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ABSTRACT

Some anticipated and present friction and wear problems in aircraft are discussed and potential approaches for mitigation of these problems are outlined. The status of atomistic and continuum methods for defining surfaces and modes of failure is described. Wear processes are explored in terms of physical and chemical considerations and basic types of wear mechanics are characterized. Solid and liquid lubricant status is generalized. Lubrication by boundary, elastohydrodynamic, and fluid films are conceptually described. Lubrication systems are discussed in regard to future applications. Attention is given to particular problems of specific mechanical components (i.e., bearings, seals, gears, fasteners, etc.). Some approaches are suggested for the study of component wear due to fretting, erosion, scuffing, pitting, and other forms of damage.

INTRODUCTION

In lubrication and wear we are concerned with load transfer from one body to another through their mating surfaces. Loads are transmitted from one body to another by three modes. These are load transfer through: (a) fluid films only, (b) combined fluid films and solid surfaces, and (c) solid surfaces only. This broad classification is not restricted to surfaces with obvious relative motion such as bearings. It also includes lubrication of space vehicle static joints that must be protected from cold welding. (Important subdivisions of the preceding classification are boundary and elastohydrodynamic lubrication.)

Maintenance records of aircraft engines reveal that component wear is one of the major reasons for nonscheduled overhaul. Other major reasons are foreign object damage and corrosion. Corrosion, in a broad sense, is a wear problem. In an aircraft there are literally thousands of parts subject to wear; such as brakes, bearings, seals, gears, actuators, pumps, flight control systems, instruments, splines, and even ''static'' joints. (A large aircraft may contain up to 4000 bearings in its various systems.)

Advances in the understanding of the mechanisms of friction, wear, and lubrication provides a basis for selection of design tolerances and materials. In some cases, lubrication and wear considerations affect basic design arrangements. For example, straddle mounted gears are preferred over cantilevered gears because deflections cause nonuniform gear loading; or, the outer race of a bearing may be flanged and bolted to prevent fretting wear of the static joint between the bearing and its mating housing.

The overall objective of this paper is to identify wear problem areas that might benefit from basic or applied research. To this end, the status of some pertinent areas of friction and wear are summarized. Details are presented in certain areas in order to establish a broad basis of reference.

Subject matter covered includes surface characterization, friction and wear in dry sliding, solid lubricants, elastohydrodynamic lubrication, and components such as bearings, gears and seals. The inherent interdisciplinary nature of friction and wear problems is stressed.

The Solid and Its Surface

Bulk structure. - The surface is the termination of a bulk material. Thus bulk properties are of direct interest in the study of friction and wear of surfaces. The materials of interest in aircraft design are crystalline solids such as aluminum alloys, titanium, graphite, etc. The understanding of the structure of bulk materials has been advanced by techniques such as X-ray and electron diffraction. (These are useful tools because of the periodic order and structural symmetry of the crystal.) As an example, studies reported in metallurgical literature have contributed significantly to the understanding of crystal transformation, work hardening, fatigue, creep and grain boundaries structure.

Also other studies have provided significant insight into the effect of dislocations (crystal defects in periodicity) on material shear and plasticity. Since friction and wear are strong functions of bulk properties, these crystallographic studies are important in constructing basic friction and wear mechanisms.

Friction and wear studies can be based on an atomistic or a continuum (macroscopic) approach (Fig. 1). Both have much to offer. The continuum approach has been very useful and, for example, is the basis upon which is founded disciplines such as elasticity and hydrodynamic lubrication. (The whole of science of areodynamics rests on the continuum approach.) Its contributions are very evident and used daily. The continuum mechanics has been developed to include anisotropic materials, dislocation movement, plasticity and multi-polar media (media with microstructure). All these areas of study are important in friction and wear. Thus these developments are a potentially useful tool for understanding and modeling the friction and wear processes. Some outstanding references in this field are publications of Truesdell, Toupin and Noll (Refs. 1 and 2).

Surface structure. - Since a surface represents an abrupt termination of the periodicity of the crystalline structure, the surface atoms are chemically active and thus have a great tendency to react with their environment. Thus metals usually have surface contaiminants such as oxide films and adsorbed gases and vapor (especially hydrocarbons). Friction and wear are both very sensitive to surface contaminants (Ref. 3). And from a fundamental standpoint, it is desirable to conduct studies without these contaminants, that is on ''clean'' surfaces. Development of vacuum techniques permitted such studies. An example of these studies is Ref. 4 which contains an extensive discussion on theoretical and experimental aspects of friction, wear and lubrication in vacuum.

The various methods for obtaining a clean surface in a vacuum are evaporation, sputtering, heating, ion bombardment, field desorption and cleavage (Ref 4). (See Ref. 5 for a discussion of these cleaning techniques.) With atomically clean surfaces in an ultra high vacuum environment, it is possible to study the effect of crystal structure on friction and wear. As an example, in the work reported by Ref. 4, wear specimens (riders) were made from single crystals of titanium and orientated to slide on the based plane and on one of the prismatic planes. Figure 2 shows how the coefficient of friction varies with crystal orientation of titanium. It was also shown that the coefficient of friction depends on the slip direction as well as the slip plane. This anisotropic behavior was shown (Ref. 4) to exist in other materials.

Experimental techniques for studying clean surfaces include field-emission microsocpy. In this technique a high electric field is applied across a sharp pointed specimen and a phosphorcoated screen electrode (usually a transparent film of tin oxide). Electrons emitted from the atoms on the specimen tip are accelerated by the field and form a magnified (10^5 to 10^7 times) image on the screen. Field-emission (Fig. 3) techniques have been very successfully used in studies of gas adsorption on metal surfaces. For example it has been shown that adsorption of oxygen on a nickel surface can cause pronounced surface structural rearrangements (Ref. 6).

Field-ion microscopy is similar to field emission except the field polarity is reversed. The chamber contains an ''imagine gas'' (usually Helium) which is ionized in the vicinity of the specimen tip and then accelerated by the field toward the phosphor coated screen. Thus an image of the specimen tip is produced. This technique allows observation of the surface atomic structure (Fig. 4). Individual interstitial atom, vacancies, dislocations and other lattice defects have been observed. Adsorption, desorption and surface diffusion are other areas in which the field-ion techniques are very powerful tools.

Reference 6 points out that of all the techniques for study of surface structure, LEED (low energy electron diffraction) holds the most promise. In this technique the surface structure is determined from the diffraction pattern of low energy electrons which penetrate only one or two atomic layers deep. Figure 5 is a sketch of a LEED apparatus. Electrons impinge on the crystal specimen and the elastically scattered electrons form an image on a fluorescent screen. This image can be viewed directly. An example of the images produced is shown in Fig. 6. A diffraction pattern of a clean copper surface is shown in Fig. 6(a) (from Ref. 4). The surface shown is the (100) plane. Figures 6(b) to (d) show the same surface after contact by gold. Gold atoms account for the additional diffraction spots not seen in Fig. 6(a). Thus gold adheres to copper, and the gold atoms are at preferred locations in an ordered fashion. Studies such as these have contributed to understanding friction and wear of surfaces.

Surface geometry. - Engineering surfaces are far from atomically flat. (Although atomically flat surfaces have been produced by cleaverage of such lamellar solids as mica, Ref. 7.) For instance, a fine ground surface will have a roughness of many thousands of atomic layers in height. Therefore two surfaces will contact only at many small high spots (Fig. 7). The actual area of contact may be only a very small fraction (such as 1/10,000) of the apparent area of contact. Characterization of this solid surface contact remains as a major problem area.

The usual approach of measuring surface texture has been the stylus profilometer (Fig. 8). Vertical magnifications to 100 000 are possible. Usually the vertical magnification is in the range of 100 times the horizontal magnification. The result is a highly distorted, but useful, picture of the surface.

A numerical method of surface description (Ref. 8) uses a stylus profilometer with the output fed into a digital computer. In a typical observation, 1800 height readings are recorded from a 0.30 centimeter (0.12 in.) long trace. Profile readings are taken over a series of parallel paths (see Fig. 8), thus in area is covered. From this input, the computer is programmed to calculate center line average surface roughness, identify peaks, calculate spacing in peaks and distrubution. The principle advantage of the method is the determination of roughness height distribution. Also topographic contour maps showing peaks and valleys can be made. It was found that many engineering surfaces had a nearly Gaussian distribution of roughness. This was an important finding for mathematical modeling. Continued effort of this type will aid in surface characterization.

Dry Sliding Contact

Theories of friction. - As previously stated, even if the two surfaces have a fine grind finish, they are not flat from an atomic viewpoint and the surfaces touch in a random manner at a number of asperities (Fig. 9). If the load is increased, the total area of asperity contact increases to a value capable of supporting the applied load.

From Leonardo de Vinci we have the first recorded laws of friction. With his keen insight he was able to formulate some fundamental generalities about friction that are true within certain limits. He concluded that every frictional body has a resistance of friction equal to 1/4 of its weight. Some 200 years later, Amontons published a paper in which he described the two main laws of friction. These laws are: first, the frictional force is proportional to the load and he said for most surfaces this frictional force is equal to 1/3 the load. The second law is that the friction is independent of the apparent area of contact.

In regard to the asperities, Amontons considered two possible models. First, the asperities are rigid and the friction is due to lifting the asperities, one over the other. This is equivalent to dragging a body up a small slope. In the second model the asperities were considered elastic and the friction was equal to the work required to press the asperities down. Coulomb, who worked almost 100 years later, concluded that friction was due to interlocking asperities. He visualized these asperities as bending, breaking and lifting over one another.

The concept that adhesion was an important part of friction was introduced by Desaguliers some 25 years after Amontons' original paper. He reasoned that if the surface asperity hypothesis were true, then smoother surfaces ought to have less friction. Yet he found that polishing surfaces increased the friction. Thus he concluded that adhesion became stronger as the surfaces of the bodies were brought nearer.

The major contribution of these early workers was that contact between two surfaces occurs between the asperities at many small discrete spots. Further significant contributions to the understanding of friction had to wait until modern times when Merchant (Ref. 9) and Bowden and Tabor (Ref. 3) showed the importance of adhesion in friction and wear.

Under the adhesion theory, the area of contact (see Fig. 9) is so small that yield stress of at least some of the asperities is reached. The elastic and plastic flow of the material is, in effect, a "cleaning" action within the contact area (Ref. 10). Surface oxides are broken up and, locally, bare metals come into contact. Since the stress is high, "cold welding" (adhesion) can occur at the junctions. Relative surface motion requires that these welded junctions be broken in the weld itself or in the bulk metal. (A break in the bulk metal appears as a wear particle or as transferred metal Fig. 9.)

In addition to shearing, ploughing of a hard asperity through a softer metal is part of the total frictional force. Thus, as given in Ref. 10, the frictional force is

$$F = S + P = \text{Shear} + \text{Ploughing} = As + A'p \tag{1}$$

where

A real area of contact
A' ploughing area
p flow pressure
S shear strength of junction

Contact area:

$$A = \frac{\text{Load}}{\text{Flow Pressure}} = \frac{W}{p} \tag{2}$$

Friction coefficient:

$$f = \frac{\text{Friction}}{\text{Load}} = \frac{As}{W} + \frac{A'p}{W} = \frac{s}{p} + \frac{A'p}{W} \tag{3}$$

When the ploughing term is negligible,

$$f = \frac{s}{p} = \frac{\text{Shear strength}}{\text{Flow pressure}} \tag{4}$$

In addition to adhesive wear, there are other wear mechanisms (Table I). These are: (a) corrosive wear (corrosion damage in helicopter transmissions is prevalent, also some extreme pressure lubricants function by a corrosive wear mechanism). (b) abrasive wear (abrasion can be caused by a hard asperity ploughing thru a softer surface or by a grit or abrasive particle caught between

2 surfaces), (c) fatigue (this is a failure mechanism of concern in gears and bearings and is covered in the sections on gears and bearings), (d) erosion (this type of damage is caused by particles entrained in a gas stream and is often found in compressor blades and in labyrinth seals.

Aircraft brakes. - From a broad view, dry sliding contact of solid surfaces includes many different combinations of mating couples. Some of these mating surface couples are: metals in vacuum, oxidized metals, solid film lubricants and viscoplastic materials. In aircraft, some components that experience dry sliding contact are brakes, engine shaft seals, air bleed door seals and bearings. As an example, Fig. 10 shows several elements of an aircraft disk brake after being run in a simulated rejected takeoff. The disks are welded together. This example is offered to illustrate a beneficial use of dry sliding contact. (It might be argued that this was not dry sliding since liquid metal was being produced during the braking action.)

Although aircraft brakes have achieved a high energy absorption per pound of weight, considerable work is continuing to improve life and reduce weight. As an example of energy absorption, Ref. 11 cites the DC-8-60 which has a kinetic energy of 55×10^6 Newton meters (40.6×10^6 foot pounds) at takeoff speed of 160 knots. This energy level was obtained by engines weighing 12 400 kg (27 200 pounds) and on a rejected takeoff must be absorbed by brakes weighing 1030 kg (2260 pounds). As compared to engine weight this is 12 times more energy per pound, however, the brakes only have to live thru one rejected takeoff. (In a rejected takeoff it is not uncommon to melt the brakes into one solid mass.) Figure 11 from Ref. 11 shows how aircraft braking energy requirement has increased from 1950 to present.

The reciprocating engine aircraft such as the DC-3 used resin bonded asbestos brake pads or disks. These were limited in life at the higher kinetic energies of the jet aircraft and sintered metal brake materials had to be used. Some earlier disk brakes for jet aircraft were often steel disks on which was sintered a copper matrix with alloying elements of oxides and graphite to control the friction and provide braking smoothness. (The subject of brake smoothness is of concern since the brake can act as a forcing function causing vibration of the entire landing gear system.) As the braking energy input increased, the pad type disk brake (Fig. 12) was introduced to alleviate problems of disk warping and chipping of the sintered facing due to differential thermal expansion. In the pad type, sintered copper matrix material in a steel cup is riveted to a steel disk. The riveting attachment mitigates the problem of differential thermal expansion between the sintered material and the steel carrying disk, also the pads have some capability for angular alignment to assembly deflection. (Integral sintered metal and steel disks do not have this alignment ability.) Most modern brakes consist of pad type disks.

