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Summary

The requirement for long-term, reliable operation of aerospace mechanisms has, with a few exceptions, pushed the state of the art in tribology. Space mission life requirements in the early 1960's were generally 6 months to a year. The proposed U.S. space station scheduled to be launched in the 1990's must be continuously usable for 10 to 20 years. Liquid lubrication systems are generally used for mission life requirements longer than a year. Although most spacecraft or satellites have reached their required lifetimes without a lubrication-related failure, the application of liquid lubricants in the space environment presents unique challenges. This report reviews the state of the art of liquid lubrication in space as well as the problems and their solutions.

Introduction

Tribology problems have grown with aerospace advances made over the past 30 years as shown in figure 1 (from Kannel and Dufrane, 1986). In the beginning of the space age life requirements were minutes or hours. In the early 1960's they were generally 6 months to a year. Early deep-space probes were notable exceptions, but these probes had relatively few mechanical assemblies with high cyclic requirements. By the mid-1960's mission life requirements had increased to 3 to 5 years. By the mid-1970's life requirements of 7 to 10 years were common (Ahlborn et al., 1975). The proposed U.S. space station scheduled to be launched in the 1990's must be continuously usable for 10 to 20 years; that is, a 10-year design life requirement with a 20-year life goal (Dolan and McMurtrey, 1985). Despite significant advances in lubrication and mechanical component technology, the demands of these aerospace systems appear to grow faster than the technology. Lubrication problems in space include

(1) Very low ambient pressure
(2) Presence of atomic species other than the normally encountered molecular species
(3) Thermal radiation
(4) Absence of a gravitational field

The absolute pressure outside the Earth's atmosphere (e.g., above 1609 km (1000 miles) altitude) is approximately 10^{-13} torr; the absolute pressure in interstellar space is approximately 10^{-16} torr. Figure 2 shows pressure as a function of altitude (Jastrow, 1960). The low-pressure environment contributes to rapid evaporation of the liquid or semisolid grease lubricants normally employed. Since lubrication ordinarily takes place by means of a film entrained between sliding or rolling surfaces, the loss of this film due to evaporation and mechanical working can result in failure of the mechanism.

With many metals the lubrication function is strongly influenced by the presence or absence of oxide films on these metals. The surface oxides frequently act as protective films and, in some cases, contribute to the final surface films through either chemical reaction or chemisorption. At altitudes greater than 89 km (55 miles) oxygen and nitrogen do not exist as the ordinary molecular species but rather in the atomic or ionic state. The reaction rates between most metals and atomic oxygen are markedly different from those with molecular oxygen. At altitudes greater than 1287 km (800 miles) atomic hydrogen and helium are the principal species present. Mechanisms in a closely sealed satellite are likely to operate in a water vapor pressure of greater than 10^{-7} torr during their first year in orbit. The atmosphere may include trace quantities of carbon monoxide, heavy hydrocarbons, and light silicone polymers with an occasional burst of ammonia from the decomposition products of the hydrazine motors. Some components on the exterior of the satellite operate in a much cleaner environment, the gas pressure generally being less than 10^{-10} torr, but with occasional bursts of ammonia (Robbins, 1975). Because of the scarcity of oxygen, oxide films are formed at a rate inadequate for lubrication.

At extremely low pressure levels (where gas conduction and convection are absent) the temperature levels will normally be dictated by thermal radiation. Heat will be absorbed by radiation from any object that the mechanism "sees," and the mechanism will, in turn, reject heat to outer space by radiation. Various mechanisms will have different temperature levels depending upon their relative rates of heat gain and loss. Robbins (1975) reports that the temperature is controlled by using carefully selected thermal blankets to balance the heat absorbed from the 6000 K (10 341 °F) radiation of the Sun, the heat radiated by the satellite at 300 K (81 °F), and the internal heat generation. He states that in order to provide an equable temperature for electronic components most satellite mechanisms are designed to operate within a narrow temperature band, usually in the range 280 to 320 K (45 to 117 °F). Lubricant evaporation from surfaces is a function of temperature. Hence, a mechanism's temperature is important to the lubrication process and thus to its survivability.

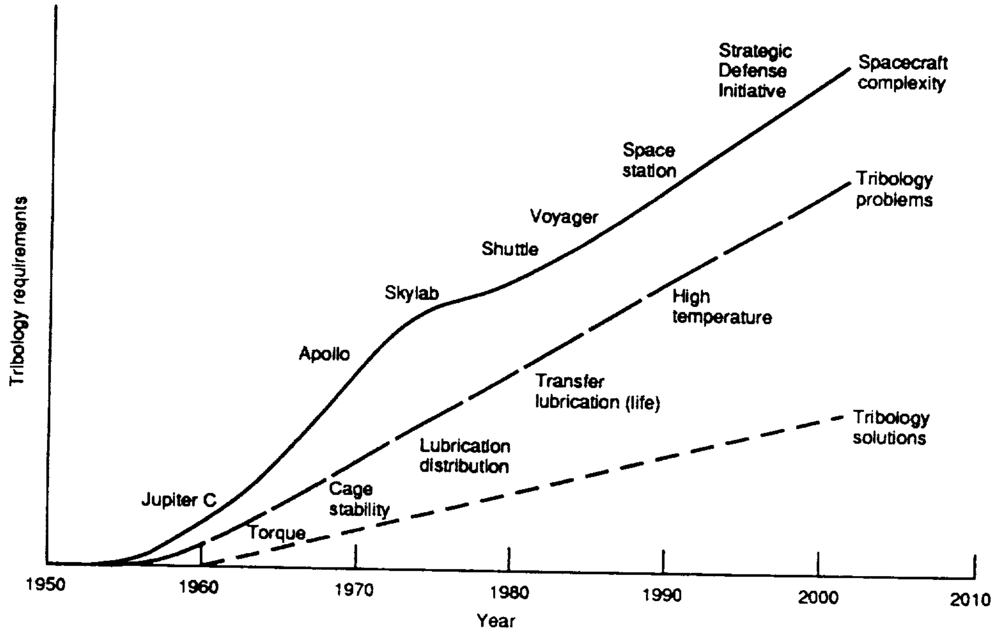


Figure 1.—Growth of tribology requirements with advances in space. Kannel and Dufrane (1986).

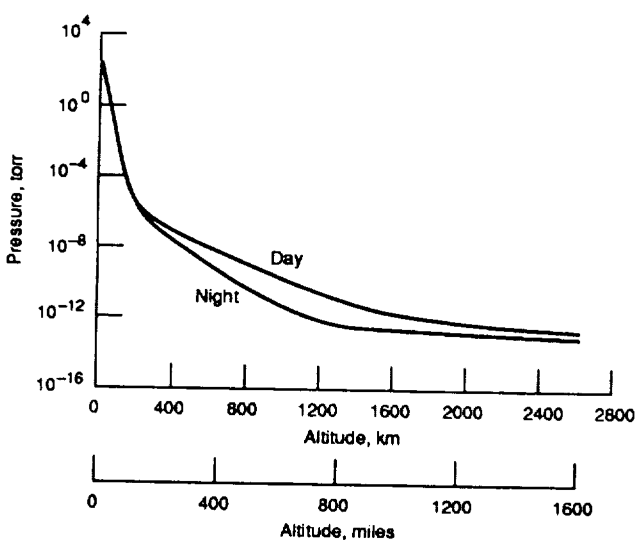


Figure 2.—Pressure as a function of altitude. Jastrow (1960).

Perhaps the only advantage of operation in space is the influence of microgravity. Eliminating the weight of a system from its bearing supports minimizes component wear and reduces the probability of rolling-element fatigue. However, the bearings are preloaded to give precise location and to avoid ball skidding (Robbins, 1975). The resultant bearing load is usually 1 percent or less of its dynamic load capacity. (The dynamic load capacity is defined as a theoretical load that, when applied to the bearing, will result in a life of 1 million inner-race revolutions.)

Spacecraft and satellites are lubricated with dry films, liquids, metallic coatings, greases, or combinations of these lubricants. Liquid lubrication systems are generally used for mission life requirements of over a year, (Ahlborn et al., 1975). The application of liquid lubricants in the space environment presents unique challenges. This report reviews the state of the art of liquid lubrication in space as well as the problems and their solutions.

Lubricant Type and Selection

Corridan (1959), Freundlich and Hannan (1961), and Clauss (1961) conducted early ball bearing tests in vacuum. They used AISI 440C stainless steel instrument bearings with bore sizes from 3.18 to 9.53 mm (1/8 to 3/8 in.) and with oil-impregnated, machined phenolic composition retainers. They obtained good performance with a silicone oil (General Electric Versilube F-50) (Bisson 1964). The operating times of these bearings were 1000 to 4600 hr, considered adequate for the early 1960's. Today they would not be acceptable. Young and Clauss (1966) tested both AISI 52100 and AISI 440C bearings with phenolic cages lubricated with chlorophenyl methyl polysiloxane oil at 10^{-7} to 10^{-9} torr and 422 K (300 °F). The bearings failed in less than 15 000 hr. Silicone-base fluids are not considered to give good lubrication for long-term operation because of their poor boundary lubricating properties and their tendency to creep (Roberts et al., 1990).

Young and Clauss (1966) conducted additional tests with dibasic-acid-ester-base lubricants: MIL-L-6085 (diester-base oil with corrosion inhibitor) and a dioctyl sebacate. Both lubricants resulted in failures in less than 3600 hr. Failure in these bearings and in the silicone-lubricated bearings were associated with either thickening or loss of lubricant. These tests suggest that the ester-base lubricants are less satisfactory than the silicone-base fluids in vacuum even though the esters are superior lubricants for rolling-element bearings in air.

Young and Clauss (1966) also performed tests with naphthenic and paraffinic mineral oils. The naphthenic-oil-lubricated bearings failed in 743 hr because of a complete loss of oil. The paraffinic-oil-lubricated bearings lasted more than 12 000 hr without failure. These tests suggest that in the mid-1960's the paraffinic-base oil had the best potential for long-life applications in space.

In the 1970's two lubricants were developed and successfully ground tested in the United Kingdom: a superrefined mineral oil (BP 110) and a triester fluid (BP 135) (Roberts et al., 1990). The problems associated with high vapor pressure and high outgassing were avoided by using molecular seals, antimigration barriers, and oil reservoirs (Parker, 1987). Hence, the key to long-life reliable application is to maintain a lubricant presence in the bearing cavity.

Superrefined mineral oils became the lubricants of choice for such devices as momentum wheels, reaction wheels, and despin mechanisms. The most popular superrefined mineral oil lubricants are KG 80 (Kendal) and Apiezon C (Shell). Interest in fluids of various viscosities also prompted the development of superrefined gyroscope (SRG) lubricants. These SRG fluids represent a homologous group that allows the user to select a fluid having specific viscosity characteristics for a given application (Kannel and Dufrane, 1986). Typical viscosity-temperature characteristics of these lubricants are given in figure 3. Kannel and Dufrane (1986) conducted torque tests on an R-6 instrument bearing with these SRG lubricants. The results of these tests (fig. 4) clearly show the torque penalty associated with increasing the viscosity of the bearing lubricant. However, there are times when the penalty of higher torque must be offset by the bearing's ability to develop a good

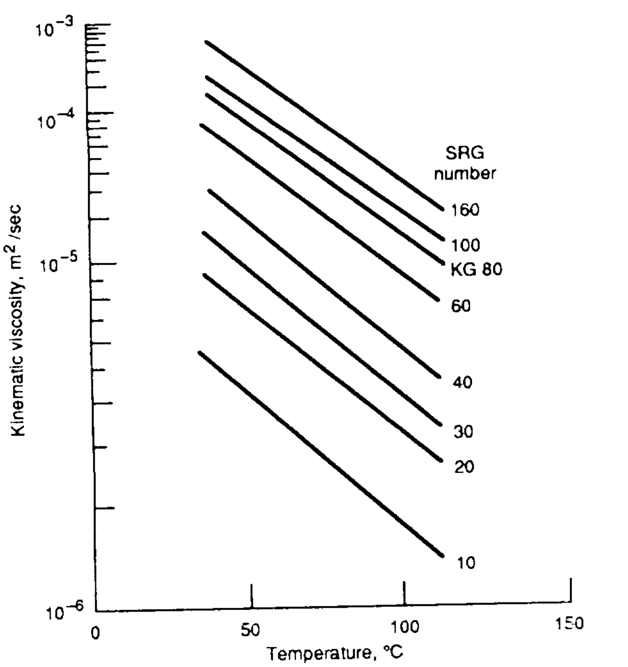


Figure 3.—Viscosity as a function of temperature for a homologous series of superrefined mineral oils. Kannel and Dufrane (1986).

