solid lubricants which reduce friction and prevent catastrophic surface damage [Mori and Morales, 1989].

Unfortunately, these fluorides are also strong Lewis acids (electron acceptors) which readily attack and decompose PFPE molecules [Herrera-Fierro et al., 1993; Carré and Markowitz, 1985; Kasai, 1992]. This causes the production of additional reactive species, which, in turn, produce more surface fluoride, resulting in an autocatalytic reaction. This can cause a lubricated contact to fail abruptly, as shown in Figure 10a. In contrast, a non-PFPE space lubricant, such as Pennzane™, has a greater lubricated lifetime and a much slower progression to failure. This is usually characterized by a gradually increasing friction coefficient as shown in Figure 10b. Therefore, the very reaction that allows the use of pure PFPE fluids in boundary contacts, eventually leads to their early destruction and accompanying bearing failure. Of course, the progression of this mechanism is highly dependent on the local contact conditions (i.e. degree of surface passivation, type and thickness of surface oxides, amount of surface contamination, temperature, load, speed, etc.).

Substantial improvements in bearing lifetimes can be obtained if ceramic or ceramic coated balls are substituted for the standard bearing steels. TiC coated balls [Boving et al., 1988] have shown considerable promise in alleviating some of these lubricant degradation problems. Gill et al., 1992 has shown a nine-fold increase in bearing lifetime with a PFPE (Fomblin™ Z25 when TiC balls were used instead of 52100 steel balls. In accelerated lifetests [Jones et al., 1999] using a spiral orbit tribometer, lubricant lifetimes with another PFPE (Krytox™ 143AC) were extended by factors of two to four, depending on the stress level.

Wear Characteristics and Relative Life.—Various Pennzane formulations have been compared to other hydrocarbons (PAOs) and PFPEs in eccentric bearing tests [Carré et al., 1995]. The data shown in Figure 11 indicated that a Pennzane formulated with antimony dialkyldithiocarbamate yielded a lifetime of several times that of a PFPE (Krytox 143AB). Linear ball screw tests at ESTL (European Space Tribology Laboratory) [Gill, 1997] compared several lubricants. These included: Fomblin™ Z-25 oil, Braycote™ 601 grease, Pennzane™ SHF X-2000 oil, sputter coated MoS₂, ion-plated lead, and Braycote™ 601 + ion plated lead. The two PFPE lubricants failed rapidly, but the use of Braycote™ 601 and ion plated lead reached the full test requirement of 2 million cycles. Ion plated lead alone failed at 400K cycles while the MoS₂ survived the full test, but some polishing of the contact zone was noted. The Pennzane™ oil completed the test but wear of the lead screw occurred and some dewetting was noted.

Earlier work at ESTL [Gill, 1994] compared the performance of several solid and liquid lubricants in oscillating ball bearings. Three liquid/grease lubricants were tested with phenolic cages: Fomblin™ Z-25, Braycote™ 601, and Pennzane™ SHF X-2000. Torque levels were measured over 10 million oscillations. For a low angle of oscillation (± 0.5 degrees), Fomblin™ Z-25 yielded average torque levels 1.5 times that of Braycote™ and approximately 3 times that of Pennzane™. At ± 5 degrees, mean torque measurements were similar for all three lubricants, but Pennzane™ did exhibit the lowest levels. Finally, at ± 20 degrees, the Z-25 yielded the highest torque, Braycote™ intermediate, and again Pennzane™ was the lowest.

Long term angular contact bearing tests were also reported by Gill [Gill and Rowntree 1995]. Pennzane™ SHF X-2000 was tested at 200 and 1400 rpm for 10^8 and 10^9 revolutions, respectively. The bearings did not fail, but they did suffer from oil starvation due to evaporation losses. It was not clear if this was due to insufficient sealing or related to a lubricant batch problem. But it was stated that similar tests with PFPE oils have never failed due to starvation by oil evaporation.

A vacuum four-ball tribometer [Jones et al., 1998] has been used to rank various space lubricants according to wear rates (Figure 12), including three PFPEs, Pennzane™ base fluid, a formulated Pennzane™, and two other unformulated fluids (a silahydrocarbon and a PAO). In general, higher wear rates represent lower lifetimes in space.

PFPE Formulations

Although no liquid PFPE additive formulations are currently being used for any space application, many additives, soluble in PFPEs, have been developed in the last few years. These include antiwear, anticorrosion and antirust additives [Gschwender et al., 1993; Helmick et al., 1997; Jones et al., 1996; Masuko et al., 1995; Nakayama et al., 1996; Sharma et al., 1990; Shogrin et al., 1997; Srinivasan et al., 1993 and Williams et al., 1994]. Several different additives exhibited antiwear behavior in a Krytox™ basestock in vacuum four ball tests (Figure 13) [Shogrin et al., 1997].

Greases

Greases based on PFPEs with PTFE thickeners (Krytox™ 240 series and Braycote™ 600 series) have been used extensively in space mechanisms. In addition, hydrocarbon greases based on Pennzane™ and marketed by Nye Lubricants under the name of Rheolube™ 2000 are also now available. Performance of various Pennzane™ based greases appears in [Rai et al., 1999]. Recently, new PFPE formulations (commercially designated as Braycote™ 700 and 701) incorporating a boundary additive have yielded much improved wear characteristics (Figure 14) [Jones, D. G. V. et al., 1999].

SOLID LUBRICANTS

A number of different solid lubricants have been used in space over the last 30 years. These include lamellar solids, soft metals, and polymers. Lamellar solids include transition metal dichalcogenides like molybdenum disulfide and tungsten disulfide. Soft metals include: lead, gold, silver, and indium. Polyimides and polytetrafluoroethylene are polymeric materials that have lubricating properties. Unlike liquid lubricants, the metals and lamellar solids are applied as thin films (i.e. less than a micron). Ion plating [Roberts, 1990; Spalvins, 1987 and 1998] and sputtering [Roberts, 1990; Spalvins, 1992; Fox et al., 1999] are preferred methods of applying these materials. Another method of applying lubricants to surfaces involves bonded films [Fusaro, 1981; Dugger et al., 1999]. In this case, lubricants are mixed with an organic binder and applied to the surface via spraying or dipping. The films, which are typically greater than 10 microns, are then cured at high temperature. Self-lubricating polymers and polymer composites [Fusaro, 1990; Gardos, 1986] are sometimes utilized. The most common form of usage is as a cage or retainer in a rolling element bearing or as a bushing. By far the most common solid lubricants used in today's space mechanisms are ion-plated lead and various forms of sputter deposited MoS₂.

Ion-plated Lead.—This is the lubricant of choice for precision spacecraft bearings in Europe. It is normally used in conjunction with a leaded bronze cage. Lead coatings and cages saw early success in cryogenic space applications [Brewer et al., 1974]. Successful applications have been used on GIOTTO [Todd and Parker, 1987], OLYMPUS [Sheppard, 1983] and GERB [Fabrizzi et al., 1999]. Although not as common in the United States, this combination has recently been applied to encoder bearings for SABER. Extensive data has been published [Arnell, 1978] on effects of speed, thickness and substrate surface roughness. One disadvantage of this lubricant is limited life in laboratory air. Much higher wear rates occur in air, producing copious amounts of lead oxide. This debris can also cause torque noise.

Molybdenum disulfide.—MoS₂ has been successfully used in space for many years [Lince and Fleischauer, 1997; Hilton and Fleischauer, 1994; Loewenthal et al., 1994 and Hopple et al., 1993]. These films display very low friction under vacuum conditions (i.e., 0.01 or less). Optimized thin films (one micron or less) are deposited by sputtering. The tribological performance of these films is extremely dependent on the sputtering conditions. The sputtering conditions control the microstructure, which in turn determines crystallinity, morphology, and composition [Didziulis et al., 1992]. For instance, the presence of oxygen in the sputtering environment can affect both friction and wear life [Lince et al., 1990]. For more details about the sputtering process see [Spalvins, 1992]. In addition, substrate surface roughness also has a pronounced effect on friction and wear. For steel bearing surfaces optimum durability occurs at a nominal surface roughness of 0.2 μm (cla) [Roberts et al., 1992].

The test environment also greatly affects the frictional behavior and life of MoS₂ films. In ultrahigh vacuum, these films display ultra-low friction (less than 0.01) as shown in Figure 15. Under normal vacuum conditions, the friction may range from 0.01 to 0.04 with exceptionally low wear and long endurance lives. When tested in humid air, these films have initial friction coefficients near 0.15 and very limited life [Roberts, 1987; Lince and Fleischauer, 1998]. Since many space mechanisms must be ground tested before launch, sometimes in room air, there has been much research to improve the performance of these films under atmospheric conditions.

One method has been to modify MoS₂ films by layering or co-depositing metals such as Au [Spalvins, 1984; Roberts and Price, 1995]. In this work, inclusions of Au doubled the film durability in dry nitrogen and tripled or quadrupled it in air. Co-deposition with other metals (i.e., Cr, Co, Ni and Ta) have also shown synergistic effects [Stupp, 1981]. Ion implantation with Ag has also been reported to be beneficial [Liu et al., 1995] but not when co-deposited [Stupp, 1981]. The properties of MoS₂ films can also be improved by co-deposition with titanium [Renevier et al., 1999]. These films were much less sensitive to atmospheric water vapor than pure MoS₂ films. Studies have shown that the problems that occur in moist air are associated with adsorption of water molecules at edge sites on the MoS₂ lattice [Fleishauer, 1984].