In military aircraft the weight of the sintered metal brake assembly is a significant penalty. Therefore, current emphasis is on development of carbon disk brakes. Such a brake would have alternate rotor and stator disks made from carbon-graphite materials, and some success in carbon brake development has been reported. Carbon graphite is a promising material since it has low density and high heat capacity. Table II shows that carbon has a heat sink capability about 3 times that of steel and copper, which are major constituents in common brake materials. However, carbon brakes are a relatively new area, and friction and wear of carbon-graphite brake materials have not been well established. Fundamental information is needed on the following:

1. Coefficient of friction and wear of various carbon-graphite materials. (Should the plate contain a high percentage of carbon or of graphite?)

2. Role of oxidation and oxidation inhibitors during stopping and during cool down after stopping.

3. Effect of wear particles on the friction process. (Are slots needed to get rid of wear debris?)

4. What is the fundamental mechanism of carbon-graphite friction and wear process?

5. Should both plates be of the same carbon-graphite formulation?

6. What is the thermal history of the braking process and how does this effect friction and wear?

In answering these questions the physical requirements of the brake disk should be kept in mind. First, a certain minimum impact strength is required and the disks must be able to withstand brake assembly deflection. Second, high heat sink capability is an essential property of a brake material since the majority of the heat is stored during the braking process. The vast majority of the heat dissipation takes place after stopping; and with frequent takeoff and landings, brake cooling could present a problem.

Carbon graphite. - Carbon-graphite materials are also used in engine shaft seals (Ref. 12). Operation is usually without liquid lubrication. Use of graphite for high temperature airframe bearings is also a possibility. Thus friction and wear of carbon-graphite materials is likely to become of increasing importance in aircraft design and maintenance.

Graphite has a hexagonal crystal structure as illustrated in Fig. 13. The atoms of each plane are held together by strong valance bonds and can be thought of as a single molecule (Ref. 13). A quantum mechanical calculation of the interlayer force gives an energy of 2500 ergs/cm^2 (Ref. 14). This is in fair agreement with an experimental value of 1750 ergs/cm^2 in high vacuum. Thus graphite is not sheared easily under all conditions. This is demonstrated by the high friction coefficient measured in vacuum (Ref. 15). Data from Ref. 15 are shown in Fig. 14. Note that the friction is markedly lower when air is admitted into the vacuum chamber. Further Refs. 16 and 17 report that graphite is a good lubricant in the presence of oxygen or water vapor. The hypothesis is that there is an interaction between the oxygen molecule and the electrons that contribute to binding the graphite layer together. Thus the binding is weakened and easy shear takes place.

It was found (Ref. 16) that under lubricating conditions a graphite film 100 A in thickness was formed on the copper disk with basal planes oriented almost parallel to the surface of the copper. And Ref. 18 reports that if the disk surface was polished and a graphite brush was oriented with its basal planes parallel to the metal surface, then low wear was possible even in dry nitrogen. However, with rougher surfaces and brushes with randomly oriented graphite it was necessary to use oxygen or condensible vapors to establish a low wear rate. This confirmed the work of Ref. 16 (except for the low wear with polished and oriented basal planes in dry nitrogen.)

In Ref. 19 it is suggested that under smooth sliding conditions in nitrogen gas, the nitrogen absorbed on the graphite surfaces would prevent their adhesion. Thus the wear would be small and no oxygen or vapor would be necessary. As surface roughness increases the orientation of the graphite layer becomes more random, then wear debris is produced and there is an increase in forces that tend to break the bond between the graphite layer and metal. In order to prevent a rupture of this graphite-metal bond it is necessary to weaken the interlayer bonding of the graphite planes so that shear occurs in the graphite planes and not at the graphite-metal bond. Also it is necessary to weaken the adhesion between the graphite body and planes of the graphite layer which are attached to the metal. Thus oxygen or some condensible molecule or vapor must be present to absorb on the exposed graphite surfaces to prevent readherence of the freshly sheared particles.

At this point it may be well to sum up the mechanism of graphite lubrication. The factors which are important are:

1. Adhesion between the metal surface and the transferred graphite film.

2. The orientation of the transferred graphite film.

3. The cohesion between the crystallite planes of this transferred graphite film.

4. The adhesion between the transferred graphite film and the graphite body. Thus the mechanism of graphite lubrication may be viewed as a graphite body sliding over a very thin orientated transferred graphite film.

Fretting. - No discussion on wear would be complete without a discussion on fretting. Fretting is associated with oscillatory movements of two surfaces in contact. Therefore an assembly of mechanical components, such as a gas turbine is often subjected to fretting wear. As an example, the desire to reduce weight leads to reducing housing wall thicknesses, and as the wall thickness decreases, its rigidity decreases. This leads to increased possibility of fretting due to movement between the bearing outside diameter and the housing bore.

Also differences in thermal response of the bearing race and the housing sometimes results in loss of fit up. The bearing can then move and this leads to fretting. Another common problem is fretting of the blade attachments caused by vibration of the stator blades. This often is mitigated through use of vibration dampers which, since they operate by friction damping, also fret and wear. The fuel pump spline is another area where fretting is often observed. In engines which use shaft face seals, fretting wear of the secondary ring is often a problem. In addition to these and others in the engines, the airframe contains numerous bearings and joints which are subject to fretting.

Fretting is different from other wear mechanisms in that debris remains in the wear area and contributes to wear. It has been concluded (Ref. 20) that the fretting process can be divided into three stages: (a) initial stage of oxide film wear, adhesion of mating surface asperities, and metal transfer, (b) debris generation stage which includes oxidation of wear debris if oxygen is present, and (c) steady state due to wear particles. It has been shown that fretting will occur under nonoxidizing as well as oxidizing conditions. Further, fully oxidized materials have been fretted. Thus the oxidation of wear particles is not a necessary condition for fretting but a secondary factor.

In Ref. 21 it is suggested that one method of reducing fretting in applications such as bolted flanges is to increase the normal load between the two surfaces. The effect of this increased pressure is to eliminate relative motion between the surfaces. This fact is soon appreciated by engineers who select interference fits for mounting rolling contact bearings. As mentioned previously this interference fit can be loosened by relative thermal growth and then movements and fretting wear can occur. Some investigators report that liquid lubricants and greases reduce or postpone fretting but are not completely effective in eliminating it. Conflicting results are reported on the effectiveness of various additives, oil types and viscosities. The general observation is that the lubricant is effective to the extent that it penetrates between the loaded areas. In high speed coupling applications it is generally accepted that oil will prevent fretting more effectively than grease. With grease, the action of centrifugal force separates the oil from the fillers and inhibits effective lubrication.

Phosphate and oxide coatings have been shown to be effective in preventing fretting but best success is obtained when used in conjunction with a solid or liquid lubricant (Ref. 22). Results suggest that effectiveness of MoS_2 depends upon the application. Dry MoS_2 was reported ineffective in preventing fretting on ball bearings (Ref. 23). Of the various types of metal coatings, nickel plate has been found to be superior (Ref. 24). This observation on nickel tends to confirm a hypothesis that fretting is reduced with metals in which the ratio of oxide hardness to metal hardness is small.

The preceding comments on fretting are only a small sampling of the literature that exists on the subject. No two studies are alike; they contain a wide variety of materials, geometries, lubricants (both liquid and solid), additives, surface roughnesses, load, frequencies, coatings, atmospheres, slip amplitudes and temperatures. Some basic principles (as pointed out) have been established. Work at higher temperatures would provide useful information for advanced applications. Some studies on an engineering level to determine the effectiveness of thin coatings may prove fruitful. Nickel and high temperature polymer coatings are one approach. On a fundamental level, the first two stages of metal fretting are the most significant in many cases. It would seem that an investigation could center on the initial phases and thus save experimental time, and more importantly, save the experiment from being damaged by the third phase which is wear due to entraped particles.

Solid Lubricants

Applications and desired capabilities. - Present aircraft contain many mechanisms that utilize solid lubricants. For example, bevel gears on a radar dome drive, cockpit quick disconnect couplings (vital in pilot ejection), control cables and airframe bearings. A large aircraft can contain about 4000 bearings (engine, airframe, instrument, auxiliary) many of which are lubricated with solids. Solid lubricants are becoming increasingly important because of increased flight speeds. At Mach 3 temperatures are near 315° C (600° F); and near 650° C (1200° F) at Mach 4 (Ref. 25). Also, space shuttle control surfaces (Fig. 15) and their associated bearings are expected to reach 760° to 871° C (1400° to 1600° F) (Ref. 26). These shuttle bearings do not operate during descent (control is by reaction jets) but must withstand the 760° to 871° C (1400° to 1600° F) temperature and operate at lower altitudes.

Reference 27 gives many examples of critical solid lubrication applications. One of these is the engine mount bearing (Fig. 16). The maximum temperature is 649° C (1200° F) and the desired life is 30 000 cycles. Due to engine thermal expansion the bearing operates through a 20° arc each cycle under a race pressure of 3448 N/cm^2 (5000 psi).

Another example of the use of solid films is in structural fasteners. It is estimated that a large supersonic airplane would contain between 2 000 000 to 4 000 000 fasteners (Ref. 27). These include riveted, threaded and interference fit types. Because of large weight saving, titanium fasteners are very attractive. However, titanium alloys have a high tendency to gall and fret. Thus, solid film lubricants are required to permit assembly and, in the case of access panels, to permit disassembly and reassembly numerous times. These fastener applications represents the largest single use of solid lubricants in the aircraft.

Solid lubricant types. - Solid lubricants are sometimes classified according to the form in which they are applied - bonded films, solid bodies (graphite, plastics), composites and powders. Molybdenum disulfied (MoS_2) is probably the most well known application of a bonded solid lubricant. Air cured and heat-cured resin-bonded MoS_2 formulations are widely used. Surface pretreatments such as phosphate and vapor blasting are important and can greatly enhance film life. These treatments tend to provide more substrate surface area and thereby increase film boiling and also provide reservoirs of solid lubricant. This last item of lubricant reservoirs is worthy of additional research. Under fretting condition these reservoirs also provide a space for wear particles.

For temperatures in the 260° to 649° C (500° to 1200° F) and above range, inorganic bonded solid films are used. The binders are ceramic or salt based, and the lubricant powders are such materials as lead oxide, lead, lead sulfide, gold, silver tellurides and selenides (Ref. 28). For temperatures to 1038° C (1900° F) Ref. 29 reports success with ceramic-bonded calcium fluoride. As pointed out in the introductory remarks of this section much higher temperatures must be faced in the future and work is needed in this area.

Chemical thermodynamics. - In considering the future of solid lubricants, it is appropriate to give attention to techniques for predicting and for measuring the thermal and chemical stabilities of candidate lubricants and bearing materials. An important tool, which has been used for predicting the chemical reactivity of solid lubricants, is the thermodynamic property known as the free energy of reaction (Refs. 30 to 32). The free energy of reaction is a measure of the chemical potential for a reaction in terms of the energy change (excluding pressure-volume work) which occurs in the reaction. If free energy is lost during an assumed reaction (ΔF is negative), the implication is that the reacting system assumes a lower energy state and the reaction has the thermodynamic potential to occur.

An early example of the usefulness of the free energy concept in explaining the chemical behavior of solid lubricants was described in Ref. 30. Free energy calculations indicated that in air, the oxidation of a high temperature solid lubricant, lead monoxide (PbO), to form nonlubricating red lead oxide (Pb_3O_4) is favored below 454° C (850° F) but PbO is the more stable compound at higher temperatures. This was experimentally corroborated in X-ray diffraction studies (Ref. 33) in which no oxidation of PbO to Pb_3O_4 occurred at 482° C (900° F), but complete conversation to Pb_3O_4 occurred at 471° C (880° F). This experiment therefore checked with the thermochemical predictions within 30° F.

In Ref. 31, the free energy relations of solid lubricants were used to predict chemical reactivity of the candidate lubricant coatings with the substrate metals. On the basis of rolling disk experimental results, softness and thermodynamic properties were judged to be important criteria in the selection of candidate lubricants. With a few exceptions those compounds which were both soft and thermodynamically favorable for chemical reaction with the substrate provided the lowest friction and wear. It was postulated by the authors that adherent solid lubricant films may be formed and maintained by chemical bonding with the substrate metal. Presumably, a tacit assumption in this postualte is that the reactivity is sufficient to form an adherent reaction layer but not sufficient to cause lubricant depletion, excessive corrosion, or bond disruption by volatile reaction products.

Chemical kinetics. - Although thermochemical calculations are useful, ultimately, experimental studies are required to more completely characterize chemical and thermal stabilities. This is essential for determining reaction rates because thermodynamics gives no insight into reaction kinetics.

Thermal dissociation rates of molybdenum and tungsten disulfides, diselenides, and ditellurides in vacuum have been systematically studied (Ref. 34). The method of thermogravimetric analyses (TGA) was used to continuously monitor sample weight during the experiments. A mass spectrometer was used to detect and characterize the volatile products of dissociation. Nonvolatiles were analyzed by X-ray fluorescence and diffraction. The major results are summarized in Table III, Ref. 35. The data indicate that the disulfides are the most stable to thermal dissociation, the diselenides are intermediate and the ditellurides are the least stable. However, the diselenides provided effective lubrication in vacuum to a higher temperature, 760° C (1400° F) than the disulfides. The friction-temperature characteristics were determined for very thin, unreplenished, burnished films. The diselenides, evaporate more slowly than the disulfides and apparently, for these very thin films, the evaporation rates were the controlling factor in determining the maximum temperatures for effective lubrication.

Shear of solid lubricants. - The importance of crystalline structure on the shear properties (and therfore friction) has been recognized for many years. Lamellar crystal structures are found in a large number of inorganic compounds. Generally, the layer lattice compounds have small positive ions and large negative ions. Crystallites of such compounds shear between well defined adjacent planes of similarly charged (negatively) atoms. The cleavage planes and shear plane of graphite and MoS_2 are the same, however, that is not true for some other solid lubricants (i.e., CaF_2).

