

**NASA**  
**SPACE VEHICLE**  
**DESIGN CRITERIA**  
**(STRUCTURES)**

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# **LUBRICATION, FRICTION, AND WEAR**

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**NATIONAL AERONAUTICS AND SPACE ADMINISTRATION**

## FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technology:

Environment  
Structures  
Guidance and Control  
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. A list of all published monographs in this series can be found at the end of this document.

These monographs are to be regarded as *guides* to the formulation of design requirements and specifications by NASA Centers and project offices.

This monograph was prepared under the cognizance of the Langley Research Center. The Task Manager was J. R. Hall. The author was J. R. Jones of Hughes Aircraft Company. A number of other individuals assisted in developing the material and reviewing the drafts. In particular, the significant contributions made by L. S. Akin of General Electric Company; A. J. Babecki and C. A. Vest, NASA Goddard Space Flight Center; F. De Laat of TRW Systems Group/TRW Inc.; K. Demorest of NASA George C. Marshall Space Flight Center; H. C. Gatos of Massachusetts Institute of Technology; J. H. Kimzey, NASA Manned Spacecraft Center; P. M. Ku of Southwest Research Institute; J. W. McKenzie, G. Walker, and C. S. Wilkinson of McDonnell Douglas Corporation; A. B. Sorkin of Jet Propulsion Laboratory, California Institute of Technology; H. I. Silversher of Lockheed Missiles & Space Company; J. W. Van Wyk of The Boeing Company; and F. J. Williams of North American Rockwell Corporation are hereby acknowledged.

NASA plans to update this monograph when need is established. Comments and recommended changes in the technical content are invited and should be forwarded to the attention of the Design Criteria Office, Langley Research Center, Hampton, Virginia 23365.

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## GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to provide a uniform basis for design of flightworthy structure. It summarizes for use in space vehicle development the significant experience and knowledge accumulated in research, development, and operational programs to date. It can be used to improve consistency in design, efficiency of the design effort, and confidence in the structure. All monographs in this series employ the same basic format – three major sections preceded by a brief INTRODUCTION, Section 1, and complemented by a list of REFERENCES.

The STATE OF THE ART, Section 2, reviews and assesses current design practices and identifies important aspects of the present state of technology. Selected references are cited to supply supporting information. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the CRITERIA and RECOMMENDED PRACTICES.

The CRITERIA, Section 3, state *what* rules, guides, or limitations must be imposed to ensure flightworthiness. The criteria can serve as a checklist for guiding a design or assessing its adequacy.

The RECOMMENDED PRACTICES, Section 4, state *how* to satisfy the criteria. Whenever possible, the best procedure is described; when this cannot be done, appropriate references are suggested. These practices, in conjunction with the criteria, provide guidance to the formulation of requirements for vehicle design and evaluation.

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# LUBRICATION, FRICTION, AND WEAR

## 1. INTRODUCTION

Whenever two contacting bodies slide, roll, or separate with respect to each other, a force referred to as *friction* is produced at their interface which opposes their movement. This friction force is usually accompanied by *wear*, the removal of material from either or both of the contacting surfaces. Both friction and wear may be aggravated by contamination, corrosion, or unusual environmental conditions, but both are minimized by the process of *lubrication*.

Improperly lubricated *bearings* (defined as contacting surfaces which experience relative motion, including merely separation) and incomplete or faulty analysis of friction and wear of bearing surfaces can affect the flightworthiness of a space vehicle by leading to destructive wear, bearing failures, and loss of operation, and thus ultimate jeopardy of the mission. For example, use of too volatile an oil in the ball bearings of a de-spun antenna could result in significant viscosity increase (by evaporation) and either total loss of operation at low temperatures or excessive power demands on the drive system. Failure to include corrosion-preventive measures during installation could result in rusting of the same bearings, thus producing abrasive wear and pitting of the races, with attendant loss of preload, and premature failure.

Examples of lubrication failures in space vehicles are presented in table I in the Appendix.

The space environment imposes several conditions that can drastically affect lubricants and the surfaces of bearings, including such conditions as:

- Extremely low pressure
- Lack of oxygen and water vapor
- Weightlessness
- Temperature extremes

- Electromagnetic radiation
- Unusual molecular species

By contamination, lubricants may adversely affect the physical and mechanical properties of other space-vehicle materials, such as elastomers, sealants, plastics, oxidizers, and fuels. Lubricants may modify electrical and optical properties of surfaces; they may also cause serious problems by their flammability or toxicity.

This monograph provides criteria for the design of the lubrication systems of bearings operating in space-vehicle mechanisms. The related processes of friction and wear are discussed, particularly as they are affected by the space environment. Various types of lubricants, self-lubricating materials, and lubrication systems for space vehicles are compared. In addition, recommendations are given for methods of solving lubrication problems.

Guides are presented for analyzing friction and wear and providing all mechanical devices with lubrication systems which do not at any stage compromise the functioning of other devices, will not be adversely affected by the range of operating conditions and environments, and which permit close control to be maintained over dimensional tolerances so that structural and mechanical integrity can be ensured.

The guides for analysis and subsequent calculations include the effects of the following on bearing surfaces and lubricants: (1) manufacturing processes, (2) assembly and handling procedures, (3) storage environments, (4) operation during test, (5) corrosive atmospheres in the launch area, (6) shock loads at launch and separation stages, and (7) the entire mission profile, including entry.

This monograph is complemented by other design criteria monographs being prepared on design-development testing, qualification testing, acceptance testing, and nuclear and space radiation effects on materials.

## **2. STATE OF THE ART**

### **2.1 Friction**

Friction may be defined as the sum of the forces acting at a bearing surface, in a direction opposing relative motion. The coefficient of friction is defined as the total friction force divided by the normal load.

### 2.1.1 Sliding Friction

As viewed through a microscope, all machined metal surfaces appear rough. The individual roughnesses are called asperities, and are of varying height and depth. Thus, when two bearing surfaces are brought into intimate contact, they actually “touch” at an extremely small number of points. In other words, their true area of contact is an extremely small fraction of their apparent contacting area, varying typically from approximately 0.005 to 0.00001 (ref. 1). Because of this microtopography in even the smoothest appearing metal surfaces, actual bearing loads reach the metal’s flow pressure (ref. 2).

When exceptionally clean metal surfaces come together under load, those asperities which actually touch can weld together, forming bonds or junctions which may be as strong as the base metal. When even minute relative motion occurs, or if load is increased, these welds or bonds tend to increase in area. This is called junction growth, and it greatly increases the force of friction because these bonds must be sheared to permit motion (refs. 3 and 4). In addition, the number of asperity contacts increases due to the plastic deformation of some asperities and the wearing away of others (ref. 5). This general process of junction formation and growth has also been termed adhesive friction, or “cold welding” (refs. 6 to 8). Not only identical alloys but all metals which are mutually soluble, or which form a continuous series of alloys, are prone to weld (refs. 9 and 10). The presence of oxide film or other contaminants can minimize the number and size of welded junctions. Therefore, the junction-forming process is vastly accelerated in high vacuum, where oxide and other protective films, once removed, cannot be regenerated.

When one bearing metal is considerably harder than the other (e.g., chrome-plated steel vs leaded brass) or when microscopic portions of the surface become work-hardened, these hard metals can cut or plow through the softer ones, and this plowing contributes to the total friction force (ref. 2). A similar frictional effect can be caused by abrasive contaminants (sand, dirt, etc.) and wear particles.

A much smaller force, by comparison, is required for metals to shear or slide over any film separating the two bearing surfaces. Such films include soft oxides, lubricants, and all forms of nonabrasive contaminants.

The effect of temperature on friction is ordinarily complicated by opposing factors. The increasing ductility of metals (with increasing temperature) facilitates junction growth and seizure (refs. 11 and 12). The rate of formation of oxides, however, also increases with increasing temperature (in air), resulting in a transition in the coefficient of friction ( $f_k$ ) at a critical value of oxide thickness (ref. 13). The effect of oxide formation is of course negligible in high vacuum.

The total friction force in any sliding state is equal to the sum of the forces required to (1) shear the metallic junctions, (2) plow through the softer metal, and (3) shear the surface films and/or lubricant (ref. 2). The problem of overcoming friction in space is the problem of controlling operating conditions, bearing materials, surface hardness and roughness, cleanliness, and lubricants. Ideally, then, the friction force may be expressed as:

$$F = AS_1 \quad (1)$$

in which  $A$  is the geometric area and  $S_1$  is the shear strength of the lubricant. In brief, friction is minimized by preventing any form of metallic contact.

### 2.1.2 Rolling Friction

Rolling friction is the condition where combined forces resist the motion of one body rolling over another. The term is commonly restricted to exceptionally smooth bodies of nearly perfect geometry. In such cases, coefficients of friction are usually quite low—on the order of  $10^{-5}$  to  $5 \times 10^{-3}$ —and the component due to surface roughness is particularly low (ref. 14). The total rolling-friction force is a composite of the following factors:

- Interfacial slip in the area of contact. This is attributable to elastic deformation of the contacting bodies; all points in the contact zone do not lie in the same plane (ref. 15). Interfacial slip is simply a form of sliding friction between rolling elements and the opposing surface.
- Elastic hysteresis losses. The energy absorbed by the elastic bodies is not entirely released back to the system when the stress is removed (refs. 16 to 20).
- Miscellaneous factors such as: (1) lack of geometric perfection; (2) presence of contaminants; (3) plastic deformation of surface asperities; and (4) the work done to create a “free” surface during rolling (ref. 14).

In a component such as a ball or a roller bearing, many other factors contribute to a composite coefficient of friction which may be considerably higher than  $10^{-3}$ . These are: (1) sliding between the rolling elements and the separator pockets; (2) sliding between the separator and the surface of the locating race; (3) shearing of oil or grease films between bearing elements; and (4) churning losses caused by excess lubricant within the bearing (refs. 14 and 15).

### 2.1.3 Range of Friction Values

Figure 1 presents the range of coefficients of friction for a wide variety of materials, conditions, and mechanisms.

## 2.2 Wear

Wear may be defined as the removal of material from a surface in bearing under dynamic conditions (sliding, rolling, and fretting). Wear is generally divided into five major categories: (1) adhesion, (2) abrasion, (3) surface fatigue, (4) erosion, and (5) corrosion.

### 2.2.1 Types of Wear

#### 2.2.1.1 Adhesion

Adhesion is the result of the direct contact of bearing metals. When the applied load is sufficient to rupture any protective surface film (oxides, etc.), the contacting asperities

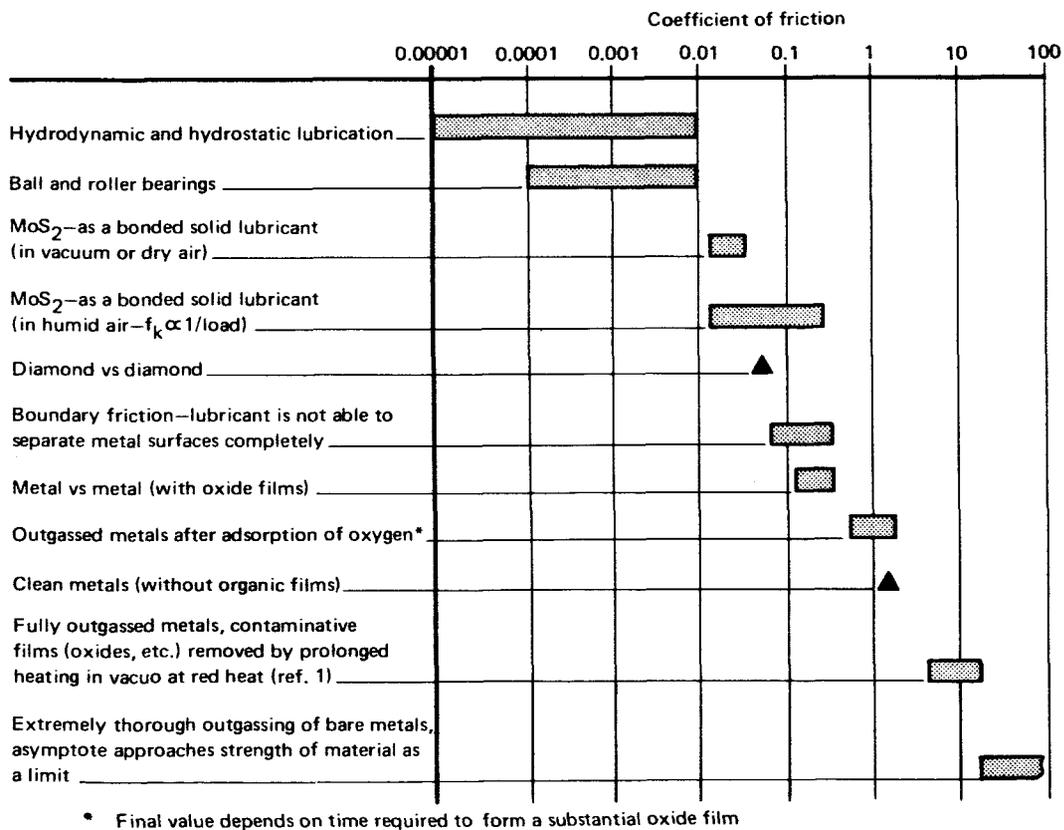


Figure 1.—Typical range of friction values.

deform elastically, then plastically. Welding of these asperities may occur on contact, but occurs more readily when relative motion takes place. The shearing of these adhesive junctions (weldments) produces wear particles (ref. 21); it is through this process also that metal is transferred from one bearing surface to the other in sliding or rolling motion (refs. 22 to 24).

When adhesive-wear damage is severe, it is referred to as scuffing. If frictional heating causes decomposition and/or desorption of protective films from the surface, the process can become destructive (refs. 25 and 26).

#### **2.2.1.2 Abrasion**

Abrasion results whenever a hard material slides against a soft one, and may be visualized best as plowing. The most common form of abrasion results when hard particles are interposed between two metal surfaces in bearing. The wear particles generated in both adhesive and in abrasive wear are deleterious because they tend to perpetuate these wear processes and they create points of localized high stress that may mark the onset of fatigue (ref. 27).

#### **2.2.1.3 Surface Fatigue**

When bearing surfaces contact each other in rolling motion, after a relatively large number of cycles the result may be surface fatigue, a condition evidenced by localized pitting or flaking. The time to onset of this form of wear is highly dependent on the stress. Local stresses are vastly increased by particulate matter; thus, it is important to minimize wear debris generated from hard-surface bearings (ref. 5). In most space applications, surface fatigue is not yet considered a serious problem because loads are light.

In future applications the number of cycles required is expected to increase drastically, and loads may be much heavier than at present. Considering the extreme difficulty of parts replacement, then, surface fatigue may become a very serious problem.

#### **2.2.1.4 Erosion**

Erosion is the removal of surface material by the impingement of fluids and/or solids, as in liquid-abrasive blasting (ref. 14, p. 119). Erosion is occasionally observed in bearings with hydrodynamic lubrication (full oil flow), particularly when filtration is not adequate to remove solid contaminants (ref. 28). One comparable condition on a space vehicle would be the damage caused by the impingement of tiny meteoroids, or space dust, a form of damage which can be minimized by shielding.