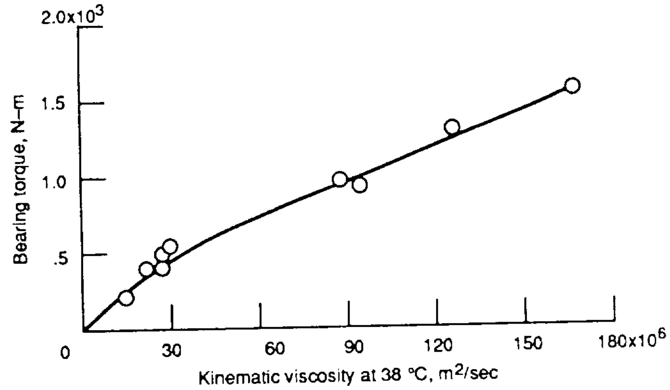


Figure 4.—Measured bearing torque as a function of lubricant viscosity for various lubricants at 25 °C for an R-6 bearing at 480 rpm. Kannel and Dufrane (1986).

elastohydrodynamic (EHD) film thickness between the balls and the races.

Perfluorinated Polyalkylethers

Roberts et al. (1990) report that in more exposed applications, or where operation at temperatures as low as 200 K (−100 °F) is required, or where a high viscosity index is deemed important, the perfluorinated polyalkylethers (PFPE) are widely used. Because minimal attenuation of infrared radiation occurs with these fluids, they have been used as lubricants within infrared scanners on spacecraft.

Table I (from Kannel and Dufrane, 1986) presents measured evaporation rate data for superrefined mineral oil and PFPE lubricants. PFPE lubricants have the lower evaporation rates. In general, evaporation rates for a given type of fluid decrease as the viscosity is increased. For longer life applications the lower loss rates associated with higher viscosities are desirable. However, the high-viscosity lubricants have the disadvantage of higher operating torques.

The general conclusion has been that the PFPE lubricants are the most desirable for spacecraft applications. Three commercial PFPE lubricants have been widely used: Krytox PR 143AC (DuPont), Fomblin Z25 (Montecatini Edison), and Brayco 815Z (Bray). These fluids generally have the lowest

TABLE I.—EFFECT OF VISCOSITY ON EVAPORATION RATE OF TWO LUBRICANTS IN A 0.133-MPa (10^{-6}-torr) VACUUM
[Kannel and Dufrane (1986).]

Fluid	Published viscosity at 40 °C, m^2/sec	Evaporation rate at 40 °C, mg/cm^2-hr
Superrefined paraffinic mineral oil (SRG 30)	14×10^6	18
Superrefined paraffinic mineral oil (SRG 40)	27	13
Perfluoro ether[a]	8	2.2
Perfluoro ether[a]	28	.19
Perfluoro ether[a]	357	.0002

[a]Type and manufacturer not specified in reference.

vapor pressure and highest viscosity index of lubricants considered for space application. Fomblin, which contains an acetal linkage, has a lower vapor pressure and a higher viscosity index than Krytox. However, as more long-term experience has been gained with these lubricants, degradation of the lubricant has been observed. This degradation has resulted in high bearing torque noise and excessive wear (Stevens, 1983).

The work of Baxter and Hall (1985) with Fomblin Z25 suggests that the degradation is caused by the presence of chemically active surfaces, wear particles combined with exposed radicals in the fluid, or both. Zehe and Faut (1989) and Carre (1987) suggest that metal oxide surfaces in contact with acetal-containing PFPE lubricants (i.e., Fomblin Z25) will inevitably result in acidic breakdown of the ethers. They conclude that little can be done to the PFPE chains to block the acidic attack without compromising the viscosity-temperature qualities of the acetal groups.

Mori and Morales (1989a,b) studied the effects of Krytox 16256 and Demnum S200 (Daikin) in addition to Fomblin Z25. Fomblin Z25, a copolymer of perfluoromethylene and perfluoroethylene oxides, is the only one of three fluids that contains an acetal linkage. Krytox 16256 is a poly(perfluoropropylene oxide) with pendent $-CF_3$ groups. Demnum S200 is a poly(perfluoropropylene oxide). The properties of PFPE fluids are summarized in table II.

Mori and Morales (1989a) found that PFPE decomposition and the resulting reaction products are dependent on the particular molecular structure of the PFPE fluid when irradiated by x-rays under ultra-high-vacuum conditions. They also studied the reaction of the PFPE fluids with AISI 440C stainless steel during sliding under 10^{-10} torr at room temperature (Mori and Morales, 1989b). All three fluids reacted with the AISI 440C material during sliding. Fomblin Z25 decomposed during sliding and gaseous products, mainly COF_2, were formed. There were no gaseous products from the Krytox or Demnum fluids.

TABLE II.—PROPERTIES OF PFPE FLUIDS
[Mori and Morales (1989a).]

Property	Fluid		
	Demnum S200	Fomblin Z25	Krytox 16256
Average molecular weight	8400	9500	11 000
Kinetic viscosity at 20 °C, cS	500 + 25	255	2717
Viscosity index	210	355	-------
Pour point, °C	−53	−66	−15
Density at 20 °C, g/ml	1.894	1.851	1.92
Surface tension at 20 °C, dyne/cm	19	25	19
Vapor pressure, torr:			
At 20 °C	5×10^{-11}	2.9×10^{-12}	3×10^{-14}
At 100 °C	1×10^{-7}	1×10^{-8}	1×10^{-9}

Solutions to the problem of thermo-oxidative breakdown should focus on the surfaces in contact with the liquid. Some success in blocking thermo-oxidative degradation of linear PFPE's below 570 K (566 °F) has been achieved with the use of additives containing nitrogen or phosphorous atoms. The action of these additives may depend upon their ability to block acidic sites on the surface (Jones et al., 1983, 1985). Other approaches may involve surface modification to limit decomposition, since it is the catalytic action of the resulting surface that accelerates the breakdown (Zehe and Faut, 1989). Table III (from Roberts et al., 1990) lists factors that both promote and retard PFPE degradation.

Performance Characteristics

Thomas (1980), in order to provide data for ball bearing torque calculations, performed sliding tests in air using AISI 52100 balls with the more common spacecraft lubricants

TABLE III.—FACTORS INFLUENCING PFPE DEGRADATION
[Roberts et al. (1990).]

Factors promoting degradation	Factors retarding degradation
Starved conditions (e.g., grease plating)	Fully flooded conditions
Low specific film thickness ($\lambda < 1$)	High specific film thickness ($\lambda > 4$)
Fomblin Z type of perfluorinated oil base (linear structure)	Fomblin Y type of perfluorinated oil base (branched structure); Krytox rarely used in Europe
Aluminum/titanium substrates at any ambient temperature	Hydrocarbon contamination
AISI 52100 bearing steel	AISI 440C bearing steel; ceramic coatings give further improvement (e.g., TiC -coated balls)
High ambient temperature (>360 °C)	Low ambient temperature
Sliding surfaces	Surfaces where rolling motion takes place
Vacuum environment	Normal atmospheric conditions

TABLE IV.—COMMONLY USED LIQUID LUBRICANTS FOR SPACE TRIBOLOGICAL APPLICATIONS

[Thomas (1980).]

Lubricant	Manufacturer	Viscosity at 20°C, cS	Vapor pressure at 20°C, torr	Description and comments
ApiezonC	Shell	250	4.0×10^{-9}	Mineral oil with no additives; used as a reference oil
BP 135[a]	British Petroleum	70	7.9×10^{-9}	Synthetic; triester base; boundary lubricant and antioxidant additives
BP 110[a]	British Petroleum	520	3.7×10^{-8}	Mineral oil base; high viscosity; refined to give low vapor pressure; boundary lubricant additives
KG 80	Kendall Refining Company	520	$<10^{-8}$	Petroleum base with boundary lubricant additive tricresylphosphate; also contains an antioxidant additive
Fomblin Z25	Montecatini Edison	240	$<5.0\times10^{-12}$	Synthetic fluorinated oil; high density, low surface tension; high temperature and high viscosity index; no boundary lubricant additives; high thermal resistance

[a]No longer marketed. Laboratory tested but never flown in spacecraft.

TABLE V.—COEFFICIENT OF FRICTION AND WEAR RATE OF STEEL ON STEEL IN LIQUID-LUBRICATED SLIDING CONTACT

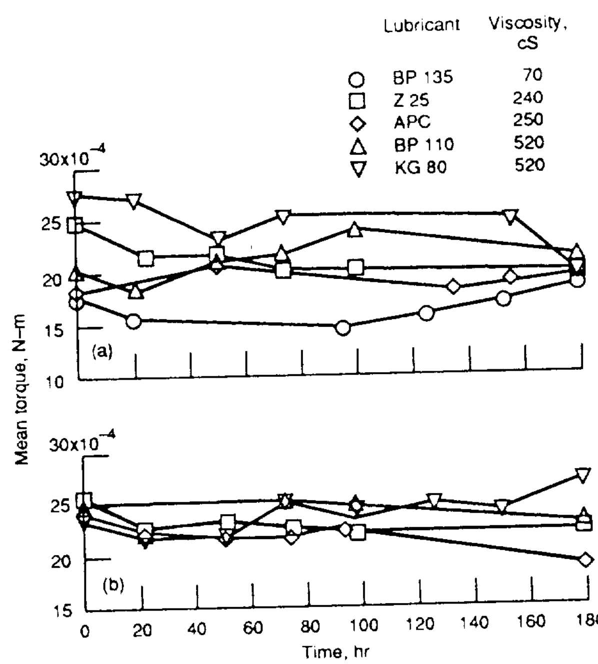
[Thomas (1980).]

Lubricant	Coulomb friction coefficient	Specific wear rate after 1000 revolutions, m^3/N-m
Unlubricated, "failed" after 100 revolutions	[a]0.46	$19.1\pm0.9\times10^{15}$
Apiezon C	$.20\pm.03$	$.93\pm.05$
BP 135	$.12\pm.015$	$1.04\pm.05$
BP 110	$.13\pm.015$	$.41\pm.02$
KG 80	$.13\pm.02$	$.69\pm.03$
Fomblin Z25	$.12\pm.02$	$.49\pm.03$

[a]Friction rose rapidly to this value.

described in table IV. The results of these tests are presented in table V. At slow speeds, where there is essentially no elastohydrodynamic lubrication, and with light loads the lubricating properties of the five oils were similar. Apiezon C, which had no additives, exhibited the highest mean friction coefficient of 0.20. The other four lubricants had lower friction coefficients, ranging from 0.12 to 0.13. With the exception of the Apiezon C, which had no boundary lubricant additives, all of the lubricants performed identically in the sliding tests in air.

Stevens (1983), using the same lubricants as Thomas (1980), performed lubrication tests to determine the wear and torque of satellite, 20-mm-bore, angular-contact ball bearings operating at 10 rpm. That speed corresponds to the boundary lubrication regime. The tests were conducted both in vacuum and in air. The results of these tests are shown in figure 5.



(a) Air.

(b) Vacuum (10^{-6} torr).

Figure 5.—Ball bearing torque as a function of lubricant viscosity. Bearing bore, 20 mm; contact angle, 15°; cage, one-piece phenolic (outer-race riding); number of balls, 10; speed, 10 rpm; thrust load, 40 N. Stevens (1983).