Another method of enhancing MoS₂ endurance in ball bearings is to use a PTFE composite as the retainer material [Suzuki and Prat, 1999]. In gimbal bearing life tests [Hopple and Loewenthal, 1994], lives in excess of 45 million cycles were demonstrated with advanced MoS₂ films combined with PTFE-based retainers.

ACCELERATED AND LIFE TESTING

Because of increasing demands of spacecraft mechanisms, new lubricants and additives are always under development. These materials must undergo ground-based testing to insure they will meet the long duty cycle requirements and stringent conditions required by today's missions.

In the past, two approaches to qualify lubricants for space were taken. The first approach is to perform system-level tests on actual flight hardware. The second is to attempt to duplicate the conditions of flight system operation [Kalogeris et al., 1993]. Although this type of testing produces results, they are costly, both in time and money. Therefore, only a few candidate lubricants are selected for testing. This has led to most lubricants being chosen through 'heritage' or actual flight experience.

Accelerated life testing is an approach that allows quick screening of several lubricants. Generally, test results cannot be extrapolated to predict lifetimes of components that are lubricated similarly but operate under different conditions [Conley, 1998], instead accelerated life testing allows for a better selection of candidate lubricants to undergo full-scale tests. Accelerated tests typically do not involve actual flight hardware, but rather utilize a setup that subjects the lubricant to an extreme condition.

Test acceleration is achieved by varying various test parameters such as speed, load, temperature, contaminants, quantity of lubricant available, and surface roughness [Murray and Heshmat, 1995]. When selecting what parameters to vary, it is important to have a thorough understanding of contact conditions and the lubricant used.

In a liquid lubricated environment, a significant change in speed or temperature could introduce a change of the lubrication regime. As speed increases or temperature decreases, film thickness also increases taking the bearing from the boundary regime, through the mixed regime, and into the EHL regime. As described earlier in this chapter, each lubrication regime has specific and different wear characteristics. Also, time dependent parameters such as creep, loss of lubricant through evaporation and centrifugal forces, and lubricant degradation are not modeled with accelerated life tests [Conley, 1998].

Accelerated life testing with solid lubricants is somewhat easier. If a system uses solid lubricants exclusively and is a fairly low speed application (< 50 RPM), then testing at increased speeds can be done with confidence, provided that the specific wear rate is not a function of speed.

Limitations with accelerated life tests of solid lubricants at higher speeds includes additional loads from inertial effects and component stability [Conley, 1998].

Advantages of accelerated life testing include: quick relative ranking of lubricants, tests which are easy to monitor, data which can be used to enhance a lubricant or additive design or model, and rapid generation of baseline data. Table 7 summarizes the strengths and weaknesses of accelerated testing.

FACILITIES FOR ACCELERATED LIFE TESTING OF SPACE MECHANISMS

Eccentric Bearing Test Apparatus

Aerospace Corporation [Kalogeris et al., 1993] has developed an apparatus based upon a 22.1 mm (0.900 in) diameter bearing. In this test, the bottom (rotating) raceway is flipped over and the flat side polished to a 0.25 μm (10 μin) finish. This allows the lower raceway to be mounted with eccentricity. The intentional misalignment accelerates the wear process and the flat surface operates at a higher stress level than the curved raceway. Both ensure that more severe tribological conditions are introduced. Also, post-test analysis can be more easily performed on a flat surface.

Figure 16 shows the eccentric bearing tester. The lower raceway is mounted to a rotating shaft and rigidly supported. The upper raceway is placed inside an assembly containing a load cell. The load is induced into the bearing by compressing springs that push on the upper raceway fixture. The eccentricity of the setup can be altered from 0 to 3.06 mm (0 to 0.125 in). The eccentricity introduces a skidding element into the motion of the ball that greatly accelerates lubricant degradation. It also provides a wide wear track for easy post-test surface analysis.

The upper housing is connected to the lower housing through a set of aluminum flexures. This allows the upper housing to flex slightly during the test for torque calculations. Deflection is measured through a proximity sensor mounted in the lower housing. The entire assembly operates under vacuum of approximately 10^{-8} Torr. A similar test device exists at ESTL [Gill et al., 1993].

Spiral Orbit Rolling Contact Tribometer (Ball-on-Plate – BoP)

NASA Glenn Research Center has developed a tribometer based upon a simplified thrust bearing design. The rig consists of two flat plates with a single ball sandwiched between them (Figure 17). A load is applied through the bottom plate, which is held stationary as the top plate rotates. As the ball moves through the orbit, it travels in a spiral orbit. The orbit pitch has a direct relationship to the friction coefficient. The guide plate is used to nudge the ball back into the original orbit diameter. The force required to do this is measured and the friction coefficient calculated. A small straight area develops on the lower plate where the ball is in contact with the guide plate and is termed the 'scrub'. The rig operates at a vacuum level of 10^{-8} Torr or better. Figure 18 shows a detailed view of the setup.

The most severe tribological conditions occur during the scrub. The scrub is easy to find on the surface of the plate and makes for simple post-test analysis. Also, all the contact surfaces being flat simplifies the analysis procedure.

Lifetimes are controlled in two ways. First, the rig is capable of operating at a variety of stress levels, either by changing the applied load or varying the ball diameter. Hertzian stress is easily calculated for a sphere-on-flat geometry. The second way to control the lifetime is by varying the amount of lubricant used. For a typical test, 50 μg of lubricant is used. With this limited amount of lubricant, almost all of it will be brought into the contacts thousands of times. Any tribologically induced changes can be more easily detected than with a large lubricant reservoir.

The features of this rig are the following:

1. It is a rolling element tribometer operating in vacuum and at a contact stress level typically found in bearings.
2. It operates in the boundary lubrication regime.

3. It permits a simple and rigorous analysis of the contact stresses and the energy loss to the lubricant.
4. It enables an easy post-test examination of the tribological elements and the degraded lubricant with surface and thin-film analysis techniques.
5. It determines the coefficient of friction.
6. It determines the system pressure rise and allows a partial pressure analysis of the gas released by the lubricant as it degrades under tribological stress.
7. It operates with an extremely small and finite lubricant charge to provide maximum opportunity for lubricant molecule to be tribologically exercised and thus to undergo and exhibit tribological degradation [Pepper et al., 1996].

Vacuum Four-Ball Tribometer

Figure 19 shows a tribometer developed at NASA Glenn Research Center based upon a four-ball configuration. It is designed to test liquid lubricants under pure sliding conditions at room temperature under a vacuum of at least 10^{-6} Torr.

Figure 20 shows a detail of the inside of the four-ball apparatus. The system consists of a rotating ball sliding against three stationary balls that are immersed in a lubricant. The system is loaded through a pneumatic cylinder, which pushes the lubricant cup and stationary balls against the rotating ball. The lubricant cup is held in position by a flex pivot. The flex pivot, which has a specific spring constant, allows for angular displacement in the direction of rotation. The amount of flex is measured to calculate torque and equivalent frictional force.

Typical tests are run with 9.82 mm (3/8 in) diameter balls, 100 RPM, room temperature, and an initial Hertzian stress of 3.5 GPa. This stress drops as a function of test time as a wear scar develops on the stationary balls. The test is stopped every hour and the wear scar diameters measured. A special platform allows for the measurement of the wear scars without removing them from the cup [Masuko et al., 1994]. This allows the test to resume exactly where it was stopped. A full test takes 4 hours. Upon completion, the wear volume is plotted as a function of sliding distance and the wear rate is calculated from the slope of the line. This rig provides quick information about the wear characteristics of lubricants and additive packages.

Vacuum Pin-on-Disk Tribometer

Figure 21 shows an ultra-high vacuum pin-on-disk tribometer [Miyoshi et al, 1999] for testing of solid film coatings. The rig is a simple setup using a stationary pin with a known end diameter pressed against a rotating disk, which is coated with the desired lubricant. Displacement of the pin in the tangential direction is measured and a friction coefficient is obtained. The rotating mechanism is enclosed in a vacuum chamber and tests are typically run at a vacuum level of at least 10^{-9} Torr. The rig can also be back filled with nitrogen or humid air to study the effects of earth's atmosphere on space lubricant films.