Future spacecraft are likely to encounter more conventional forms of erosion, for example in tubing walls, valve seats, baffles, or nozzles due to the entrainment of solid contaminants by fuels, oxidizers, breathing gases, and heat exchanger fluids.

### **2.2.1.5 Corrosion**

Corrosive wear is the removal of surface material by chemical action, or by a combination of chemical action and relative motion (refs. 14, pp. 186 to 190; and 28, chs. 4 to 6). The rate of corrosive wear may be increased by increasing the load. Of primary concern in space vehicles, with respect to corrosion, are (1) the inadvertent trapping of some corrosive substance within a bearing, housing, or reservoir; (2) unnoticed corrosion that occurs prior to launch, usually during storage or shipment; and (3) fretting corrosion, the wear resulting from low-amplitude oscillation in the presence of oxygen. The iron oxide formed from the presence of oxygen subsequently causes abrasive wear. Rust, the most common form of corrosion, may be aggravated in earth operations by rolling or sliding motions in air if the lubricant does not contain suitable additives or if the mating surfaces have not been properly selected (ref. 15, pp. 16 and 17).

### **2.2.2 General Process of Wear**

Wear takes place even in properly lubricated mechanisms (ball bearings, gears, bushings, cams, etc.) according to a definite pattern that is shown generalized in figure 2. The initial-wear rate (run-in) is relatively high because the microscopic surface asperities penetrate the lubricant film, particularly during the low speeds of start-up and shut-down. This rate is aggravated by oscillating motion and by increasing loads. If conditions are not sufficiently severe to cause scuffing in this stage, the wear rate is termed "mild" (ref. 15, pp. 16 and 17; and ref. 29).

Normal wear begins when the true area of bearing contact (total asperity-contact area) has been substantially increased by plastic deformation and wear to the extent that the lubricant is fully able to support the load (ref. 30); there will then be only occasional metal-to-metal contacts. From this point on, wear occurs at a negligible rate, modified only by such events as lubricant breakdown, sudden rises in temperature (which reduce oil viscosity), shock loads, or a significant reduction in speed. Normal wear may proceed indefinitely, depending on the several controlling variables, until sufficient debris has accumulated to cause one of the following failure modes:

- Stress risers in the path of motion, ultimately initiating fatigue
- Abrasive wear sufficient to change surface roughness or dimensions appreciably

- Physical blockage (e.g., in gear-teeth roots)

When severe wear or scuffing sets in, the mechanism is so close to failure that the rates of wear are only of academic interest.

The foregoing discussion is only partially applicable to bonded solid lubricants (Sec. 2.4.3).

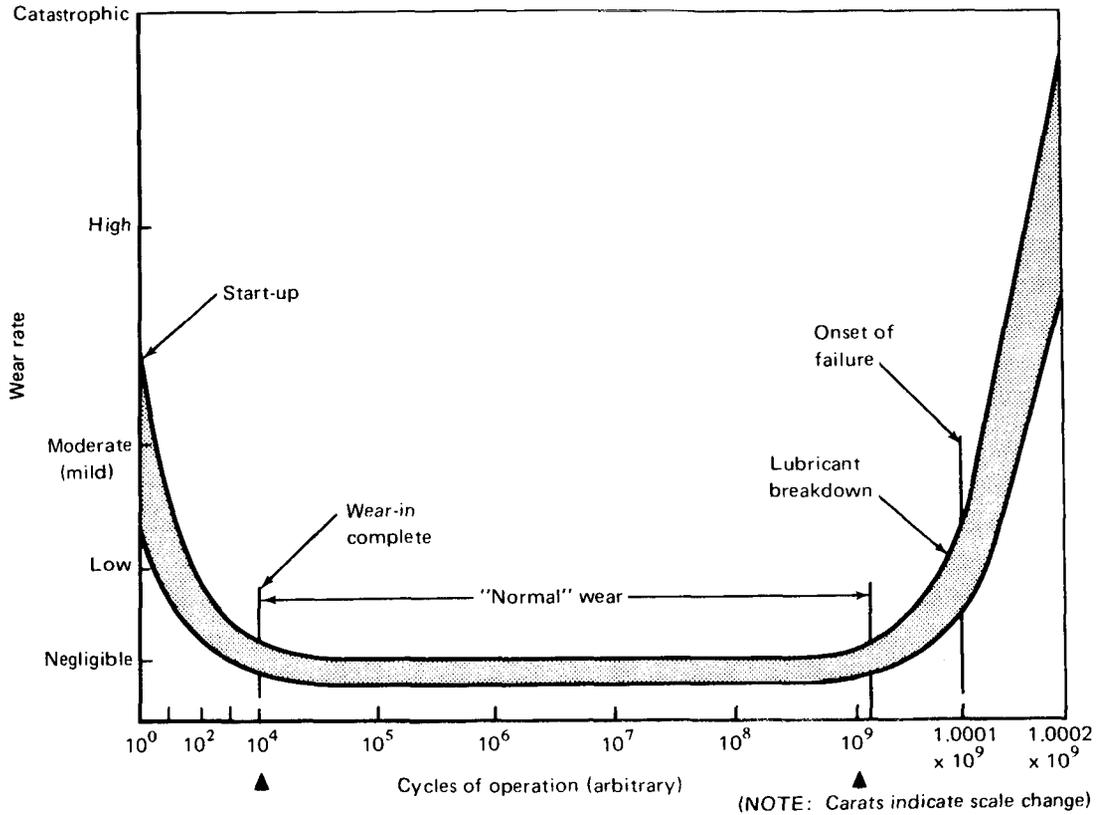


Figure 2.—Generalized pattern of the wear process.

### 2.2.3 Wear Rates

The areas of practical interest in determining rates of wear are the mild-wear regime (run-in) and the normal-wear regime. The wear-rate formula given below is generally applicable to dry conditions, boundary lubrication, mixed lubrication, or partial elastohydrodynamic lubrication. (See Sec. 2.3.3.)

$$V = \frac{KWL}{P} \quad (2)$$

in which  $V$  is the wear volume;  $W$ , the load;  $L$ , the sliding distance;  $P$ , the indentation hardness of the softer material; and  $K$ , a constant for the lubricant and system involved. This formula predicts the wear rate of the softer of two bearing materials. The wear rate  $V_2$  of the harder material is given by:

$$V_2 = \left(\frac{P}{P_2}\right)^2 V \quad (3)$$

in which  $P_2$  is the indentation hardness of the harder material. These formulas do not predict the effect of the wear particles on the given system; they merely determine the amount of wear which will be formed (refs. 31 and 32).

Whether wear will be mild or severe is determined by the type of deformation which the asperities undergo during contact and shear. The actual contact area is closely proportional to the load, but the question of elastic or plastic deformation has been found to be more directly related to material properties and surface topography than to load (refs. 33 and 34). This is illustrated in the following equation:

$$\psi = (E'/H) \sqrt{(\sigma/R)} \quad (4)$$

where  $\psi$  is defined as a plasticity index depending on  $E'$  (an elastic constant);  $H$  is hardness;  $\sigma$  the standard deviation of the asperity heights; and  $R$ , the average radius of the asperity tips.

It has been found that for  $\psi$  values of 0.6 or lower, the deformation is essentially elastic, while values above 1.0 indicate plastic deformation. A borderline range exists between the two values.

## 2.3 Lubrication

Lubrication is defined as the process by which any foreign substance is interposed between contacting surfaces undergoing relative motion. These substances include adsorbed layers of oxygen, oxide films (including rust), contaminants, dirt, oil, grease, various chemical-conversion coatings (sulfides, phosphates, etc.), and numerous other solids. Many of these materials, however, would not normally be chosen as lubricants because of their abrasiveness, because they cannot be controlled during the manufacturing process, and because of their unpredictable friction and wear life. Nevertheless, extremely thin oxide films, as distinguished from rust or other more severe forms of corrosion, have occasionally been found to be highly beneficial in the operation of lubricated ball bearings (refs. 15 and 20), even though the replenishment rate of these films is quite slow in the vacuum of space.

### **2.3.1 Hydrodynamic**

Although hydrodynamic lubrication is in extremely common use in machinery on earth where it is possible to assure a full, constant supply of oil to the inlet of the contact zone, it is not frequently used in space.

Hydrodynamic, or thick-film lubrication, utilizes a physical property (viscosity) of fluids and semifluids to develop a wedge-shaped film between bearing metals in sliding motion, completely separating them. The film is formed at some minimum sliding velocity, depending on applied load and lubricant viscosity. Thus, it is ineffective during start-up, shut-down, and reversal of motion (ref. 15, pp. 63 to 95; and ref. 35). The exception is when external lubricant pressure is supplied directly to the bearing, as in the case of hydrostatic bearings (ref. 28, pp. 3-1 to 3-10). The oil supply may be provided by gravity feed from a reservoir or by various types of pumps.

Hydrodynamic lubrication offers four desirable characteristics: (1) it completely separates the bearing surfaces at full-speed conditions; (2) it has a cooling capacity to maintain operating temperatures at low levels; (3) it provides very low friction; and (4) it washes away wear particles, thus maintaining cleaner bearings. However, the disadvantages listed below generally far outweigh these advantages in space applications:

- Severe weight penalty (extra equipment required)
- Not effective at low speeds
- Complexity of mechanisms
- Gravity cannot be used to bring fluid into the bearing area
- Potential contamination of adjacent systems by evaporation or migration

### **2.3.2 Elastohydrodynamic**

Elastohydrodynamic (EHD) lubrication refers to the oil films formed between rolling bearing elements. Examples are the contact-area films formed in ball and roller bearings and spur gears with involute-tooth form (refs. 15 and 36). EHD lubrication is actually an extremely special case of fluid film formation because of (1) the elastic deformation of metals in contact under load and (2) the effect of pressure on oil viscosity. Depending on the speed, the load, the lubricant viscosity under pressure, and the surface roughness of the bearing elements, a film can be developed that completely

separates the rolling parts. Increasing viscosity and speed cause the effects shown in the following equation:

$$\xi \propto (N\nu)^n \quad (\text{ref. 16}) \quad (5)$$

where  $\xi$  is a dimensionless parameter;  $N$ , the speed in rpm; and  $\nu$ , the lubricant viscosity. The exponent  $n$  depends on the geometry of the system. The parameter  $\xi$  is related to film thickness and surface roughness by the following equation:

$$\xi = \frac{h_0}{\sqrt{\sigma_1^2 + \sigma_2^2}} \quad (\text{ref. 37}) \quad (6)$$

where  $h_0$  is the film thickness, and  $\sigma_1$  and  $\sigma_2$  are the surface-roughness values of the mating parts. Obviously, at higher values of  $\xi$ , less wear can take place. In general, increasing the load decreases film thickness and permits more metallic contact, thereby increasing wear.

Thus, it is important to determine not only the evaporation rate of the lubricant, but an accurate value of  $\xi$  or  $h_0$  from equations in references 16 and 37. The film thickness and  $\xi$  itself can be checked experimentally by measurements of electrical resistance through the bearings or by utilizing X-rays and other special equipment (refs. 36 and 38). It is possible to select a liquid lubricant that will permit antifriction (ball and roller) bearings and spur gears to function in space mechanisms for long periods of time if the following variables can be controlled:

- Temperature range (for viscosity determination)
- Load
- Speed
- Surface roughness of balls and races (or gear teeth)

The film thickness  $h_0$  is generally considered to be adequate to prevent metallic contact at 4 microinches (0.1016 microns). It is borderline at 3 microinches (0.0762 microns) and inadequate at 2 microinches (0.0508 micron) or less. (See table II in the Appendix for conversion from U.S. customary units to SI units.)

Very low speed ball-bearing operation, in which EHD films cannot be established, has been significantly improved by treating the balls and races with commercial tricresyl phosphate (ref. 39). The process has been applied to both 52100 steel and 440-C

corrosion resistant steel. The life of ball bearings operating at 1 rpm has been increased from approximately two to hundreds of hours.

### 2.3.3 Boundary

Boundary lubrication is defined as a process in which (1) the friction and wear characteristics are determined by physiochemical interactions between the lubricant and the bearing surfaces, and (2) the lubricant is unable to separate the opposing surfaces completely (ref. 28, ch. 2).

Boundary lubrication is distinguished from hydrodynamic lubrication primarily because (1) the bearing-surface topography has an important effect and (2) the predominating effects of the lubricant are caused by properties other than its viscosity (ref. 28, ch. 2). Boundary lubrication inevitably occurs in applications where speeds are too low and loads are too high to permit establishing a hydrodynamic film.

Boundary lubricants function by one of four different means:

1. Physical adsorption - nonpolar
2. Physical adsorption - polar
3. Chemisorption
4. Chemical reaction

The tenaciousness of the lubricant molecules to the bearing surfaces is greatest in chemical-reaction products and weakest in nonpolar adsorption. (In the latter case, the process is reversible with increasing temperatures.)

Boundary lubricants, alone or dissolved in other substances, function by separating contacting metallic asperities; they establish tenacious films of high strength directly on bearing surfaces. However, boundary lubrication has these disadvantages:

- Limited lubricant supply (thin-film lubrication)
- Limited cooling capacity
- No means of handling (disposing of) wear debris

Wear life under thin-film boundary-lubrication conditions is extremely difficult to predict because of the continually increasing amounts of debris, the lack of system control over temperature, and the loss of lubricant due to volatility. Therefore, boundary lubrication has not been used extensively in design of space vehicles.

Boundary lubricants are frequently added to fluid or semifluid lubricants to increase load-carrying ability or lubricity. This is possible because of the affinity of boundary lubricants for metal surfaces, which is stronger than that of the base fluids. When loads are decreased and speeds increased, liquid lubricants are able to act more and more in the hydrodynamic (or EHD) mode. Metallic contacts are fewer, although they still occur, and load carrying is partially taken over by the viscous properties of the lubricant. This condition, between boundary and hydrodynamic, is known as mixed or mixed-boundary lubrication. In such cases, wear is generally lower than in boundary conditions (ref. 5).

### **2.3.4 Solid**

In the last few decades several natural and synthetic solid lubricants have become available, such as molybdenum disulfide ( $\text{MoS}_2$ ), tungsten disulfide ( $\text{WS}_2$ ), and niobium diselenide ( $\text{NbSe}_2$ ). They are soft, lamellar, crystalline materials which form low shear-strength films on metal surfaces, and prevent seizure and wear by their capacity to withstand high-unit loads (psi). Ideally, their film thickness is adequate to separate the bearing surfaces completely, so that wear occurs only within the lubricative film itself (ref. 30).