Figure 5 shows the mean torque, or the average of the torque variations, as a function of time for each lubricant. Although the torque differences between the five lubricants are not considered significant, there appears to be a trend of increasing torque with increasing lubricant viscosity. This trend correlates with the trend shown in figure 4 by Kannel and Dufrane (1986). These results strongly suggest that the magnitude and differences in bearing torque are due to viscous effects. The variation in torque about the mean value was reported to be similar for the five lubricants. The highest torque was obtained with the KG 80 lubricant, which has a viscosity of 520 cS at 20 °C. The lowest torque was obtained with the BP 135 lubricant, which has a viscosity of 70 cS at 20 °C. On the basis of work by Todd and Stevens (1978) with a cageless three-ball bearing, Stevens (1983) predicted that the torque due to spin, hysteresis, and microslip would be 14×10^{-4} N-m. In tests by Stevens (1983) with KG 80 and Apiezon C at speeds of 0.2 and 1 rpm and a thrust load of 40 N, the torque of these bearings measured at 10 rpm ranged from 15×10^{-4} to 17×10^{-4} N-m. At 10 rpm and a higher load of 400 N the torque was less with KG 80 (120×10^{-4} N-m) than with Apiezon C (200×10^{-4} N-m). These results suggest that at the higher load the torque is dependent on the sliding friction coefficients reported in table V by Thomas (1980). Hence, it may be reasonably concluded that at fractional speeds bearing torque is related to ball spin, hysteresis, and microslip. At light loads and extremely low speeds the torque is related to lubricant viscosity, primarily because of cage friction. At low speeds and high loads the torque is related to the Coulomb friction coefficient (table V). In satellite applications the only load the bearing experiences in the space environment is the applied bearing preload, which is generally low. If the speed is high enough for a sufficient elastohydrodynamic film to form, the bearing torque will be a function of the lubricant's viscosity at the operating temperature of the bearing.

In the 1960's Young et al. (1963) and Young and Clauss (1966) performed extensive lubricant research in vacuums ranging from 2×10^{-7} to 4×10^{-9} torr. The following types of liquid lubricant were endurance tested with both AISI 52100 and AISI 440 ball bearings:

(1) Paraffinic and naphthenic mineral oils
(2) Chlorophenyl methyl polysiloxane oils
(3) Mixed isomeric five-ring polyphenyl ethers
(4) Diphenyl bis-n-dodecyl silane
(5) Dibasic-acid-ester-base oils

These lubricants are listed here in the order of the endurance lives obtained with the bearings. The best results and the longest lives were obtained with bearings having a paper-based phenolic cage lubricated with a paraffinic-base mineral oil containing additives used for gyroscope bearings. The phenolic cage is used to retain and supply the oil. Linen-based phenolic cages with the same lubricant did not provide as good results. The linen phenolic cages tended to dry out and crack. Tests were run with the chlorophenyl methyl polysiloxane "as received" and after the 50 percent most volatile fraction was removed by distillation. These tests were run to determine whether removing the most volatile material would lengthen the operating time in vacuum. The as-received material gave at least twice the life (Young et al., 1963).

Flom and Haltner (1968) list early satellite applications using chlorophenyl methyl silicones. The most common of these fluids was General Electric Versilube F-50. However, the use of F-50 for space applications appears to have met disfavor in the 1970's. The fluid polymerizes both in vacuum and in air because of microasperity interaction. Experience with chlorophenyl methyl silicones has also shown that, where there are high sliding velocities, large amounts of wear can occur in angular-contact ball bearings and sliding surfaces. Meeks et al. (1971) conclude that the estimated life before significant torque fluctuations (greater than two times average) for F-50-lubricated, small bearings run at 55 rpm below 10^{-8} torr is less than 1 year. This result is due to thermally induced chemical changes in the oil. Meeks et al. (1971) predict that the ultimate failure life of these bearings will be less than 2 years.

Fleischauer and Hilton (1990) discuss the use of polyalphaolefin (PAO) and polyolester (PE) oils as possible substitutes for linear perfluorinated polyalkylethers. They suggest that these oils can be synthesized and blended to produce in a controlled way viscosities, vapor pressures, pour points, and other properties that can meet various spacecraft requirements. These oils can also be blended with standard additive packages similar to those in conventional mineral oils. However, there does not appear to be a definitive data base that would allow using these oils in a satellite or spacecraft system at this time with a reasonable degree of confidence.

Elastohydrodynamic and Boundary Lubrication Effects

Lubricant Function

The primary function of a liquid lubricant in space applications is to separate surfaces in relative motion so that the surfaces do not sustain major damage and to keep the coefficient of friction between the surfaces relatively low. A number of lubrication regimes, depending on the type of intervening film and its thickness, can be identified. These lubrication regimes can be depicted by the Stribeck-Hersey curve shown in figure 6 (from Jones, 1982). This curve takes the form of the friction coefficient as a function of the parameter ZN/P, where Z is the viscosity of the liquid, N is the velocity, and P is the load.

At high values of ZN/P, which occur at high speeds, low loads, and high viscosities, the surfaces are completely separated by a thick (0.25 μm; 10^{-5} in.) lubricant film. This is the hydrodynamic lubrication regime, 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 fluid film regime, elastohydrodynamic lubrication, can be identified. In this regime film thickness may range from 0.025 to 2.5 μm (10^{-6} to

10^{-4} in.). As the film becomes progressively thinner, surface interactions begin. This regime of increasing friction, which combines asperity interactions and fluid film effects, is referred to as the "mixed-lubrication regime." Finally, at low values of the ZN/P parameter is the boundary lubrication regime. This regime is highly complex, involving metallurgy, surface topography, physical and chemical adsorption, corrosion, catalysis, and reaction kinetics. Its most important aspect is the formation of a protective surface film to minimize wear and surface damage. The formation of surface films is governed by the chemistry of the film-forming agent as well as by the surface of the solid and other environmental factors. The effectiveness of surface films in minimizing wear is determined by their physical properties, which include shear strength, thickness, surface adhesion, film cohesion, melting point or decomposition temperature, and solubility.

Besides the Stribeck-Hersey curve (fig. 6) already described, an idealized plot of wear rate as a function of relative load can also delineate the various lubrication regimes and some wear transitions. Region O–A of figure 7 (from Beerbower, 1972) encompasses the hydrodynamic and elastohydrodynamic lubrication regimes, the latter as point A is approached. Since no surface interactions occur in this region except during startup and shutdown, little or no wear occurs, except that caused by rolling-element fatigue, which can occur without surface interactions. Region A–X is the mixed-lubrication regime, where surface interactions begin to occur at A and become more prevalent as point X is approached. Wear is low because fluid film effects still exist.

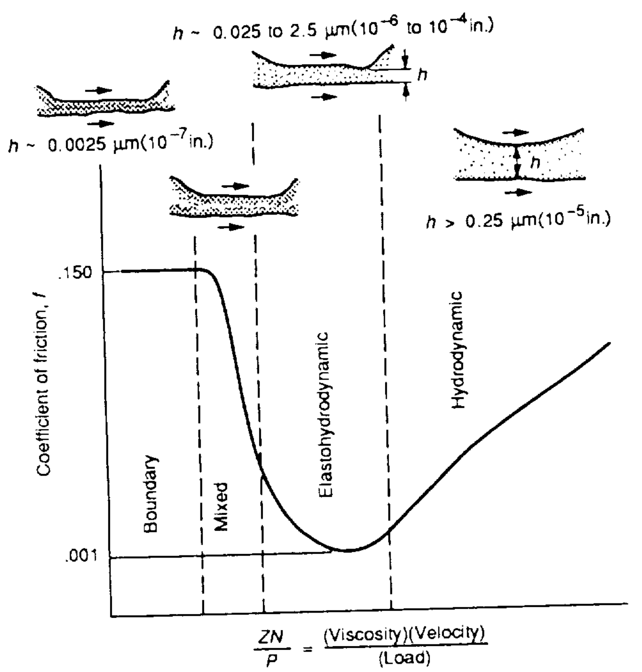


Figure 6.—Coefficient of friction as function of viscosity-velocity-load parameter (Stribeck-Hersey curve). Jones (1982).

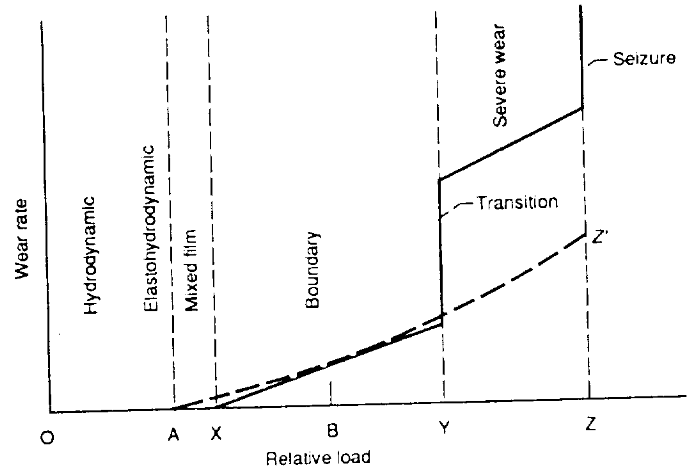


Figure 7.—Wear rate as a function of relative load, depicting various lubrication regimes. Beerbower (1972).

Region X–Y is the boundary lubrication regime. The degree of metal-to-metal contact and the wear rate increase as the load increases. Wear is mild and tends to be corrosive to the left of B and adhesive to the right of B. The location of B is quite variable and depends on the corrosivity of the lubricant formulation. For a noncorrosive lubricant adhesive wear can occur at X. On the other hand, a corrosive additive can extend the boundary regime to Z′ before boundary film failure occurs. Region Y–Z is the severe wear regime, where severe adhesion and scoring occur. Mechanisms cannot operate successfully in this regime, and therefore the location of this transition point is quite important. At Z total surface failure occurs, followed by seizure.

In the boundary lubrication regime many properties of the liquid lubricant become important: shear strength, film thickness, melting point, and chemical reactivity with the surface. Operating variables that affect lubricant film performance include load, speed, temperature, and atmosphere, as already discussed. Additives present in the lubricant to serve specific functions, such as antiwear, antifoam, and antioxidant additives and viscosity improvers, also affect behavior (Jones, 1982).

Elastohydrodynamic Theory

Elastohydrodynamic (EHD) lubrication theory is well documented by Dowson and Higginson (1966) and Hamrock and Dowson (1981). The Grubin (1949) EHD film thickness formula is as follows:

$$H = 1.95\ G^{0.73}U^{0.73}W^{-0.091}$$

where

H	film thickness, h/R_x
G	materials parameter, $\alpha E'$

U speed parameter, $u\eta_0/E'R_x$

W load parameter, $F/E'R_x^2$

α pressure-viscosity exponent, GPa^{-1} (psi^{-1})

E' reduced modulus of elasticity,

$$\frac{1}{E'} = \frac{1}{2}\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)$$

u average surface speed, $(u_1 + u_2)/2$, m/sec (in./sec)

η_0 atmospheric viscosity, N-sec/m^2 (lb-sec/in.^{2})

R_x equivalent radius in rolling direction, m (in.),

$$\frac{1}{R_x} = \frac{1}{R_{x,1}} + \frac{1}{R_{x,2}}$$

F normal applied load, N (lb)

E_1, E_2 Young's modulus for body 1 and body 2, GPa (psi)

h film thickness, m (in.)

ν_1, ν_2 Poisson's ratio for body 1 and body 2

The variations in EHD theory and the resulting formulas are beyond the scope of this report. Suffice it so say that the variations between the EHD film thicknesses calculated by the different formulas are less than the variations between the various sets of experimental data used to verify the theories (Coy and Zaretsky, 1981). This is illustrated in figure 8, which shows the film thicknesses calculated by the theories of Hamrock and Dowson (1977), Grubin (1949), Cheng (1972), and Chiu (1974). A theoretical model for oil starvation in a rolling-element bearing was formulated by Chiu (1974). Figure 8 also compares the results from Chiu's (1974) analysis with his experimental data and experimental data of Coy and Zaretsky (1981). Oil starvation in the Hertzian contacts even under flooded conditions appears to be the primary cause of the deviation of the experimental data from classical EHD theory (Coy and Zaretsky, 1981).