Graphite and boron nitride have similar lamellar crystal structures. Their performance as solid lubricants however, depends on extraneous adsorbates to reduce the shear resistance between adjacent surface layers. The same adsorbates can have adverse influence on shear with more ionic compounds like MoS_2 and WS_2.

The structural hypothesis for shear of solid lubricants is predicated on the anisotropic nature of such materials. To properly understand structural processes it is necessary to study the shear process of pure materials with no or controlled adsorbates present as in ultra-high vacuum facilities (Ref. 36).

Surface adherance. - Establishing chemical bonds such as those in the burnished MoS_2 films is very desirable. Experience with lightly burnished graphite fluoride films in Ref. 37 showed very good film adherence in the absence or presence of contaminating vapors. The utilization of more frictional energy in the burnishing process would be likely to further improve film adherance.

It is suggested that nonstoichiometric solid lubricant compounds such as graphite fluoride (as with unsaturated organics) provide greater capability to form chemical bonds with substrate material. Chemical bond strengths can then be used to postulate relative adherance of the compound to a known substrate. Such bonds occur with the surface atoms of the substrate. New tools such as the LEED and Auger spectrometer as well as field ion-emission microscopy may allow a less speculative appraisal of physical and chemical adherance mechanisms for lubricating films in the near future. Those tools allow study of surface atoms with the capability to build up by adsorption or strip successive atomic layers on surfaces. Pertinent mechanisms can only be explained on an atomistic mechanics basis.

The deposition of highly adherant surface films by vacuum methods has unusual promise (Refs. 38 and 39). Either evaporation or sputtering processes in conjunction with an ionized plasma may be used to deposit pure metals and inorganic compounds that can provide lubrication. The combined conditions of an extremely clean substrate, accelerated high velocity coating material, and ionization of that material provides several of the requirements needed for strongly adherent films. In particular, it should be noted that mechanical bonding does not have a major influence on adhesion in vacuum film formation processes as it does when bonding agents are used. Much remains to be resolved with vacuum deposition methods. It has been demonstrated however, that highly adherent coatings of desired stoichiometry can be deposited in closely controlled and uniform thickness. Coatings around 2000 Å (8×10^{-6} in.) thick are sufficient for good performance with MoS_2 while the optimum thickness for films with bonding agents would be 50 000 Å (200×10^{-6} in.) or more (25 times greater thickness).

The temperatures at high flight speeds present a severe environment for operation of sliding and rolling mechanisms. Table IV shows what the present needs are and what the anticipated future needs are. Temperatures to 1371° C (2500° F) is an ambitious goal. And the design approaches taken in the future will be influenced by the degree of success attained in solid lubricant research.

Liquid Lubricants

Ester type lubricants will most likely continue to dominate the usage in military aircraft turbines. Additive technology has made possible some advanced (MIL L-27502) formulations. One formulation appears to be capable of operation with bulk temperatures of 232° C (450° F) and bearing temperatures above 260° C (500° F) with an open system, and at 287° C (550° F) and 343° C (650° F), respectively, with an inert system.

More stable lubricants than the esters are available but special lubrication system design may be needed to utilize their capability; for example, changing bearing loads to compensate for varied lubricating ability, inerting with nitrogen, performing the lubricant and coolant functions separately, and lubricating by mist. Certainly, different additive technology will apply to the varied liquids.

As an example of the different performance of fluids, Fig. 17 shows the effect of using various lubricants on the endurance of bearings operating at 316° C (600° F) in an inerted environment. The synthetic paraffin showed superior performance compared with other high temperature lubricants giving bearing lives many times the AFBMA (Anti-Friction Bearing Manufacturer Association) predicted life for that bearing under more normal conditions. Properties that must continue to be sought are (1) higher bulk temperature stability (2) greater hot spot temperature stability, (3) greater viscosity or film strength and thickness in the lubricated contacts, (4) improved low temperature pumpability, (5) lower vapor pressure, (6) greater thermal conductivity and film coefficients for heat transfer, (7) greater specific heat and (8) minimum density.

Increased bulk oil stability has substantial impact on aircraft performance and weight. With reduced heat rejection to the engine oil, the fuel temperature can be allowed to increase by removing tank insulation. On a large supersonic aircraft, tank insulation can weigh several hundred pounds. Further, a 260° C (500° F) bulk temperature oil will allow continuous operation at over Mach 3.0. At the present, if tank insulation is reduced it would be limited by bulk oil stability. Several oils with good high temperature stability have relatively poor low temperature pumpability. The low temperature problem should be realistically assessed so as not to unduly penalize the high temperature performance capabilities.

High vapor pressure of the lubricant can complicate the lubrication system or cause excessive oil consumption. Further, excessive vapor in the lubricant can impede the critical cooling function of the lubricant. The present considerations for density, specific heat, thermal conductivity and film coefficients are also mostly related to the coolant function. Oil flow quantities in engines are determined more by cooling than by lubricating needs. A lubricant that is a more efficient coolant can reduce oil flows with attendant reductions in systems weight. Another role of density that could become important in high temperature systems is the convective inertia effect on lubricating film pressures. While viscosity decreases rapidly with higher temperatures the density changes little. It has been shown in fluid film seals that convective inertia is very important and we need to learn to utilize convective inertia film pressure generation in lubrication designs. Convective inertia influences will be increased by the trend to higher rotative speeds as well as to higher temperatures.

In high performance engines the sump fire problem will become more critical with increasing temperatures. Experimental study of design consideration to minimize sump fires are in process (NAS3-14310). The perfluorinated ethers are the most fire resistant lubricants studied under simulated engine conditions (Ref. 40). In closed systems that material performs very well. Also, since they are a high density class of materials the probability of convective inertia effects are greater than for most lubricating fluids.

Some concepts of lubrication, which may have dominate effects, deal with properties needing further study. The bulk of this discussion will deal with such concepts.

The concepts of Reynolds provide a generally satisfactory basis for thick film lubricant behavior. However, the greater pressures and shear rates common for thin films of concentrated contacts (Ref. 41) impose regimes of fluid behavior not adequately defined. Preliminary data by Winer (NASA P.O. No. C-57357-B) suggests that at modest pressures of less than 6.9×10^4 N/cm^2 (100 000 psi) synthetic fluids of interest to present and advanced aircraft show flow lag properties indicative of phase transformations. Those flow changes are analogous to the "wax formation" reported in early pressure viscosity data for mineral oils. Also, measurements of viscosity at 10 000 psi with shear stress to 10^6 dynes per cm^2 in a capillary viscometer indicate that several lubricants of interest are non-Newtonian. Analysis of optical interferometry data obtained by A. Cameron and associates at Imperial College with rolling and sliding contacts (NASA contract NAS3-12418) also show non-Newtonian behavior.

Additional studies by Dr. Cameron's group (Ref. 42) that eliminated particulate effects, confirm the questioned separate observations of Ficks, Needs and Deryaguin that a surface film as thick as about 20 000 A does exist on oil lubricated solid surfaces. Chemisorption appears to be essential for such film formation. Shear of the film and increased pressure quickly reduces its thickness. Similar measurements have also been made by Battelle Memorial Institute (Ref. 43). The means for practical utilization of such films in the mixed thin film and boundary regimes merits close study. It is likely that the use of high molecular weight paraffinic resin as a lubricant additive (Ref. 44) may function according to the above film formation mechanism; the resin is a complex molecule capable of molecular entanglement. Molecular entanglement would retard the orientation process. Orientation occurs during shear and could be the reason surface films suffer thickness reductions.

Several lubricants have lubrication properties at high temperatures that are much better than one would deduce from the physical and chemical characteristics of the fluids. The friction polymer concept (Ref. 45) would seem to explain the performance of several fluids including a modified polyphenyl ether. That fluid in spite of lower viscosity gave endurance lives in engine type bearings that were at least as long as the best advanced esters and at 50° F higher bulk oil temperature (Ref. 46). The surface polymers tend to form on high energy metal surfaces activated by the friction and wear process to catalyze decomposition of the base fluid. The final reaction product is a function polymer that forms a solid film (its chemistry analogous to the liquid lubricant). Reference 45 ranks various metals and hydrocarbon types in regard to their tendency to form friction polymer surface films and suggests the descriptive term "surface resin".

The effects of pressure on the formation of crystalline and solid phases in lubricant films for concentrated contacts need careful study. The high pressure viscosity measurement flow problem and films of greater than predicted thickness in elastohydrodynamic contacts as determined by optical interferometry both indicate substantial change in rheological properties of the high pressure fluids. Also, the rheology of solid lubricants and those of liquid lubricants under the pressures of concentrated contacts may be very similar. The importance of rheological studies of both liquid and solid lubricating materials cannot be over emphasized.

Cooling performance of lubricants is responsive to the various factors mentioned previously. One aspect of that performance not yet adequately studied is the importance of wetting ability of the lubricant on the lubricated solid surface. Temperature instabilities in bearings have been found with silicones, polyphenyl ethers, perfluorinated ethers, some esters and other liquid lubricants. Those liquids have been found incapable of wetting the friction polymer or surface resin that developed on lubricated parts. Where there is gaseous or vapor phase in the fluid stream, nonwetting of the solid by the liquid can greatly reduce cooling ability by allowing a discontinuous coolant film. That possibility of reduced cooling ability offers a reasonable explanation for the temperature instabilities on bearings cited above. In one case a modified polyphenyl ether was formulated with an additive to improve wetting. While it is possible the additive may have had other beneficial effects besides improved wetting, significant gains in performance were noted. In one type of ball bearing test at 600° F the original fluid gave failure in a few minutes; the same fluid with a wetting additive performed satisfactory for periods from 40 to 100 hours. Similar experiments in other components (e.g., engine scavenge pump bearings) have been gained.

Historically most lubricant studies have pointed to either the physical concepts (e.g., viscosity) or the chemical concepts (e.g., functionality) but rarely both. Fine work on physical concepts of hydrodynamic and elastohydrodynamic regimes is continuing and the chemistry of lubricant synthesis, boundary lubrication and reactive lubrication has many capable advocates. It is the rare interdisciplinary type studies that need be encouraged.

Greases are convenient products that offer not just a lubricating material but often an adequate lubrication system. Virtually all the advanced fluid lubricants can be utilized in greases. Surface transport phenomena, such as surface energy and wetting or spreading can be more important for the liquid phase of greases than for use of the same liquids in circulating oil systems. Starvation concepts in lubrication discussed in regard to elastohydrodynamics are relevant to grease lubrication. Rheological properties of grease structures are unique to its application as well as lubricating capabilities. The use of solid lubricants in grease formulations has gained wide acceptance but there is question if such materials contribute to performance except in very special circumstances. Reference 47 provides a detail review of recent grease technology in the United States. The greases as a class have sufficient commercial appeal that lubricant manufacturers perform needed formulations. Unique applications need special study and limitations are frequently associated with cooling problems.

Lubrication Systems

The discussion on lubricants described performance advantages from the use of increased bulk temperatures. Reference 48 indicates that the primary limitation, on higher temperature use of lubricants, is oxidation, and shows that inerting allows the lubricant bulk temperature to be increased by 66° to 93° C (150° to 200° F). The limiting temperature for lubricated surfaces can approach the temperatures of thermal decomposition of the lubricant. That limiting useful temperature depends on the mechanism by which the lubricant functions. Surface chemistry as well as the bearing surface conditions (e.g., stress, speed and ambient temperature) and physical properties of the lubricant are important. The development of a thermodynamic approach to lubricant failure is badly needed but other elements of the system beside the lubricant need to be included and especially the oxygen availability. The International Research Group on Lubrication, started under the sponsorship of OECD, has a current cooperative project on lubrication failure that emphasizes the thermodynamic aspects.

Considering the weight penalty in insulation for the fuel cells of high speed (Mach 3) aircraft, the real cost of inerting the lubrication systems with nitrogen from a cryogenic reservoir needs to be evaluated. The availability of cryogenic nitrogen for other coolant functions may also be important. Further, experience in advanced experimental programs shows that a nitrogen blanket is a most effective method of preventing oil sump fires and of putting them out. A simple commercial system for handling liquid nitrogen has been developed.

Internal fires are caused by high temperature compressor air leaking through seals and by friction rubs on shaft assemblies which create hot sparks capable of igniting the vapor in the lubricant system. A consideration is the selection of shaft area materials such that inadvertant friction rubs will not create hot sparks when function rubs inadverently occur.

With engines operating at higher rotating speeds, the use of mist systems have advantages. Figure 18 shows the effect of lubricant viscosity or power loss in a simulated engine type sump. Also, the effect of increased speeds is shown for the type II ester from 1.8 to 2.5×10^6 DN with substantially greater power loss at the higher speeds using the conventional circulating oil systems. Data using the most viscous of the lubricants in a mist system shows the power loss to be reduced to about one-third that for the recirculating system at even the normal operating speed.

The use of once through mist lubrication for aircraft turbine engines has been utilized on a limited basis by several engine manufacturers. The most notable success has been by SNECMA. The greatest success has been on roller bearings at the engine extremities. Mist lubrication of outboard bearings can save substantial system weight over a recirculating system. The use is limited to bearings with modest radial loads. Experiments with thrust bearings using diester lubricants were not successful (Ref. 48), whereas service experience with radial load bearings using the same type lubricant has been successful.

Advanced technology for mist lubrication is being gained (Ref. 49) that should allow less restrictive use. Optimum applicators and lubricant types are sought for enhanced wetting. It is clear that particle size; impingement velocity, and surface chemistry are very important as well as the lubricant properties. The minute amounts of lubricant used can have only a minor cooling effect. With optimized application methods, only small carrier gas flows may be needed. Thus, it will be important to provide separate cooling designs for the lubricated components. Fortunately, at the higher rotative speeds anticipated, increased turbulence makes gas cooling more efficient. Separate and optimized designs of the oil system, the cooling system and the lubricated components are indicated.

Bearings with under race cooling, open flow passages and vapor pumping capabilities are needed. At present typical operating speeds (e.g., 1.8×10^6 DN), bearings lubricated with mist should be expected to run at higher temperatures than with recirculated bulk oil. With the higher speeds anticipated (e.g., 3.0×10^6 DN), there could well be cooling advantages with mist lubrication and gas cooling.