In general, solid lubrication offers the following advantages over fluid and semifluid materials (refs. 40 and 41):

- Stability and ability to function over a wide temperature range (fig. 3)
- Design simplification (no recirculation system, filtration, heat exchangers, seals, etc.)
- High resistance to radiation
- Good performance in high vacuum (i.e., negligible volatility)
- Unusually high load-carrying ability in many cases

Some important disadvantages should also be noted:

- More difficult to apply than oils or greases

- Virtually no cooling capacity
- Little or no corrosion protection
- In a given application, wear life extremely difficult to predict from available data in another application
- No simple mechanism for replenishment of lubricant
- Failure is rapidly catastrophic once the film is degraded

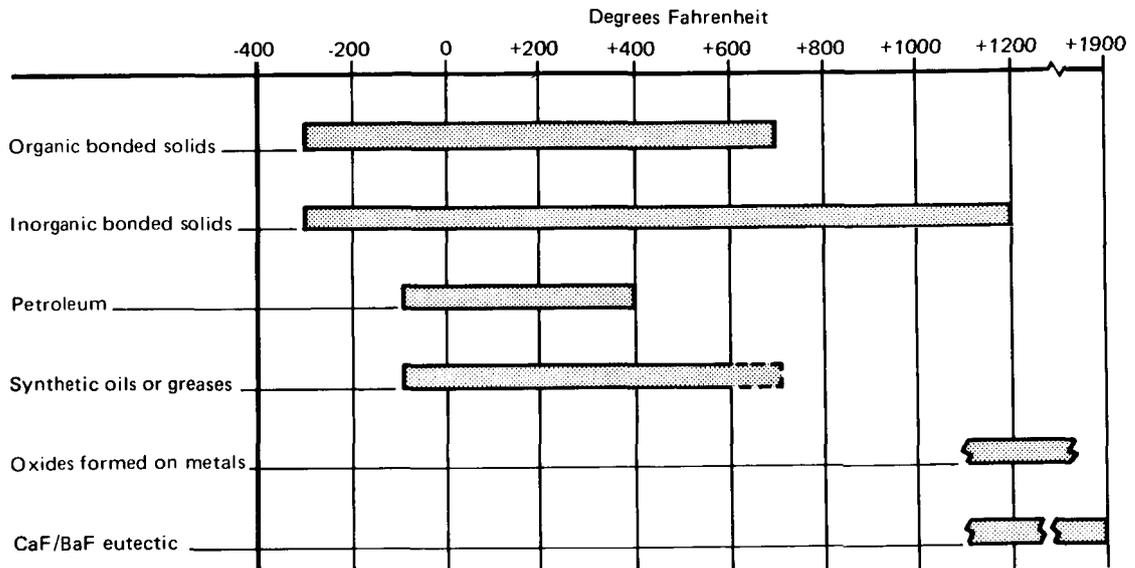


Figure 3.—Temperature range of various lubricants at atmospheric pressure.

### 2.3.5 Transfer

A special form of lubrication by solids consists of extremely thin films from a “solid lubricant reservoir” applied to a bearing surface by rubbing contact under light to moderate pressures. This method is called transfer lubrication (refs. 42 and 43). The principle is illustrated in figure 4, which shows the most common application, a ball-bearing retainer fabricated from a composite containing a solid lubricant. The ball rubs against the retainer and adheres to a trace of the solid lubricant, transferring it to the races (ref. 44). The mechanics seem simple enough, and numerous space-vehicle bearings have utilized this principle (ref. 41), but data relating to the volumes transferred and their ultimate destination are lacking. It should be noted that excessive bearing loads may cause rapid wear of the retainer and failure of the bearing.

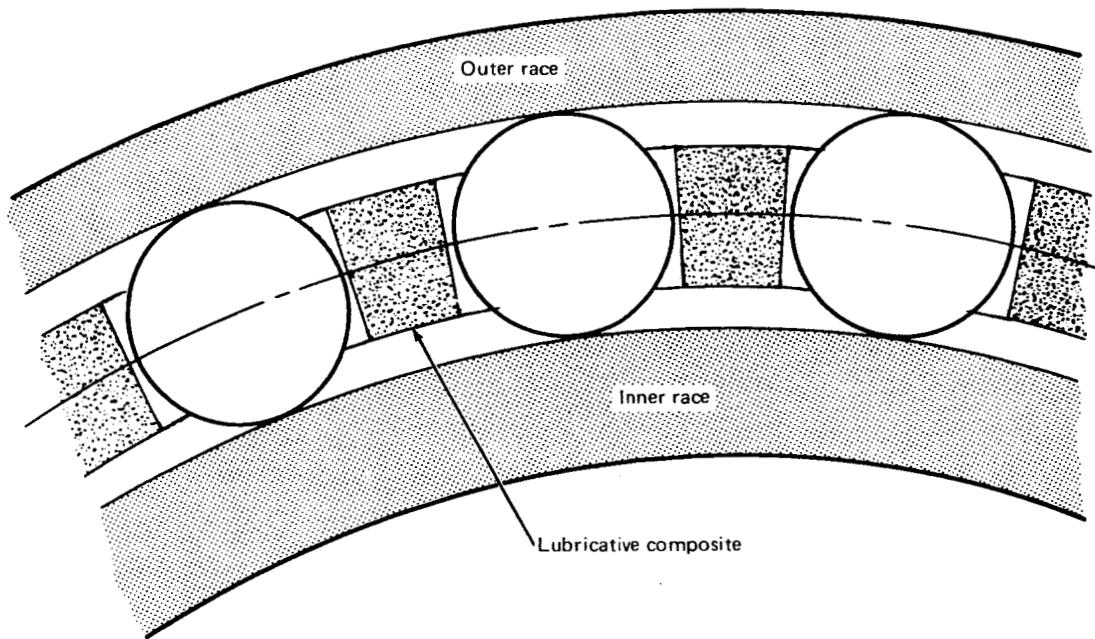


Figure 4.—Composite ball-bearing retainer used in transfer lubrication.

In addition to ball-bearing retainers, lubricative composites have been used to fabricate idler gears (refs. 45 and 46), which transfer lubricant to the other gears in a train, and shoes of various configurations, which rub against shafts, cams, etc., to provide thin, solid films. One concern in the use of transfer lubrication in space has been that portions of the bearing surfaces will contact each other in the “bare” condition before establishing the transfer film. In most applications it is possible to prevent any conceivable surface damage from this contact by applying the solid lubricant itself by hand to the bearing surfaces prior to any run-in (ref. 47). This problem is not ordinarily encountered in on-earth application if run-in is accomplished at low loads because the continued regeneration of surface oxides will prevent damage until the film is established.

## 2.4 Lubricants

Defined functionally in terms of preventing metal seizure, reducing wear and friction, and in general providing long wear life, a lubricant is a substance which is deliberately placed between contacting bearing surfaces. Ideally, a lubricant (1) has lower shear strength than either bearing material, (2) adheres to the surfaces strongly, (3) prevents contact of opposing surface asperities, (4) is stable under all operating conditions, and (5) maintains friction at a level consistent with available power. In addition, lubricants may be required to prevent corrosion and provide cooling of metal parts.

Materials used to reduce friction and wear are generally divided into three groups: oils, greases, and solids. This section discusses various materials in each of these categories with respect to their use in space vehicles.

## **2.4.1 Oils**

In spite of the wide variety of liquid lubricants (oils) commercially available today, including highly refined petroleum and synthetic materials, only a few are suitable for space operations. The properties which determine such suitability are:

- Volatility (the tendency of an oil to evaporate)
- Viscosity (the internal frictional resistance)
- Surface energy (creep tendencies)
- Lubricity (the capacity of an oil to minimize metal-to-metal contact)
- Stability (resistance to chemical degradation).

### **2.4.1.1 Volatility**

Volatility is defined as the tendency to evaporate, and is an extremely important factor in the selection of lubricants for space applications. It is generally expressed either in "absolute" terms (the vapor pressure at a given temperature) or in practical terms (how much is lost by evaporation after X hours at Y degrees under Z millimeters-of-mercury pressure). Both of these methods are valuable, and both may be used to predict the amount of fluid in the vicinity of a bearing at any time (ref. 41).

Vapor pressure is defined as the pressure exerted when a substance is in equilibrium with its own vapor, and must therefore be measured in a closed system. Petroleum oil and various polymeric lubricants having a range of molecular weights do not exhibit single-point vapor pressures; as soon as lower molecular-weight fractions volatilize, the vapor pressure becomes slightly lower.

Single-point vapor-pressure fluids are available, usually in extremely small quantities and at very high cost. Several available classes of lubricating fluids, however, have narrow ranges of molecular weight, and therefore relatively narrow ranges of both volatility and viscosity.

The amount of fluid escaping from a container with an opening of well-established geometry can be estimated with a reasonable degree of accuracy from the formulas

given by Dushman in reference 48. The vapor pressure of pure substances at any given temperature can be calculated by using Langmuir's equation. If the fluid is a mixture, Langmuir's equation may be used to approximate the vapor pressure only if the range of molecular weights of the components is narrow enough to calculate an acceptable average molecular weight (ref. 49).

The effects of volatilization on lubricants of wide molecular-weight range (polymeric oils or petroleum) are to (1) increase the viscosity, (2) raise the pour point or minimum-service temperature, and (3) decrease the volatility of the remaining fluid.

#### 2.4.1.2 Viscosity

The degree of viscosity (internal friction) of a lubricating oil determines the frictional drag of a hydrodynamic bearing and a significant part of ball- and roller-bearing friction under EHD conditions. [See eq. (5).] It is a much less significant factor in boundary lubrication, but becomes more important as speed increases and load decreases (mixed lubrication). The effects of viscosity on total friction under boundary conditions are discussed in references 15 to 17 and 35; a full discussion of viscous drag under hydrodynamic conditions is given in references 15, 28, and 35.

Figure 5 presents the temperature-viscosity characteristics of several types of oils (refs. 50 to 53). Two terms are frequently used to express the change in viscosity with temperature:

1. Viscosity index (limited usage outside the petroleum industry).
2. Viscosity-temperature coefficient (VTC); preferred over the viscosity index. This coefficient is expressed as:

$$\text{VTC} = 1 - \frac{\text{Viscosity at } 210^{\circ}\text{F (372K)}}{\text{Viscosity at } 100^{\circ}\text{F (310.9K)}} \quad (7)$$

As has been stated, the performance of oil in an antifriction bearing is determined primarily by its viscosity. This in turn is a function of both temperature and pressure. The pressure effects are given in table III in the Appendix, taken from reference 28.

#### 2.4.1.3 Lubricity

The lubricity (lubricating ability) of an oil (or grease) – that is, its capacity to keep metals apart under dynamic conditions – is a function of its viscosity and its chemical

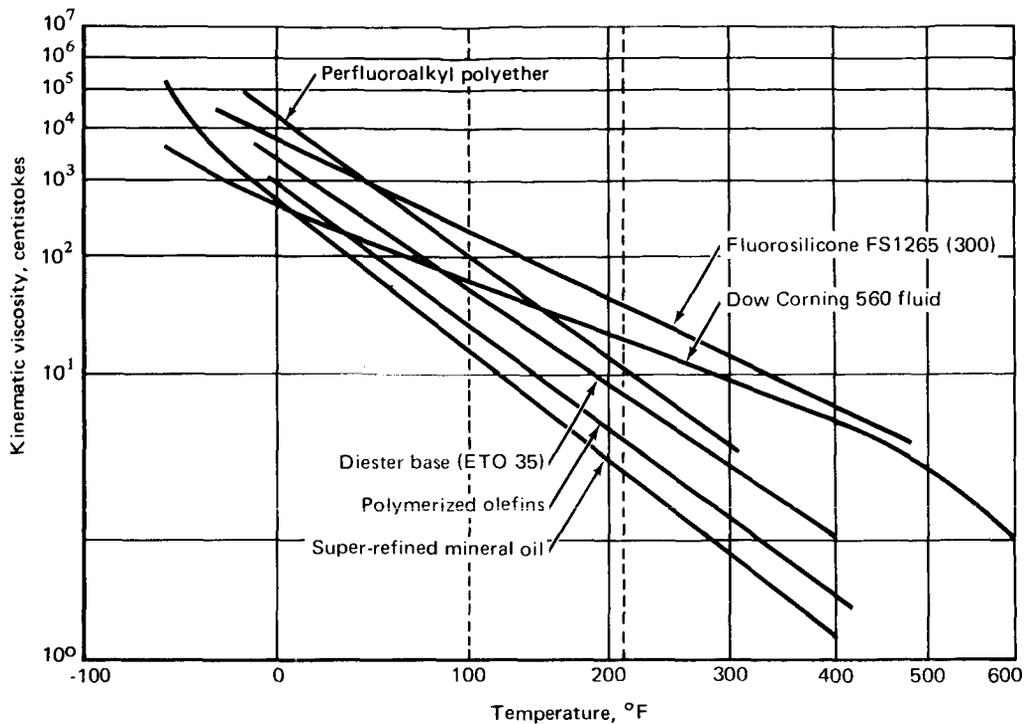


Figure 5.—Viscosity vs temperature for various oils.

activity. For example, uninhibited petroleum oils (those containing no additives) show no tendency to orient themselves with respect to the bearing surface. Neither do dimethyl silicones. Phosphate esters and organic acids, on the other hand, show exceptionally strong surface orientation, thus exhibiting a higher “film strength,” or lubricity.

Because of the light loads in space applications, lubricity is not always a premium property as are volatility and viscosity. Thus, dimethyl silicone oils may exhibit sufficient lubricity for extended periods in some applications, particularly in high-speed, lightly-loaded ball bearings. There are indications, however, that their life is limited in comparison with ultrarefined petroleum or fluorosilicone oils (refs. 54 and 55).

#### 2.4.1.4 Stability

Viscosity and volatility are reliable physical properties only if the oil is inherently stable; that is, if it is highly resistant to chemical (oxidative and thermal) breakdown. Mere traces of metallic or acidic contaminants can significantly affect such chemical degradation. Of greatest concern is the amount of exposure to oxygen during earth operations, particularly at the high surface temperatures which may be generated during tests. Thus, after the test period, the particular supply of lubricant in a system

may not be suitable for the mission. Table IV (Appendix) indicates the resistance to oxidation (useful temperature range) for several families of lubricants, along with a listing of other important properties.

Lubricant stability is strongly influenced by contamination, which includes all forms of atmospheric dust and dirt, wear debris, outgassing products from other systems, and various forms of manufacturing debris inadvertently introduced during assembly. With the exception of the copper- and lead-bearing alloys occasionally used in retainers (separators), the metals used in ball and roller bearings are not strongly catalytic to oxidation of oils. Many types of metallic debris (chips, filings, etc.) caused during manufacturing, however, can catalyze (speed up) the destructive process of oxidation. Examples of such metals are platinum, silver, aluminum, tin, and brass (refs. 56 to 58).

Synthetic fluids composed of halogenated compounds are generally more resistant to oxidation than hydrocarbons. Under conditions of high frictional energy (for example high shear), however, and in combination with certain metals, they have been known to ignite explosively.

#### **2.4.2 Semifluids (Greases)**

A grease is defined as a stable and homogenous dispersion of a thickener in an oil. Greases constitute the most important category of semifluids that are used as lubricants. They may contain various fillers for thickening (increasing viscosity), improving load-carrying ability, or increasing thermal conductivity.