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TABLE 1.—RELATIVE MERITS OF SOLID AND LIQUID SPACE LUBRICANTS
[ROBERTS AND TODD, 1990]

Dry Lubricants	Wet Lubricants
Negligible vapor pressure	Finite vapor pressure
Wide operating temperature	Viscosity, creep and vapor pressure all temperature dependent
Negligible surface migration	Sealing required
Valid accelerated testing	Invalid accelerated testing
Short life in moist air	Insensitive to air or vacuum
Debris causes frictional noise	Low frictional noise
Friction speed independent	Friction speed dependent
Life determined by lubricant wear	Life determined by lubricant degradation
Poor thermal characteristics	High thermal conductance
Electrically conductive	Electrically insulating

TABLE 2.—TYPICAL PROPERTIES FOR THREE COMMERCIAL POLYALPHAOLEFINS

		OIL 132	OIL 182	OIL 186
Viscosity At:	210°F, SUS	39	62.5	79.5
	210°F, cs	3.9	10.9	15.4
	100°F, SUS	92	348	552
	100°F, cs	18.7	75.0	119
	0°F, cs	350	2700	7600
	Flash Point	440°F	465°F	480°F
	Pour Point	-85°F	-60°F	-55°F
	Evaporation 6 Hours at 350°F	2.2%	2.0%	1.9%
	Specific Gravity @ 25°C	0.828	0.842	0.847

TABLE 3.—TYPICAL PROPERTIES OF NYE SYNTHETIC OIL 2001A (SHF X-2000)

Viscosity at 100°C	14.6 cSt
Viscosity at 40°C	108 cSt
Viscosity at -40°C	80,500 cSt
Viscosity Index	137
Flash Point	300°C
Fire Point	330°C
Pour Point	-55°C
Specific Gravity at 25°C	0.841
Specific Gravity at 100°C	0.796
Coefficient of Thermal Expansion	0.0008 cc/cc/°C
Evaporation, 24h at 100°C	None
Refractive Index at 25°C	1.4671
Vapor Pressure at 25°C	10 ⁻¹¹ to 10 ⁻¹⁰ Torr

TABLE 4.—PHYSICAL PROPERTIES OF FOUR COMMERCIAL PFPE LUBRICANTS AND PENNZANE SHF X-2000

Lubricant	Average Molecular Weight	Viscosity At 200°C, cSt	Viscosity Index	Pour Point °C	Vapor Pressure, Pascal	
					At 20°C	At 100°C
Fomblin™ Z-25	9500	255	355	-66	3.9x10 ⁻¹⁰	1.3x10 ⁻⁶
Krytox™ 143AB	3700	230	113	-40	2.0x10 ⁻⁴	4.0x10 ⁻²
Krytox™ 143AC	6250	800	134	-35	2.7x10 ⁻⁶	1.1x10 ⁻³
Demnum™ S-200	8400	500	210	-53	1.3x10 ⁻⁸	1.3x10 ⁻⁵
Pennzane SHF X-2000	1000	330	137	-55	2.2x10 ⁻¹¹	1.9x10 ⁻⁸

TABLE 5.—PRESSURE VISCOSITY COEFFICIENTS AT THREE TEMPERATURES α' , PA⁻¹X10⁸ FOR SEVERAL LUBRICANTS [JONES ET AL, 1975]

	38°C	99°C	149°C
Ester	1.3	1.0	0.85
Synthetic Paraffin	1.8	1.5	1.1
Z Fluid (Z-25)	1.8	1.5	1.3 (Extrapolated)
Naphthenic Mineral Oil	2.5	1.5	1.3
Traction Fluid	3.1	1.7	0.94
K Fluid (143AB)	4.2	3.2	3.0

TABLE 6.—MEASURED VISCOSITIES AND CALCULATED PRESSURE VISCOSITY COEFFICIENTS (α VALUES) FOR SEVERAL SPACE LUBRICANTS [SPIKES, 1997]

Temp °C	Pennzane 2001 – Synth. Oil	PAO-186 – Synth. Oil	NPE UC-7 – Ester	Pennzane 2001+5% Pb Napthenate	Pennzane 2001+3% Pb Napthenate
<i>Viscosity cP</i>					
40	88	90	37	98	96
80	19	21	10	21	21
100	12	13	6	12	12
120	8	8	4	8	8
<i>Alpha GPa⁻¹</i>					
40	11.0	12.5	6.5	12.0	10.0
80	9.5	9.0	5.0	9.0	9.0
100	7.0	7.0	5.0	6.5	7.0
120	7.0	5.0	5.0	6.0	7.0

TABLE 7.—STRENGTHS AND WEAKNESS OF ACCELERATED LIFE TESTING [MURRAY AND HESHMAT, 1995]

Weakness	Strengths
Wetting condition with temperature	Easy to monitor
Chemistry changes with temperature and pressure	Use to enhance design
Oxidation changes with temperature and pressure	Use to validate model
Hydrodynamics region changes	Rapid baseline data generation
Wear/friction polymers changes	
Coating fracture under high load (solid lube)	
Non-standardized	
Dynamic changes in cages and components	
Low confidence	

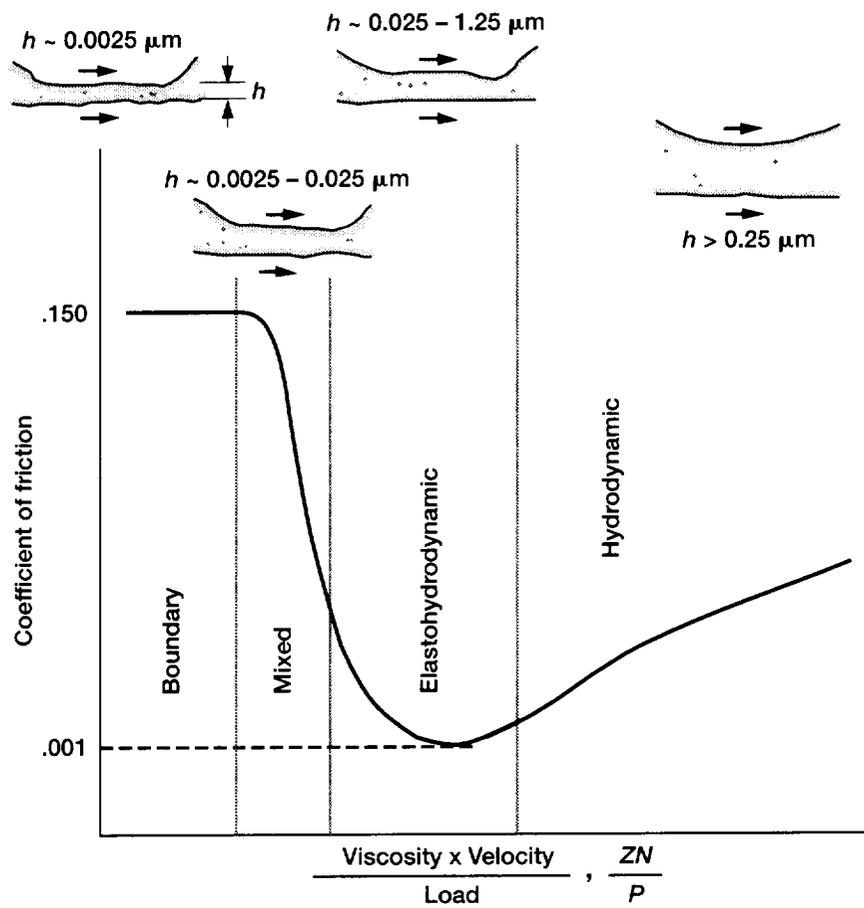


Figure 1.—Coefficient of friction as a function of viscosity-velocity-load parameter [Stribeck, 1902].

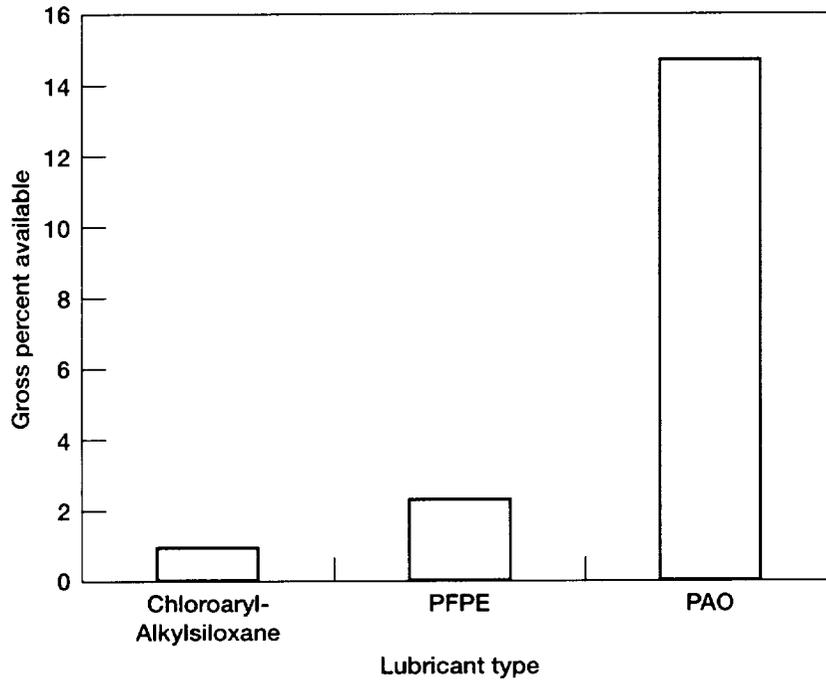


Figure 2.—Screening tests results (scanner mechanism) [Kalegeras et al., 1993].