The heat generated in high power reduction gearing is a critical problem (Ref. 50). The largest reductions in systems weight can be gained by optimizing the cooling. Some preliminary data has suggested slight load capacity advantage is possible with gas cooling and mist lubrication for high speed aircraft gearing. Again, as for bearings and other components, separate and optimized designs of the oil delivery system, the cooling system and the lubricated parts are essential. Basic heat transfer studies are needed that encompass the total component geometries, the surface macrogeometries, the lubricant film effect, and the convective gas film effects for cooling.

In aircraft gas turbines fine dirt particles in the lubricant degrade the life of the engine bearings. Dirt in oil is one reason why some engine designers prefer inner race riding cages. Outer race riding cages tend to trap dirt within the bearing. Analysis of lubricants reveals that 95 percent of the dirt is carbonaceous; and these are probably products of lubricant degradation. The remainder is metallic (wear particles) or silaceous (sand). (Engines some times ingest sand and some of this can find its way into the lubricating system.)

Centrifugal filters for lubrication systems are being developed and have been applied to some engines now in use (Ref. 51). Figure 19 shows a typical system. The lubricant and entrained dirt enters in one end of the rotating shaft, and centrifugal action forces the dirt into the rotating collection canister. The canister, on typical models in use, has a capacity of 490 cm^3 (30 cubic in.). Performance comparison of static and centrifugal filters is shown in Fig. 20.

Centrifugal filtration represents one approach to obtaining a cleaner lubricating system. These efforts are important since damage by dirt in a system is a problem of some significance. And surface deformation caused by dirt in bearings and gears can lead to fatigue failures. The centrifugal filter removes metallic debris from the main flow path of the lubricant; an important feature that can significantly reduce catalysis of lubricant degradation. In this regard perhaps more attention should be paid to protecting the lubricant in the highly stressed and vital helicopter transmission. (Vital because loss of transmission function can be catastrophic.)

The value of spectrographic diagnostic tools to forcast impending engine and transmission failures has been conclusively demonstrated in both commercial and military applications. For example, the U.S. Army Spectrometric Oil Analysis Control Center in St. Louis, Missouri has accumulated much data proving the success of that activity for aircraft.

In the 1968 fiscal year they saved 268 engines and transmissions from impending failure. There are sampling and personnel training as well as equipment problems inherent ~~in the wide~~ use of this valuable method of predicting wear failures. As experience is developed, perhaps the data requirements can be simplified and sensing concepts advanced so as to make continuous in-flight analysis feasible as a part of aircraft lubication systems. Centrifugal filtration should enable more sensitive spectrographic analysis of engine oil. In the meantime, the laboratory studies for this program should be expanded.

Elastohydrodynamic Lubrication

EHD theory. - It has been long known that gears would operate under some conditions with a film of lubricant separating the surfaces. This was based on the observation that the machining marks remained after hundreds of hours of operation. Attempts to explain this mode of lubrication were not successful until Grubin's paper published in 1949 (Ref. 52).

The first attempts to explain gear lubrication considered the gear surfaces as rigid bodies and the fluids as isoviscous. On this basis the predicted film thicknesses were too small to account for the apparent lubrication of gears. Later investigators included, in the analysis, elasticity of the surfaces but kept the isoviscous assumption. The resulting film thickness predictions were still too small. Another group of investigators introduced different pressure viscosity relations but kept the rigid body assumption. This approach was not successful either. Grubin published a solution for a mathematicl model which provided for elastic deformation and for pressure viscosity effects. The predicted film thicknesses were much larger than those of the previous attempts cited and could therefore, account for the apparent thick coherent films that existed in some gear applications. The Grubin model is illustrated in Fig. 21. Note that the elastically deformed mating surface forms a parallel channel. Other features of the mathematical model are: (a) infinitely long cylinder (no end effect) and (b) the deformation is the same as the Hertzian shape for dry contact of a cylinder (inlet deformation is the same as the Hertzian shape). Crook (Ref. 53), in a series of experiments on rotating disks, obtained convincing evidence that the Grubin solution provided accurate film thicknesses.

Later investigators (Dowson and Higginson, Ref. 54) in 1959 published a numerical method for obtaining film thicknesses and pressure distribution as a function of speed, load and lubricant parameters. This method determines the actual EHD shape which includes the local restricted at the exit of the contact area. (See exaggerated illustration in Fig. 22.) Dowson and Higginson describe the EHD film thickness in terms of the following dimensionless parameters:

$H^* = \frac{h_{min}}{R}$ film thickness parameter

$W = \frac{w}{E'R}$ load parameters

$U = \frac{\mu_o u}{E'R}$ speed parameter

$G = \alpha E'$ material parameter

where

h_{min} = minimum film thickness

R = effective roller radius

w = load per unit length

E' = equivalent elastic modulus

μ_o = inlet viscosity at ambient pressure

u = one-half the sum of surface speeds

α = lubricant pressure-viscosity coefficient

From a wide variety of theoretical results Dowson and Higginson derived the following very useful formula for minimum film thickness

$$H^* = 1.6 \frac{G^{0.6}U^{0.7}}{W^{0.13}} \tag{5}$$

Reference 55 extended the elastohydrodynamic analysis to include thermal effects. This was accomplished by solving the energy equation in conjunction with the Reynolds equation for the inlet section. Such analysis is useful for high speed in which appreciable shear and fluid compression is to be expected.

Actually the minimum film thickness for a ball on a flat geometry is not at the exit as shown in Fig. 22, but is at the sides of the contacts. This was shown in a series of experiments reported in Ref. 56 and can be illustrated with a series of photographs from Ref. 57. In the experiment, optical interferometry was used to measure the film thickness between a ball rolling against a flat glass disk. The set up is illustrated in Fig. 23. By observing the color interference pattern the film thickness could be calculated (with the proper calibration) for the entire contact region. The surprising observation is that EHD (elastohydrodynamic) films are very easily established. (This probably accounts for the dependability and ruggedness of concentrated contact elements such as gears and bearings.)

Figure 24 shows the light interference patterns obtained with a 2.54 cm (1 in.) diameter ball rolling against a glass disk. Note that the film thickness is rather constant in the inlet and central regions. The exit is restricted as previously pointed out in Fig. 22. But note also that a restriction occurs on the sides of the central region and the film thickness is actually the smallest at these lateral restrictions. Representative plots of the film thickness are given in Fig. 25 (from Ref. 57). In this case (Fig. 25) the exit film thickness is 46.5×10^{-6} cm (18.3×10^{-6} in.). However, the film thickness at the lateral restrictions is only about 20×10^{-6} cm (8×10^{-6} in.). The important point is that the mathematical model does not account for these lateral restrictions and these are the thicknesses that determine the closest approach of the two surfaces.

It is of interest to compare the EHD film thickness and contact diameters with the surface topography discussed in one of the preceding sections. In general the film thicknesses are in the same range as the surface variations. Thus a clear concept of the capabilities and limits of EHD lubrication is a powerful tool for guiding designers in tolerance and design selection.

Effect of inlet starvation. - A significant point in the cited mathematical models is that the film thickness is primarily determined in the inlet region. And the model assumes that the inlet region is fully flooded with lubricant. However, as shown by Ref. 57, a very important point in operation is a partially flooded inlet in the contact area. This partial flooding could occur in a bearing since the action of one element (ball or roller) effects the inlet condition on the one following it. A series of very significant experiments were made (Ref. 57) showing how the starvation determines the film thickness. Figure 26 (from Ref. 57) shows a series of pictures typical of increasing starvation. In Fig. 26(a), the extent of the oil in the inlet is sufficient for development of an EHD film thickness as predicted by the mathematical models. Note the extensive wake at the exit of the contact. For a bearing this wake could effect the inlet condition of the next element in line, and this is an example of an important factor not revealed by mathematical models. The next pictures show decreasing film thicknesses as the inlet starvation increases. The EHD contact pattern "degrades" to the Hertzian shape shown in the final photograph, Fig. 26(f).

EHD entrapment. - Reference 57 also reported some interesting data on lubricant entrapment when the rotating ball is brought to rest. This is shown in Fig. 27. The central region contains lubricant and only the edges of the Hertzian region are in contact (or nearly so). The color fringes indicated the shape in Fig. 27(b). This photograph was taken 5 minutes after the ball stopped and some fluid had leaked out of the entraped region. These results imply that it may be possible to maintain a full EHD film under a velocity reversal during which the relative velocity may be zero for an instant. Further, this entrapment also suggests that normal approach of two surfaces may be very significant in EHD lubrication. The normal approach model which has not been developed for combined rolling and squeeze films may prove to be a fruitful area of research.

Effects of surface roughness. - The preceding mathematical models all assume a perfect surface with no surface roughness. Also the experimental results were obtained with very smooth ball and disk elements. The effect of surface roughness is important from an operating viewpoint. Penetration of the film should be expected when a film thickness to surface roughness ratio reaches a certain value. This is the transition from elastohydrodynamic to boundary lubrication and is sometimes referred to as mixed lubrication. Many machine elements operate in this regime especially during a "break-in" or "wear-in" period.

Because of the thin films in EHD or mixed lubrication regime it is apparent that tolerances such as ball or shaft out of roundness, tooth surface profile waviness and small deflections can also lead to severe penetration of the lubricating films. Thus a better understanding of the lubricating mechanism and its limits provides designers with an invaluable design guide.

Gears

Gear applications. - EHD lubrication, which was previously discussed in a general manner, is very prominent in gear operation. Low power gearing is used in accessory drives of conventional turbojets and turbofans for fixed wing aircraft. The gears are generally sized to make up the center distance necessary for the various accessories and the gear weight is a minor fraction of engine weight. Therefore, there is no strong incentive to operate at Hertz stresses over 8.27×10^4 N/cm^2 (120 000 psi).

In helicopter transmissions, however, the transmission weight is a significant item. These transmissions are built with a weight to horsepower ratio of about 0.0002 kilograms per watt (0.4 pounds per horsepower). The overall reductions on some are in a range of 70:1 Usual gear materials are AMS 9310 carburized and hardened to produce a case of 58-62 Rockwell C with a finish of 20 micro-inch or better. Efficiencies of these transmissions are near 98 percent. However, for a 3.73×10^6 watt (5000 horsepower) system this amounts to a dissipation of 7.46×10^4 watt (100 horsepower) that must be handled by the lubrication system.

Since the weight is critical in aircraft, it is desirable to optimize the gears and operate at full gear capability. The computer has allowed exacting and detailed analysis of gear mechanics such as deflections, load distribution, stress and dynamics; and these analyses have contributed significantly to gear design. The difficult remaining problem is predicting and eliminating the principle modes of failure: scoring, wear, and contact fatigue pitting.

Dudley (Ref. 58) illustrates gear failure regions as shown in Fig. 28. In the low speed region, (1), the gear is not running fast enough to develop elastohydrodynamic films. The wear that occurs in this region can be mitigated by more viscous oil, additives, surface finish improvements, and metallurgy. In region 2, tooth breakage occurs due to high loads and associated bending fatigue, and in region 3 fatigue pitting is apt to occur. Region 4 is the regime of no wear. Here the speed is high enough to develop a thick EHD film. However, as the speed increases, too much heat is developed and scoring becomes a problem (region 5).

Scoring occurs in high speed gears and can be visualized as asperity adhesion (momentary seizure) in which asperity fracture results in metal transfer. And then because of the sliding motion the surface is scored. Scoring has been described (Ref. 59) as "the rapid removal of metal from tooth surfaces caused by tearing-out of small contacting particles which have welded together as a result of metal-to-metal contact." Figure 29 shows scoring on a roller used to simulate gear rolling and sliding action.

Scoring has received considerable attention and is a controversial subject amongst research groups. Some groups approach scoring from EHD viewpoints and others through the critical temperature hypothesis of Blok. A marriage between these two viewpoints with additional data on lubricant rheology and chemistry may provide a better predictive model.

In 1937 Blok (Ref. 60) published a critical temperature theory. Under this hypothesis, scoring takes place when the temperature in a contact zone reaches a critical magnitude (for nonreactive lubricants). Blok (Ref. 61) considers a number of possible mechanisms of the loss of lubricant protective capacity through: (a) inability to weaken asperity welds through contamination, (b) inability to prevent the conjunction temperature from becoming too high, and (c) lubricant film collapse. This critical temperature is sometimes called flash temperature (T_t) and is related to the gear material temperature (T_b) and instantaneous computable temperature (T_f) as follows (Ref. 62).

$$T_t = T_b + T_f \tag{6}$$

The application of the preceding formula requires care: Defining the temperature of the gear blank is critical since it controls the inlet temperature of the conjunction. This inlet temperature, in turn, controls the film thickness and the film thickness influences the friction and instantaneous temperature. Thus T_b and T_f are interdependent.

To improve this model Kelley (Ref. 63) modified it to include surface roughness effects. His formula is:

$$T_t = \frac{\left(T_b + T_f\right)}{\left(1 - \frac{S}{55}\right)}$$

where

$$S = \text{surface finish (rms microinches)}$$

Kelly's experimental data shows that this experimental formula works for surface finishes between 7 to 25 rms microinches. Later variations of this formula provided for other effects such as gear inaccuracies and profile modification near the tip.

As stated previously in many gear applications, the EHD film thickness is not great enough to separate the mating asperity tips. Thus some boundary lubrication should be expected. Under these conditions the critical temperature model assumes importance. The picture, however, is clouded by the relative contributions of EHD and boundary lubrication, lubricant rheology under high shear and pressure, lubricant chemistry, gear dynamics and gear accuracy. Also the normal approach in EHD entrapment observed (see Fig. 27) may be important in asperity contact.

The loads at which scoring starts are higher for reactive types of lubricants containing extreme pressure additives such as compounds of sulfur, phosphorous and chlorine. However, the magnitude of increase is not well defined. And as Ref. 62 points out, although scoring may not occur, the wear rate may increase due to increased chemical activity. The preceding comments point out the need for predictive models for additive effects.