The most outstanding difference between oils and the greases derived from them is their shear behavior (i.e., their viscosity). The viscosity of an oil is generally a logarithmic function of the temperature, and, at a given temperature, the shear stress is directly proportional to the shear rate (i.e., the viscosity is constant). Most lubricating oils are therefore Newtonian fluids, but this is not the case with greases. Their shear stress is inversely related to shear rate: as speed increases, the viscosity of a grease approaches that of the base fluid at the particular temperature. In addition, greases exhibit a slow recovery to their original viscosity if allowed to stand after shearing. Many colloidal suspensions have this thixotropic property.

Greases offer certain advantages over oils:

- They tend to occupy a more-or-less fixed volume and adhere better to metal surfaces
- Their volatility is somewhat lower

- They generally have higher load-carrying ability
- They tend to seal the bearing and exclude dirt

Their disadvantages are:

- The effects of evaporation are more serious. Greases tend to dry out or harden noticeably as the base oil evaporates
- The characteristics of a grease are affected by adjacent absorbent surfaces, which also cause drying out
- They retain dirt and wear debris within the bearing area
- They are not efficient in removing heat from the bearing area
- They cause higher bearing torque than oils

Because of the severe environmental conditions, only a few greases have been used in space applications, these primarily in enclosed ball bearings operating at relatively high speeds (5000 to 8000 rpm) (ref. 15).

Several greases have been used in spacecraft. One, having exceptional temperature range,  $-100^{\circ}\text{F}$  to  $+450^{\circ}\text{F}$  (200K to 505.4K), in air is a silicone product, Versilube G300. In numerous laboratory vacuum tests this grease has lubricated small electric motor bearings for thousands of hours (Ref. 54). The principal cause of ultimate failure has been evaporation of the base fluid. A much longer lasting grease might be made available if the base fluid were "stripped" of its lower boiling fractions; for example, by vacuum distillation.

Another synthetic grease which has seen spacecraft service is Krytox 240 AC, based on a perfluoroalkyl polyether. This grease has very superior load-carrying ability, toxicity and flammability properties, as well as resistance to oxidizers and hydrazine-type compounds.

A number of candidate greases and their properties are shown in the Appendix in table V (refs. 59 to 61).

### **2.4.3 Solids**

A wide variety of solid substances has been used to lubricate bearings under environmental conditions too severe for oils or greases. These materials are conveniently

divided into five categories: (1) lamellar solids, (2) metal platings, (3) miscellaneous inorganic substances, (4) polymers, and (5) bearing surface-conversion coatings. These are discussed separately in the following sections.

### 2.4.3.1 Lamellar Solids

Lamellar solids are so named because of the layered lattice-like crystal structure of the various members. Materials in this category range from graphite and molybdenum disulfide through cadmium iodide and lead iodide to mica and boron nitride. In space, the most widely used member of the group is molybdenum disulfide because it exhibits exceptionally high load-carrying ability, extremely low friction in vacuum ( $f_k = 0.04$  to  $0.05$ ), and excellent thermal stability (more than  $600^\circ\text{C}$  in vacuum). Certain synthetic powders have properties quite similar to molybdenum disulfide ( $\text{MoS}_2$ ); some of them are electrically conductive. The most important are niobium diselenide ( $\text{NbSe}_2$ ) and molybdenum diselenide ( $\text{MoSe}_2$ ). Their properties are listed in the Appendix in table VI (refs. 62 to 65).

Molybdenum disulfide (and other metal salts) are potentially capable of contributing to the corrosion of bearing metals, especially steel. This corrosion depends on the presence of moisture and in the case of  $\text{MoS}_2$  is further affected by the presence of the acidic products of oxidation.

The lamellar solid, graphite, although quite similar to  $\text{MoS}_2$  in physical appearance, exhibits vastly different behavior in the space environment (vacuum). The coefficient of friction of  $\text{MoS}_2$  in humid air is strongly affected by load and slightly affected by speed. Its performance is greatly improved in vacuum. The opposite is true of graphite, which requires moisture to behave as a lubricant, and which actually becomes fast-wearing and abrasive in vacuum.

There are bonded solid lubricants (table VI in the Appendix) which combine relatively small quantities of graphite with major portions of  $\text{MoS}_2$ . Their behavior in vacuum is dominated by the  $\text{MoS}_2$ , and they perform satisfactorily with respect to friction.

Certain new compounds of carbon (graphite) and fluorine (ref. 66) have been found potentially useful in space vehicle mechanisms. They exhibit vacuum friction and wear characteristics similar to  $\text{MoS}_2$ , and are stable in air to  $842^\circ\text{F}$  ( $721.3\text{K}$ ).

The lamellar solids can be utilized by a variety of methods. They have been used principally in the form of rubbed-on films, resin-bonded coatings, conversion coatings, and ion-sputtered films. A comparison of several coatings is given in the Appendix in table VII (refs. 40 and 67). "Temperature range in air" in this table should be considered in the light of diminishing wear life with increasing temperatures, particularly

temperatures higher than 100°F (310.9K). This may be attributed to the rapidly increasing rates of oxidation of both solid lubricants and organic-resin binders with increasing temperatures.

Rubbed-on and “burnished” film of MoS<sub>2</sub> and/or other solid lubricants are generally considered useful for relatively short-term applications. Their wear life is not comparable to that of bonded solid lubricants containing the same pigments. In certain applications, however, they may be the only suitable lubricant. For example, NASA Lewis Research Center (ref. 68) reported 92 million cycles (1-degree oscillations) of a ball-bearing operating in ultrahigh vacuum with a transfer-lubricant ball retainer. Because of the small arc of oscillation, the retainer could not be depended on to provide a transfer film, and MoS<sub>2</sub> was burnished onto the races in an argon atmosphere prior to operation. The same technique is useful with rotating and oscillating bearings, even when the rotation is sufficient to establish a transfer film, because the initial relative motion under load is potentially capable of causing damage to clean metal surfaces.

The variables associated with the process of burnishing MoS<sub>2</sub> into metal surfaces and the characteristics and the endurance life of such films are discussed in considerable detail in references 64 and 69 to 71. Most recently, certain investigators have observed that MoS<sub>2</sub>, both in powder form and in bonded solid lubricants, is capable of abrading hard metal surfaces (refs. 72 and 73) before the surface of the film is “worn in.” This strongly suggests that such films should be run in prior to assembly, particularly if the opposing surface also is not coated.

Resin-bonded solid lubricants are essentially coats of paint in which the pigment is a solid lubricant. They are much tougher and longer lasting than rubbed-on films for several reasons:

- Much more pigment (lubricant) is available. The coating thickness is approximately 0.00005 in. (12.7 microns), compared to the ~0.00002 in. (0.508 micron) of rubbed-on films.
- The metal surface is cleaned, roughened, and prepared for maximum adhesion of the binder.
- The coating itself has higher cohesion due to the binder, and is self-healing to a significant degree (ref. 72).
- The coating properties can be modified (by incorporating additives) to provide maximum toughness and durability in a variety of conditions and atmospheres.

The wear life of bonded solid lubricants is a subject of considerable disagreement among the numerous investigators; no two seem to agree on the comparative endurance of various coatings, or the best method or test machine for evaluating them (ref. 75). There is general agreement, however, that for space applications, the advantages of bonded coatings containing  $\text{MoS}_2$  outweigh their disadvantages (Sec. 2.3.4), their chief advantages being their low volatility (ref. 76) and their high load-carrying ability. With good design practice, the latter is higher than the strength of the bearing materials (ref. 77). Coatings containing  $\text{MoS}_2$  exhibit the same air-to-vacuum friction transition as that observed for the powder: a five- to ten-fold drop. This is attributed to the effect of moisture on the bonding energy between adjacent crystallites (ref. 78).

Inorganic bonded solid lubricants, in particular MIL-L-81329, have been found to be extremely effective for ball bearings. They are especially suitable when applied in small cavities or reservoirs machined into the ball retainers and lands (ref. 79).

“Conversion coating” refers to a process by which the lubricating substance is formed directly (in situ) on the bearing surface. The best known material is Molykote E3C, which can be applied in controlled thicknesses ranging from 50 to 250 microinches (1.27 to 6.35 microns) (ref. 78). After the surface has been suitably cleaned, the application process consists of two steps: (1) dipping the material in aqueous molybdenum trioxide ( $\text{MoO}_3$ ) and (2) baking it at  $\sim 450^\circ\text{F}$  (505.4K) in an autoclave pressurized with hydrogen sulfide, which converts the  $\text{MoO}_3$  to  $\text{MoS}_2$ . The coating offers excellent performance when applied completely under control. One serious disadvantage is the required application temperature of  $450^\circ\text{F}$  (505.4K), which restricts use of the coating to alloys such as stainless steels, tool steels, and nickel-cobalt alloys.

NASA Lewis Research Center (refs. 81 and 82) recently developed a vacuum-deposition (sputtering) technique for applying  $\text{MoS}_2$  to bearing surfaces. As a result, wear life has been lengthened and friction reduced; but there are some practical difficulties associated with its use, primarily those of quality control and cost of manufacture.

### **2.4.3.2 Metal Platings**

Several soft metals have been successfully used as low-shear-strength solid lubricants on gears, plain bearings, ball bearings, and slip rings. The principal materials have been lead, indium, silver, and gold (refs. 83 and 84). For metal platings to provide low friction for extended periods, they must be applied very thin. If they are applied much thicker than 0.0002 in. (5.08 microns) they become quite rough in texture because of plastic deformation, and can contribute to fatigue due to stress risers. This is of greatest importance in ball bearings. Thickness is not as important in plain bearings,

except so far as excess material contributes to wear debris. The principal difficulty with the use of thin metal platings in ball bearings is the formidable amount of process control required to obtain uniform coatings. Highly specialized deposition techniques have been employed for both lead and gold (refs. 83 and 84).

Thin-film metal platings have been used extensively on electrical slip rings and indium and gallium (refs. 85 and 86) have been investigated, both with and without supplementary oil lubrication. Both metals improve the wear life of slip rings. Gold plate rubbed with  $\text{MoS}_2$  has also been effective, in spite of the fact that  $\text{MoS}_2$  is a semiconductor.

#### **2.4.3.3 Miscellaneous Inorganic Substances**

Numerous inorganic substances (other than lamellar solids) are used as solid lubricants. One of the most prominent is lead monoxide ( $\text{PbO}$ ), frequently used as a supplementary "pigment" in bonded solid lubricants. By itself,  $\text{PbO}$  exhibits high friction (in air) up to about  $800^\circ\text{F}$  ( $700\text{K}$ ); its friction is relatively low from  $800^\circ\text{F}$  to  $1250^\circ\text{F}$  ( $700\text{K}$  to  $950\text{K}$ ). Silica has been used as a binder for  $\text{PbO}$  in high-temperature bonded solid lubricants (ref. 87).

Because of the thermal limitations of  $\text{MoS}_2$  coatings [(rapid oxidation above  $750^\circ\text{F}$  ( $672\text{K}$ ), a short service life in air above  $400^\circ\text{F}$  ( $477.6\text{K}$ )], large numbers of high-melting materials have been combined with various ceramic binders such as boric acid and cobalt oxide ( $\text{CoO}$ ). One solid in particular, calcium fluoride ( $\text{CaF}_2$ ), plus  $\text{CoO}$  binder, gives reasonably low friction ( $f_k = 0.1$  to  $0.2$ ) over the range of  $75^\circ\text{F}$  to  $1500^\circ\text{F}$  ( $297\text{K}$  to  $1088.7\text{K}$ ). It is generally used on nickel alloys at temperatures of  $500^\circ\text{F}$  to  $1900^\circ\text{F}$  ( $553.1\text{K}$  to  $1310.9\text{K}$ ) (ref. 88).

#### **2.4.3.4 Polymers**

Polymeric substances (organic and inorganic resins, lacquers, copolymers, etc.) are suitable for application in space primarily in the form of filled materials or composites (Sec. 2.4.4), and as binders for solid lubricants. The notable exception is a baked-on coating of polytetrafluoroethylene (PTFE) or fluorinated ethylene/propylene (FEP). These coatings have the disadvantages of cold flow under load and a high-temperature application ( $\sim 700^\circ\text{F}$ ) ( $644\text{K}$ ) which severely limits the selection of base material.

#### **2.4.3.5 Bearing-Surface-Conversion Coatings**

Some of the special alloys used from  $1000^\circ\text{F}$  to  $1500^\circ\text{F}$  ( $810.9\text{K}$  to  $1088.7\text{K}$ ) form a protective and relatively low-friction oxide coating that can prevent galling at high

temperatures. For example, Haynes Stellite 25 has quite low friction in air at temperatures in excess of 1000°F (810.9K), but exhibits high friction and galling at lower temperatures (ref. 89).

Another alternative to the use of special lubricants at extremely high temperatures is to make bearings entirely from temperature-resistant materials, such as aluminum oxide and tungsten carbide (refs. 15 and 90). However, these materials have the disadvantage of being quite brittle, and their coefficients of thermal expansion are significantly lower than those of most structural metal alloys.

## **2.4.4 Composites**

### **2.4.4.1 Plastic-Base Composites**

For years, a few polymeric organic substances have been applied as lightly loaded bushings or bearings. The low-friction polymers, polytetrafluorethylene (PTFE) and fluorinated ethylene/propylene (FEP), exhibit cold flow even under very light loads, and unless special design provisions are made to contain them, are not suitable for extended applications. This disadvantage can be overcome, however, by incorporating fillers such as fiberglass, vermiculite, and diatomaceous earth (ref. 91). PTFE and FEP may also be improved by the addition of MoS<sub>2</sub>. The best-known lubricative composite (Duroid 5813) is a mixture of PTFE and fiberglass, with about 15-percent MoS<sub>2</sub> (refs. 54, 83, and 92).

A special family of extremely high-strength composites consists of a multistrand, interwoven cloth of PTFE fibers and fiberglass with (usually) a phenolic resin binder (ref. 93). One example is Fabroid, which has been used as oscillating and rotating bushings at loads in excess of 90,000 psi ( $621 \times 10^6$  N/m<sup>2</sup>). These materials are not used in tension, and have a tendency to unravel in line-contact applications, but their performance in area contact is outstanding. They have been used extensively in space vehicles, particularly in rocket-engine gimbals.

The polyimide resins have been evaluated (filled and unfilled), and compare favorably with PTFE-base composites with respect to friction, wear, and weight loss at elevated temperatures in high vacuum (refs. 47, 94, and 95).

Composites having a polymeric (plastic) base have been successfully used as ball-bearing retainers, lightly loaded gears, and plain bearings and other sliding surfaces (refs. 96 and 97). The properties of several different materials in this category are compared in tables VIII and IX (Appendix).

#### **2.4.4.2 Metal-Base Composites**

Lubricating composites containing large portions of various metals are usually much stronger in compression and tension than polymeric-base materials, but are frequently more brittle. Because of the metal constituent, they have good heat-transfer properties and can occasionally function as brushes in electrical equipment. For example, the "hot compact" 046-45 (ref. 45) was used for brushes on the drive motor of the Surveyor Surface Sampler (ref. 98). Properties of several metal-base composites are given in the Appendix in table X (refs. 45 and 99 to 101).