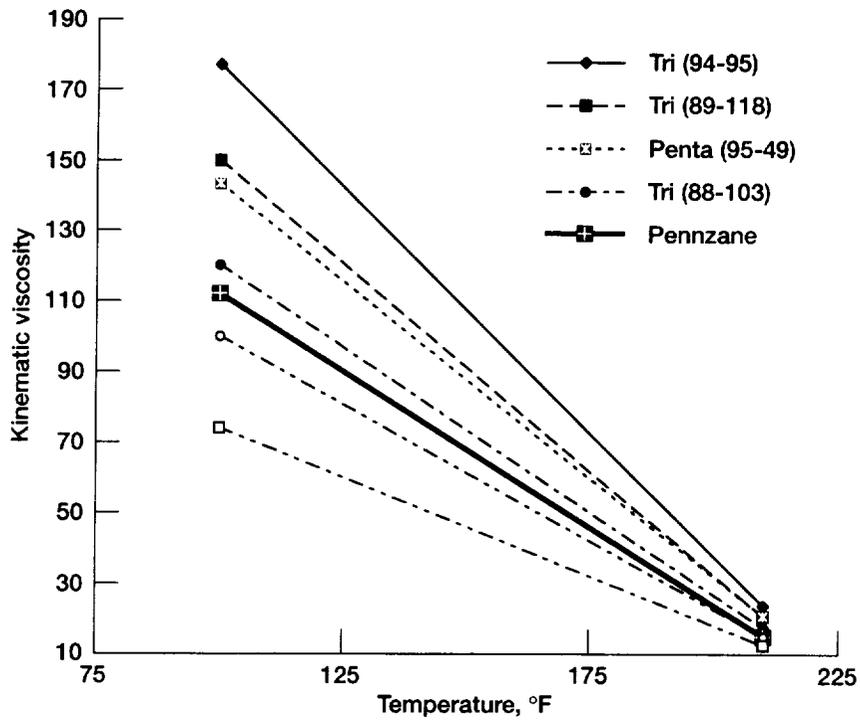


Figure 3.—Kinematic viscosity as a function of temperature for a series of silahydrocarbons [Jones et al., 1998].

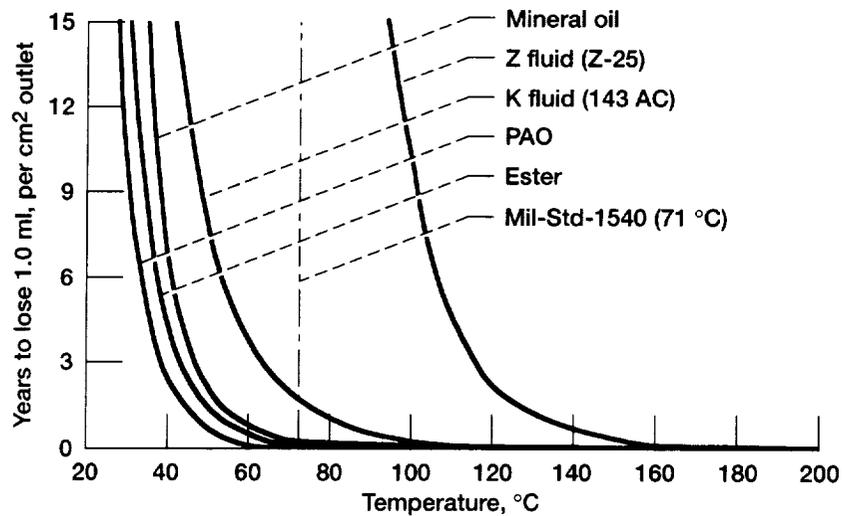


Figure 4.—Relative evaporation rates of aerospace lubricants [Conley and Bohner, 1990].

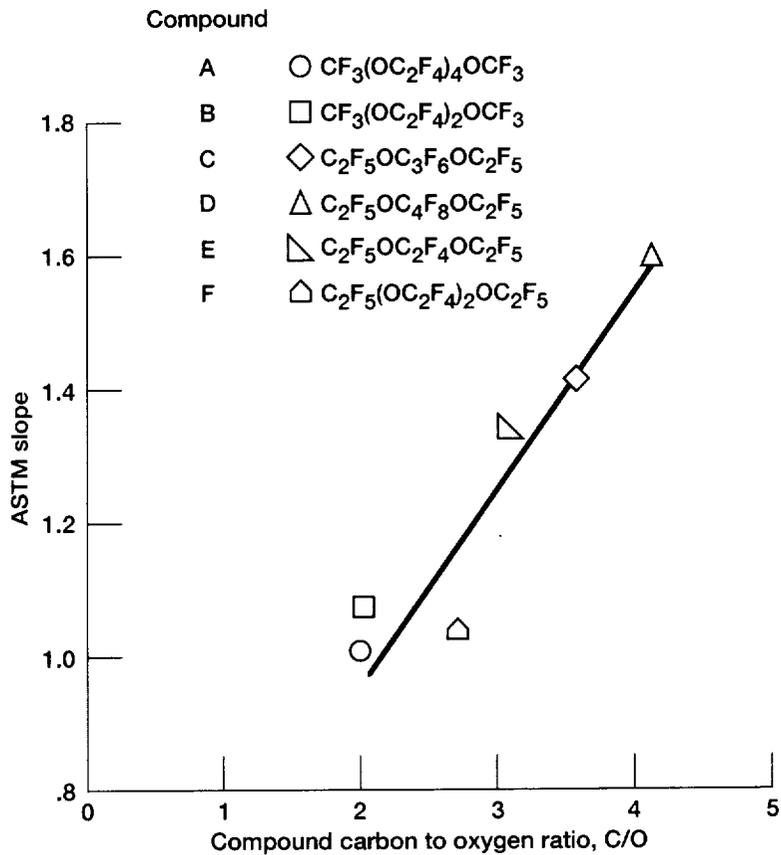


Figure 5.—Viscosity-temperature slope as a function of carbon to oxygen ratio [Jones, 1995].

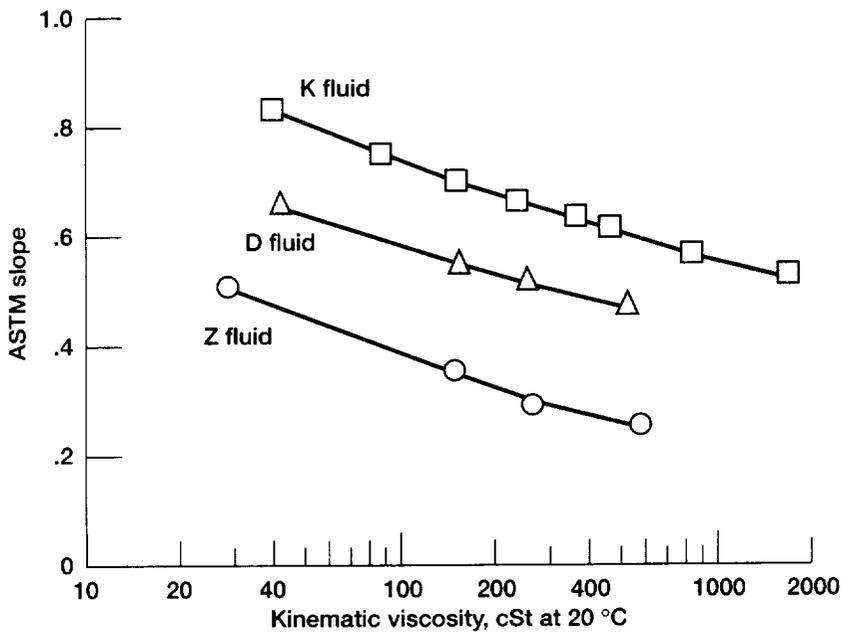


Figure 6.—Viscosity-temperature slope (ASTM D 341-43) as a function of kinematic viscosity at 20 °C for Krylox™ (K), Denmum™ (D), and Fomblin™ (Z) fluids [Jones, 1995].

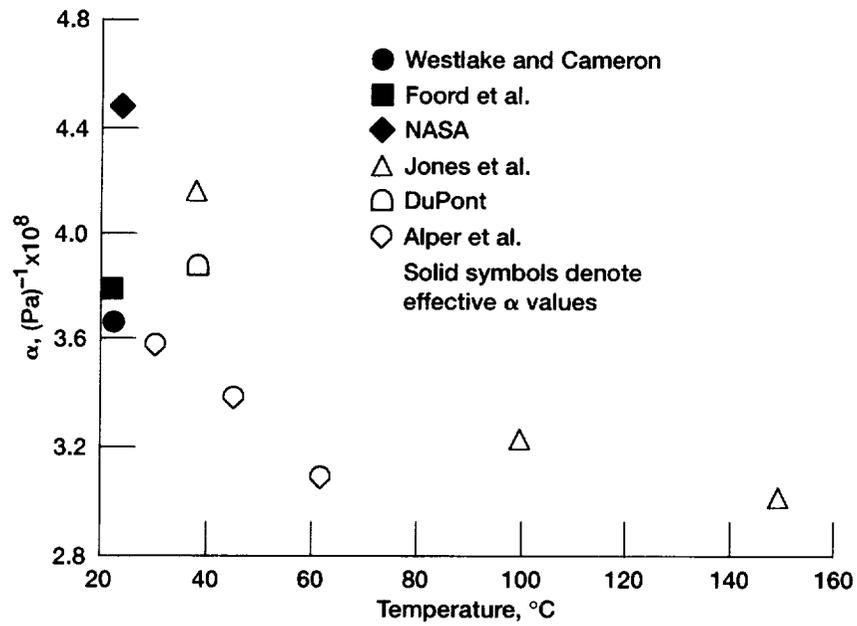


Figure 7.—Pressure-viscosity coefficients for PFPE Krytox™ 143AB as a function of temperature [Jones, 1995].