The wear failure can be discussed in terms of the work of Ref. 64. In this study a series of roller tests were made simulating the combined rolling and sliding of gear teeth. Carburized and hardened steel (SAE 8620) rollers with a 5 to 10 microinch centerline average surface finish were operated at Hertz stresses of 2.07×10^9 N/m^2 (300 000 psi) with straight mineral oil at 96° C (205° F). The film thickness, h, was calculated by the Dowson equation. Table V contains a summary of the data from Ref. 64, and total wear data is shown replotted in Fig. 30. The important point of Table V and Fig. 30 is that the wear decreases as the film thickness, h, increases. This demonstrates that the calculated EHD film thicknesses are a powerful guide. This work (Ref. 64) is indicative of the useful wear information that a systematic approach can produce. Certainly additional work of this type is needed since only a small range of variables were covered.

In the past, contact fatigue or pitting was associated primarily with the contact Hertz stress. But there are other primary factors such as: EHD film thickness, surface microtopography and temperature. With improved materials and heat treatment, significant increases in contact fatigue resistance have been made. However, some (Ref. 62) are of the opinion that the current performance level is far from that possible.

Some investigators view contact fatigue pitting as starting at some subsurface inclusion in the zone of maximum shear stress. It is now generally accepted that contact fatigue can nucleate and propogate from many different types of stress raisers at the surface or subsurface. Surface stress raisers include nicks, surface inclusions, machining flaws, corrosion pits and debris dents. There is a question whether surface tangential forces are necessary for surface cracks to propagate (Ref. 65). More data are needed to answer this question especially in high hardness gear steels.

It has been shown that surface cracking is suppressed if the EHD film is thick enough to separate the asperities (Refs. 66 and 67). However, as stated previously many practical applications operate with some asperity contact, at least at the start of operation. For these applications surface initiated cracks may become the predominant mode when premium quality steel eliminates subsurface inclusions.

Reference 68 demonstrated with fatigue studies on 52100 balls that crack formation is a function of lubricant composition. Microcracking of undissolved carbide-matrix interfaces was found. Further 0.5 percent sulfur additive was sufficient to remove surface microcracks by chemical wear. The important point is that the lubricant chemistry and metallurgical microstructure are significant variables in this question of crack formation. Further, the real contact stress distribution in a concentrated contact is not known. As pointed out in a prior section, better tools for characterization of surface microtopography are needed. Sources of high stress concentration not considered in usual stress analysis are: debris dents, and high deformation gradients associated with EHD entrapment or at contact area restrictions.

Effect of gear errors. - The preceding is rather a complicated picture that includes the previous discussions on EHD, surface topography, boundary lubrication, lubricants and lubricant additives. However, the steps that can be taken to improve gear operation are many and can be classified as follows: (1) metallurgy - sources of stress raisers must be investigated. (2) fluid rheology - possibility of improving EHD lubrication needs study. (3) surface topography - previous discussion indicates need for surface definition and characterization. (4) mounting accuracy - these errors principally affect load distribution. Elastic deformation of the shaft can also cause nonuniform loading. (5) gear accuracy - a gear has many possible sources of error. Each affects the lubrication process and control of errors must be weighed against the cost of more accurate gears. Dynamic load is affected by accuracy of the tooth profile and tooth spacing. Load distribution is affected by lead error and elastic deformations.

As an example on gear errors, profile errors are deviations from the true involute profile which appears as a straight line in Fig. 31, curve A. Of course, a machining tolerance is required and a question arises what are the allowable variations from the true involute form. Note that if the variation is stated to be 0.0005 cm (0.0002 in.) from the true involute form then the trace shown in curve B would be acceptable. Since the lubricant film thickness is much less than the 0.0005 cm (0.0002 in.) variation, a "lubricating" problem might be expected. Noise might also be expected since it has been long known that a hollow in a profile at the pitch line causes noise. To avoid wavy profiles some gear manufacturers specify that the profile must fall within a specified range, and more importantly must also be a smooth curve (fig. 31, curve C). Tip modifications of the profile are handled in a similar manner.

Rolling Element Bearings

Bearing applications. - Helicopter gear transmissions contain numerous rolling element bearings that are subjected to high gear separating and tangential forces. The input speeds on some transmissions are in the range of 6000 to 7000 rpm and this is reduced to rotor speed through a set of bevel gears and two planetary sets in series. Rolling element bearings are used for shaft and planet gear support.

In modern gas turbine engines rolling element bearings are used to support the main shaft(s). This is illustrated schematically in Fig. 32. In typical construction ball bearings near the center of the engine carry the thrust. The axial aerodynamic and pressure loads on the rotor are not completely balanced out and the ball bearing must carry this residual thrust. As a typical example of size, one large engine has 135 mm bore bearing that operates at 0.92×10^6 DN (DN = diameter in mm × N in rpm). The DN value has reached 2×10^6 on some current engines. And for advanced engines bearings are needed that are capable of operating to DN values of 3×10^6 (Ref. 69). Other projections on future requirements on bearing capability are given in Ref. 70 and shown in Fig. 33. These projections are bearing temperature to 316° C (600° F) and speeds of 3 to 4 million DN.

AISI M-50 material is commonly used for critical gas turbine mainshaft bearings. This material will maintain a hardness above Rockwell C 58 to temperatures of approximately 316° C (600° F) (Ref. 71). Another bearing material is SAE 52100. It is often used in helicopter transmission bearings, and is limited to about 177° C (350° F) since the hardness decreases above this temperature. Case carburized material is also used in helicopter bearings.

Bearing life rating. - In regard to materials, the present AFBMA standard (Ref. 72) which is used to provide a measure of fatigue life does not take into account the differences between materials. Comparison of materials given in Ref. 73 reveals that case carburized materials have better life in the film thickness region where appreciable contact of asperities occur.

The conventional fatigue life analysis is based on subsurface nucleation of cracks in a through-hardened air-melt material. In helicopter transmission bearings, the use of vacuum melt process materials (also not covered by AFBMA standard (Ref. 72)) has reduced the number of classical subsurface fatigue failure to a small percentage of the total failure population. As a result, another competing mode of failure, surface initiated cracks predominates in actual operation (Ref. 65). These observations suggest that the material factor should be included in life prediction. This would be an extensive program since it involves critical quality control and accurate extensive fatigue testing.

The AFBMA life rating does not take into account the very important effects of EHD lubrication. However, a recent paper (Ref. 73) shows how to apply EHD theory to predict bearing fatigue life.

Figure 34 (from Ref. 74) illustrates how a highly loaded bearing can have full EHD separation of the ball and race surfaces at high speed and yet have asperity contact and short life at low speeds. (When the speed is low the film is thin and the asperties touch.) The obvious solution is to:

a. Increase the lubricant viscosity,
b. decrease the lubricant temperature, and
c. improve the surface finish.

A determining factor in the prediction of bearing fatigue life is a parameter λ, the ratio of film thickness to composite surface roughness:

$$\lambda = \frac{h_{min}}{\sigma}$$

where h_{min} is film thickness as calculated by the Grubin equation (see Eq. (5)), and σ is a composite surface roughness (Ref. 73).

Figure 35 (Ref. 73) shows how life factor varies with this film parameter λ. At film parameter of 4 the B-10 life is nearly twice the conventional AFBMA calculated life; at $\lambda = 1.6$ the factor is unity which means life is equal to the conventional AFBMA value. (At $\lambda < 1.6$ the surface asperities contact with high enough frequency to produce possible surface distress.) Below $\lambda = 1.6$ the possibility of surface related distress increases. The preceding parameter gives the designer some insight into the extent of lubrication at the contact. From this insight he can better specify the necessary tolerances, lubricant, surface finish, run-in coatings, etc., that are necessary. However, more work is required in this area was pointed out in the earlier section.

Contact fatigue is affected by material type, temperature, contact geometry, surface microtopography, stress state, lubrication, and lubricant type. Thus, this is a complicated problem. The fatigue life of a single bearing cannot be predicted. But the distribution of life in a large group of identical bearings can be predicted. The usual approach used is the B_{10} life rating: that is, 90 percent of bearings will exceed a given number of hours for a selected load and speed.

In the literature the terminology for classifying the types of fatigue damage varies. Reference 65 summarized the damage descriptions used by various sources and proposed the following classification:

A. Subsurface Origin Spalling (or pitting)

   1. Nonmetallic inclusion
   2. Unconfirmed origin

B. Subcase Fatigue or "Case Crushing"

C. Surface Origin

   1. Point surface origin (PSO)
      (a) Debris dent
      (b) Handling nick
      (c) Surface flaw (grinding furrow, etc.)
      (d) Surface inclusion (intersecting or parallel and coincident with surface)
      (e) Peeling + hydraulic pressure propagation (HPP)
      (f) Corrosion pit
   2. General surface distress

D. Geometric Stress Concentration (GSC)

   1. End of "line" contact
   2. Edge of peeled area

E. Peeling (or Superficial Pitting, sometimes called "Frosting" or "Glazing"

F. Section Fracture

In the above classification it is of interest that one method of crack propogation is by hydraulic pressure (HPP). The observations listed in Ref. 65 to support this hypothesis are:

1. Evidence of a critical crack length for rapid propagation. The crack must extend beyond the compressive field of the contact stressed zone for the HPP mechanism to operate.

2. The only cracks which propagate by HPP are those inclined downward in the rolling direction at a small angle.

3. Relief of the pressure by branching cracks to the surface, or by drilling a hole, or by removing the fluid from the system, prevents the propagation or greatly retards it.

4. Low viscosity or low speed promotes HPP. The quantity of fluid entering the crack between stress cycles is governed by the atmospheric viscosity of the fluid and the access time.

Figure 36 shows various types of surface damage. In the first frame the bearing race has the normal appearance associated with a full EHD film. For thinner films the race becomes glazed. The next frame shows the superficial pitting and the last one shows gross damage.

Bearing cage materials. - In conventional rolling-element bearings, both metallic and nonmetallic cages have found widespread use. Precision bearings, such as those used for aerospace applications, are usually equipped with cages machined from metal alloys or nonmetallic phenolic materials. In some applications, where marginal lubrication exists during operation, such as at high temperatures, silver plating on the metal alloy has been used.

Standard bronze cage materials with lead plate over silver plate have proven inadequate to meet the long life, high speed requirements of modern engines (Ref. 75). Military requirement for operation up to one minute after oil loss adds to the severity of cage operation. In a study on a number of cage materials (Ref. 75) it was concluded that a silver plated AMS 6415 cage material had the best performance. This cage was able to survive 50 one-minute oil cutoff cycles. The optimum silver plate thickness was 0.0025 to 0.0050 cm (0.001 to 0.002 in.).

Phenolic materials are limited to temperatures of approximately 121° C (250° F), while some metal alloys are suitable for operation to approximately 316° C (600° F). Above 316° C (600° F), some success has been obtained with low-carbon steel or cast-iron cages, but, generally, the most successful high-temperature cages have been nickel-base alloys (Ref. 70).

Seals

Applications. - In some areas of mechanical technology, seals are so troublesome that great effort is made to design out the seals under the philosophy that the best seal is no seal. However, eliminating seals is not always feasible and analysis of some power systems reveals very serious performance penalties if sealing problems are not solved. For example, a serious seal problem had to be solved before the Wankel engine was marketable. Low leakage seals remain to be demonstrated in tip driven lift fans (a candidate for VTOL aircraft) and in closed cycle Rankine systems (a low pollution engine).

Rotor tip seals. - Various types of contact and labyrinth seals are used in the aircraft engine and power train systems. As an example, Fig. 37 is a simplified schematic of a three-bearing gas turbine engine showing typical seal locations. Many engines have abradable shroud materials on the housing over the tips of the compressor and turbine. The purpose of this abradable material is to achieve operation at small radial clearances between the tips and the surrounding housing. Gas leakage over the rotor tips disturbs the flow in the outer portion of the blades, and this flow disturbance has a significant impact on performance. The fact that these losses are significant makes a good case for using abradable seals. Abradable polymer seals are restricted to temperatures below 260° C (500° F). Other abradable materials, such as nickel-graphite are applied by flame spray technique. A nickel-aluminum alloy is applied by sintering to a backing plate. For temperatures to 760° C (1400° F) felt metal and honeycomb structures are also used.

Experience revealed that the principal problem, of abradable materials containing metal, is galling. This is a form of adhesive wear and affects the rotor blade tips as well as the abradable material. The current need is a nongalling abradable material suitable for operation to 1073° C (2000° F). Since turbine temperatures are expected to continue to increase it is anticipated that even higher wall temperatures will be operational. These operating temperatures suggest some type of porous oxide, such as aluminum oxide with micro-ball voids to provide the abradable feature.

The abradable wear mechanism is not well defined. And the honeycomb materials, rather than abrade, suffer permanent deformation when contacted by the blade tips. Some abradable materials, when rubbed, compress and also release wear particles; in others (brittle materials) release of wear particles is dominate mode of establishing clearance. In general this abrading action may be viewed as a form of wear in. Development is needed in this area; especially in the higher temperature applications. Some obvious desirable features of an abradable material are:

1. nongalling and little wear of blade tips
2. low density
3. wear debris not harmful to engine
4. dimensional stability
5. oxidation resistant
6. suitable life

<u>Labyrinth seals</u>. - The labyrinth seals are used at various locations, such as between compressor stages, at the end of the compressor and at the turbine. These seals have an impact on efficiency.

In one case, the seal leakage (one seal) amounts to a 0.6 percent efficiency loss. This is a significant loss and in future engines much greater losses are predicted. Thus there are strong incentives for devising more effective seals. (A seal concept for replacement of the labyrinth seal is discussed below.)

Because it is desirable to run as close clearance as possible, abradable type materials are also used in labyrinth seals. The wear problems are similar to those discussed in the preceding section.

<u>Bearing sump gas sealing</u>. - Various types of contact type shaft seals are shown in Fig. 37. These seals restrict gas leakage into the bearing sumps. (It should be noted that labyrinth type seals are also used for this purpose and the comments in the preceding section apply.)