#### **2.4.5 Liquid Metals**

In nuclear-reactor systems lubricated with a cycle-working fluid, the bearings can be expected to operate in the range of 400°F to 1500°F (477.6K to 1088.7K). These temperatures accentuate the chemical activity of the working fluids, particularly of their solvency. The most common fluids are mercury and the alkali metals (sodium, potassium, etc.). It is the fluid activity of these liquid metals, far more than the severity of the environment, that causes problems in nuclear applications. The subject is discussed in detail in references 15 and 102 to 105.

### **2.5 Effects of the Space Environment**

#### **2.5.1 Pressure**

The concentration of gas molecules in outer space is so low that the phenomenon of pressure is not experienced as such. The effects are of individual particles behaving according to the laws of diffusion.

Table XI in the Appendix summarizes the characteristics of the space environment. For theoretical purposes, oxygen must be considered as available to form oxides on metal surfaces as far out as 600 miles ( $966 \times 10^3$  m) in space, but its concentration is 14 orders of magnitude less than at sea level. Figure 6 illustrates the effects of the partial pressure of temperature and oxygen on the time required to form a theoretical monolayer of oxide; that is, a layer one angstrom ( $10^{-10}$  m) thick. Note that drastic increases in temperature have not nearly the effect of an order of magnitude of oxygen pressure. A limited amount of experimental work has been done to incorporate chemicals in lubricants to regenerate an oxide film in space (ref. 106).

Of equal consideration is the effect of low pressure on the transfer of heat from bearing surfaces or mechanisms exposed to the atmosphere. The "pressure" of the atmosphere surrounding the spacecraft is actually much higher than that of the normal

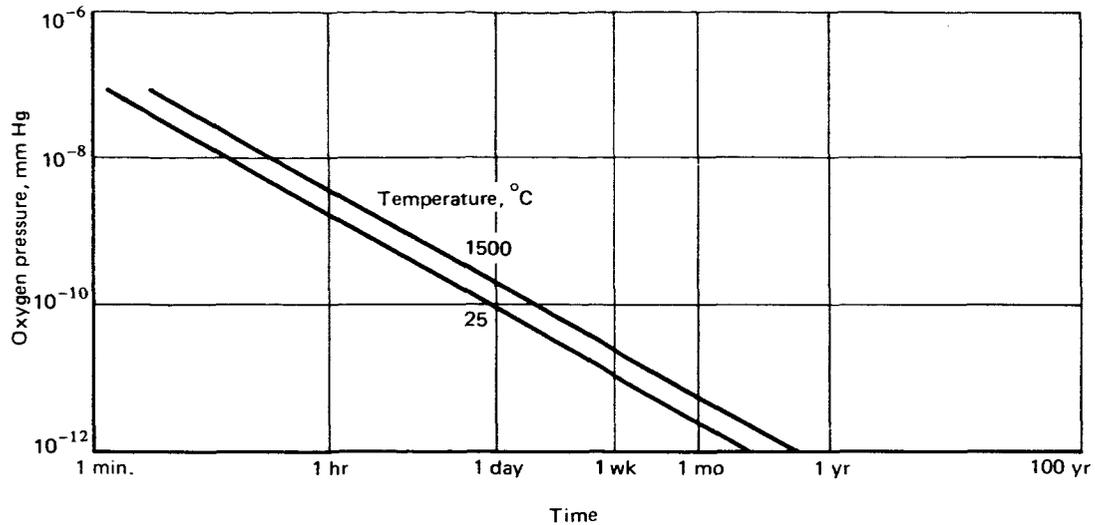


Figure 6.—Time required to form a one-angstrom ( $10^{-10}$  m) thick oxide film (FeO on iron).

space environment because of the outgassing of lubricants, elastomers, and various structural materials. Thus, the ambient pressure experienced by the spacecraft is an “average” of the vapor pressures of all the contributing materials.

## 2.5.2 Radiation

Generally, the radiation commonly occurring in space does not pose a serious threat to surface materials and lubricants properly selected for use on space vehicles (refs. 15 and 107). Radiation from nuclear-propulsion systems, however, and repeated exposure to Van Allen-belt radiation can cause significant degradation of organic liquids and most plastics (table IV in the Appendix). The critical factor in radiation effects is the accumulated dosage. Thus, the level which even personnel find harmless on short missions may prove harmful to materials on space stations designed for ten years' use. Resistance of solid lubricants to radiation is discussed in detail in references 108 and 109.

## 2.5.3 Weightlessness

During the “space” portion of all missions (essentially the entire mission for orbiting vehicles), the lack of gravitational effects means that many bearings are not required to support structural loads (except those that are internally applied). Bearing loads imposed by acceleration and deceleration, unbalanced dynamic forces, and the centrifugal forces of balls against races in rotating ball bearings are relatively light. Thus, the net effect will generally be reduced wear (ref. 54). In certain space vehicles, however, weight may become critical. Bearings may therefore be required to operate at high preloads for extended periods, and premature failures may occur with parts difficult to repair or replace.

Lubricants per se may be affected by weightlessness. For example, greases and oils may develop “voids” not readily filled, giving rise to higher friction, heat, and wear.

The weightlessness experienced by orbiting vehicles causes design difficulties because (1) gravity-fed systems of lubrication and filtration are ineffective and (2) the characteristics of the flow of lubricants in “weightless” bearings are not fully understood. The latter is not yet considered to be a serious disadvantage; oil-lubricated ball bearings have been able to function in space for extended periods.

## **2.5.4 Temperature**

Extremes of temperature encountered during space missions make lubricant selection extremely difficult, and in fact usually require that special forms of insulation or thermal-control coatings be incorporated on the outer skins of spacecraft. Direct radiation from the sun can produce surface temperatures as high as 250°F (394.2K), but when positioned in the sun’s shadow, mechanisms can experience temperatures as low as -280°F (99.8K). In cryogenic application the lubricants must withstand the extremely low cryogen temperature and the presence of the cryogen, often at elevated pressures. If deep space probes and entry vehicles are considered, these temperature limits must be extended to as low as -427.7°F (20K) and as high as 2510°F (1650K) (ref. 110). If the entire range of temperature must be endured by the lubricant, only solids can be considered as candidates (fig. 4).

## **2.5.5 Special Conditions**

Certain operations among the many space vehicles and missions have imposed unusual requirements upon lubricants. Examples are:

- Oxygen-pressurized personnel cabins in which humidity is at a high level (e.g., 65-percent relative humidity or higher in Apollo). These conditions combine to form a corrosive atmosphere, especially when a high carbon dioxide atmosphere is added to make the condensate acidic or when perspiration and urine contribute electrolytes.
- Entry of an orbiting vehicle, particularly when the vehicle is intended for reuse, as the space shuttle. The enormous skin temperatures generated at control surfaces will impose correspondingly higher temperatures on support bearings and lubricants.
- Use of lubricants in the vicinity of highly reactive propellants and oxidizers.
- Nuclear radiation for electrical power.

## **2.6 Earth-Operation Effects**

At all times on earth, space-vehicle mechanisms must be protected from corrosion and from contamination, from metal chips, solder, atmospheric dust and dirt, acids and alkalis, and lubricants not compatible with the flight lubricant. This protection is equally important throughout all stages of manufacturing, storage, and testing.

Corrosion effects may be particularly insidious when items are coated with solid lubricants and have no other means of corrosion protection (Sec. 2.3.4).

In certain environments (e.g., smog), the corrosive elements may be of great concern. Such acidic substances as sulfur dioxide can rapidly cause corrosion of steel and bronze bearing surfaces, particularly when combined with moisture, although this is one of the easier effects to control.

Thermal and chemical procedures are used to sterilize spacecraft. The temperatures involved may be in excess of the maximum which certain fluid and semifluid lubricants can withstand without degradation, although the temperatures are generally not high enough to affect bonded solid lubricants or lubricative composites.

Because it poisons viable organisms, ethylene oxide gas has been used for sterilization of spacecraft. Liquid lubricants can absorb and dissolve varying quantities of this gas, but the gas has no effect on bonded solid lubricants (ref. 111).

## **3. CRITERIA**

All contact surfaces designed to undergo relative motion (including sliding, rolling, or separating) shall be lubricated to prevent excessive wear or seizure. Such surfaces shall be lubricated to minimize friction except for specifically designed high-friction devices. All material properties and environmental and operational conditions shall be accounted for in design. Lubricants shall be defined in conjunction with the choice of bearing materials. Lubricants shall not induce detrimental interactions with materials or with other space-vehicle systems.

Analysis or tests shall be conducted to determine friction and wear throughout the vehicle's service life and to demonstrate that the effects of friction and wear will not impair the functioning of vehicle mechanisms. Tests shall be conducted to characterize lubricants and material surface properties when available data are inadequate.

### **3.1 Lubrication System**

The system selected for lubrication of any bearing shall ensure that:

- Friction does not exceed the limitation of available power.
- Temperature resulting from frictional heating does not exceed allowable limits.
- Wear does not occur at a rate that compromises the function of any mechanical device.
- No detrimental interaction occurs between the lubricant and any other material or space-vehicle system.

### **3.2 Selection of Lubricants**

When selecting lubricants for mechanical devices, the following should be accounted for:

- Operating conditions
  - (a) Relative speed
  - (b) Load (stress levels)
  - (c) Long periods of inactivity
  - (d) Desired life
- Environmental conditions
  - (a) Temperature
  - (b) High humidity and condensation during storage
  - (c) Radiation
  - (d) Ambient pressure (vacuum)
  - (e) Gravity
  - (f) Detrimental medium
- Materials properties
  - (a) Bearing-material characteristics
  - (b) Lubricant characteristics
  - (c) Surface topography
- Effects of use, including test
  - (a) Wear, including metallic and lubricant debris
  - (b) Contamination of the lubricant

- (c) Degradation of the lubricant
- (d) Temperature gradients due to frictional heat

### **3.3 Analysis**

Analyses shall be performed on all mechanical devices to demonstrate that:

- The actual friction force will not exceed the limitations of available power .
- The effects of environment and use have been adequately accounted for .
- The wear will be so low that no failure could occur within the total mission duration .
- Their operation will not impair the functioning of any other system, either by chemical or physical interaction .

Calculations to determine the amount of wear which will occur shall be based on published data or laboratory measurements. Allowable wear shall be at least five times the anticipated (calculated) wear.

### **3.4 Tests**

When available friction and wear data cannot be demonstrated to be reliable, adequate, and applicable to the bearing/lubricant combination, data shall be obtained by tests that simulate earth operations and mission profile as closely as possible, and in chronological sequence.

## **4. RECOMMENDED PRACTICES**

### **4.1 Lubrication System**

The lubrication system for any mechanical device should be selected only after thorough review of the following:

- Total operational requirements, including ground handling and storage, mission life, and environmental conditions
- Characteristics of the mechanical function (load, speed, type of motion)
- Interaction between the lubricant and other systems

Suitable calculations should be performed to determine:

- Total friction permissible in each device, based on the power available under the most adverse conditions
- Local temperature rise resulting from the calculated friction
- Maximum wear that can occur without compromising the function of the mechanical device
- Rate of use of lubricant

The overall requirements of the lubrication system should then be tabulated and the lubricant properties determined; for example, the volatility, viscosity, stability, and chemical properties. These properties should evolve from a thorough study of all the expected operating and environmental conditions, and the effects of use. The physical and chemical properties of materials in adjacent systems should then be studied to preclude the possibility of adverse interactions with the selected lubricant. In the event that the only solution includes use of a lubricant which will inevitably produce an undesirable interaction, seals or other physical barriers must be provided to prevent contact between the lubricant and the particular material or surface.

## 4.2 Selection of Lubricants

A stepwise approach to the selection of lubricants is presented in figure 7. As an example, for an enclosed pair of support ball bearings, the following requirements have been determined:

- Oil lubrication is desirable.
- The vapor pressure of the oil at the maximum chamber temperature must be no more than  $1 \times 10^{-8}$  torr ( $1.33 \times 10^{-6}$  N/m<sup>2</sup>) [for a 5-year mission in a 4-in. (10.16 cm) diameter labyrinth seal with a clearance of 0.002 in. (0.0508 mm) and a length of 0.75 in. (1.9 cm)].
- The oil must contain a corrosion preventive because there is no feasible way of maintaining an inert atmosphere during the entire earth-operations stage.
- To maintain an EHD film thickness of 4 microinches (0.1016 microns), the viscosity of the oil at 130°F (327.6K) (maximum temperature) must be no less than 140 centistokes ( $140 \times 10^{-6}$  m<sup>2</sup>/sec).
- The pour point must be a maximum of 10°F (260.9K).

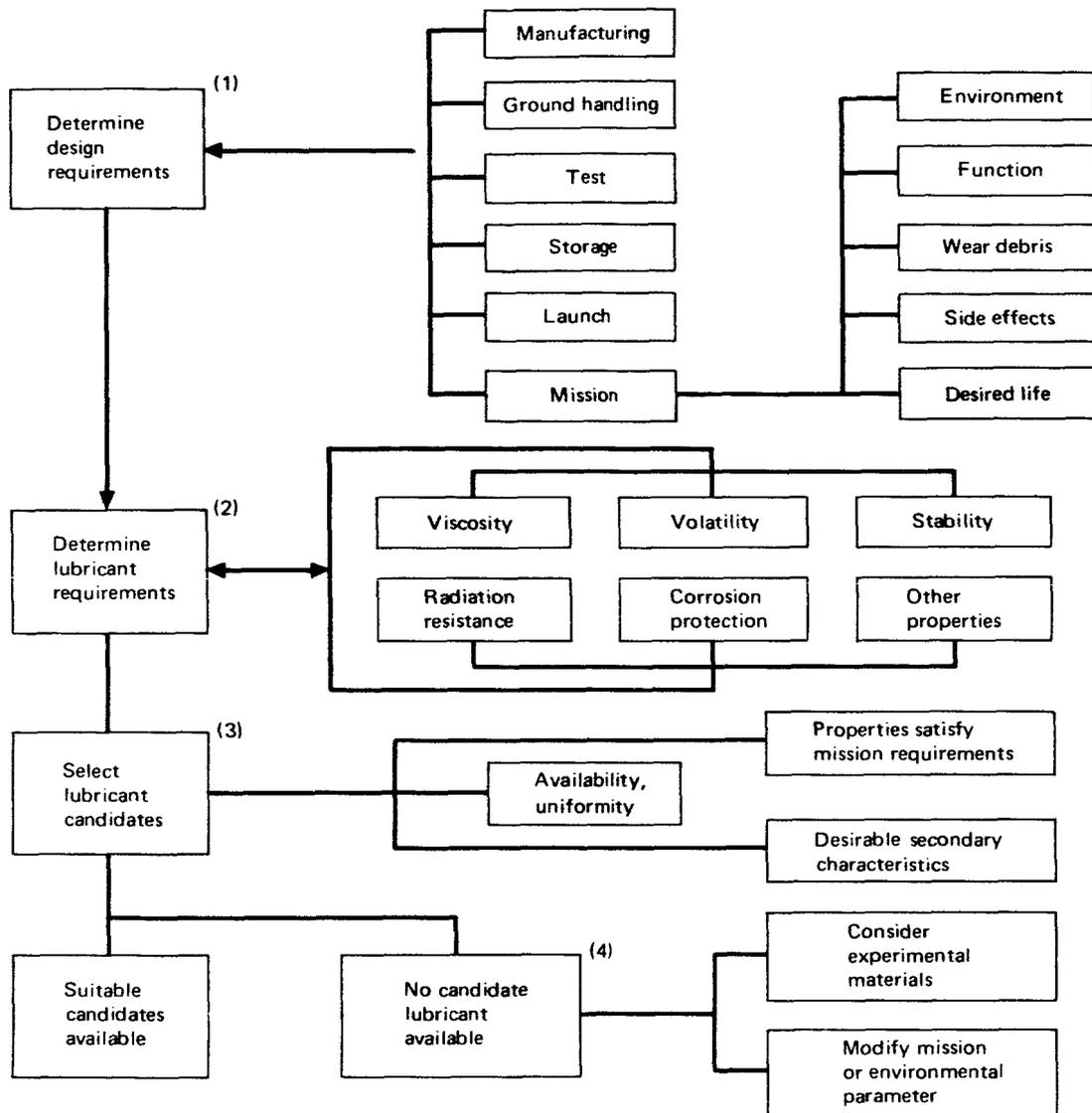


Figure 7.—Lubrication problem solving.