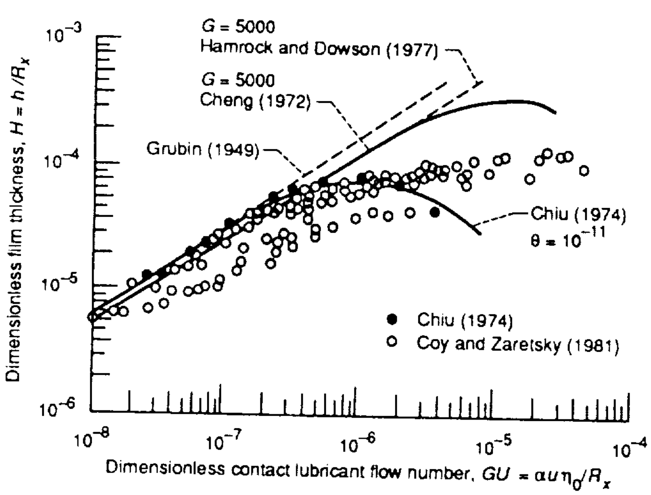


Figure 8.—Theoretical effect of kinematic starvation and inlet shear heating on film thickness and comparison with experiment. Coy and Zaretsky (1981).

The measure of the effectiveness of the lubricant film is the λ ratio (i.e., the central film thickness divided by the composite surface roughness of the rolling-element surfaces h_c/σ). Usually the root mean square (rms) surface finishes of the contacting bodies σ_1 and σ_2 are used to determine the composite surface roughness as follows:

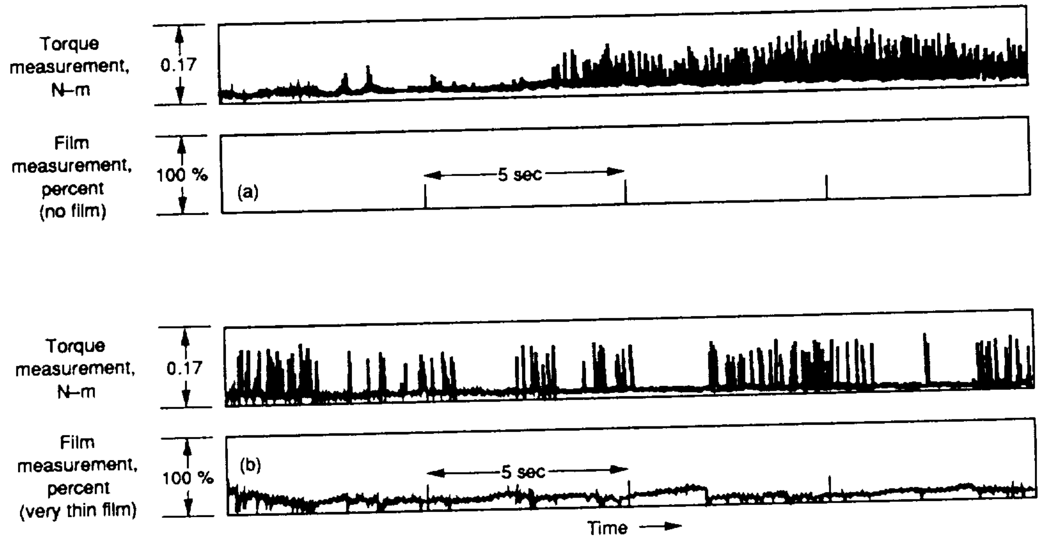
$$\sigma = \left(\sigma_1^2 + \sigma_2^2\right)^{1/2}$$

The λ ratio can be used as an indicator of rolling-element performance and life. For $\lambda < 1$, surface smearing or deformation, accompanied by wear, will occur on the rolling surface. For $1 < \lambda < 1.5$, surface distress may be accompanied by superficial surface pitting. This type of pitting was observed by Stevens (1983) in oil-lubricated satellite ball bearings. For $1.5 < \lambda < 3$, some surface glazing can occur with eventual roller failure caused by classical subsurface-originated rolling-element fatigue. At $\lambda \geq 3$, minimal wear can be expected with extremely long life; failure will eventually be by classical subsurface-originated rolling-element fatigue. The most expedient, although not the least expensive, way of attaining a high λ ratio is to select a high-quality surface finish. The effect of film thickness is to reduce the magnitude and instability of the torque, as shown in figure 9.

A good example of an effective lubricant film is provided by the NASA Solar Maximum Mission satellite, known as Solar Max. The 3-ton spacecraft came crashing into the atmosphere on December 2, 1989, after 9 years in orbit. In 1984, Solar Max became the first satellite to be captured and repaired in space. It had been in orbit for 4 years when NASA space shuttle astronauts replaced its inertial reference unit and returned the original unit to Earth. The unit had three gyroscopes, each with two ball bearings. One of these gyroscopes was disassembled and its bearings examined. The AISI 440C bearings had a 7-mm bore. They had run continuously at 6000 rpm. The cages were made from a phenolic material impregnated with KG 80 lubricant. The two bearings were found to be well lubricated, with oil still in the cages. Iron and chromium debris particles were found in both bearings. The surfaces of the bearing raceways and balls had superficial pits and scratch-like deformations. The wear debris, which was considered small, was attributed to the pits. This micropitting appeared similar to and confirmed that reported by Stevens (1983). It was concluded that these bearings would have lasted for their predicted life of 5 years (Uber, 1985), and this conclusion was borne out by the additional 5-year operation of the Solar Max satellite.

Design Criteria

Ahlborn, et al. (1975) report that they designed their bearings to operate at $\lambda < 1$ and maximum Hertz stresses less than 1.03×10^9 N/m^2 (150 000 psi). However, their test data indicate that failure to meet these criteria does not mean that long life is precluded. They conducted two separate tests



(a) Dry bearing.
(b) Two drops of oil in bearing.

Figure 9.—Angular-contact ball bearing torque and film thickness measurement with no lubricant and marginal lubrication. Bearing bore, 90 mm; contact angle, 15°; thrust load, 267 N (60 lb); number of balls, 21; speed, 100 rpm; lubricant, Apiezon C. Torque spikes imply instability. Kannel and Dufrane (1986).

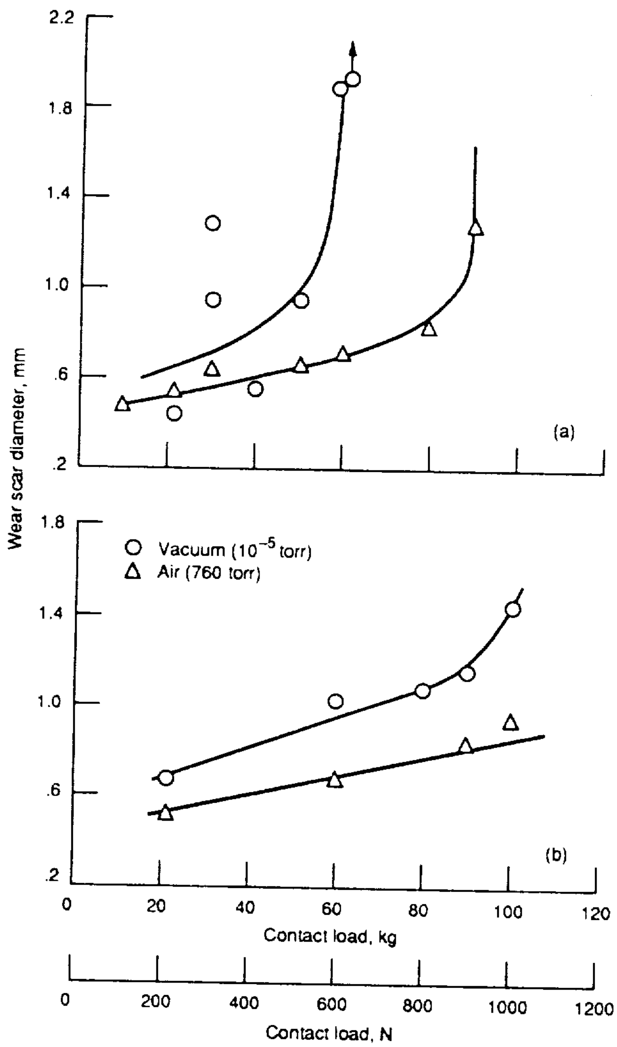
in vacuum with R-6 ball bearings (six bearings in each test, with different lubricants) at a maximum Hertz stress of 8.27×10^8 N/m^2 (120 000 psi) and λ of 0.3 and 0.4. There was no detectable wear on the balls or races after 20 and 14 million revolutions, respectively. In another test they operated larger bearings for more than 2 years at 90 rpm at a λ of 0.5 with no change in performance. Stevens (1983) concludes that the predominant form of damage that occurs on the races and balls is micropitting, some of which is caused by repeated indentation of the steel surface by small, hard features. (Micropitting is also characteristic of microasperity interaction under boundary lubrication conditions.) In tests conducted by Stevens the scale of the damage was small and had no significant effect on the torque characteristic of the bearing, even down to zero speed.

Boundary Lubrication

No definitive research has been published comparing the effect of lubricant additives with various base stocks on rolling-element torque characteristics and wear both in and out of vacuum. Reichenbach et al. (1964) conducted experiments with fluid lubricants in air and in vacuum to determine the influence of a modest vacuum of 10^{-5} torr on the wear behavior of AISI 52100 steel. The experiment consisted of crossed cylinders in contact, with one of the cylinders rotating. After the cylinders had rotated in contact under fixed load for a set time, the wear scars on the cylinders were measured. The wear scar diameters on the cylinders lubricated with a superrefined paraffinic mineral oil are shown in figure 10(a) as a function of contact load. The lubricant contained no additives. The wear scar increased with increasing contact load. The wear scar was also greater in vacuum than in air because the vacuum environment (10^{-5} torr) caused the mineral oil to degas. Gases entrained in conventional oil lubricants include oxygen, water vapor, and nitrogen. Oxygen and water vapor help protective surface films to form on the steel surfaces. These films minimize wear for two solid surfaces in contact. When the fluids are degassed, the beneficial effects of the oxygen and the water vapor are lost, and the wear of the surfaces in contact increases. The lubricant becomes less effective in a vacuum environment than in air (Buckley, 1971).

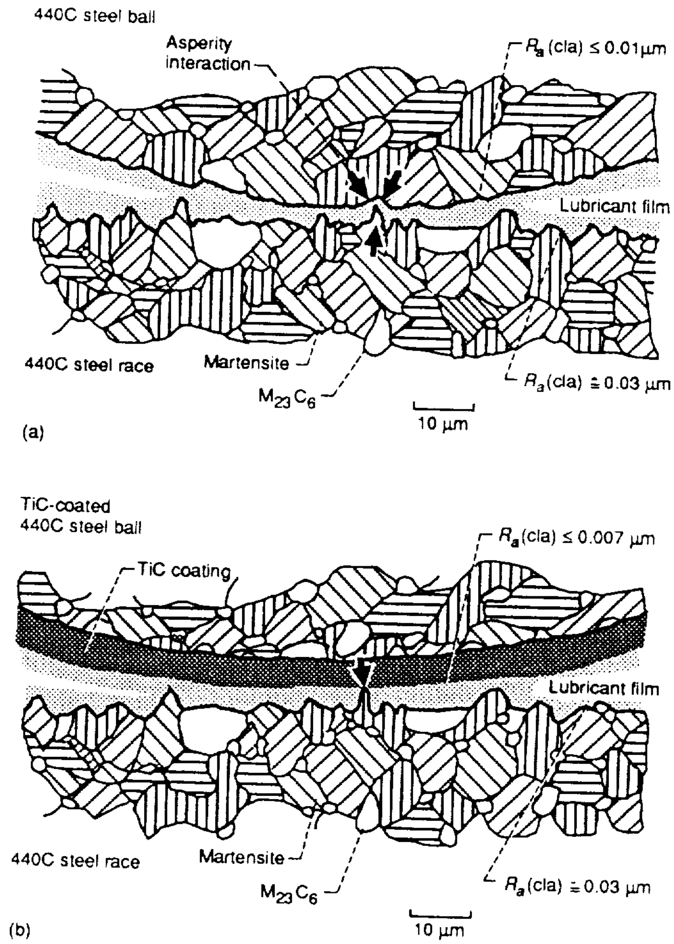
Reichenbach et al. (1964) repeated these tests with SAE 90 EP gear oil (fig. 10(b)). SAE 90 EP is a high-viscosity oil containing a full commercial additive package. Here again, the wear was higher in vacuum than in air. Buckley (1971) concludes that the wear data indicate that the additives improve the load-carrying capacity of the fluid. However, it can be argued that if the additives are effective in vacuum, there should be no significant difference between the amount of wear obtained in air and in vacuum.