The sump seals at the front of the compressor usually present no problem since the sump can be surrounded by compressor bleed air of moderate pressure and temperature, and a shaft seal of many types is adequate. As an example, ring seals have been used at 37 N/cm^2 (54 psi) and 166° C (330° F); circumferential seals are more than adequate for this location, and in the present state of development, have been used to 58 N/cm^2 (85 psi), 371° C (700° F) and 73 m/sec (240 ft/sec) sliding speed; face seals are also adequate as some are operating at 86 N/cm^2 (125 psi), 427° C (800° F) and 107 m/sec (350 ft/sec) sliding speed (Ref. 76).

At the middle of the engine (Fig. 37) and at the turbine bearing sump, the seal operational requirements becomes more severe. The bearing(s) and seals tend to be large because of shaft size; and because the thrust bearing must be large enough to carry the net thrust load on the rotating parts. In addition, the sump is usually surrounded by higher pressure gas (and corresponding high temperature) than the front of the engine.

Figure 38 shows modern shaft seal arrangements (schematics) for the turbine bearing sump location. Here the basic problem is protection of the bearing sump from the turbine cooling gas. In early engines the cooling gas pressure and temperature were relatively low and a single labyrinth seal, which restricted turbine cooling gas leakage into the sump, was adequate. At these pressures, the efficiency loss due to seal leakage was insignificant. However, a disadvantage of the labyrinth seal, as compared to the close-clearance (ring, circumferential and face) seals, is easier passage of air borne water and dirt into the sump. In addition, for labyrinth seals, reverse pressure drops must be avoided to preclude high oil loss. As turbine cooling gas pressure requirements increased with engine development, the single labyrinth seal was no longer suitable, and the seal systems such as illustrated in Fig. 38 were used (no scale is intended by these schematic drawings). Figure 38(b) illustrates a seal system in which a labyrinth seal restricts the leakage of relatively high pressure (P_1), high-temperature turbine cooling gas to an overboard vent (P_0') (or to a low-pressure region). Low-pressure (and corresponding low temperature) compressor bleed (P_1') surrounds the sump, hence, provides thermal protection, and leakage (through the labyrinth) into the sump provides required sump pressurization. The labyrinth seal system (Fig. 38(b)) has higher operating temperature and speed capability than face seal systems. However, as the turbine cooling gas pressure (P_1) increases, the leakage has a significant affect on efficiency. Conventional face seal technology can be used up to pressures of 86 N/cm^2 (125 psi), to temperatures of 427° C (800° F), and to sliding speeds of 107 m/sec (350 ft/sec) (Ref. 76). These seal studies showed that operation at higher pressures, temperatures, or sliding speeds was unsatisfactory from a wear and leakage standpoint; and subsequent analysis revealed that the seal limitations were primarily due to thermal deformations that induced seal imbalance, which in turn, caused high wear and leakage.

Results of various studies (Refs. 76 and 77) reveal that of all the various contact seal types the face contact seal has the highest pressure and speed capability. However, the face contact seal cannot meet the temperature, pressure and speed conditions of some modern engines. Therefore, labyrinth seals are used. A look into the future (Fig. 39) shows an even greater gap between contact seal capability and engine temperature, pressure, and speed requirements. These projections are based on past history of engine development and on presently desired operating goals. During the past 20 years, seal sliding speeds have doubled, and sealed air pressures and temperatures have had corresponding increases.

It is anticipated that advanced engines will have compressor discharge temperature of approximately 760° C (1400° F). The seals for these engines will having sliding speeds of 152 to 183 m/sec (500 to 600 ft/sec).

Recent studies on self-acting seals have demonstrated these improved face seals overcome the major drawbacks of a conventional seal; these drawbacks are, rubbing contact and a gas film with inadequate film stiffness (Ref. 78).

Figure 40 is a cross section of a face seal with self-acting lift geometry. As with a conventional face seal, it consists of a rotating seat that is attached to the shaft (all rotating parts are shaded) and a nonrotating seal head assembly that is free to move in an axial direction and thus accommodate engine thermal expansion (axial). The secondary ring (piston ring) is subjected only to the axial motion (no rotation) of the head assembly, and several springs provided mechanical force to maintain contact at start and stop. In operation, the sealing faces are separated a slight amount (in the range of 0.0013 cm (0.0005 in.) by action of the self-acting lift geometry, and gas leakage is from the high-pressure side (inside diameter of carbon primary ring) across the sealing dam into the bearing sump. This gas leakage pressurizes the sump and assures proper scavenging of the bearing lubricant. It should be noted that, although the sealed gas temperature is high, tests have shown that considerable gas temperature drop occurs in the leakage flow so that the leakage into the sump does not pose a fire hazard when the seal is operating properly.

The self-acting lift pads consist of a series of shallow recesses arranged circumferentially around the seal under the sealing dam as shown in Figs. 41 and 42. An important point is that the lift pads are bounded at the inside diameter and outside diameter by the sealed pressure, P_1. (This is accomplished by feed slots that communicate with the annular groove directly under the sealing dam.) Therefore, a pressure gradient due to gas leakage occurs only across the seal dam. Thus the effects of force changes due to seal face deformation are minimized (Ref. 78).

The self-acting lift pads develop high forces at operating speeds to prevent rubbing contact (Ref. 78). For instance, calculations of lift pad force (Fig. 43) indicate a 176 kg (80-pound) lift force at a 0.0005 cm (0.0002 in.) gap height with 217 N/cm^2 (315 psi) air and a 152 meter (500 foot) per second sliding speed. However, if the gap opens (e.g., to 0.0025 cm (0.001 in.)), the lift force drops markedly to 2 kg (4 pounds). Thus, there is little tendency for the lift pads to hold the seal open at large gaps where leakage will be high. The lift pads, therefore, have a force-against-gap-height characteristic (high-gas film stiffness) which makes lift pad incorporation inherently suited to seal operation. Thus the lift pads provide the following important features generally missing in conventional face seals: (1) high-gas film stiffness that allows the head to dynamically track the seat face motions without rubbing contact and (2) the ability to operate with divergent face deformation (see Ref. 78).

Figure 44 shows the condition of the primary seal ring after 338 hours of running. The original polishing scratches are still visible on the land areas of the pads. Thus, the seal was operating without rubbing contact (except at start and stop). The seal had a total of 40 starts and stops during the 338 hours.

Accessory and Transmission Seals (Oil Lubrication)

The usual concern in contact seals is the wear caused by sliding motion at the sealing surfaces. However, oil coking (and related seal failure) due to heat generation in the seal is increasingly an area of concern. The higher heat generation is a result of increasing speeds coupled with higher ambient temperatures.

A transmission face contact seal is basically the same as the mainshaft seal except that it is oil lubricated (most mainshaft seals are constructed to run dry).

It should be noted that several mechanisms by which an oil seal is lubricated have been proposed. None have been fully accepted as applicable over the wide range of conditions at which oil seals are run. However, there is general agreement that the primary faces are separated by an oil film. High pressure industrial compressors have face seals operating at 121 m/sec (400 ft/sec) and apparently with an oil film in the primary seal.

Even before startup an oil film exists in the seal primary face. This is due to surface energy. What happens to the oil film after startup is still a matter of investigation especially for those applications in the high speed regime 91 m/sec (300 ft/sec) and higher. Some of these theories proposed as a mechanism to account for load support of an oil film are:

1. Wobble of the seal faces with respect to each other
2. Elastohydrodynamic lubrication
3. Hydrodynamic lubrication by surface waviness
4. Liquid boiling at the interface
5. Microasperity lubrication
6. Cavitation

Each of the mechanisms has been subjected to analytical studies and it has not been determined which one or ones predominate. The important point is that all theories agree that there is a rather tenacious oil film in the primary seal. This has been confirmed by experiment.

The other factor of concern is the oil coking on the seal caused by the additional local heat input due to shear of the thin oil film in the primary seal and/or rubbing contact. What is needed here is information on how to reduce heat generated by the seal.

Heat generation will be of increasing concern as the sliding speed increases in advanced engines. It is suggested that future work consider the extent and nature of the fluid film in high speed oil lubricated seals. This investigation should also include ways and means to reduce heat generated by the seal.

SUMMARY

Friction and wear in aircraft remains a problem area that deserves both basic and applied research. Apparatus such as LEED and the ion-field emission microscope are powerful tools for observing the surface on an atomic level. There is no question that adhesion and material transfer can occur with any material combination in contact. Adsorption of vapors and lubricants (monolayers and even partial monolayers) are being studied for their effects on adhesion of surfaces. Chemisorption occurs more readily than has been anticipated. Effects of crystal structure on friction and adhesion have been determined. It is possible to use trace quantities of additives in solid materials, that migrate to and concentrate at the surface, and dominate adhesion and surface chemistry. Such studies provide a method of characterizing a surface on an atomic level. On a macroscopic level, advances have been made in characterizing the dry contact between two surfaces. Coupling a computer to a surface profilometer has provided significant contact data but more convenient definition methods are required.

The adhesion theory for friction and wear is a generally accepted model. Information is needed on the role of adhesion in "run-in" wear and in fatigue damage.

Carbon-graphite composites are promising for aircraft brakes and for high-temperature self-lubricating materials; more fundamental data are needed on their friction and wear. Further, improved brake materials are needed with heat dissipation problems being more critical than friction properties.

Calcium fluoride solid lubricants in bearings continue to show promise. Graphite fluoride is a new material requiring study. Current and anticipated projects would benefit from solid lubricants like CaF_2 that are effective to >871° C (1600° F). Sliding mechanisms at 1371° C (2500° F) (in turbine exhaust) would probably be attempted if developments look feasible. In general, upgrading solid lubricant capacity in load, speed and temperature is needed along with more basic understanding of atomistic mechanisms for performance.

The ester type liquid lubricants will probably dominate the usage in aircraft turbines. A continuing effort is required to optimize the ester lubricants. In advanced engines running at higher temperatures, the autogeneous ignition temperature of esters is of concern. Therefore, more stable lubricants such as the polyphenyl ethers are attractive. Use of these fluids at higher bulk temperatures than possible with esters offers advantages in reduced weight and complexity as compared to design compromises (such as inerting) necessary for esters at higher temperatures. The key to extended capabilities for lubricants is in greater chemical stability and

improved understanding of rheology on a molecular scale. Coolant effects are becoming increasingly critical. Friction polymers or surface resins need careful study.

Elastohydrodynamic lubrication theory has provided a significant mechanistic insight on the lubrication of gears and bearings. Optical measurements of EHD film thickness and inlet starvation provide a direct measure of the lubricating effectiveness of candidate oils. EHD continues to be a very fruitful area of research.

Scoring, wear and contact fatigue remain problem areas in mechanical components like gears and bearings. A combined thin film lubrication and Blok's critical temperature hypothesis may provide basis for an improved scoring model that must include complete thermodynamic and atomistic concepts. It is suggested that the effect of gear tooth errors on lubrication be reviewed.

Future requirements for bearings are operation to 3×10^6 DN or higher and at temperatures to 316° C (600° F). A rational explanation of surface distress is now possible with the introduction of the EHD concepts including film thickness and surface roughness parameters.

Abradable seal materials are needed for operation at temperatures greater than 649° C (1200° F). Conventional contact seals cannot meet current speed, pressure and temperature levels in advanced engines. Future engines will have even more severe operating conditions. Predictions are speeds to 183 m/sec (600 ft/sec), pressures to 345 N/cm^2 (500 psi) and temperatures of 760° C (1400° F). Promising developments have been made in face seals with lift geometry. Improved oxidation resistant carbons are needed.

RECOMMENDED AREAS FOR STUDY

1. The Solid and Its Surface

   a. Definition of surfaces from atomistic to macroscopic scale of observations (includes computer program analysis of surfaces in contact).

2. Dry Sliding Contact

   a. Effect of small alloy additions on adhesion
   b. Development of carbon-graphites for brakes

3. Solid Lubricants

   a. Development of solid lubricants for operation to 871° C (1600° F); also solid lubricants to 1371° C (2500° F).
   b. Investigation of wear mechanism on atomistic scale.

4. Liquid Lubricants

   a. Rheology of lubricants at high pressures and shear rates.
   b. Synthesis and formulation for high temperature and improvement of fluid film rheology.
   c. Additive and surface chemistry to control corrosion and dirt and improved boundary and E.P. films.
   d. Investigate friction polymers.

5. Lubricant Systems

   a. Methods to improve cooling (includes techniques for improved application of lubricant, mist lubricating, and separate cooling and lubricating flows).
   b. Improved filtration
   c. Controlled atmosphere

6. Elastohydrodynamic Lubrication

   a. Investigation of entrapment and normal approach effects
   b. Differences among various lubricants

7. Gears

   a. Improved scoring model
   b. Review of the effect of gear errors on lubrication
   c. Investigation of run-in effects on surface and on profile geometry.

8. Bearings

   a. Information on differences in fatigue life between case carburized and through-hardened materials.
   b. Improved method for predicting bearing life taking into account material, lubricant and film thickness effects.
   c. Reduced stress at high speeds.

9. Seals

   a. Improved abradable material and a model for predicting abradability property.
   b. Seal carbons with improved erosion and oxidation resistance.
   c. Materials for reducing secondary seal fretting.

Also, it is recommended that support be lent to the IRG (OECD) group in work on: a. terms and definitions (and translations)
b. thermodynamics of failure.

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TABLE I

TYPES OF WEAR

I. ADHESIVE - INTERFACE WELDS

II. CORROSIVE - CHEMICAL SURFACE REACTIONS

III. ABRASIVE - ROUGH SURFACE AND FREE PARTICLE CUTTING

IV. FATIGUE - SUBSURFACE REPETITIVE STRESS FAILURE

V. EROSION - IMPACT DISPLACEMENT BY FLUID, AND ENTRAINED SOLIDS

VI. IMPACT CHIPPING - SOLID SURFACE IMPACT STRESS FAILURE

VII. MISC. - INCLUDES ELECTRICAL ATTRITION AND OTHER PHENOMENA

TABLE II. - APPROXIMATE HEAT GAIN FROM 25° C (77° F) TO 650° C (1202° F) IN BTU

Material	Btu/lb	Btu/ft^3 (vol at 77° F)
carbon	399	3.74×10^4
graphite	399	3.60×10^4
copper	113	6.21×10^4
steel	165	7.98×10^4

TABLE III. - RESULTS OF THERMAL STABILITY AND FRICTIONAL EXPERIMENTS IN VACUUM OF 10^{-9} TO 10^{-6} TORR

Compound	Probable onset of thermal dissociation (as detected by TGA) °F	Dissociation products first detected by mass spectrometry, °F	Maximum temperature at which burnished films provided effective lubrication, °F
MoS_2	1700	2000	1200
WS_2	1600	1900	1350
$MoSe_2$	1400	1800	1400
WSe_2	1300	1700	1400
$MoTe_2$	1300	1300	1000
WTe_2	1300	1300	(a)

[a]Friction coefficient greater than 0.2 at all temperatures.