After following steps 1-4 shown in figure 7, if two or more lubricant candidates are available, the best should be selected on a basis of superior lubricity, increased thermal and oxidative stability, compatibility with space-vehicle materials, or other considerations.

If no lubricant candidate is available, two options are open:

1. Several experimental materials are available in each of the general categories: oils, greases, and solids. The materials are usually quite expensive, and, since

they are not in production, may vary considerably from batch to batch. It is therefore recommended that sufficient quantities of the selected material be obtained at the outset to conduct all tests and to lubricate all parts involved, plus spares.

2. If it is not feasible to use experimental lubricants, a study should be conducted to determine whether some parameter of either the performance (mission) or the environment can be relaxed. Obviously, this is a last resort because it starts a trend toward compromising the mission or diminishing its value.

The lubricating value of a grease or an oil may be estimated from the data obtained on the numerous bench testers which have been used to predict lubricity, load-carrying ability, and antiseize and antiwear properties (refs. 17 to 19 and 37). Such data should be studied carefully, used with great discretion, and considered as preliminary information only.

#### 4.2.1 Ball and Roller Bearings

Lowest friction in an antifriction (ball or roller) bearing is achieved by lubricating it with an oil of the lowest viscosity consistent with maintaining a full EHD film [e.g., table VI in the Appendix and eqs. (5) and (6) in Sec. 2]. The following oils are recommended for consideration in the ground and space environments, with the reservations noted.

<u>Lubricant</u>	<u>Reservations</u>	<u>Ever used in space?</u>
Apiezon C	High pour point; relatively poor viscosity-temperature characteristics	Yes
Versilube F-50	Traces of acid or alkali, or exposure to high temperatures (at metallic asperity contacts) may cause polymerization in vacuum and in air (ref. 110); low boiling fractions should be removed; poor additive susceptibility	Yes
Krytox 143AC	Poor additive susceptibility; additives do not form stable solutions with this oil under all conditions; very little experience	Yes

<u>Lubricant</u>	<u>Reservations</u>	<u>Ever used in space?</u>
FS-1265	Poor additive susceptibility; very little experience except with high-speed ball bearings	No
Polyolefins	Experimental. Excellent temperature range, lubricity and additive acceptance	No

In calculating volatility losses of lubricants from the housing of ball bearings (and other mechanisms), equations derived from reference 48 should be used. (See Sec. 2.4.1.1.)

It is recommended that where long life is a prime consideration, the most volatile fractions of certain candidate lubricants be removed by nondestructive distillation (e.g., molecular distillation). This method is effective only when a range of molecular weights is present in the lubricant; in such cases, the vapor pressure may be reduced by two or more orders of magnitude. Note that this has the effect of increasing the viscosity and raising the pour point.

When ball bearings are required to function at speeds so low that a satisfactory EHD film cannot be established, it is recommended that a tricresyl phosphate (TCP) treatment be considered for balls and races (ref. 39).

The greases listed in table V (Appendix) should be considered for use in ball bearings.

One form of solid lubrication in antifriction bearings has been extensively used in space: the self-lubricating retainer, principally exemplified by the Bartemp Bearing, which incorporates a Duroid 5813 retainer (PTFE/MoS<sub>2</sub>/fiberglass) (ref. 96). A similar material (Rulon A + 5-percent MoS<sub>2</sub>) has undergone considerable vacuum-system testing, as have several other composites (ref. 112). The solid materials in table XII (Appendix) are recommended where liquid lubricants may cause contamination of adjacent mechanisms. Brand names are listed in table XIII. Loads and preloads should be drastically reduced when low tensile strength composite retainers are used.

#### **4.2.2 Gears**

Oil or grease lubrication is recommended for gears to be used in the space environment whenever the gears can be enclosed in a housing, and preferably when provision can be made for some form of reservoir. With respect to volatility, the same considerations apply to gears as to ball bearings, and the same calculations apply for rate losses through seals. Lubricative composites are recommended for gears only when stress

levels are quite low; these materials are especially sensitive to shock loading. They should be considered for “idler” gears to provide lubrication by transfer in systems where otherwise unlubricated gears transmit the power.

Bonded solid lubricants are recommended for use in low-speed gears (below 1000 rpm) with moderate loads. They should be applied thin – under 0.001 in. (25.4 microns). The gear train should be given a moderate run-in and the lubricant debris should be removed afterwards. This is important because buildup in teeth roots can cause stoppage. Debris should be removed by (1) blowing the gears with clean, dry air or dry nitrogen; (2) vacuuming; or (3) brushing with clean camel hair or sable brushes. The following bonded solid lubricants are recommended for evaluation:

<u>Lubricant</u>	<u>Description</u>
Lubeco 905	High pigment, inorganic binder
Electrofilm 2396	Sodium silicate/MoS <sub>2</sub> /graphite (Mil-L-81329)
Molykote X-15	
Vitrolube 1220	Ceramic binder/MoS <sub>2</sub> /graphite
NPI-14	Phenolic binder/MoS <sub>2</sub> /graphite (Mil-L-8937)
NPI-425 (MLR-2)	Polyimide binder/MoS <sub>2</sub> /Sb <sub>2</sub> O <sub>3</sub>

#### 4.2.3 Plain-Bearing Surfaces

Solid lubrication is recommended for plain-bearing surfaces, even when the endurance life required is extremely short or the total number of cycles very small. When it is expected that corrosive environments will be encountered, bearing materials should be selected from corrosion-resistant alloys and combinations of metals. If oils or greases must be used, consideration must be given to the safety requirements of the mission (spacecraft).

Composites are recommended for plain-bearing applications, with the following reservations:

- Filled PTFE materials are quite limited with respect to load, and their friction increases with speed.

- Metal-base composites are generally brittle, and should be kept in compression.
- PTFE-fiber/fiberglass resin-bonded weaves will unravel if used in line- or point-contact applications.

## 4.2.4 Electrical Contacts

### 4.2.4.1 Potentiometers

Thin films of silicone and diester greases and oils are recommended for lubrication of the bearings and sliding contacts of wire-wound potentiometers. The limiting factor is lubricant volatility, or loss of lubricant through the bearing shield. Whenever escaping oil molecules can degrade the functioning of any adjacent surface, it is recommended that solid lubricants be used (ref. 100). With the exception of those used in the ball bearings, these solid lubricants must be electrically conductive. Both niobium diselenide ( $\text{NbSe}_2$ ) and molybdenum diselenide ( $\text{MoSe}_2$ ) are excellent materials, but their purity should be carefully controlled. They should be applied to the slip ring and the wire windings (resistance element) with a Q-tip dipped in the lubricant. The excess lubricant should be removed either by vacuuming or blowing with dry, filtered nitrogen or with argon.

### 4.2.4.2 Slip Rings

Because the critical factor in signal slip rings is noise generation, which can be of greater magnitude than the signal itself, it is recommended that very thin films of low-viscosity, low-volatility oil be used. For lubrication of slip rings, Versilube F-50 and Apiezon C have both been used successfully. Temperature of F-50 or any other silicone should be watched carefully because of the polymerization potential (refs. 58 and 113). Unit load should be kept as low as possible to minimize wear, but not so low that hydrodynamic conditions are reached. (In this case, an open circuit would result.) This is unlikely when wires in V-grooves are used, but becomes a consideration with conforming brushes.

Power slip rings, conducting up to  $200 \text{ A/in.}^2$  ( $3.1 \times 10^5 \text{ A/m}^2$ ) generate considerable heat because of power losses, and the danger of polymerization of silicones becomes even greater. When semiporous brush materials such as carbon graphite are used in conjunction with oils, it is recommended that the brush be impregnated with the oil before assembly. Several types of composite brushes are recommended. The most satisfactory (with lowest friction and electrical resistance) are comprised chiefly of silver; for example, silver/ $\text{MoS}_2$ /graphite (85/12.5/2.5 percent) and silver/ $\text{MoS}_2$ /copper

(85/12.5/2.5 percent). Many experimental electrical-contact materials are becoming available; their use should be based on full-scale testing, including wear tests and determination of volume resistivity.

#### **4.2.4.3 Motor Brushes**

Since the primary considerations in drive motors are to get efficient commutation and to minimize brush wear, the volume resistivity of the brushes may be permitted to increase two or three orders of magnitude over 85/15-percent silver/MoS<sub>2</sub> (~10<sup>-5</sup> ohm-cm). The Boeing "hot compact" 046-45 is particularly recommended (ref. 114).

### **4.3 Analysis**

The formulas and references provided in Section 2 should be used to determine actual friction forces, environmental effects, wear rates, and any interactions of lubrication with other systems.

#### **4.3.1 Friction**

The analysis of friction should cover all conditions to be encountered and at the highest levels of severity. Maximum friction should be determined on a basis of anticipated overloads plus increases caused by various types of degradation (Sec. 2.4.1.4). It is recommended that literature values of friction be used only to assess probable ranges for actual power requirements. In this sense, the following types of data are considered sufficiently accurate:

- Viscosity of oils and apparent viscosity of greases. (Variations in batches should be accounted for.)
- Manufacturer's friction values for antifriction bearings. (Determine the friction from the viscosity of the oil at lowest temperature to be encountered. For polymeric oils or mixtures, this viscosity should be measured in advance on a sample of the oil from which the predetermined volume has been evaporated.)
- General relationships between viscosity, speed, and film thickness in antifriction bearings or gears. (Note the change from a condition of full EHD to significant metallic contact with increasing temperature, when original viscosity is marginal.)

Coefficients of friction of bonded solid lubricants in vacuum should be multiplied by a factor of 2.0 to account for such variables as (1) pigment/resin ratio, (2) impurities and contaminants, and (3) degree of cure. To calculate power requirements in earth-ambient environments or high humidity, the value of  $f_k$  must be obtained at the applicable load and humidity. The effect of temperature on friction of MoS<sub>2</sub> (and bonded solid lubricants which are composed chiefly of MoS<sub>2</sub>) is negligible up to approximately 200° F (277.6K) in air, and negligible in vacuum.

Coefficients of friction for composites (Sec. 2.3.5) are generally a function of both load and speed. In cases where MoS<sub>2</sub> or a similar material is a significant part of the composition, the value of  $f_k$  will again be influenced by humidity (refs. 115 and 116), as in bonded solid lubricants. If literature values do not take these factors into consideration, they should not be used.

### 4.3.2 Wear

Before making calculations from available data or laboratory measurements, it should be ascertained that none of the following conditions will exist:

<u>Condition</u>	<u>Causing</u>
<ul style="list-style-type: none"> <li>• Excessive temperature</li> <li>• Excessive contact stress</li> </ul>	} Adhesive wear
<ul style="list-style-type: none"> <li>• Presence of dirt and grit</li> <li>• Rough bearing surface</li> </ul>	} Abrasive wear
<ul style="list-style-type: none"> <li>• Presence of chemically active substances (acids, alkalis, mercury, gallium, sulfur, phosphorus, oxidizers, etc.)</li> </ul>	} Corrosive wear

To estimate rates of wear, available data and literature should be used where possible. Actual wear should be measured by the most sophisticated means available that are consistent with accuracy. Statistical methods should be used to determine the significance of projections based on measurements.

### 4.3.3 Interactions

The possibility of deleterious effects on other systems by the lubrication system should be accounted for by analysis supported by tests where appropriate examples of

deleterious effects include fogging of optics, evolution of flammable or toxic material, and undesirable electrical properties.

## 4.4 Tests

### 4.4.1 Friction and Wear

If the literature data obtained are insufficient or if they were not obtained under conditions closely resembling the application under consideration, both the static and kinetic (dynamic) friction and the wear rate should be measured experimentally. The tests should be conducted under conditions that simulate environmental and operational conditions during earth and space phases as closely as possible. In simulating the space environment, the molecular species outgassing from all applicable substances should be considered as constituting part of the atmosphere. (See Sec. 2.5.1.)

Although many test machines give a reasonable approximation of friction, none is considered suitable for providing more than a comparative indication of wear life. For the screening of materials to select candidate lubricants, area-contact testers are more likely to be indicative of actual life than point- or line-contact testers, providing loads and speeds are carefully controlled. Examples of sliding friction testers are:

<u>Area Contact</u>	<u>Line Contact</u>	<u>Point Contact</u>
LFW-1 (low-pressure blocks)	LFW-1 (high-pressure blocks)	Shell 4-ball EP tester
Boeing galling tester	Falex tester	Shell 4-ball-wear tester
NAA plain bearing tester (ref. 115). Uses actual plain spherical bearings as test specimens.	Hohman A-6 friction and wear tester	Pin-and-disc (Bowden-Leben) tester
Hohman A-6 friction and wear tester		Rolling contact test machine (Industrial Techtonics, Inc.)
LFW-5 plain bearing tester (unidirectional motion only)		

The Bowden-Leben Tester is the only vacuum tester in general use, but it cannot be considered a "standard." Whatever apparatus is selected for making the final evaluation, the measurements must be made not only under simulated space conditions but under earth atmosphere to ensure that there will be no unsuspected anomaly that might jeopardize subsequent functioning of the part under consideration.

The thermal loads imposed by friction on the mechanism and surrounding structure should also be determined. On the basis of thermal load, both the outgassing of the lubricant and the effects of lubricant volatiles on adjacent systems should be determined.