Stevens (1983) concludes that the amount and type of damage change very little with the type of oil used in rolling-element bearings in space applications. As a result, the choice of lubricant for variable-speed bearings in spacecraft applications can be made on the basis of vapor pressure and viscosity without regard to boundary lubricating properties. Most space mechanisms operate at relatively low speeds. From conventional industrial experience, even though the bearings are operating in the boundary lubrication regime, no significant damage or wear will occur at low speeds. This is not the case at moderate to high speeds. In a space application, without effective coatings or boundary lubrication films or both, it can reasonably be expected that adhesive wear and plastic deformation or smearing of the surfaces will occur at these speeds. Figure 11(a) shows a ball-to-race contact region for



(a) Superrefined mineral oil.
(b) SAE 90 EP gear oil.

Figure 10—Cross-cylinder load-carrying tests. Material, AISI 52100; diameter, 0.64 cm; cylinder speed, 95 rpm; experiment duration, 1 hr; ambient temperature, 20 °C (293 K). Reichenbach et al. (1964).



(a) Uncoated ball.
(b) TiC-coated ball.

Figure 11.—Representation of lubricated ball-race contact with and without titanium carbide coating on ball. Boving et al. (1987).

a conventional liquid-lubricated bearing. The surface roughness is an arbitrary scale. The steel microstructure corresponds to the indicated magnification and is to scale. The race surface (R_a(cla) ≤ 0.01 μm) is approximately three times rougher than the ball surface (R_a(cla) = 0.03 μm). While the bearing operates in the mixed or boundary lubrication regime, or even when the lubricant film mostly prevents the balls and the races from coming in contact, collisions of the surface peaks cannot be avoided. Such collisions lead to microwelds, which instantly rupture and roughen the surface by material transfer. The resulting heat leads to lubricant breakdown. The magnitude of this breakdown depends on the component surface roughness, the applied load, and the lubricant properties (Boving et al., 1987).

In order to overcome this potential problem, Boving et al. (1981, 1987) coated both balls and races with titanium carbide (TiC), using a chemical vapor deposition process. Boving et al. (1981) tested TiC-coated AISI 440C stainless steel, oil-and-grease-lubricated, 6-mm-bore ABEC 5 ball bearings under a radial load of 1 N and an axial load of 10 N at a speed of 24 000 rpm. Comparisons were made with uncoated bearings operated under the same conditions. The average pretest vibration level of the coated bearings was between 0.20 and 0.25 g, which corresponded to the pretest vibration level of the uncoated bearings. At 24 000 rpm and after several thousand revolutions of operation, the TiC-coated bearings operated more smoothly than the uncoated bearings. The deceleration time of the TiC-coated bearings at different intervals during the experiment, from 100 to 25 000 hr, remained between 100 and 120 sec. After 3000 hr of operation wear tracks could be observed on the races and balls of the uncoated bearings. The lubricant blackened because of wear particles caused by adhesive wear and their reaction to the oil and the grease. However, when the TiC-coated races were examined after 25 000 hr, no traces of wear were observed. The lubricant remained clear.

Boving et al. (1987) tested both TiC-coated and uncoated AFBMA grade 3 AISI 440C balls in uncoated races (fig. 11(b)). The race surface was approximately five times rougher than the ball surface. Gyro spin axis ball bearings with the uncoated steel balls running uninterrupted at 30 000 rpm for 20 000 hr underwent a steady lubricant breakdown with increased noise and vibration levels. McKee (1987) tested TiC-coated balls under identical conditions; there was practically no lubricant breakdown and the noise and vibration levels remained low during the entire operating period.

Lammer et al. (1980) tested TiC-coated balls lubricated with Fomblin Z25 oil (perfluorinated polyether), again with no lubricant deterioration or surface reaction. These tests were run at low rotationless angle movements under high loads and motionless modes for extended times. Therefore, we can conclude that TiC coatings may be able to extend the useful life of liquid-lubricated bearings in a space environment.

Bearing Torque Instability

Cage Instability and Noise

One type of failure observed in space despin mechanical assemblies (DMA) is ball bearing cage (separator) instability. This instability has caused abnormal torque variations and serious pointing errors in the antenna. An associated problem is the maintenance of low electrical noise in the interface between the stationary and rotating elements, such as the slip ring assemblies shown in figure 12. Typical DMA mission requirements have been 1 to 5 years. For missions of these durations it has been possible to conduct real-time life tests to generate the desired level of confidence in achieving the mission goal and to define the operating characteristics of the system. A number of laboratory life tests of these systems have been conducted (Parker, 1987; Ward, 1984; Feuerstein and Forster 1975). In typical DMA systems the angular-contact ball bearings are ABEC 7 specification and have bore sizes ranging from 90 to 150 mm. They generally operate at speeds from 30 to 60 rpm and are oil lubricated. The current trend is for longer missions of 7 to 10 years. Tests have been run for 7 to 10 years with little or no bearing wear and no torque or torque noise problems. Generally, power system failures occur before any lubrication system problems become apparent (Fleischauer and Hilton, 1990).

The ability to control gross bearing torque and bearing torque variations (or noise) is essential to achieving long life and reliability. Gross bearing torque can be controlled by properly selecting the lubricant and the lubricant quantity in the bearing. Bearing torque variations are much more difficult to control. One major source of torque variations is erratic motions of the cage separating the balls. Under ideal conditions the cage simply rotates with the balls. However, under some conditions the cage incurs rapid secondary motion, which is known as cage instability. Cage instability generates erratic friction in a bearing, which is observed as erratic torque. The torque fluctuation measured in figure 9 is an example of a bearing operating in an unstable mode (Kannel and Dufrane, 1986).

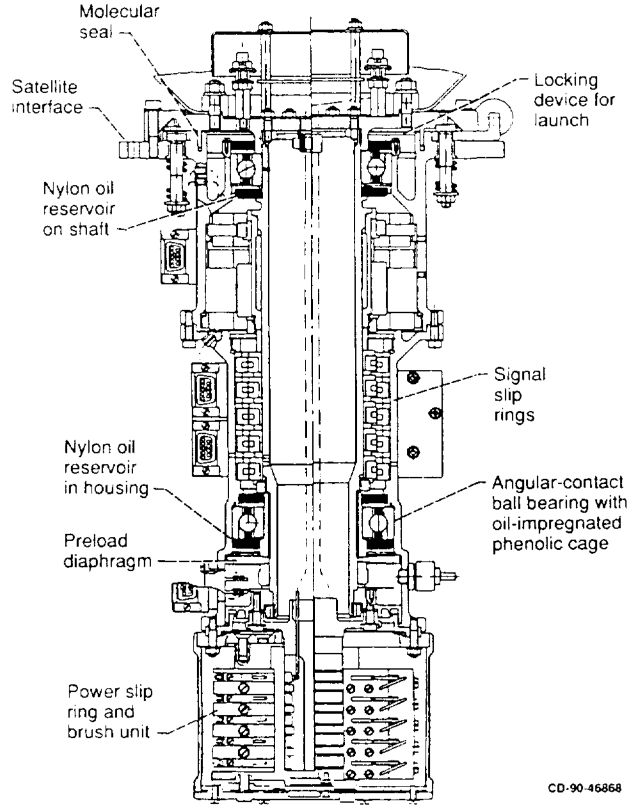


Figure 12.—Schematic of medium-speed (despin) mechanism. Hostenkamp (1980).

The key to cage stability lies in the tangential stiffness of the ball-race interfaces and the rate at which energy can be absorbed at these interfaces. When the cage strikes a ball, the ball will slip and generate a reactive force against the cage. If the slip occurs easily, such as when the balls and races are completely separated by a film of low-viscosity fluid, the reaction force is small. Under such conditions the cage energy is absorbed by shear losses in the lubricant film. Conversely, if the balls and races are separated by a highly viscous fluid, the ball slippage will be small, the shear losses will be much less, and the cage motion will be undamped. Depending on the parameters of the particular bearing, cage instability can be a serious problem (Kannel and Dufrane, 1986).

Cage Stability Criterion

Kannel et al. (1976) developed a cage stability criterion given by the following equation:

$$\mathrm{DP} = \frac{32C_\mu^2}{MC_{sl}}$$

where

DP damping parameter, dimensionless

C_μ ball-race traction parameter, N-sec/m (lb-sec/in.)

M cage mass, N-sec^2/m (lb-sec^2/in.)

C_{sl} ball-cage spring rate (linearized), N/m (lb/in.)

$$C_\mu \approx \mu_{av} \left(\frac{A}{h}\right)$$

where

μ_{av} average contact zone viscosity ($\sim 10^6$ cP for a mineral oil), cP

A contact area, m^2 (in.^{2})

h film thickness, m (in.)

As h gets small or μ_{av} gets large, C_μ gets large.

The larger the value of DP, the more likely that the cage will be unstable. An approximate criterion for cage stability is given in figure 13. To check stability, values of the ball-cage friction coefficient f and a cage restitution factor e_c must be known, where

$$e_c = \exp\left(\frac{-\pi}{\mathrm{DP} - 1}\right)$$

An accurate assessment of cage stability requires analyzing the cage motions with comprehensive computer models.

Cage instabilities reported for DMA bearings have typically occurred in bearings lubricated with a Vackote lubricant (basically Apiezon C). At room temperature this lubricant produces a damping parameter DP greater than unity, implying an instability. The damping parameter can be greatly reduced by using a lower viscosity lubricant, which hypothetically should produce a stable cage (Kannel et al., 1976). In order to evaluate this hypothesis, Kannel et al. (1976) conducted bearing tests with an SAE 10 mineral oil (50 cP), Vackote (192 cP), and Apiezon C (176 cP). The cage for these tests

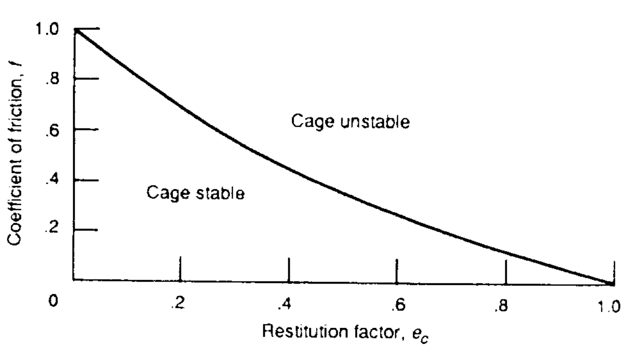


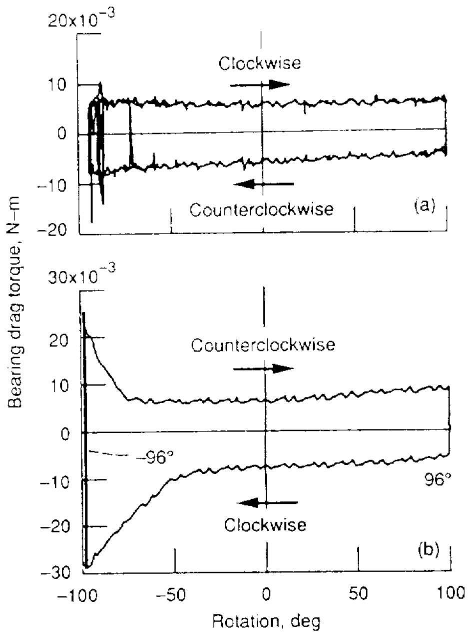
Figure 13.—Cage restitution factor for determining cage stability. Kannel and Dufrane (1986).

was ball guided. The bearings were completely stable with the lower viscosity SAE 10 but were continuously unstable with the other conventional, higher viscosity spacecraft lubricants. Changing the bearing nominal contact angle from 23° to 28° did not affect cage stability. Other tests run by Kannel et al. (1976) with race-guided cages did not result in any instability. Varying the cage-race clearances up to 0.018 mm (0.007 in.) did not produce bearing cage instability. However, cage instability can occur in bearings with race-guided cages. Changing the bearing contact angle as before did not affect cage stability. Kannel et al. (1976) also varied the quantity of lubricant in the bearing. They showed that cage stability can be achieved by increasing the lubricant supply, but overall bearing torque is increased. These tests suggest that bearing torque instability may be controlled by using race-guided cages, by decreasing lubricant viscosity, or by increasing the lubricant supply to the bearing.