TABLE IV. - LUBRICATION AND BEARING REQUIREMENTS

	Present	Future Anticipated
Temperature	-300 to 1500° F	-450 to 2500° F
Pressure	atmosphere to 10^{-3} torr	atmosphere to 10^{-6} torr
Load	0 to 100 000 psi	0 to 200 000 psi
Speeds	1 to 10 000 rpm	1 to 100 000 rpm
Duration	up to 500 hours	up to 30 000 hours

TABLE V. - SLOW SPEED ROLLER WEAR AS A FUNCTION OF FILM THICKNESS

Calculated film thickness, h, in.	Wear Description	
	Rate	Surface Appearance
$\simeq 0.2\times10^{-6}$	constant	scratches
$\simeq 0.5\times10^{-6}$	constant	diffuse scratches
$\simeq 0.8\times10^{-6}$	constant	rippled
$\simeq 1.1\times10^{-6}$	small transient then constant	rippled on faster surface
$\simeq 1.2\times10^{-6}$	large transient then constant or zero	continuous colored film on faster surface
$> 2\times10^{-6}$	small transient then constant or zero	continuous colored film on faster surface

E-6313

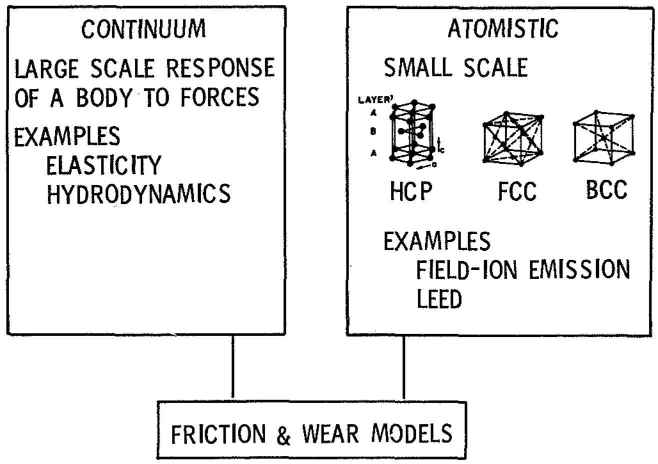

Figure 1. - Approaches for development of friction and wear models.

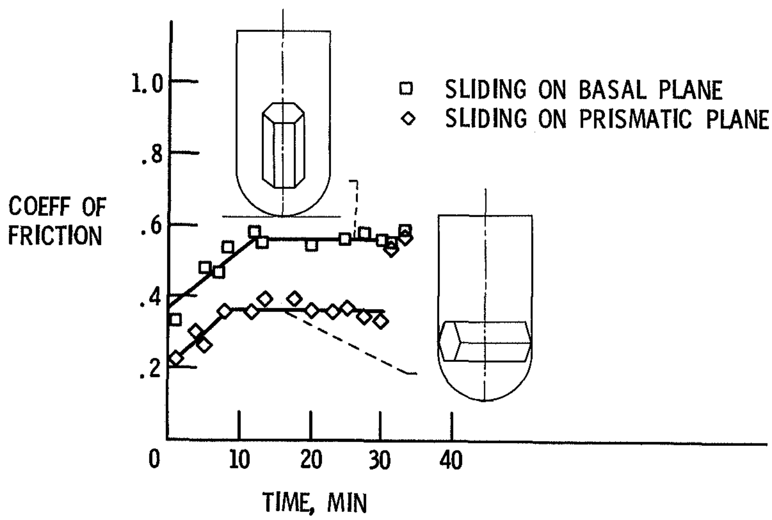

Figure 2. - Effect of crystal orientation on coefficient of friction. Single crystal titanium sliding on polycrystalline titanium (ref. 4).

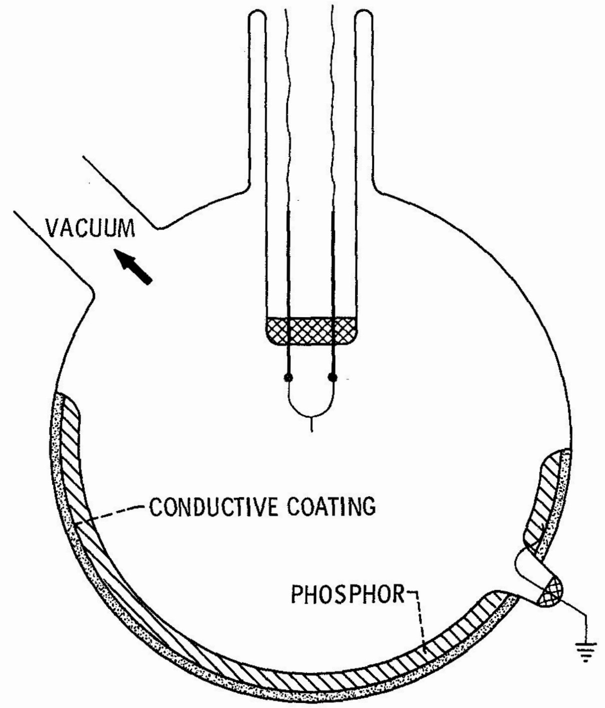

Figure 3. - Schematic representation of field-emission microscope (from ref. 6).

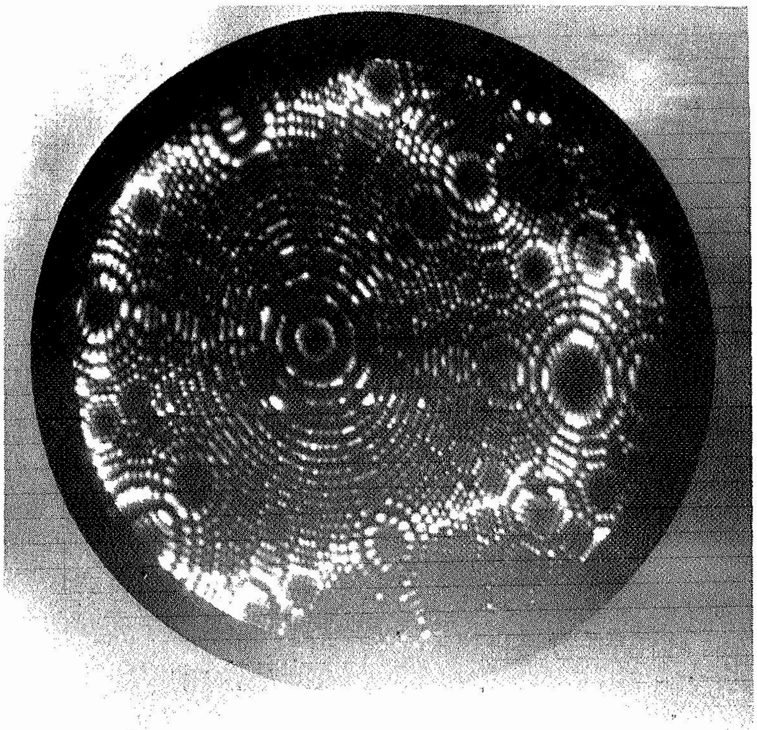

Figure 4. - Field-ion pattern of a metal tip.

E-6313

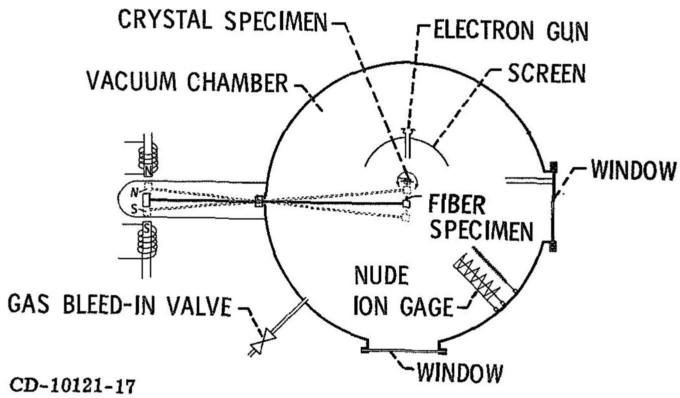

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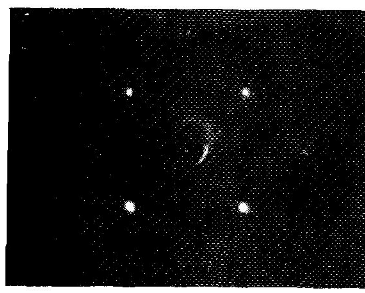
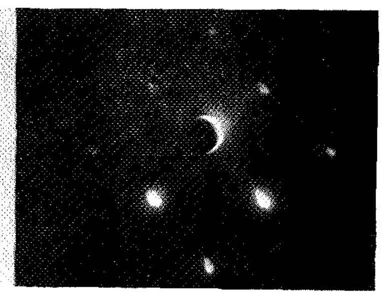
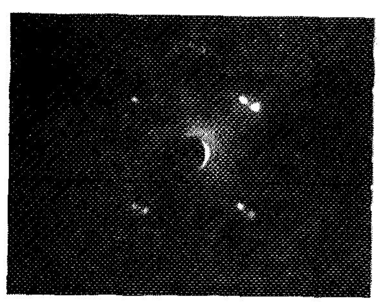
Figure 5. - Leed apparatus (ref. 4) for study of adhesion of surfaces.



(A) CLEAN COPPER (100) AT 110 V.



(B) AFTER CONTACT WITH GOLD (100) UNDER 20-DYNE FORCE AT 75 V.



(C) AFTER CONTACT WITH GOLD (100) UNDER 20-DYNE FORCE AT 140 V.

(D) AFTER CONTACT WITH GOLD (100) UNDER 20-DYNE FORCE AT 75 V AFTER HEATING TO 200^{o} C FOR 30 MIN.

CS-50759

Figure 6. - Copper (100) surface before and after adhesive contact with gold (100) surface (ref. 4).

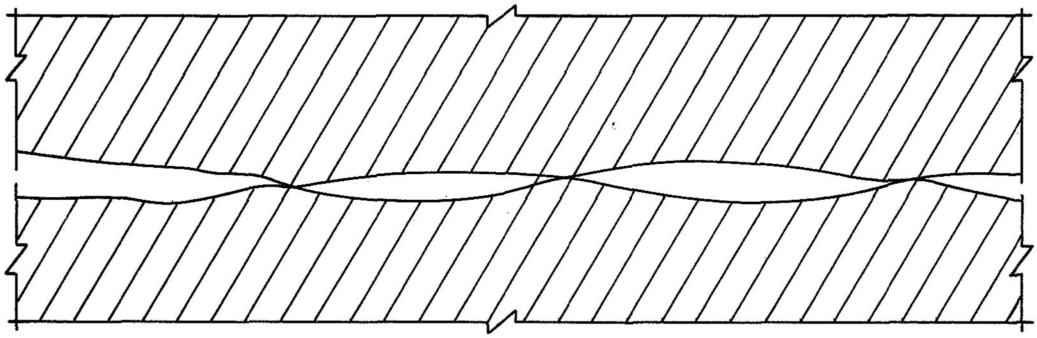


Figure 7. - Solid surfaces in contact.

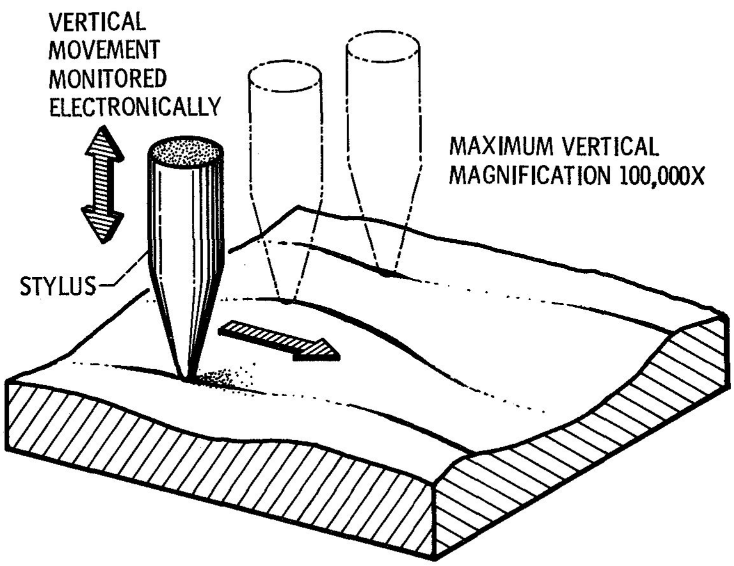


Figure 8. - Surface profile measurement by stylus method.

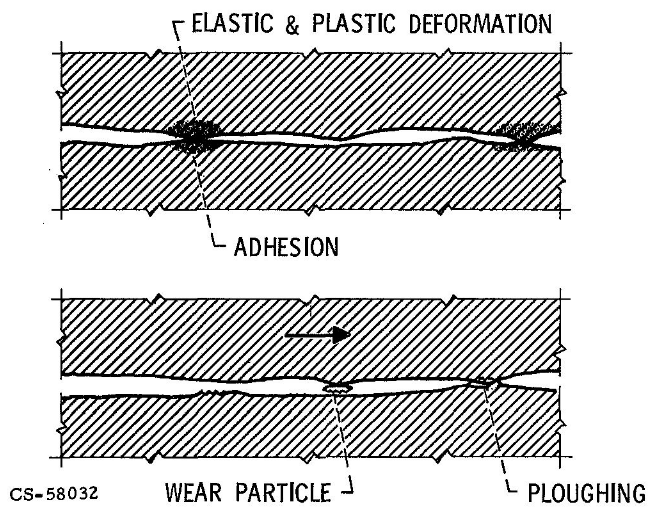


Figure 9. - Friction and wear model.