The following techniques for measuring wear are recommended:

- Use of radioactive tracers
- Weight-difference determinations (on an analytical balance accurate to  $\pm 0.2$  mg)
- Weight of accumulated debris. This method requires painstaking steps to collect actual wear debris (rather than lubricant debris); it also requires a calibrated analytical balance, plus chemical analysis of the ingredients of the wear debris
- Metallographic examination

The following devices are recommended as useful auxiliaries to the foregoing techniques:

- Dimensional changes
- Surface profilometer
- Surface replica (nondestructive method utilizing microscopy or interferometry)
- Scanning electron microscope

Since the performance of a bearing will be affected by the cumulative debris formed, there are only two acceptable methods of assessing the performance after extended periods. The preferred method is to run the bearing under actual conditions for the total period; the other method is to run the bearing for some shorter period and introduce suitable artificial means for accelerating the wear without affecting other

parameters out of proportion (ref. 113). The wear may be projected beyond this point, but performance estimates cannot be considered truly reliable unless the mechanism is actually operated while containing the ultimate quantity of wear debris, simulated if necessary.

#### 4.4.2 Wear of Mechanisms

The following paragraphs present recommendations for accelerated-wear determinations of various mechanisms.

*Ball and Roller Bearings and Gears* (oil lubricated). The performance of lubricating oils in ball bearings should be assessed initially from their behavior in EHD film testers. Examples are the Rolling 4-Ball Tester and the Precision Rolling Disc Machine (refs. 16, 18, and 19). Both are in the category of expensive research tools and will not be available in many laboratories, but their data are considered significant for comparison of lubricants. The amount of wear is a function of the time that metallic asperities are in contact; therefore, it is desirable to have a long test-period. The test period can be increased as follows:

- From equation (5), more revolutions can be validly accumulated in a given test period by increasing the speed while lowering the viscosity to keep the product ( $N\nu$ ) constant; that is, the EHD film thickness should not be varied.
- Viscosity can be decreased by using lower molecular-weight materials of the same chemical family. This is preferred to raising the temperature, which might induce undesirable chemical reactions.
- Since the amount of wear generated is a direct function of the amount of time that metallic asperities are in contact, the rate of wear should be monitored by electrical-conductivity determinations. By this technique, the accumulation of wear debris in the bearing will also be sensed since metallic particles will interfere with the formation of the EHD film.

*Ball and Roller Bearings and Gears* (dry lubricated). The life of dry-lubricated bearings is not established; it depends on so many variables (thickness, uniformity, density, composition, etc.) that each application should be tested for enough time to establish confidence that its individual mission will be completed (ref. 117).

*Plain Bearings and Simple Sliding Surfaces* (dry lubricated). Plain sliding surfaces coated with solid lubricants may be related to wear-life data available from conforming surface-test machines operated in air. Vacuum operation should be confirmed by test.

The formation of debris is not nearly as great a concern as in the case of ball bearings, where unit loads are in the Hertz stress region.

*Plain Bearings and Simple Sliding Surfaces* (oil or grease lubricated). Again, testing is recommended, even in cases where special wick-feed mechanisms and reservoirs are provided. Where possible, oil-barrier coatings should be used to prevent migration of oil from the bearing areas.

## **4.5 System (Design) Practices**

In order to minimize wear, it is recommended that the following conditions prevail in all bearings (refs. 33 and 34):

- High elastic modulus
- High hardness
- Smooth, uniform surface topography

All of these conditions cannot be achieved at all times, but the surface roughness can usually be controlled by good machining practices. The run-in of lubricated bearings invariably decreases the surface roughness to some degree, provided severe wear conditions are not approached during this period.

### **4.5.1 Ball Bearings**

Cleanliness cannot be overstressed as a condition for lubricating ball bearings, regardless of the lubrication system selected. It is recommended that all processing be carried out in a Class 10 000 clean room, or better. Fluid and semifluid lubricants should be filtered with 0.45-micron millipore filters for oils and fine-mesh screens for greases. An alternate for greases is to specify dirt count based on microscopic inspection, FTM No. 3005.3 (from Federal Test Method Standard No. 791a).

When selecting a lubricating oil for a low-speed ball bearing, it is recommended that the EHD film thickness be held to a minimum of 4 microinches (0.1016 micron) (Sec. 2.3.3). Maximum load and temperature and the minimum speed to be encountered should be used in calculations. Ball-bearing surface roughnesses should be 1 microinch (0.0254 micron) or less for both balls and races.

No matter which lubrication system is used, it is desirable (though not always possible) to clean the bearings and relubricate them after a substantial run-in period. This step is

simply to remove any metal particles and/or all excessive solid-lubricant debris. It can be rather time-consuming when angular contact bearings are involved, and an analysis should be made to assess the possible risk of reassembly, after running in the bearings under several different loads to create a sufficiently wide ball path.

In those special missions where moderate to high loads are encountered or when angular-contact bearings are preloaded, it is recommended that Hertz stresses (between ball and race) be kept under 200 000 psi ( $1.38 \times 10^9$  N/m<sup>2</sup>).

In any oil or grease system for antifriction bearings, if the mission is extended (approximately six months), the bearing retainer should be fabricated from porous phenolic or nylon and impregnated with the oil (or base fluid, in the case of greases). Prior to use, the retainer should be thoroughly extracted (solvents) to remove any contaminants. It is also recommended that porous reservoirs be incorporated in the bearing housing to provide a long-term lubricant supply, available by evaporation or by migration to the bearing surfaces. Loss of lubricant from the bearings can be restricted to evaporation by incorporating a barrier coating such as Braycote 806 or the fluorochemical coating FX-106.

An alternate method of bringing oil from the reservoir to the bearing is by the use of a wick. Since little vacuum experience and no space data are available on use of wicks, it is recommended they be made as short as possible if used.

In the case of all transfer-lubricant systems (solid), it is recommended that, before assembly, the races be burnished with MoS<sub>2</sub> in an inert atmosphere (ref. 47). Excess powder should then be removed by rubbing the surfaces with clean, lint-free nylon or cotton cloth, or a clean cotton Q-tip.

#### **4.5.2 Gears**

When lubricative composites are to be used for gears, it is recommended that they be incorporated as idlers and loaded lightly against another pinion in a manner that will transfer lubricant through the train (refs. 45 and 46). The other (metal) gears should be burnished with MoS<sub>2</sub> before assembly. Because of imperfections in machining and assembly, it is recommended that gear trains be at least partially disassembled after the run-in period, and the wear debris removed.

Width of a composite idler gear should be the same as the gear it contacts. If too narrow, the transfer film will not adequately protect contacting metal surfaces. If too wide, the edges may break due to wear of the center portion. Shock and vibration tests should take into account the lower shock resistance of many composites. In addition,

thermal stresses must be considered, and any possible unbalance due to differences in wear rates (composites versus metals).

### **4.5.3 Plain-Bearing Surfaces**

The following design recommendations apply to the use of bonded solid lubricants in plain sliding applications:

- Surface roughness should be 32 microinches (0.8128 micron) or better, before coating.
- All sharp edges (which may contact the bearing area) should be eliminated. Radiused or bell-mouthed hole edges are preferred over chamfers.
- Contamination of coated surfaces should be prevented, particularly contamination by fluid or semifluid lubricants (organic substances).
- Corrosion protection should be provided by plating (with nickel or chrome) or by the use of stainless alloys.
- Both bearing surfaces should be lubricated for longer life.
- If only one surface can be lubricated, it should be the surface having the greatest contact area, the harder part, or the cooler part.

# APPENDIX

## TABLES CITED IN TEXT

TABLE I. —MECHANICAL FAILURES ON SPACE-VEHICLE DEVICES

Failure	Cause of failure	Cause Code (a)
Door latch stuck	Excessive wear of improperly selected bonded solid lubricant during ground tests	A
Camera thermal door mechanism stuck, preventing closure	Bonded solid lubricant (on anodized aluminum drive gears) wore excessively during ground tests	B
Potentiometer ceased to function	No lubricant on either slider or wiring windings	A
Solar array drive failed	Motor bearings became too hot (> 300°F); grease became stiff (due to oil evaporation) and jammed motor	A
Tape recorders failed early	Bearings experienced fretting corrosion or became contaminated with iron oxide from tape	C
Bearing failure in mechanical power transmission	Inadequate clearance caused overheating	A
Heat shield failed to separate; mission aborted	Bonded solid lubricant permitted rusting of hinge pins, resulting in excessive friction; actuator mechanism could not pull pins to release shields	B
Torque tube bearing inoperable	Spacecraft tumbled and became cold; turn-on time delayed; grease not suitable for low temperature	A
Antenna deployed late	Vibration of rolled-up antenna against cup (housing) removed gold plate, permitting cold welding to occur	B

<sup>a</sup>A — known cause

B — probable cause

C — possible cause

APPENDIX

TABLE II. -CONVERSION OF U.S. CUSTOMARY UNITS TO SI UNITS

To convert from:	To:	Multiply by:
centipoise	newton second/meter <sup>2</sup>	10 <sup>-3</sup>
centistoke	meter <sup>2</sup> /second	10 <sup>-6</sup>
Fahrenheit	kelvin	$t_K = (5/9)(t_F + 459.67)$
inch	meter	0.0254
mile (U.S. statute)	meter	1609.344
pound force/inch <sup>2</sup> (psi)	newton/meter <sup>2</sup>	6894.757

TABLE III. -RATIO OF VISCOSITY IN RELATION TO PRESSURE AND TEMPERATURE

Oil	Ratio of viscosity at 60 000 psig to viscosity at 0 psig at 210°F, centipoises
Diocyl sebacate	$\frac{72}{2.8}$
Paraffinic oil	$\frac{2300}{14}$
Gear-test oil	$\frac{5100}{8.0}$
Naphthenic oil (V.I. = 8)	$\frac{18\ 000}{8.3}$
Naphthenic oil (V.I. = -42)	$\frac{180\ 000}{16}$
Aromatic extract from above oil	$\frac{1\ 000\ 000}{13}$

## APPENDIX

### TABLE IV. —PROPERTIES OF SEVERAL OILS

Oil	Vapor pressure	Evaporation rate	Viscosities (centistokes)		Pour point (°F)	Radiation resistance (rads)	Useful temperature range in air (°F)
			100°F	210°F			
Molecularly distilled hydrocarbon oil (Apiezon C)	$1 \times 10^{-8}$ torr at 68°F		100	10	+15	$< 1 \times 10^8$	+20 to +120
Dimethyl polysiloxane (DC 200, Grade 100)		$< 2\%$ at 392°F after 48 hours (atm. press.)	80	40	-67	$< 1 \times 10^8$	-40 to +400
Chlorophenyl methyl polysiloxane (F-50)		0.4% at 302°F after 24 hours (atm. press.)	52	16	$< -100$	$< 1 \times 10^8$	-100 to 450
Tetraalkylortho- silicate ester (Coolanol 45, Monsanto)	4.7 torr at 400°F		12	4	$< -85$	$< 1 \times 10^8$	-65 to 400
Perfluoroalkyl polyether (Krytox 143 AC)	$1.3 \times 10^{-6}$ torr at 150°F	0.3% at 300°F after 6-1/2 hours (atm. press.)	270	26	-30	$< 1 \times 10^8$	-25 to 500
Fluoroalkyl polysiloxane (FS-1265, GR 300)	$1 \times 10^{-3}$ torr at 212°F	2% at 392°F after 4 hours (atm. press.)	160	30	-55	$< 1 \times 10^8$	-40 to 500
Polyphenyl ether (5 ring) (OS-124)	$7 \times 10^{-1}$ torr at 500°F		363	13	40	$< 1 \times 10^9$	+45 to 700
Chlorotrifluoro ethylene (KEL-F No. 3)	$1.28 \times 10^{-8}$ torr at 212°F		25	3	-45	$< 1 \times 10^8$	-40 to 500

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TABLE V. -PROPERTIES OF SEVERAL GREASES

Product designation	Base fluid	Thickener	Useful temp. range in air, °F	Specific gravity	Evaporation rate	Bearing performance	Oxidation stability
Versilube G-300	Versilube F50 (chlorophenyl methyl polysiloxane)	Lithium soap	-100 to +450	1.08	2% max. after 50 hr at 300°F (modified FS 791, method 321.1)	9600 hr at 8000 rpm, 1/6-lb radial load; R-3 bearing at 130-220°F; $2 \times 10^{-7}$ torr avg. pres.	2 psi drop in 50 hr at 300°F under 110 psig O <sub>2</sub> (ASTM D942)
Krytox 240 AC	Krytox 143 AC (per-fluoroalkyl polyether)	Vydax 1000 fluorocarbon telomer solids	-30 to +550	1.93	3% after 22 hr at 400°F; 6% after 22 hr at 500°F; (FTMS 791-351)	> 500 hr at 20 000 rpm at 400°F; > 2000 hr at 10 000 rpm at 500°F; > 500 hr at 10 000 rpm at 550°F (all per FTMS 791-333)	No pressure drop in 600 hr at 210°F (ASTM D942)
PL-631	Perfluoroalkyl polyether stocks	Ammeline (2, 4-diamino -6-hydroxy 1,3,5-triazine)	-30 to +500		0.4% after 22 hr at 400°F; 3.6% after 22 hr at 500°F; 23.2% after 22 hr at 600°F (per ASTM D972)	9117 hr at 10 000 rpm 3-lb radial load, R4 bearing at 300°F under $9 \times 10^{-11}$ torr average pres.	No data; probably very similar to Krytox 243AC
FS-1292	Fluorosilicone fluid	High melting-point soap	-40 to +400	1.28	7% after 22 hr at 400°F (per ASTM D972)	800 hr at 10 000 rpm, 5-lb radial or thrust load, 204 bearing at 400°F. 750 hr at 10 000 rpm, 197-lb thrust load, 204 bearing at 400°F (in air)	No pressure drop (psi) in 100 hr at 210°F (ASTMD942)

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TABLE VI.—PROPERTIES OF LAMELLAR SOLID LUBRICANTS

Lubricant	Formula	Specific gravity (g/cm <sup>3</sup> )	Coefficients of friction				Volume resistivity (ohm-cm)	Thermal stability		References
			Air		Inert gas	Vacuum		Air (°F)	Vacuum (°F)	
			High humidity	Low humidity						
Graphite	C	2.25	0.15 to 0.25	0.5	0.5	0.45	$2.64 \times 10^{-3}$	1000		99, 100, 116, 118
Molybdenum disulfide (natural)	MoS <sub>2</sub>	4.80	0.10 to 0.25	0.03 to 0.04	0.02 to 0.11 (N <sub>2</sub> ) 0.08 (A)	0.04 to 0.05	851	750	2000	100, 118, 119, 120, 121, 122
Molybdenum diselenide	MoSe <sub>2</sub>	6.9	0.22	0.10 to 0.22	0.11 (N <sub>2</sub> ) 0.12 (A)	0.04 to 0.08	$18.6 \times 10^{-3}$	800	1800	65, 100, 119, 121
Tungsten disulfide	WS <sub>2</sub>	7.5	Tests at 38% relative humidity and above lead to breakdown of the rubbed-on film in less than two hours	0.07 to 0.20	0.03	0.15	14.4	800	1900	115, 116, 118, 119, 122
Tungsten diselenide	WSe <sub>2</sub>	9.0	0.20	0.17	0.05 to 0.11 (N <sub>2</sub> ) 0.09 (A)	0.09	114	400	1700	65, 99, 119, 123
Niobium diselenide	NbSe <sub>2</sub>	6.25	0.12 to 0.20			0.08 to 0.20	$0.535 \times 10^{-3}$	2500	650	65, 100