Blocking

Loewenthal (1988) describes another form of torque variation experienced in oil-lubricated, oscillating gimbal bearings such as those used in the Hubble Space Telescope high-gain antenna (HGA) drive. The bearings were a 66.7-N (15-lb) preloaded pair of A541 (27-mm bore) size, duplex angular-contact ball bearings mounted face to face. Each bearing contained 24 balls with a diameter of 3.18 mm (0.125 in.), an inner and outer ball-race conformity of 51.8 percent, and a one-piece, inner-land-guided, phenolic-laminated cage. The bearing was lubricated by KG 80 oil. During testing under repeated cycling the gimbal drag torque increased from a nominal 14 N-mm (2 in.-oz) to as high as 127 N-mm (18 in.-oz). This drag approached the stall torque of the drive motor. The torque trace for these bearings is shown in figure 14. The bearing oscillated ±96° at a cycling rate of 0.5 deg/sec. The highest torques only occurred at the end-of-travel reverse point shown at the −96° location in figure 14(b).

Loewenthal (1988) reports that this torque anomaly was observed with six HGA gimbals. The gimbal design was essentially identical to an earlier gimbal that exhibited no such torque anomaly. Todd (1981) had reported the same phenomenon (termed "blocking") in hard-preloaded pairs of ball bearings oscillated over a 90° arc. Loewenthal (1988) discusses the various causes of blocking. Among these are a ball speed variation caused by bearing misalignment: as the balls advance or retard from the average speed they squeeze the cage's ball pockets. In oscillatory bearings the distance errors between the balls and the resulting cage loading or "windup" increase with rotation, reaching a maximum at the end of travel and then decreasing as rotation is reversed. A common approach to reducing this problem is to use alternating toroid ball separators so that the balls can more freely adjust their spacing. The toroidal material can be Teflon or polyimide



(a) Torque characteristics at startup.
(b) Torque characteristics at failure.

Figure 14.—Startup and failure torque characteristics for preloaded pair of 27-mm-bore, duplex angular-contact ball bearings under oscillatory motion. Cycling rate, 0.5 deg/sec; bearing nominal preload, 66.7 N (15 lb); lubricant, KG 80. Loewenthal (1988).

impregnated with oil. Loewenthal (1988) reports that although the toroidal separators reduced the "runaway" torque at the end of travel, drag torque continued to increase over time by 60 percent from startup values.

Todd (1981) suggests that not only misalignment (hence, ball speed variation) but also transverse creep of the spinning ball are necessary ingredients of blocking. "Tight" (51.8 percent) ball-race conformity readily produced blocking because of the sensitivity of ball speed to changes in contact angle. However, under "loose" (57 percent) ball-race conformity, where the ball speed was less sensitive to contact angle, blocking was never observed. A disadvantage of using the loose conformity is that at satellite launching the resultant Hertz (contact) stress on the bearing raceway can exceed acceptable limits. However, Loewenthal (1988) determined that the conformity for the Hubble Space Telescope HGA bearings could be increased from a baseline of 51.8 to 54 percent without exceeding allowable Hertz stress limits.

Lowenthal (1988) performed a transverse creep analysis, considering the elastohydrodynamic film in the contact to behave elastically at low strain rates until the shear stress reached some limiting value. At this limiting value the film was considered to shear like a plastic material. By integrating the local traction forces across the contact, he computed the net traction forces in the direction of rolling as well as in the transverse direction. Loewenthal (1988) concluded that increasing the percentage of conformity reduces both spin torque and drag torque. At the 54-percent conformity the predicted spin torque was 45 percent less and the predicted drag torque was 39 percent less than that with the baseline bearing having a 51.8-percent conformity.

Results of tests run with the baseline bearing and the 54-percent-conformity bearing with and without toroidal separators are shown in figure 15. The measured torque reduction was greater than had been predicted. From these data the blocking phenomenon appears to be the primary result of spin-generated, transverse ball creep, which increases the balls' tendency to climb the bearing raceway shoulder, rather than of ball speed variation (Loewenthal, 1988). But according to M.J. Todd of the European Space Tribology Laboratory, Risley, England (in a letter to this author dated Jan. 15, 1990), if all balls crept equally, there would be no blocking. This would then suggest that both misalignment and transverse creep must be present for blocking to occur.

Raceway Deposits

Phinney et al. (1988) also studied the blocking phenomenon in oscillating duplex gimbal bearing applications. In addition, they investigated torque spikes, another common occurrence. They concluded that torque spikes were caused by debris piling up just before the ends of the ball paths, under repeated cycling, at fixed angles too short to overlap the ball tracks. Debris piles up behind the balls by compaction before a low wave builds up and carries any particles off to the side and back into the running track behind the ball. The debris could only build up when the ball tracks did not overlap. Once the ball tracks overlapped, the debris was redistributed such that it could no longer build up when travel was shortened. Phinney et al. (1988) attribute the source of the debris to particles generated during assembly by press fits of the hardware and the tools used for the installation. They report that bearings are run in a 360° rotation in the gimbal to distribute any assembly debris. This is in addition to the run-in they receive before assembly (Phinney et al., 1988).

There can also be other causes of torque spike. An example is the formation of solid friction polymer from the lubricant, which becomes deposited in the running track and the cage pockets (Hunter et al., 1987). Meeks et al. (1971) discuss the oil polymerization of F-50-lubricated bearings. They suggest an inverse correlation between the theoretical oil-film thickness and the amount of polymer observed. Since the F-50 oil polymerizes (or crosslinks) at approximately 589 K (601 °F), they speculate that this temperature is reached at the microasperity contact points with very low-viscosity oil (fig. 11(a)). A polymer deposit will act like solid debris. Additional debris is generated from the ball pockets and the land-guided faces of the cage. The accumulation of this debris will cause torque fluctuation and can eventually jam the bearing (Harris, 1969).

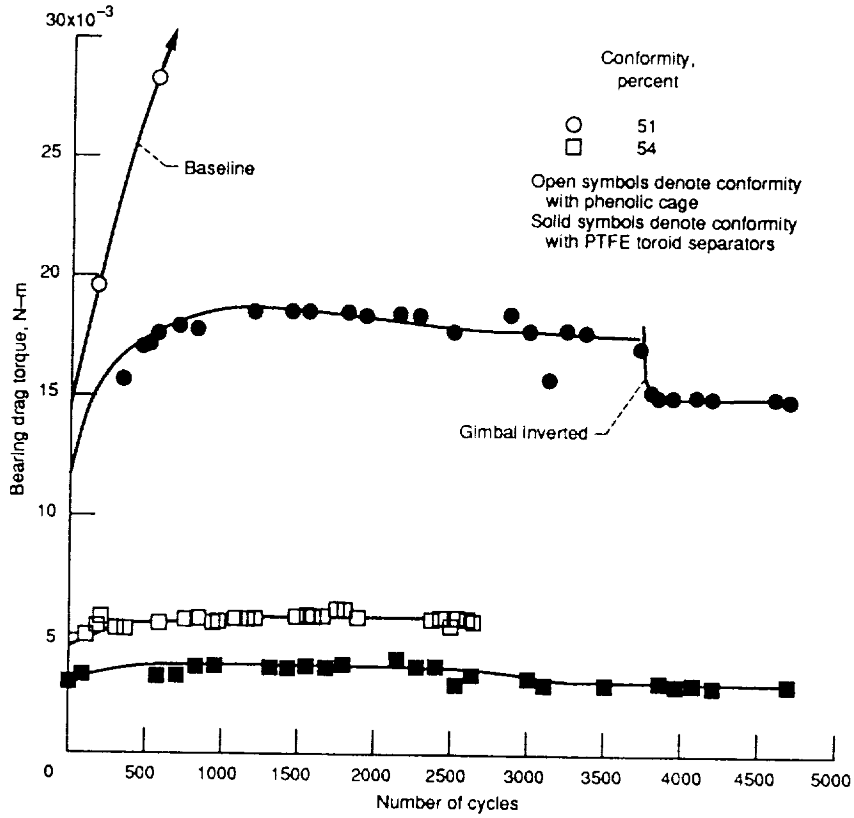


Figure 15.—Bearing torque for preloaded pair of 27-mm-bore, duplex angular-contact bearings under oscillatory motion with combinations of ball-race conformity and separator type. Speed, 0.5 deg/sec; oscillatory motion, 96°; bearing nominal preload, 66.7 N (15 lb); lubricant, KG 80. Loewenthal (1988).

Lubrication Methodology and Supply

Controlled Leakage

Ahlborn et al. (1975) discuss the design criteria for space liquid lubrication systems. They state that, whenever possible, sealed systems should be avoided. Seals, whether static or rotating, offer lower reliability than an open system design because the seals may fail. However, sealed systems are used when considerations such as preventing the contamination of optics or planetary surfaces are important.

If a lubricant film that provides reasonably constant friction force is maintained on moving contacting surfaces, a space mechanism can last for an indefinite time. Therefore, the lubricant evaporated from the contacting surfaces into the space environment must be replenished. Ahlborn et al. (1975) recommend that the lubrication system be designed to lose only 10 percent of the internal lubricant supply to space during the life of the mission. This goal can be achieved by controlling leakage.

The premier work related to controlled leakage was reported by Weinreb (1961) for the bearings in a mechanism for the TIROS II meteorological satellite. The mechanism, shown in figure 16, was a five-channel infrared radiometer that consisted of five optical mirrors mounted on five gears and eight ball bearings driven by a low-power motor whose output torque was 2.12×10^{-4} N-m (0.03 in.-oz). The design was based

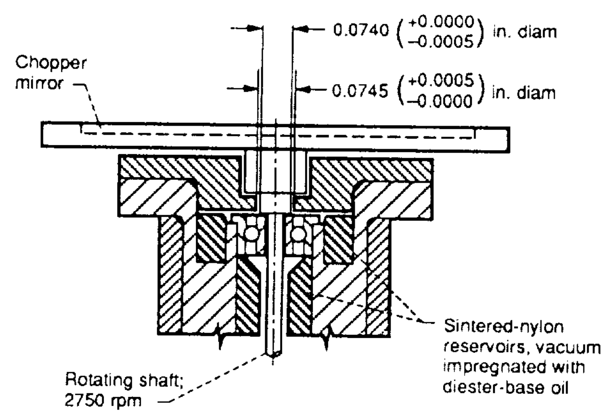


Figure 16.—Radiometer spindle assembly for TIROS II meteorological satellite. Weinreb (1961).

on the fact that, on a molecular scale, even smooth surfaces appear rough, and according to Knudsen (1950) the direction in which a molecule rebounds after a collision with a wall is statistically independent of the angle of incidence. For this reason the molecular flow resistance of small orifices can be made relatively high. The vapor pressure inside the chamber can be maintained and vaporization of the lubricant can be minimized.