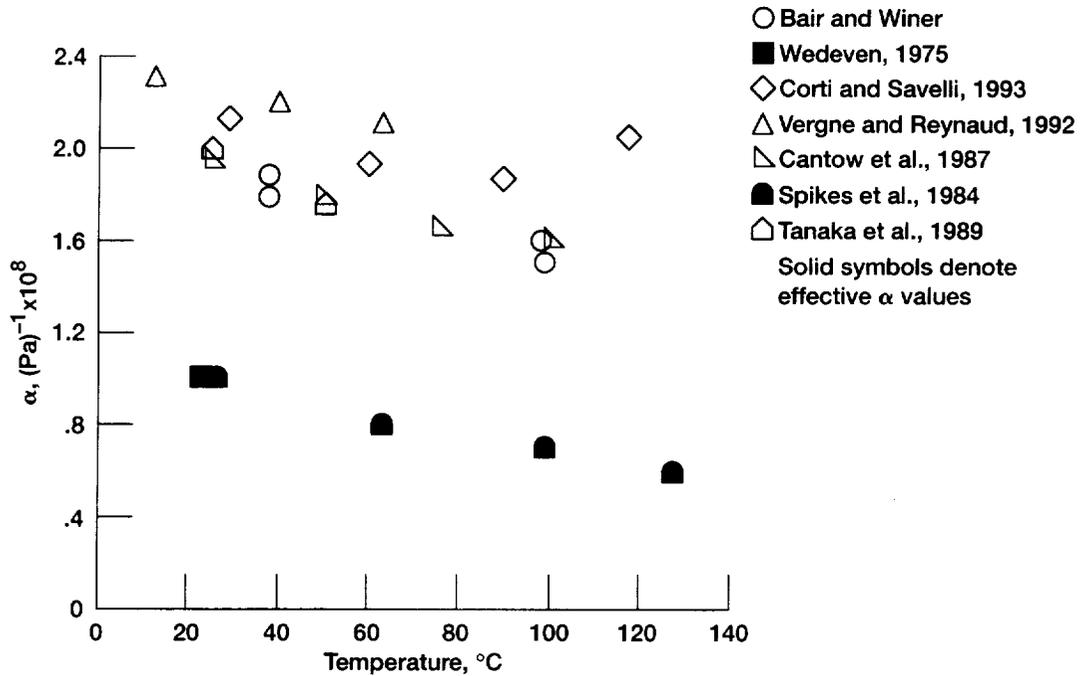


Figure 8.—Pressure-viscosity coefficients for PFPE Fomblin™ Z-25 as a function of temperature [Jones, 1995].

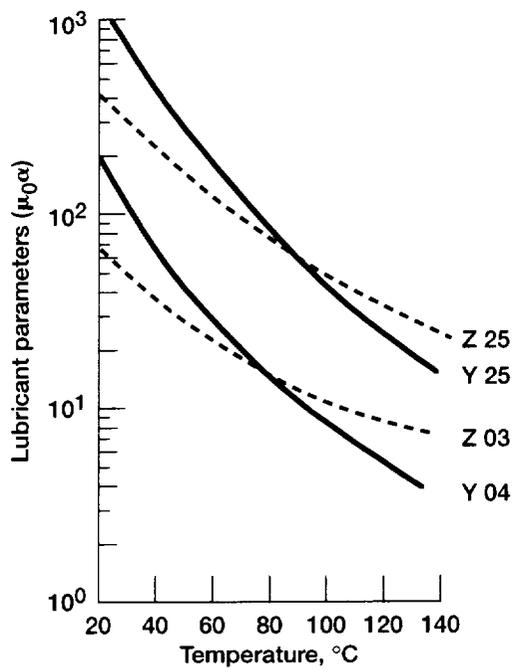


Figure 9.—Lubricant parameters for PFPE Y and Z fluids [Spikes et al., 1984].

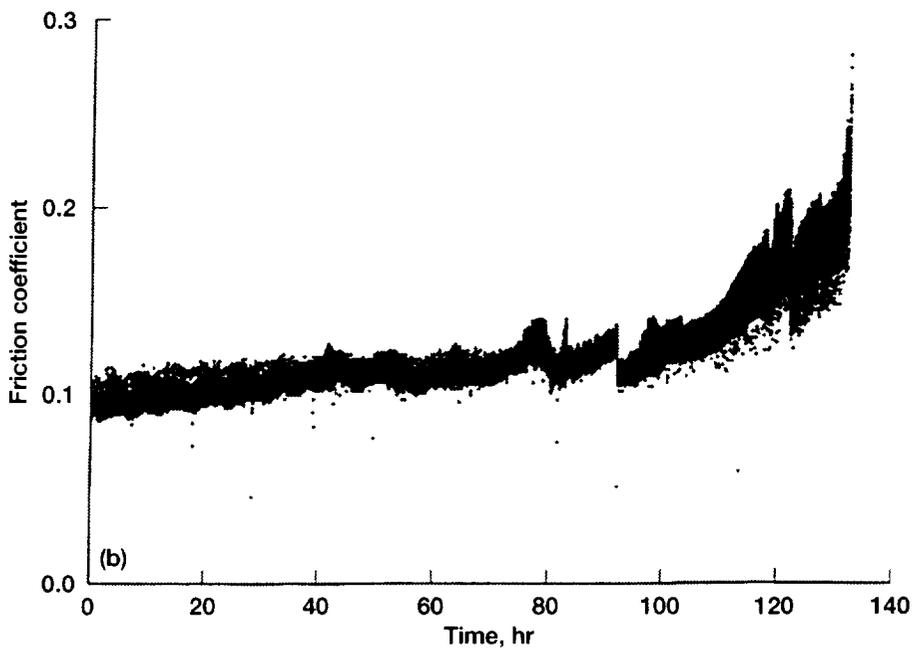
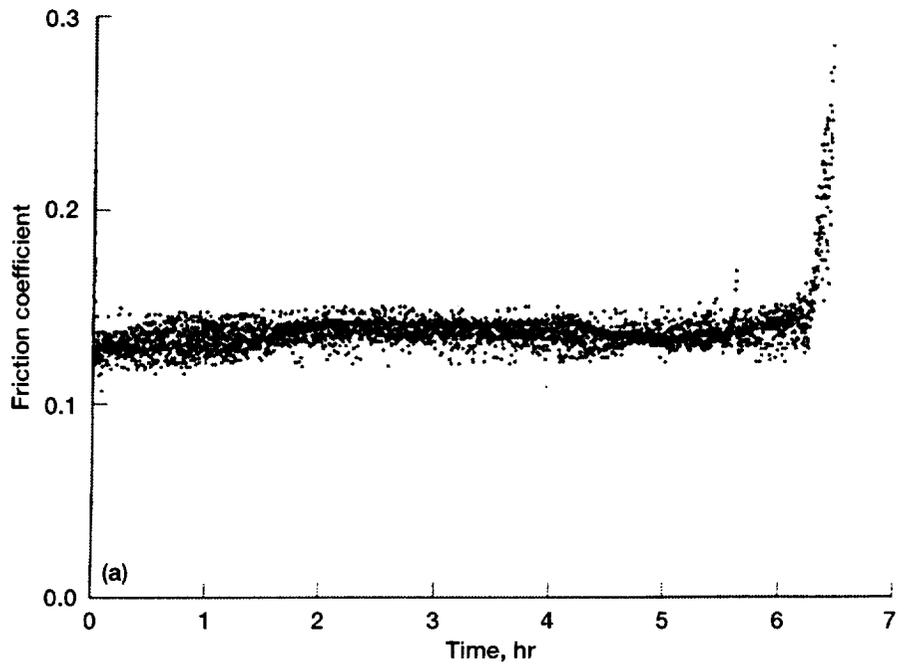


Figure 10.—Coefficient of friction as a function of time. (a) Krytox™ 143AC. (b) Pennzane™ 2001A (Spiral Orbit Rolling Contact Tribometer).

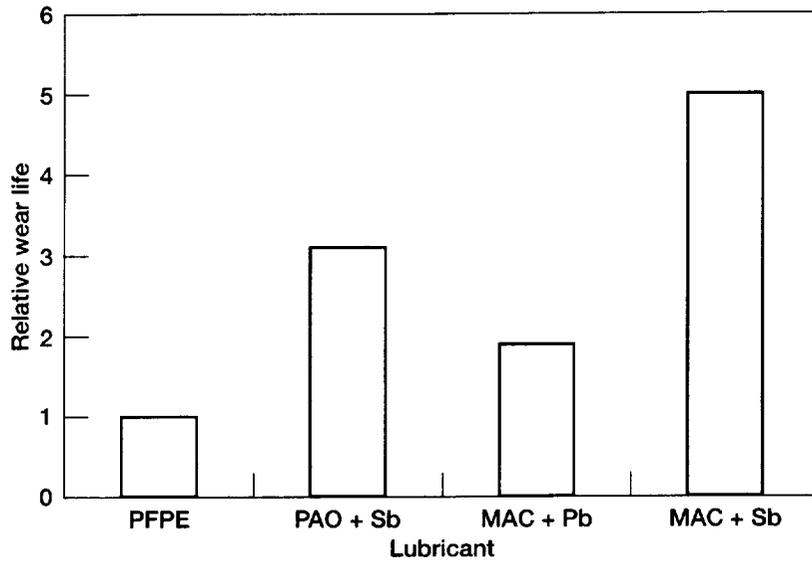


Figure 11.—Eccentric bearing screening test results for PFPE, PAO, and MAC oils, [Carré et al., 1995].