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TABLE VII. —PROPERTIES OF SEVERAL BONDED SOLID LUBRICANTS

Type	Coating designation	Temperature range in air, °F	Pigment-binder ratio	Relative wear-life ratings in air (a)	References
MoS <sub>2</sub> / phenolic	Lubeco M-390 (example)	-352 to +350	1	2	40, 124
MoS <sub>2</sub> -Sb <sub>2</sub> O <sub>3</sub> / polyimide	MLR-2	-100 to +500	2	1	40, 125
MoS <sub>2</sub> (graphite) / sodium silicate	Electrofilm 2396 MLF-5	-300 to +750	2.5 to 4.0	3	79, 126
MoS <sub>2</sub> (graphite) / aluminum phosphate	Lubeco-905	-452 to +500	> 9	4	127, 128
	MLF-9		2.5	4	124
MoS <sub>2</sub> (graphite) / ceramic binder	Vitrolube 1220	-200 to +650	—	1	129
MoS <sub>2</sub> diffused into a soft, multilayered, plated metallic film	Hi-T-Lube	—	—	2	67

<sup>a</sup>Numerically in descending order of merit

APPENDIX

TABLE VIII. —PROPERTIES OF PTFE-BASE COMPOSITES (ref. 126)

Parameters	Teflon (PTFE)	Rulon A	Rulon A +5% MoS <sub>2</sub>	Rulon 175	Duroid 5813	Teflon +25% fiberglass	Rulon J (Teflon + polyimide)
Specific gravity	2.18	2.26	2.32	2.4-2.5	2.3	2.3-2.3	1.87
Tensile strength, psi (ASTM D1457)	1000	1450	1500	1200	4000-7000	3600	2000
Coefficient of thermal expansion (in./in./°F)	$6.9 \times 10^{-5}$	$3.3 \times 10^{-5}$	$\sim 3 \times 10^{-5}$	$\sim 3 \times 10^{-5}$	$15 \times 10^{-5}$	$3.6 \times 10^{-5}$	$5 \times 10^{-5}$ to $7 \times 10^{-5}$
Coefficient of friction (air)							
Vendor data $f_k$	—	0.12 to 0.19	—	—	0.02	0.16	0.15
Vendor data $f_s$	0.02	0.04 to 0.16	—	—	0.02	0.08	—
LFW-1 (vs S/S)							
200 psi - 1 ipm	0.03	0.02	0.21	0.09	0.08	—	0.09
200 psi - 33 ipm	0.095	0.08	0.21	0.10	0.14	—	0.14
2000 psi - 1 ipm	distortion	0.03	0.21	0.09	0.06	—	0.16
2000 psi - 33 ipm	distortion	0.09	0.21	0.10	0.11	—	0.17
Wear factor ( $\times 10^{10}$ ) <sup>b</sup>	25 000	2 to 4	—	—	9	8 to 10	3 to 7
Rated PV at 10 fpm	1000	4500	—	—	11 000	10 000	10 000
1000 fpm	1000	8000	—	—	17 500	16 000	—
Temperature range (°F)	-400/+550	-400/+550	-400/+550	-400/+550	-400/+550	-400/+550	-400/+550

<sup>a</sup>Thickness direction: longitudinal value is  $1.6 \times 10^{-5}$

<sup>b</sup>Wear factor of 1.0 = 1 in.<sup>3</sup> of wear in 1 hr with 1-lb load at 1 fpm

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TABLE IX. —PROPERTIES OF MISCELLANEOUS POLYMERIC COMPOSITES

Parameters	Nylon 6/6 +15% PTFE +30% fiberglass	Polycarbonate +15% PTFE +30% fiberglass	Meldin PI-30X polyimide +30% PTFE	Vespel SP-3 polyimide +15% MoS <sub>2</sub>
Specific gravity	1.51	1.55	1.58	1.59
Tensile strength, psi (ASTM D1457)	22 000	12 000	3320	10 500
Coefficient of thermal expansion, in./in./°F	1.6 × 10 <sup>-5</sup>	1.1 × 10 <sup>-5</sup>	—	—
Coefficient of friction (air)				
Vendor data f <sub>k</sub>	0.26		0.20 to 0.25	0.11 to 0.37
Vendor data f <sub>s</sub>	0.19			
LFW-1 (vs. S/S)				
200 psi - 1 ipm		0.15	--	0.10
200 psi - 33 ipm		0.15	--	0.14
2000 psi - 1 ipm	--	0.06	--	0.25
2000 psi - 33 ipm	--	0.06	--	0.24
Wear factor (× 10 <sup>10</sup> ) <sup>a</sup>	--	30	--	--
Rated PV at 10 fpm	17 500	27 500	--	
1000 fpm	13 000	13 000		
Temperature range, °F	--	--	-400/+500	-400/+500

<sup>a</sup>Wear factor of 1.0 = 1 in.<sup>3</sup> of wear in 1 hr with 1-lb load at 1 fpm

## APPENDIX

**TABLE X. – PROPERTIES OF SOME METAL-BASE COMPOSITES**

Parameters	Boeing lubricant compacts				Westinghouse compacts		
	108	108M	046-52	046-45	Ag/PTFE/MoS <sub>2</sub>	Ga/In/WSe <sub>2</sub>	
Density, g/cc at 77°F	5.40	6.45	7.77	5.65	–	–	
Thermal conductivity, <sup>a</sup> BTU/hr-ft <sup>2</sup> · °F-ft.	16	14	12	12	–	–	
Coefficient of thermal expansion, in./in./°F	3.5 x 10 <sup>-6</sup> (-420 to 1800°F)	3.7 x 10 <sup>-6</sup> (RT <sup>b</sup> to 1800°F)	3.9 x 10 <sup>-6</sup> (RT <sup>b</sup> to 1800°F)	3.7 x 10 <sup>-6</sup> (RT <sup>b</sup> to 1800°F)	–	–	
Volume resistivity, ohm-cm	4.8 x 10 <sup>-2</sup>	2.3 x 10 <sup>-3</sup>	5.1 x 10 <sup>-4</sup>	1.02 x 10 <sup>-2</sup>	–	–	
Operating temperature range (°F)							
	In nonoxidizing atmosphere, inert gas, or vacuum	-420 to 1500	-420 to 2000	-420 to 2500	-420 to 2500	–	–
	In air	-420 to 750	-420 to 750	-420 to 750	-420 to 750	–	-225 to 1500
Tensile strength, psi	2700 <sup>a</sup>	4500 <sup>a</sup>	8400 <sup>a</sup>	4500	–	–	
Compressive strength, psi	20 000	35 100	175 000	27 800	–	20 000	
Maximum surface speed, ft/min.	12 000	3 000	500	12 000	–	–	
Maximum operating stress, psi	2 000	5 000	15 000	2 000	–	–	
PV at value (psi x ft/sec)	24 x 10 <sup>6</sup>	15 x 10 <sup>6</sup>	75 x 10 <sup>5</sup>	24 x 10 <sup>6</sup>	–	–	
Friction coefficient range	In air	~0.3	~0.3	~0.3	0.09 to 0.31	0.16 to 0.21	0.15 to 0.20
	In vacuum	–	–	–	0.02 to 0.07	0.08 to 0.16	0.06 to 0.18
References	45, 99	45, 99	45, 99	45, 96, 98, 111	111	97, 111	

<sup>a</sup>Estimated value

<sup>b</sup>Room temperature

## APPENDIX

TABLE XI. -CHARACTERISTICS OF THE SPACE ENVIRONMENT (ref. 130)

Mission level	Altitude, miles	Pressure, torr	Concentration, particles/cm <sup>3</sup>	Dominant composition								
Sea level	0	760	$9.17 \times 10^{16}$	78%N, 21%O <sub>2</sub> , 1%Ar								
Low-earth orbital	100	$3 \times 10^{-6}$	$3.67 \times 10^7$	N <sub>2</sub> , O <sub>2</sub> , O, O <sup>+</sup>								
	600	$10^{-10}$	$3.67 \times 10^3$	N, O, O <sup>+</sup> , H								
High-earth orbital	4000	$4 \times 10^{-13}$	3.67	H, H <sup>+</sup> , e <sup>-</sup>								
Lunar	--	$2 \times 10^{-13}$	$6.1 \times 10^{-2}$ $3.05 \times 10^{-1}$ to	H <sup>+</sup> , e <sup>-</sup>								
Interplanetary	> 250 000	$10^{-15}$	--	H <sup>+</sup> , e <sup>-</sup>								
<p><b>SYMBOLS</b></p> <table style="width: 100%; border: none;"> <tr> <td style="width: 50%;">N<sub>2</sub> nitrogen gas</td> <td style="width: 50%;">O<sup>+</sup> oxygen ion</td> </tr> <tr> <td>O<sub>2</sub> oxygen gas</td> <td>H hydrogen atom</td> </tr> <tr> <td>Ar argon gas</td> <td>H<sup>+</sup> hydrogen ion (proton)</td> </tr> <tr> <td>O oxygen atom</td> <td>e<sup>-</sup> electron</td> </tr> </table>					N <sub>2</sub> nitrogen gas	O <sup>+</sup> oxygen ion	O <sub>2</sub> oxygen gas	H hydrogen atom	Ar argon gas	H <sup>+</sup> hydrogen ion (proton)	O oxygen atom	e <sup>-</sup> electron
N <sub>2</sub> nitrogen gas	O <sup>+</sup> oxygen ion											
O <sub>2</sub> oxygen gas	H hydrogen atom											
Ar argon gas	H <sup>+</sup> hydrogen ion (proton)											
O oxygen atom	e <sup>-</sup> electron											

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TABLE XII. –SOLID LUBRICANTS USED WITH BALL BEARINGS

Solid lubricant form	Comments	Used in space?
Rubbed-on powder (MoS <sub>2</sub> )	Very limited life, compared to other forms; difficult to obtain reliable and repeatable data	Yes
Bonded solid lubricants	Generally too thick a film for ball bearings; apply to races only; use very thin (<0.0001 in.) coatings and burnish prior to assembly	Yes
Inorganic bonded	Lower volatility, slightly lower friction than organic; higher temperature capability than organic-bonded coatings; use to fill machined-out reservoirs in retainers (ref. 44)	Yes
Organic bonded	Pigment/resin ratio (weight) should be greater than 1.0; has longer wear life than inorganic	Yes
“In-situ”	Processes lacking in control; varying wear life; recommend 100% inspection	Yes
Transfer (lubricative composites)	Mechanism and rate of transfer are fully understood; materials must be carefully machined to remove “burrs”; must be inspected at 40x magnification	Yes
Duroid 5813	Anisotropic with respect to strength and coefficient of thermal expansion; extensively used in ball bearings (retainers)	Yes
Rulon A +5% MoS <sub>2</sub>	(No other trade name) Not anisotropic, but weaker than Duroid 5813	Yes
Rulon 175	Similar to above, but lower $f_k$ ; less experience; (used as long-life friction brake)	No
Gallium-Indium/WSe <sub>2</sub> (Westing-house)	Very low friction, but weak and crumbly, especially at edges; must be enclosed in a shroud (rings pressed into holes); extensive experience with liquid hydrogen at high speeds	No
“Hot compacts” (Boeing)	Very low friction in vacuum, but $f_k$ in air 10 times as high; very brittle, but hard; strong in compression; must be enclosed in shroud	Yes

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TABLE XIII.—BRAND NAMES

Brand name	Manufacturer	Description
BarTemp	Barden Bearing Co.	Ball bearing utilizing a Duroid 5813 ball retainer
Duroid 5813	Rogers Corp. Rogers, Conn.	Composite of PTFE fortified with fiberglass and containing 15% MoS <sub>2</sub>
Rulon A	Dixon Industries Bristol, R.I.	PTFE (70%) + inert filler
Rulon A + 5% MoS <sub>2</sub>	Dixon Industries Bristol, R.I.	Rulon A + 5% MoS <sub>2</sub>
Rulon 175	Dixon Industries Bristol, R.I.	PTFE + inert filler + 15% MoS <sub>2</sub>
Rulon J	Dixon Industries Bristol, R.I.	Composite of PTFE filled with polyimide
Fabroid	Transport Dynamics Santa Ana, Calif.	Phenolic-bonded woven PTFE cloth
NPI-14	National Process Industries South Gate, Calif.	Phenolic-bonded MoS <sub>2</sub> /graphite (meets MIL-L-8937)
NPI-425	National Process Industries South Gate, Calif.	Polyimide-bonded MoS <sub>2</sub> /Sb <sub>2</sub> O <sub>3</sub>
Vitrolube 1220	National Process Industries South Gate, Calif.	Ceramic-bonded MoS <sub>2</sub> /graphite
Lubeco M390	Lubeco Inc. Compton, Calif.	Phenolic-bonded MoS <sub>2</sub>
Lubeco 905	Lubeco Inc. Compton, Calif.	"Inorganic"-bonded MoS <sub>2</sub> + proprietary pigments
Hi-T-Lube	General Magnaplate Corp.	MoS <sub>2</sub> diffused into soft metal plate
Electrofilm 2396	Electrofilm, Inc. No. Hollywood, Calif.	Sodium silicate-bonded MoS <sub>2</sub> /graphite
Molykote E3C	Dow Corning Corp. Midland, Mich.	Conversion Coating of MoS <sub>2</sub>
FS-1292	Dow Corning Corp. Midland, Mich.	Grease based on fluorosilicone oil + high-melting soap
Molykote X-15	Dow Corning Corp. Midland, Mich.	Sodium silicate bonded MoS <sub>2</sub> /graphite
FS-1265	Dow Corning Corp. Midland, Mich.	Fluorosilicone polymer
DC-560	Dow Corning Corp. Midland, Mich.	Chlorophenyl silicone
Versilube F50	General Electric Waterford, N.Y.	Methylchlorophenyl silicone polymer
Versilube G-300	General Electric Waterford, N.Y.	Grease based on F50

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TABLE XIII.—BRAND NAMES — Concluded

Brand name	Manufacturer	Description
PL-631	Dupont Wilmington, Delaware	Grease based on perfluoroalkyl polyether and ammeline
Krytox 143 AC	DuPont Wilmington, Delaware	Perfluoroalkyl polyether
Krytox 240 AC	DuPont Wilmington, Delaware	Grease based on Krytox 143 AC, thickened with TFE
Coolanol 45	Monsanto St. Louis, Mo.	Tetraalkylorthosilicate ester
OS-124	Monsanto St. Louis, Mo.	5-Ring polyphenyl ether
KEL-F No. 3	Minnesota Mining and Manufacturing Co. Minneapolis, Minn.	Chlorotrifluoroethylene
Apiezon C	James G. Biddle Co. <sup>a</sup> Plymouth Meeting, Penn.	Molecular distilled hydrocarbon fraction

<sup>a</sup>Distributor in USA

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