The mechanism shown in figure 16 was designed with lubricant reservoirs of oil-impregnated sintered nylon. The lubricant was a MIL-L-6085A diester oil with a vapor pressure of approximately 10^{-4} torr. When the outside pressure was below 10^{-2} torr, a molecular flow occurred around the shaft through the small clearance. The clearance was a nominal 0.0127 mm (0.0005 in.). By using the equation derived by Knudsen (1950), it was possible to calculate the rate at which oil escaped from the bearing assembly. The amount of oil was determined in the reservoirs from the required life of the satellite and the escape rate of the lubricant. A problem with this type of lubricating system is the potential of condensing oil vapor on the optical mirrors.

Silversher (1970) reports a simplified equation, based on a modification of kinematic gas theory, that accounts for the escaping oil vapor loss through an aperture:

$$w = 0.0583\ \pi P\left(\frac{M}{T}\right)^{1/2}\left(\frac{a}{\ell}\right)A$$

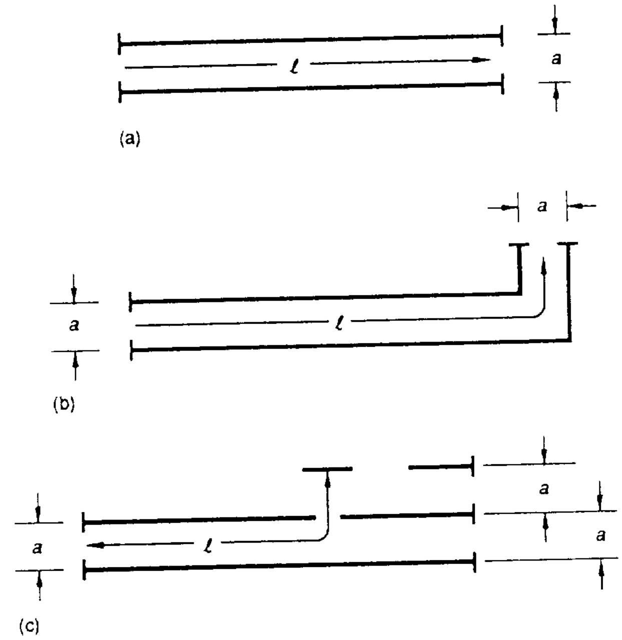
where

- w weight loss through aperture, g/sec
- P vapor pressure of gas, torr
- M molecular weight of oil vapor
- T temperature of oil vapor, K
- a gap width of aperture, cm
- ℓ channel length; or escape path, cm
- A area of aperture, cm^2

This equation is valid if the ratio ℓ/a is 16 or greater.

Several examples of escape paths are shown in figure 17. Controlled-leakage sealing was augmented by using liquid lubricants with low vapor pressure. Silversher (1970) reports that it is sometimes difficult to correlate a high-average-molecular-weight product with a uniform low vapor pressure. Many liquid lubricants contain a wide range of molecular weights. The low-molecular-weight fractions generally volatilize first, so that the rate of fluid loss will not necessarily be constant. This is illustrated in the data of figure 18 (from Ahlborn et al., 1975). The rate of evaporation is greater from the porous nylon reservoirs than from the bearing surfaces or the phenolic cage. As a result Ahlborn et al. (1975) assume that only 30 percent of the total oil in the reservoirs is available to lubricate the bearings or other contacting surfaces.

The most extensively documented use of evaporation as a method for space lubrication was with a medium-speed despin mechanism manufactured by Dornier System GmbH in 1973 (Hostenkamp, 1980). A schematic of the bearing and lubrication system is shown in figure 12. The launch loads bypassed the bearings through an off-load device. The lubrication system for the bearings was a combination of BP 2110 grease with BP 110 base oil (table IV) thinly applied to both balls and races and the BP 110 base oil impregnated in porous phenolic cages and nylon reservoirs. There was approximately 0.25 g of oil per cage (an effective porosity of approximately 2.3 percent). Each reservoir contained



(a) Straight escape path.
(b) Elbow escape path.
(c) T-shaped escape path.

Figure 17.—Measurable escape paths. Silversher (1970).

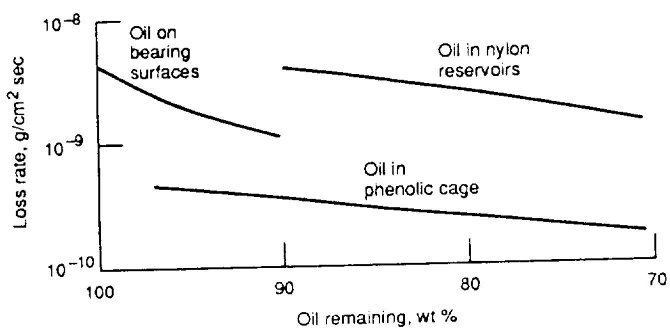


Figure 18.—Oil evaporation rate variations in vacuum at 40 °C. Ahlborn et al. (1975)

approximately 3.3 g (an effective porosity of approximately 25 percent). The reservoirs (two for each bearing) were placed close to each bearing, one on the shaft and one on the housing, to ensure that at least one reservoir was always at a higher temperature than the bearing. Narrow clearance relative to the moving counterpart was provided for the aluminum-backed reservoirs to limit oil loss from the immediate bearing vicinity. Barrier films (Tillan M-2, derived from FX706) were applied to the clearances to aid in this purpose.

A thermal vacuum real-life test of 7 years was successfully conducted by the European Space Tribology Laboratory (Parker, 1987). During the 7-year test period the thermal conditions were changed at 4-week intervals to simulate flight conditions; the mechanism speed was 60 rpm. Earth eclipses were simulated at 6-month intervals. The testing was successfully continued for an eighth year but at normal ambient temperature.

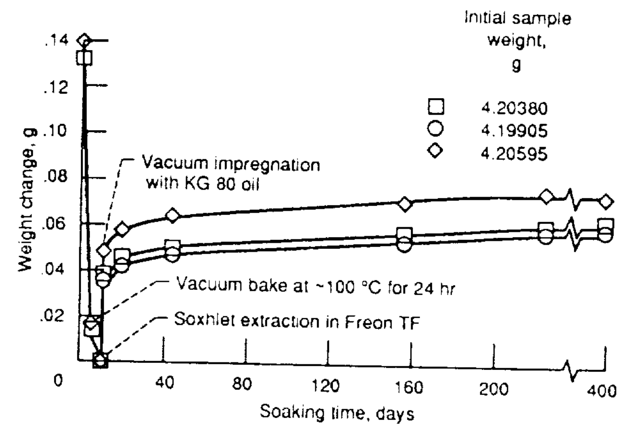
The vacuum pressure took several years to stabilize because of continuous outgassing from the mechanism. This outgassing resulted in the pressure substantially increasing during the warmer test cycles. Residual gas analysis by mass spectrometry suggested that a gradual increase in the relative pressure of hydrocarbons was due to small amounts of oil being released into the test environment from the mechanism (Parker, 1987). During testing the current demanded by the motor to enable the speed control system to maintain a constant speed gradually increased. This was not, however, attributed to an increase in friction torque within the mechanism but to aging of the motor rotor's permanent magnets. The thermal distribution within the mechanism remained substantially constant except during eclipse simulations. Toward the end of these simulations a small increase in motor power dissipation was detected. Changes of thermal gradient across the bearings in the various test modes did not result in any detectable change in mechanism performance. A one-per-revolution ticking noise was present during the early part of the life test. This was attributed to relaxation of the flexible diaphragm used to preload the bearings (Parker, 1987).

Post-test examination of the dismantled mechanism revealed its components to be in excellent condition. The oil-lubricated bearings were virtually like new without any measurable wear. The oil and grease lubrication was highly satisfactory although some oil was lost from the bearings because of the inadequate application of barrier films. The oil had marginally contaminated the power brush and slip ring components without any adverse effect (Duvall, 1985). In a letter to this author dated Nov. 1, 1989, H.M. Briscoe, Consultant in Space Mechanisms and Tribology, Tykes Barn, Great Britain, suggested that the oil film on the slip rings was beneficial because the brush wear was less than expected. He speculated that oil leakage from the bearing chamber may be a reasonable means of lubricating slip rings.

Lubricant-Impregnated Cages

Fleischauer and Hilton (1990) state that the phenolic-based cage materials commonly used in satellite bearings have irreproducible, time-dependent oil absorption properties (fig. 19). These data were obtained from samples that had been vacuum impregnated with oil and then allowed to soak in the oil for the times indicated. From the data it appears that the cage material continuously absorbs oil. This would suggest that the phenolic material acts more like an oil sink than a source.

Tyler et al. (1976) suggest that it is possible for lubricant feed to occur from the phenolic cage to the balls but not from the phenolic cage to the bearing race lands. Layers of cotton material in the phenolic are parallel to the bearing race lands. Lubricant feed, they conclude, cannot occur in a direction perpendicular to these layers. As a result, the lubrication of the interface must depend on the initial oil coating on the two



(a) Vacuum bake at ~ 100 °C for 24 hr.
(b) Soxhlet extraction in Freon TF.
(c) Vacuum impregnation with KG 80 oil.

Figure 19.--Absorption of oil by phenolic retainers as a function of time. Fleischauer and Hilton (1990).

surfaces. Hence, it may be concluded that long-term effective lubrication using phenolic cages as lubricant reservoirs can best be achieved if the cage is ball guided rather than race guided. However, as previously discussed, ball-guided cages are more likely to be unstable (Kannel et al., 1976).

From late 1984 into early 1985 the scanners of the U.S. Earth Radiation Budget Experiment (ERBE) and NOAA-9 satellites, after several months in orbit, began experiencing problems (Watson et al., 1985). A motor torque increase was detected on several of the units. This suggested an impediment to the free rotation of the instruments, which were mounted on a ball bearing assembly. The motor drove the instruments back and forth at 1 rps. The scan angle was 176°. On one occasion the rotation of an instrument stopped completely. The torque increase was attributed to the ball bearing assembly.

The operating conditions of the ball bearing assembly were duplicated in a laboratory bearing endurance test at 10^{-3} torr. The bearing was an AISI 440C, 40-mm-bore, angular-contact ball bearing with 34 balls and a nominal contact angle of 20° to 25°. The preload was 89 N (20 lb). The bearings porous phenolic cage, which was inner-land guided, was saturated with Brayco 815Z, a perfluoroalkylether. The phenolic cage was the only source of liquid lubricant for the bearing. Both the races and the balls were passivated and chemically treated with tricresylphosphate (TCP). The test duration was 8.5 million cycles. The results of these tests revealed ineffective transfer of lubricant from the phenolic cage to the contacting surfaces. Also, the use of TCP to chemically treat the ball and race surfaces resulted in the inability of the PFPE to wet the bearing surfaces (Morales et al., 1986).

Barrier Films

In order to minimize lubricant loss or migration, both from the bearing and the satellite, creep barrier coating films are being used on noncontacting bearing surfaces and on the surfaces of molecular seals. However, normal procedure is to avoid the application of barrier films within the bearing and apply them only to escape gaps. The U.S. Naval Research Laboratory developed a barrier film for instrument bearings. The film, a fluorinated methacrylate, acts as a low-energy barrier across which most lubricants cannot creep. It is stable in air to 423 K (302 °F) and in vacuum to 373 K (212 °F). It outgasses at 373 K (212 °F) at a rate less than 2×10^{-11} g-cm^2/sec. Ahlborn et al. (1975) observe that because there are differences between the various commercial barrier films, they should be evaluated for a specific application before being used. They observe that some lubricant formations appear to "poison" the surface of the coatings and render them ineffective. They state that the mechanical and chemical stability of these films is improved by high-temperature vacuum baking.

Positive Feed Systems

Large spacecraft, starting with Skylab (fig. 1), have brought a new set of tribological problems. Smaller spacecraft could be stabilized by spinning the spacecraft or by using control jets. Large, extended-mission spacecraft require large control moment gyroscopes (CMG's) that are capable of handling the large slew loads resulting from astronauts moving around the craft and from changing the orientation. The large CMG's can be heavily loaded and operate at high speeds. Small bearings using lubricant-impregnated cages may no longer suffice for these units, and positive lubrication of large bearings may be required. The lubrication system has to be highly reliable but extremely compact to meet weight and space requirements (Kannel and Dufrane, 1986).