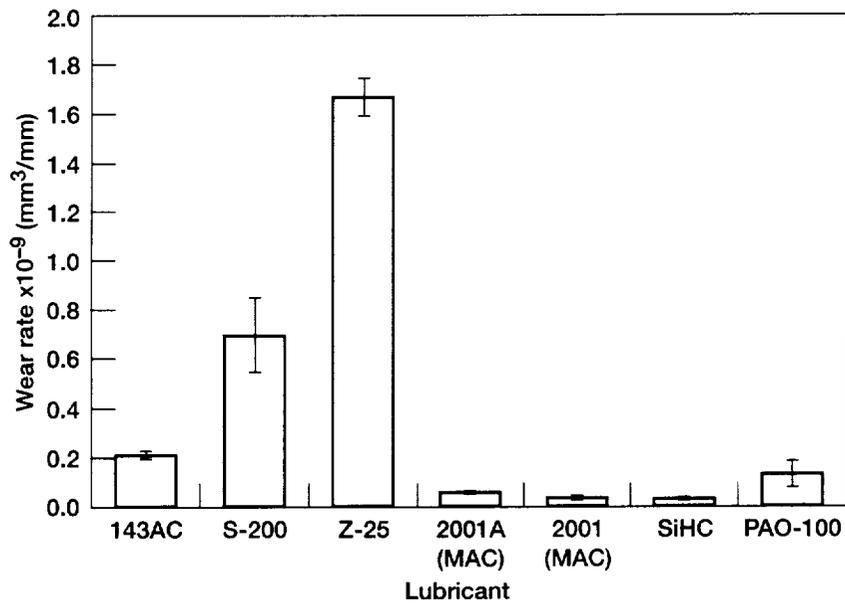


Figure 12.—Mean wear rates of various space lubricants using a vacuum 4-ball tribometer [Jones et al., 1998].

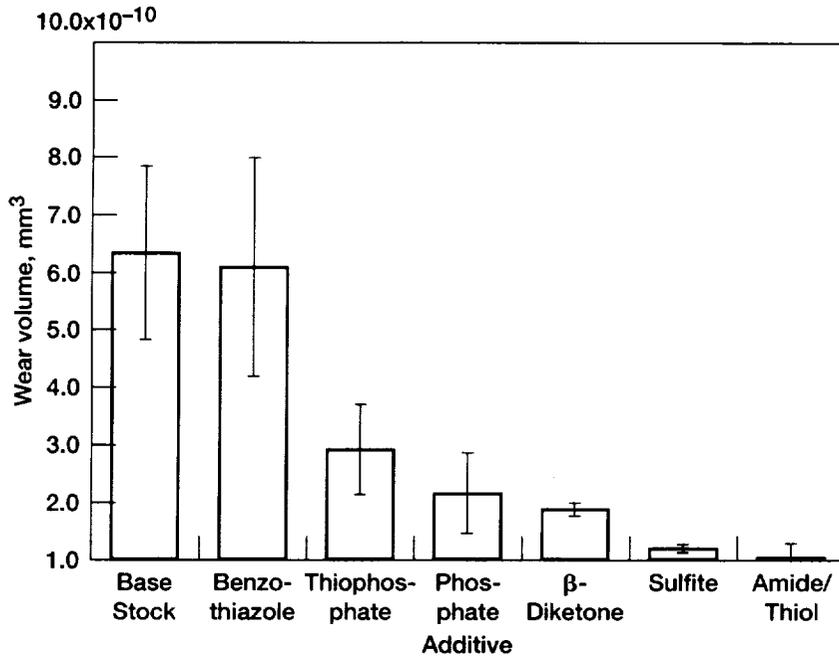


Figure 13.—Antiwear behavior of several additives in Krytox™ base stock using a vacuum 4-ball tribometer [Jones et al., 1997].

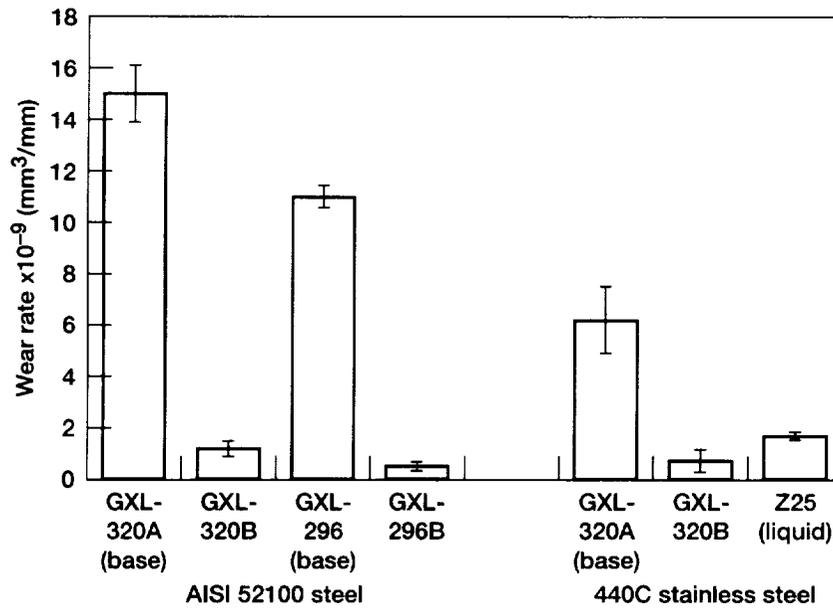


Figure 14.—Mean wear rates of various PFPE formulated greases using a vacuum 4-ball tribometer [Jones et al., 1999].

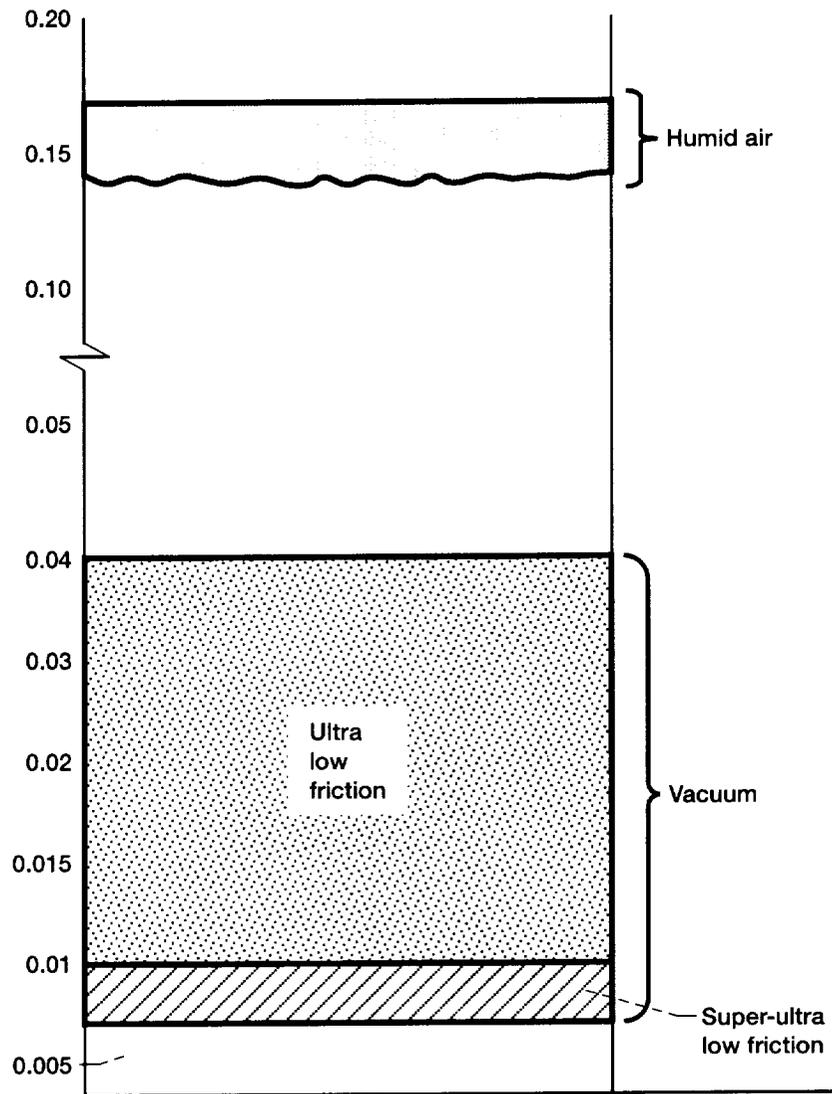


Figure 15.—Frictional variation of sputtered MoS₂ films [Spalvins, 1992].

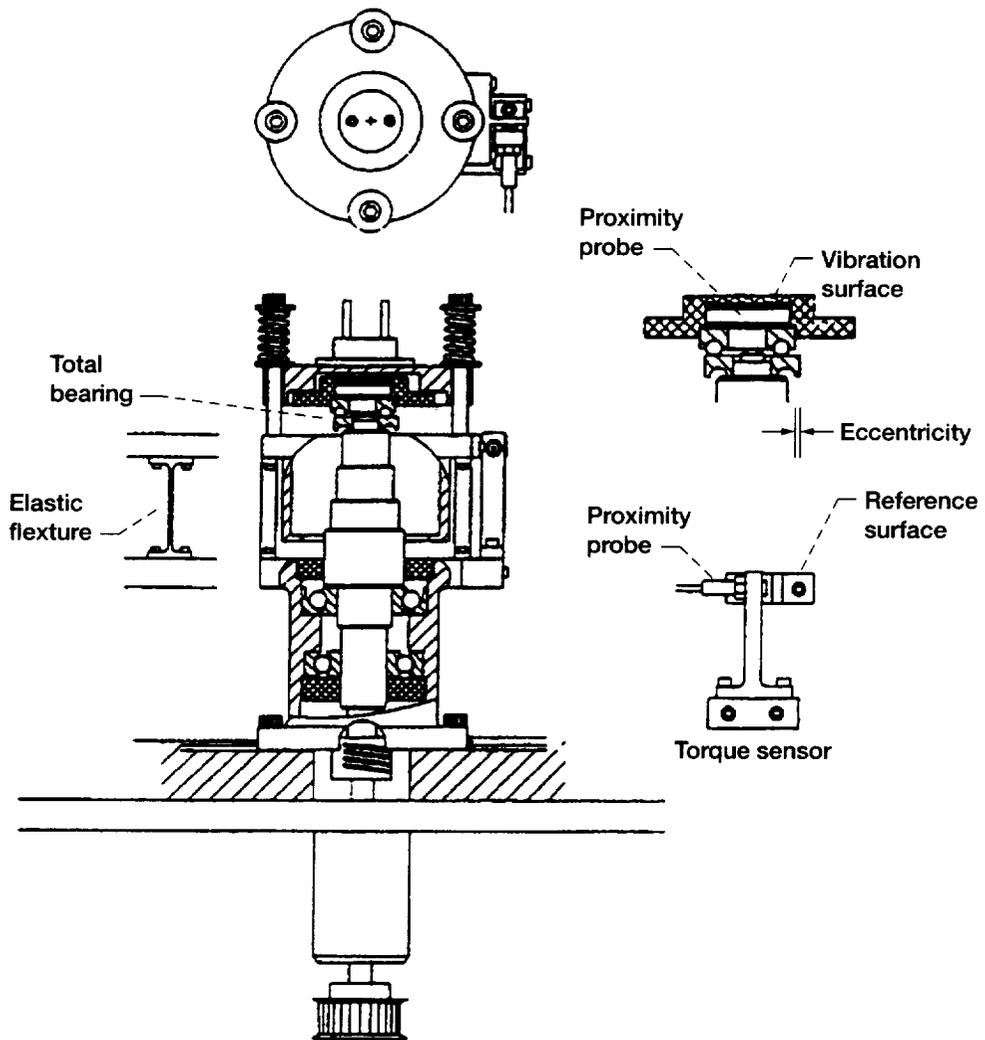


Figure 16.—Aerospace Corp.'s Eccentric Bearing Test Mechanism [Kalogeris et al., 1993].

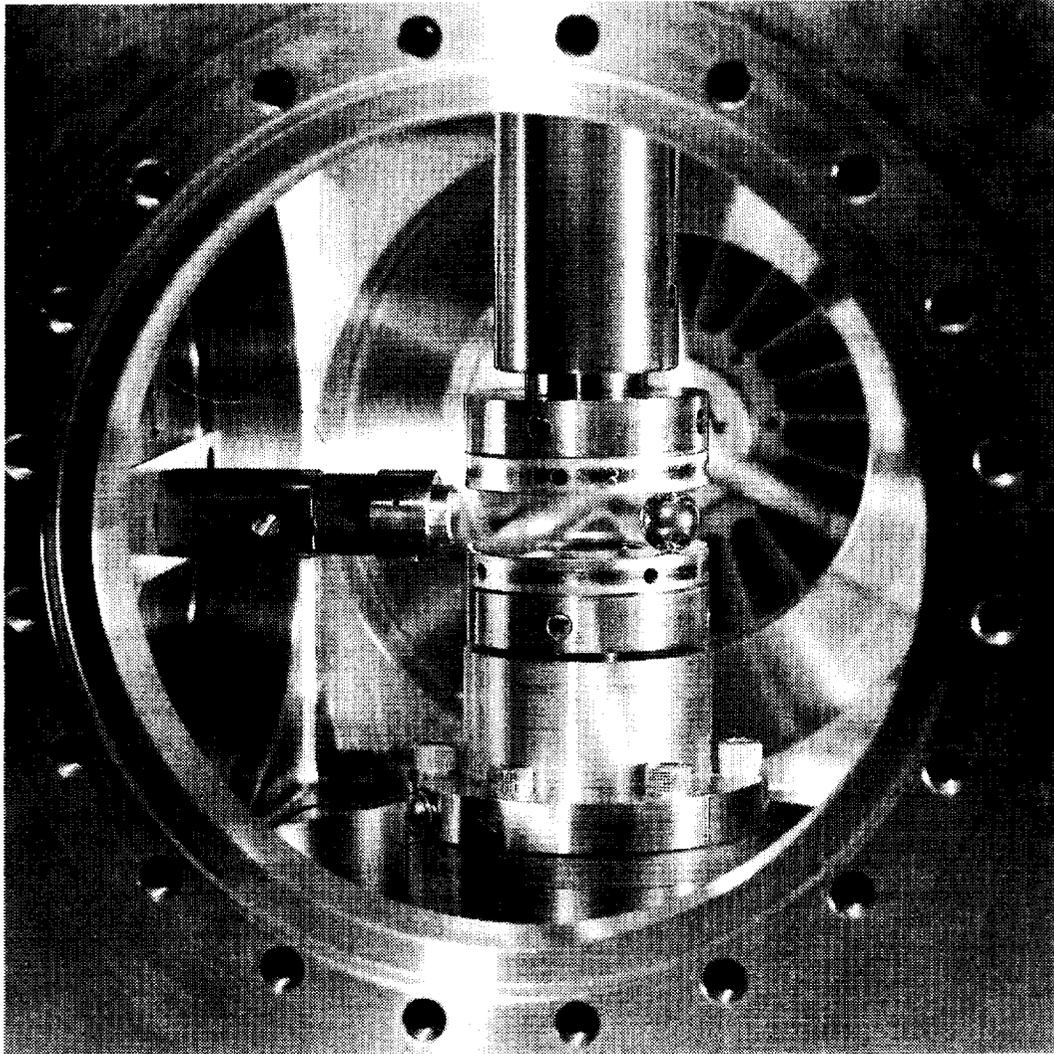


Figure 17.—Spiral Orbit Rolling Contact Tribometer [Pepper et al., 1996].

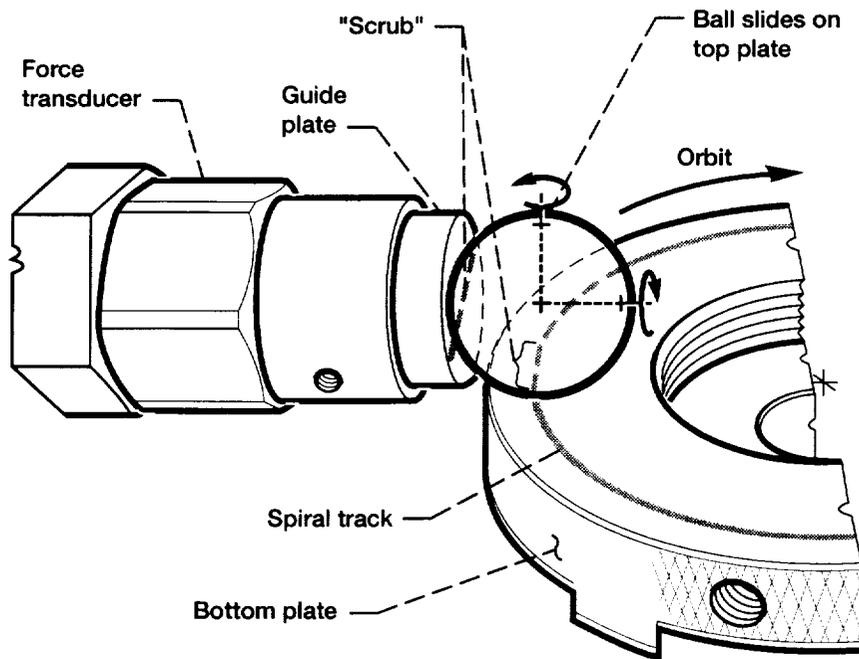


Figure 18.—Detailed view of spiral orbit rolling contact tribometer.