A major tribological challenge is ensuring that the lubricant gets into the bearing to lubricate the ball-race interfaces. Devices such as centrifugal oilers attached to the rotating shaft have been successfully used in some applications. Another method (Glassow, 1976) uses a single-stroke pump immersed in a vented oil reservoir to pump the lubricant to the bearing. James (1977) proposes the entirely different approach shown in figure 20. In this system, a positive commandable lubricator, the reservoir is sealed to prevent lubricant contamination. The degassed lubricant supply is stored in a flexible metal bellows. Pressure is maintained by an external spring pack. Opening the release valve permits oil to inflate an adjustable-stroke metering bellows. Subsequent closing of the release valve and opening of the metering valve starts the flow to the applicator. Metering pressure is sufficient to overcome the characteristic backpressure of the applicator and to provide the desired flow rate. A metering orifice provides flow rate adjustment capability. The applicator is supported rigidly in the space between the outer race and the inner-race-guided retainer. In a 110-mm-bore bearing with 12.7-mm (0.5-in.) diameter balls the space is approximately 3.81 mm (0.15 in.) wide. A standoff distance, typically five times the expected launch-induced axial ball movement, separates the applicator tip from the ball path. A combination of toroidal tip shape and Teflon coating enables the applicator to support an oil droplet that spans the standoff distance. In the operating bearing the passing balls wipe off a portion of the distended hemispherical droplet. In this manner oil is slowly and uniformly transferred from the balls to the retainer ball pockets and to both races. The droplet is continuously replenished by flow from the metering system during the 2- to 4-min relubrication cycle. The positive commandable lubricator is designed for high-surface-energy oils such as Apiezon C. The high surface energy of hydrocarbon oils provides the stabilizing force for the lubricant droplet on the applicator tip. The system is best suited for large bearings, where there is sufficient space between the retainer dynamic envelope and the race to accept the applicator (James, 1977).

NASA research (Loewenthal et al., 1985) with a terrestrial experimental 46-cm (19-in.) diameter, 58-kg (128-lb) flywheel showed the feasibility of using a wick lubrication system in a vacuum environment to lubricate moderate-speed bearings. The flywheel with its lubrication system is shown in figure 21. In this system a lightly spring-loaded wick saturated with oil contacts a conical sleeve adjacent to the bearing inner race. Frictional contact against the sleeve causes a small amount of oil to be deposited. This oil migrates along the sleeve to

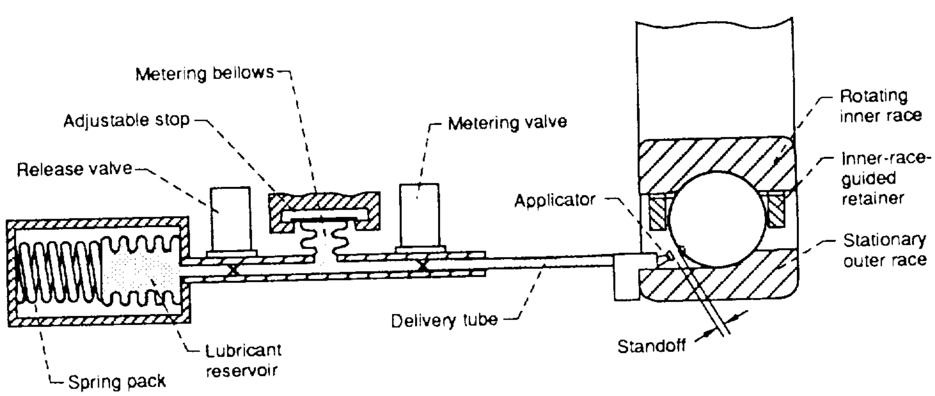


Figure 20.—Positive commandable lubricator for satellite bearing application. James (1977).

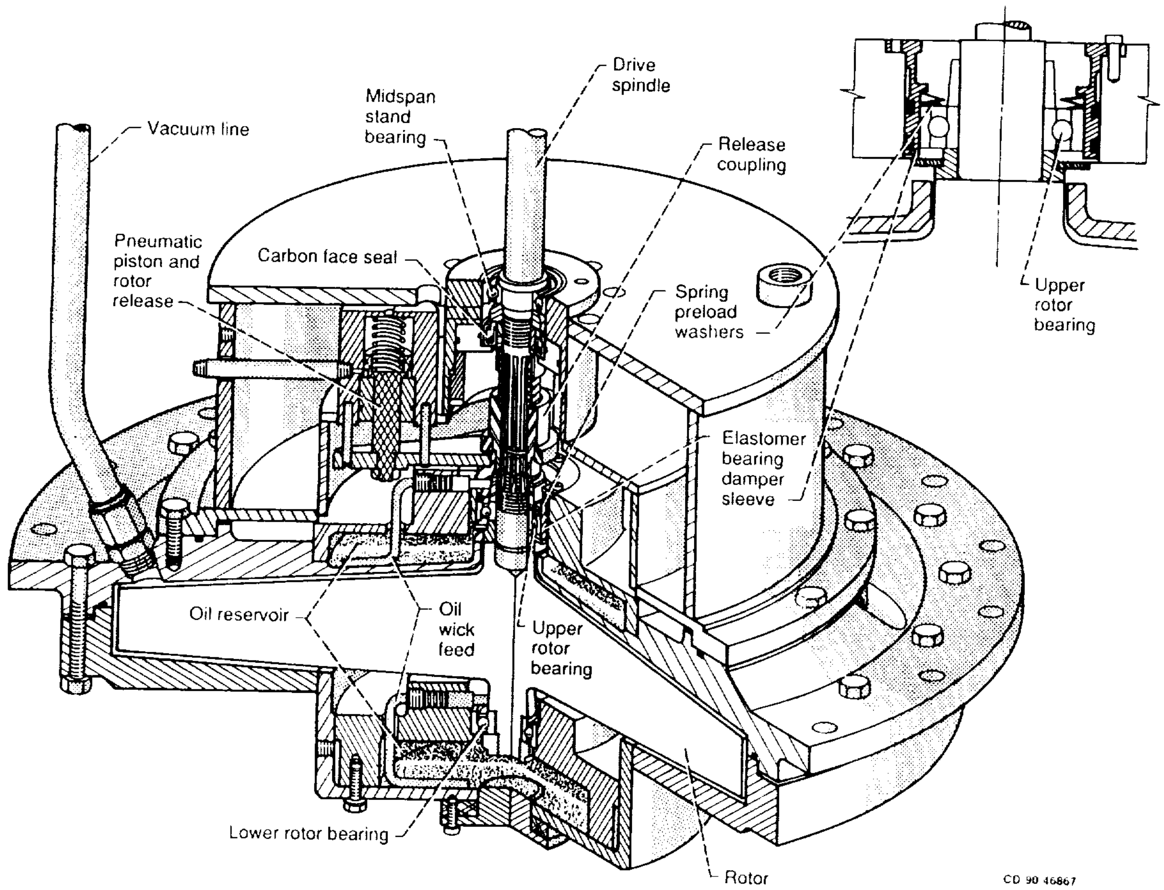


Figure 21.—Schematic of NASA flywheel rotor module. Loewenthal et al. (1985).

its large end and into the bearing under the centrifugal force field. The wick absorbs oil from the reservoir; the oil travels through the wick by capillary action to replenish the oil entering the bearing. In a terrestrial system, because of gravity, the oil droplets leaving the bearing and the oil vapor that coalesces against the cooler housing walls eventually return to the reservoir, thus closing the cycle. Whether condensation and reuse of the oil can be accomplished in a space system is an open issue.

The rotor system was operated at speeds from 10 000 to 20 000 rpm at absolute housing pressures from 195 down to 45 millitorr. The peak rated power of the flywheel, 120 kW at 20 000 rpm, corresponds to a maximum discharge torque of 57 N-m (504 in.-lb). The bearing outer-race temperature profiles are shown in figure 22. These data show that the bearings will operate at lower temperatures when the housing pressure is low: the aerodynamic heating of the rotor is less at lower pressure. The data confirm that the wick feed system provides adequate cooling, particularly at the lower vacuum levels (Loewenthal et al., 1985).

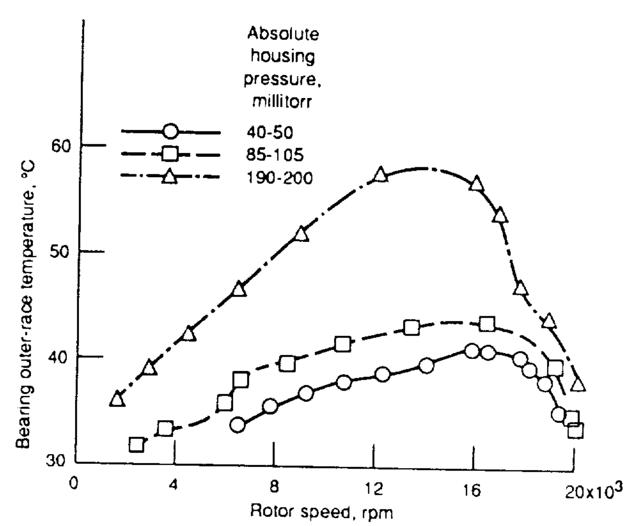


Figure 22.—Flywheel lower support bearing temperature characteristics with speed and housing pressure. Bearing type, 20-mm-bore, deep-groove ball bearing; thrust load, 712 N (160 lb). Loewenthal et al. (1985).

Concluding Remarks

A search of the literature revealed that spacecraft lubrication systems have, for the most part, performed adequately over the past three decades. That is, spacecraft and satellites have reached their required lifetimes without a lubrication-related failure. However, this success was achieved because the life of satellite systems is generally limited by their batteries, electronics, and thermal and optical systems. However, as these technologies are improved, spacecraft systems will probably become limited by the lubrication system.

Problems related to spin-stabilized satellites involve the despin mechanical assemblies and the slip ring assemblies, which are required to operate with low torque ripple and low electrical noise. Tests have verified current designs, and in many satellites power system failures have occurred before lubrication failures have surfaced. However, the ability to control gross bearing torque and noise is essential to achieving high reliability and long life.

Spin bearings for control moment gyros, operating at speeds up to 12 000 rpm, have been lubricated successfully for current design lives with oil-impregnated phenolic cages. Future systems will require much longer lives as well as higher load capabilities.

Gimbal bearings, which operate in an oscillatory mode and rarely make a full revolution, are a major concern because of the blocking phenomenon and debris buildup discussed herein. Careful consideration must be given to the design and use of these bearings in order to achieve long lifetimes.

Boundary lubrication with antiwear and extreme-pressure additives in the lubricant in a satellite (vacuum) environment is not well defined. There is an issue as to whether these additives are at all effective in the absence of oxygen. The role of additives needs to be defined for these applications.

The long-term effects of atomic oxygen and other space radiation on lubricant behavior and stability remain unknown. The effects can be important for space missions longer than 10 years.

Alternative cage materials with lower wear rates need to be considered as replacements for phenolic-base materials, especially in oscillating bearings, in order to inhibit the buildup of wear debris and the potential jamming of the bearing.

Alternative lubrication systems providing for positive lubricant feed to the contacting surfaces may need to be considered for future space missions. In addition, alternative lubricants need to be seriously examined as replacements for the perfluorinated polyalkylethers (PFPE) for long-term space missions.

In the fourth decade of the space age tribologists and aerospace design engineers face continuing challenges to achieve longer lifetimes with liquid lubrication in space.

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16. Abstract

The requirement for long-term, reliable operation of aerospace mechanisms has, with a few exceptions, pushed the state of the art in tribology. Space mission life requirements in the early 1960's were generally 6 months to a year. The proposed U.S. space station scheduled to be launched in the 1990's must be continuously usable for 10 to 20 years. Liquid lubrication systems are generally used for mission life requirements longer than a year. Although most spacecraft or satellites have reached their required lifetimes without a lubrication-related failure, the application of liquid lubricants in the space environment presents unique challenges. This report reviews the state of the art of liquid lubrication in space as well as the problems and their solutions.

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