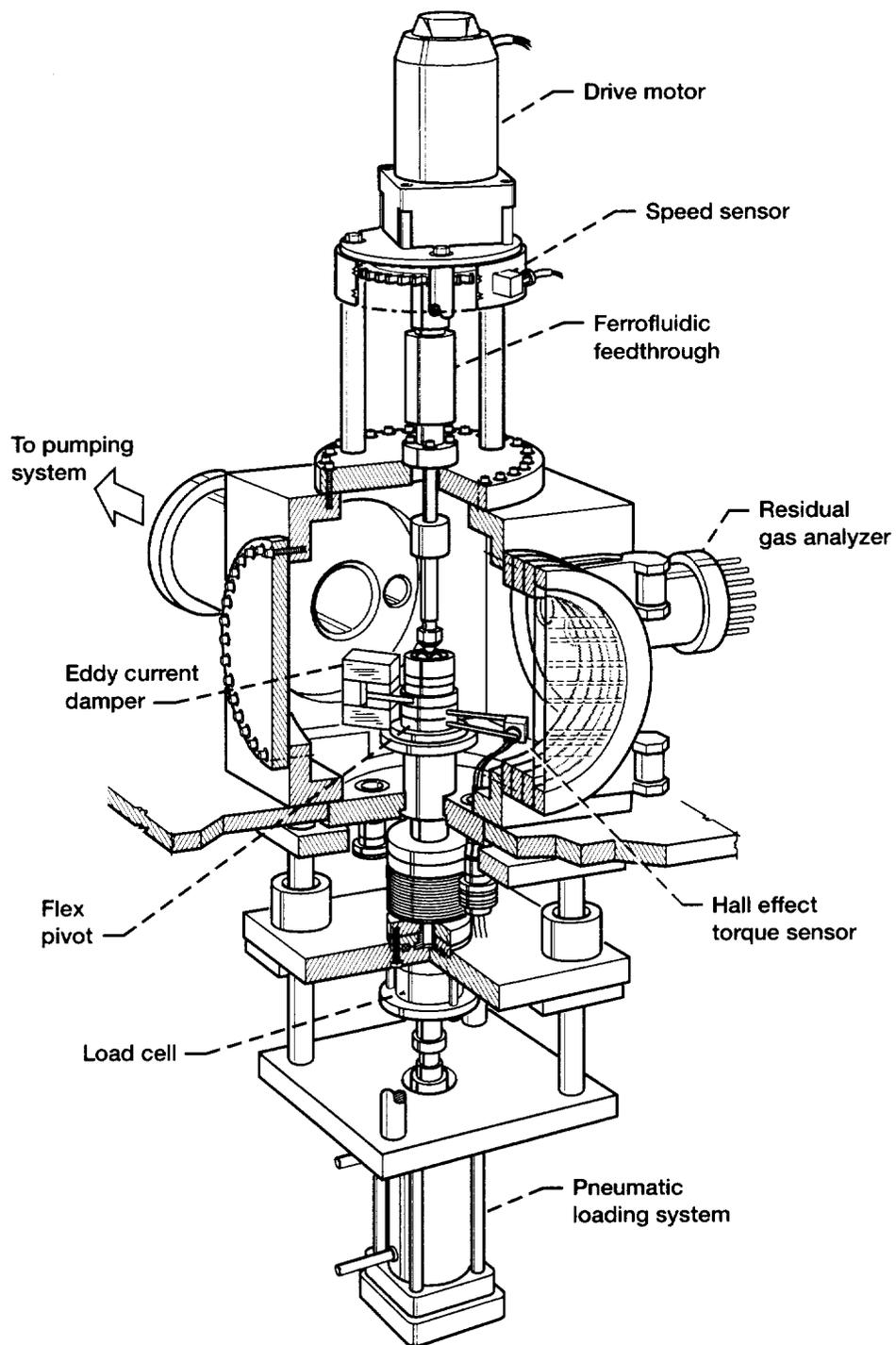


Figure 19.—Overview of 4-ball vacuum tribometer [Jones, 1995].

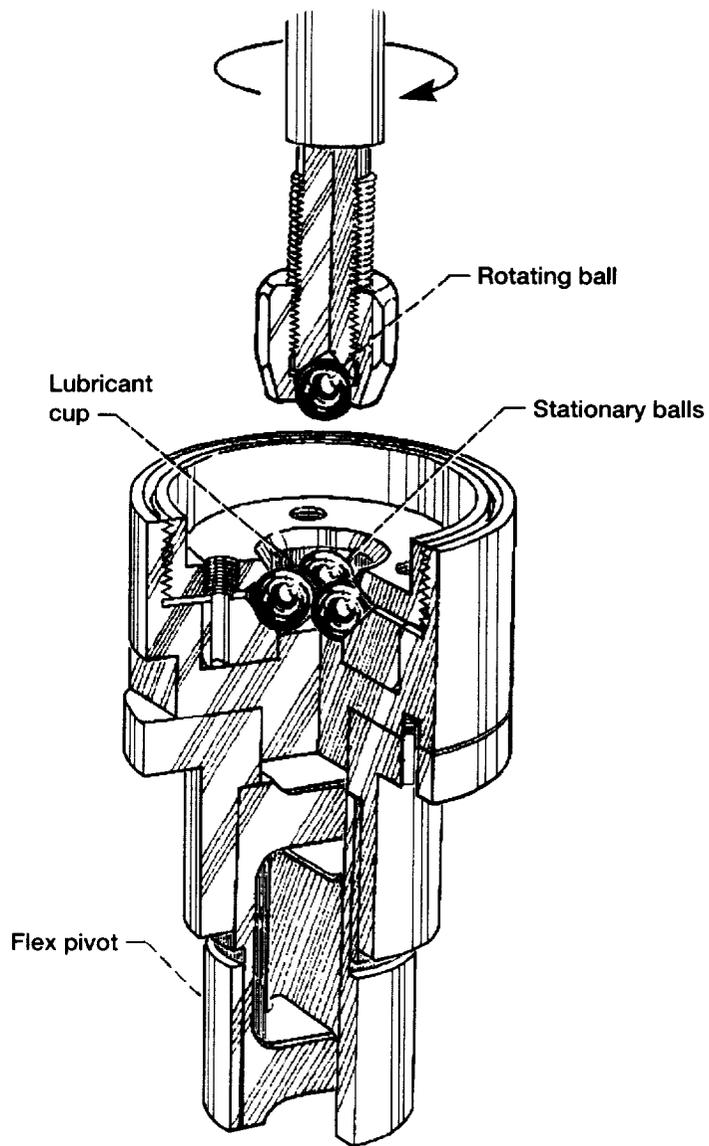


Figure 20.—Detail of 4-ball vacuum tribometer [Jones, 1995].

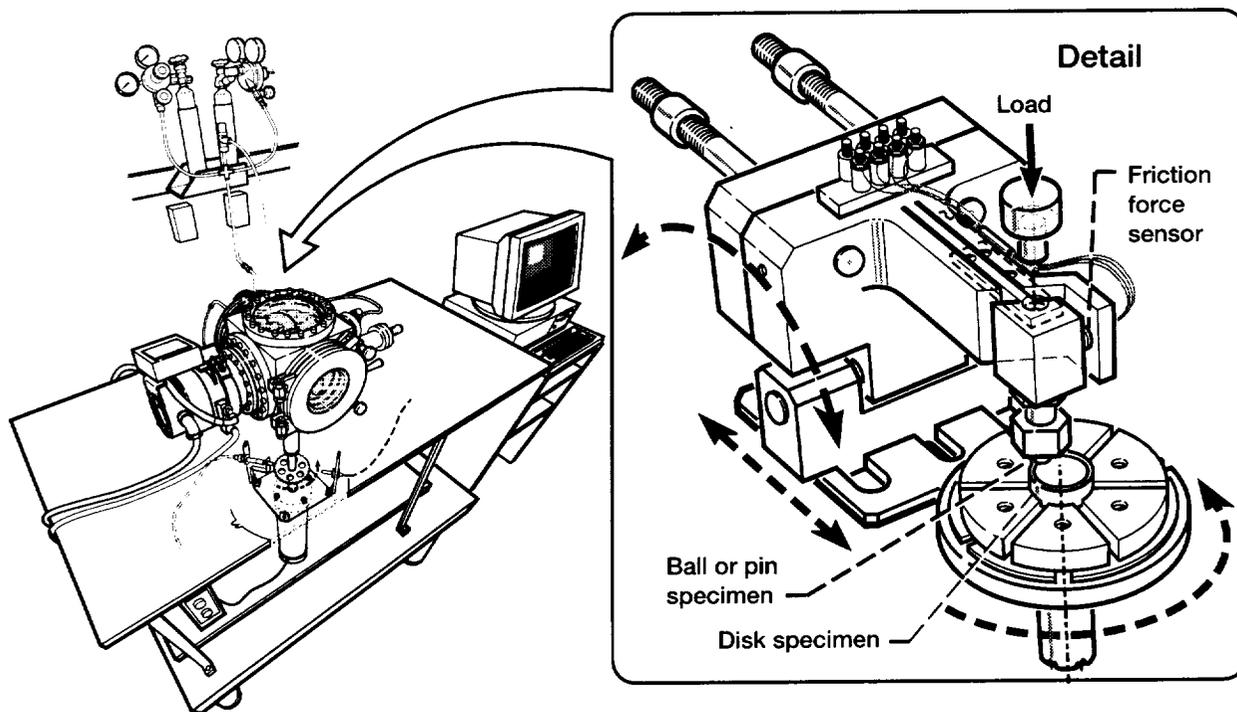


Figure 21.—Ultra-high vacuum pin-on-disk tribometer [Miyoshi et al., 1999].

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