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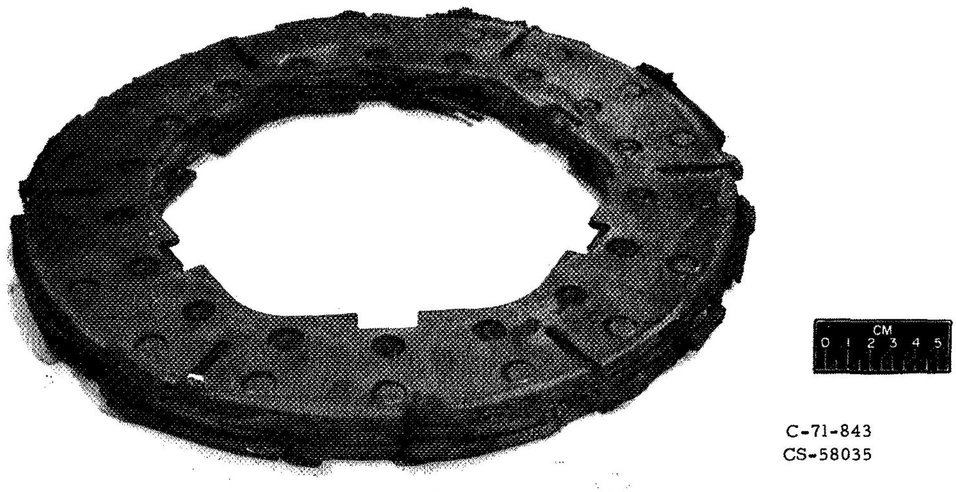

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CS-58035

Figure 10. - Brake disks after simulated rejected takeoff.

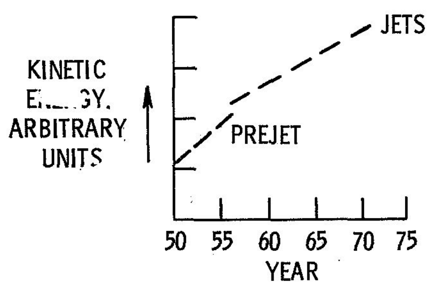
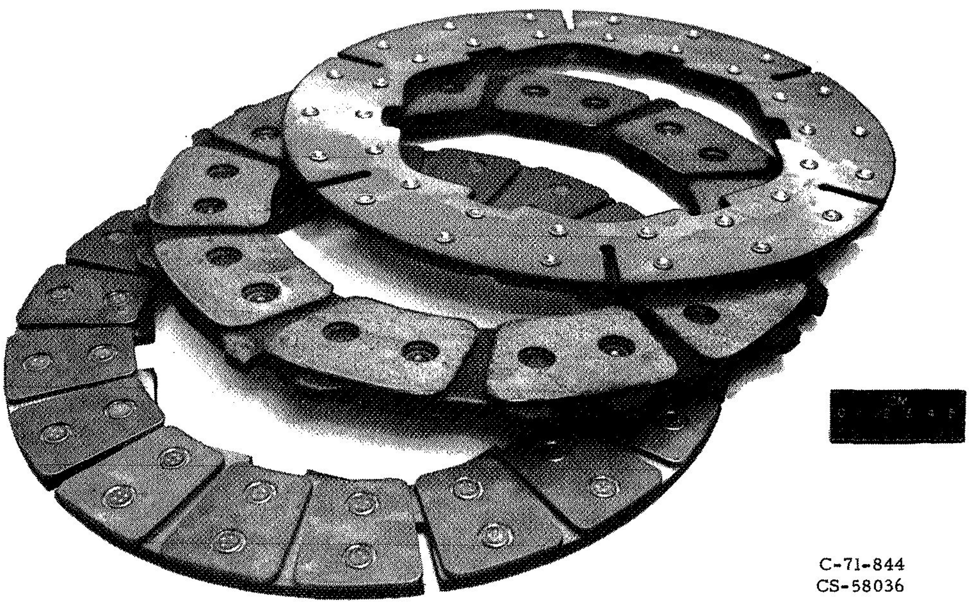


Figure 11. - Braking energy increases with time (ref. 11).

CS-58030



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CS-58036

Figure 12. - Disk brake elements.

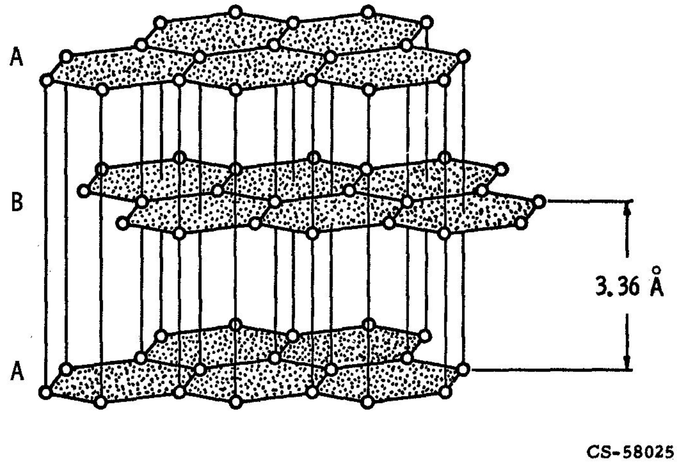
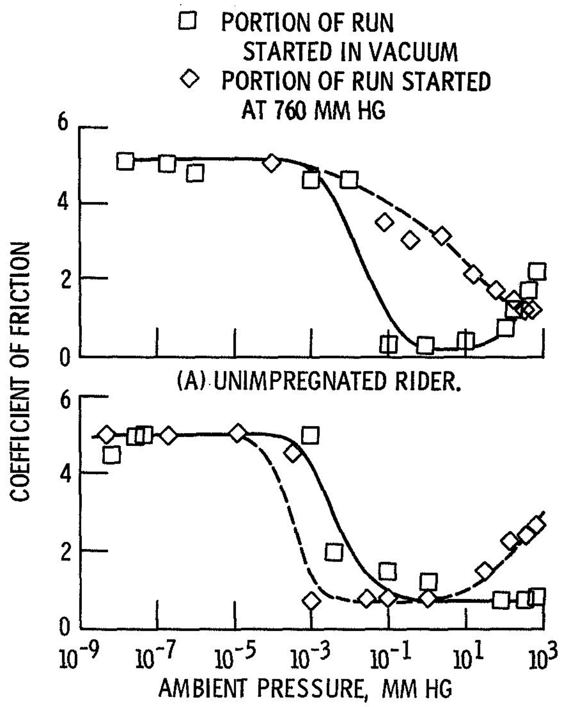


Figure 13. - Structure of graphite.



(B) RIDER IMPREGNATED WITH BARIUM FLUORIDE.

Figure 14. - Effect of ambient pressure on coefficient of friction for impregnated and unimpregnated 20 percent graphite carbon sliding on electrolytic silver at various ambient pressures. Sliding velocity, 390 feet per minute; load, 1000 grams; temperature, 75^{0} F.

E-6313

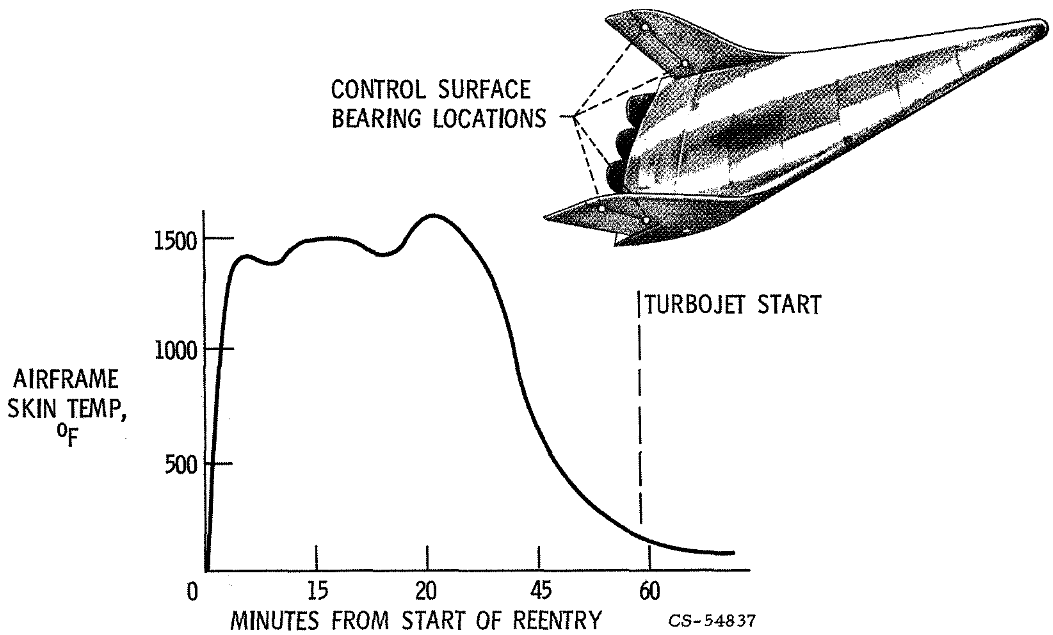


Figure 15. - Airframe temperature of shuttle reentry vehicle (ref. 26).

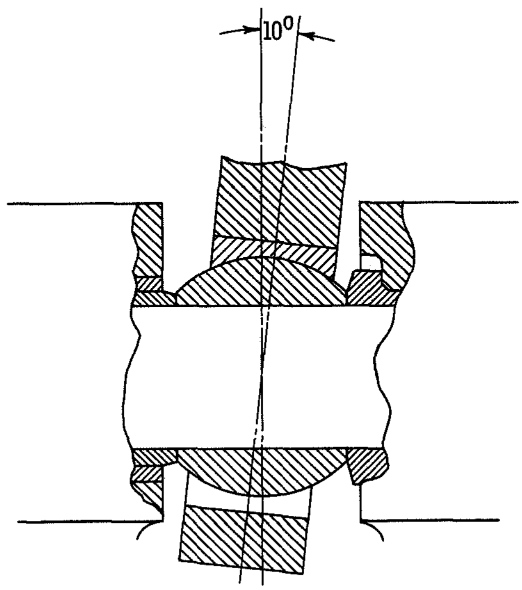


Figure 16. - Solid film lubricated engine mount support bearing.

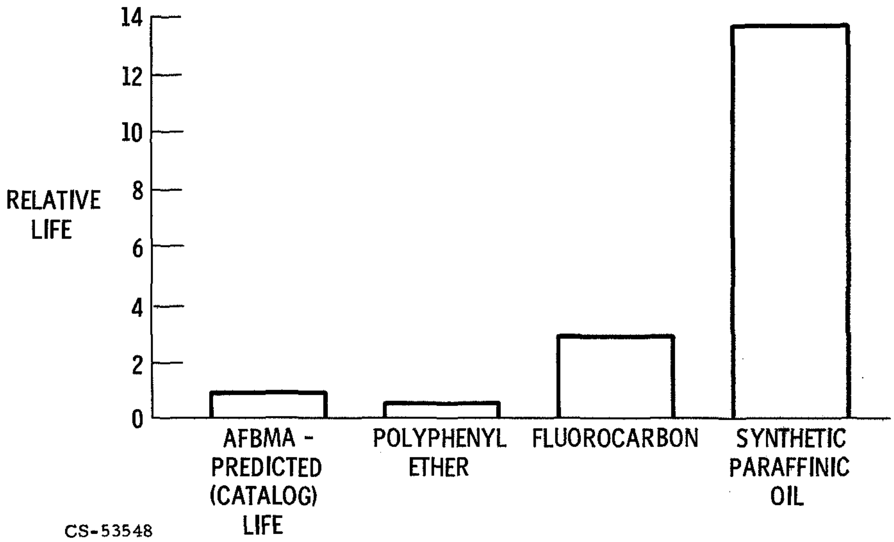


Figure 17. - Relative bearing lives with various lubricants.

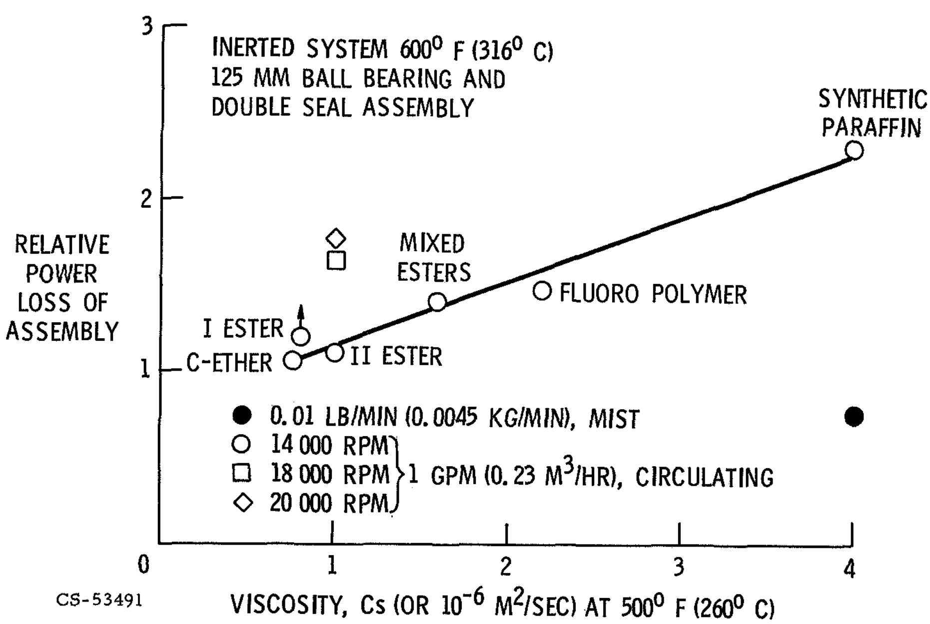


Figure 18. - Effect of viscosity, speed and lubrication method on power loss.

E-6313

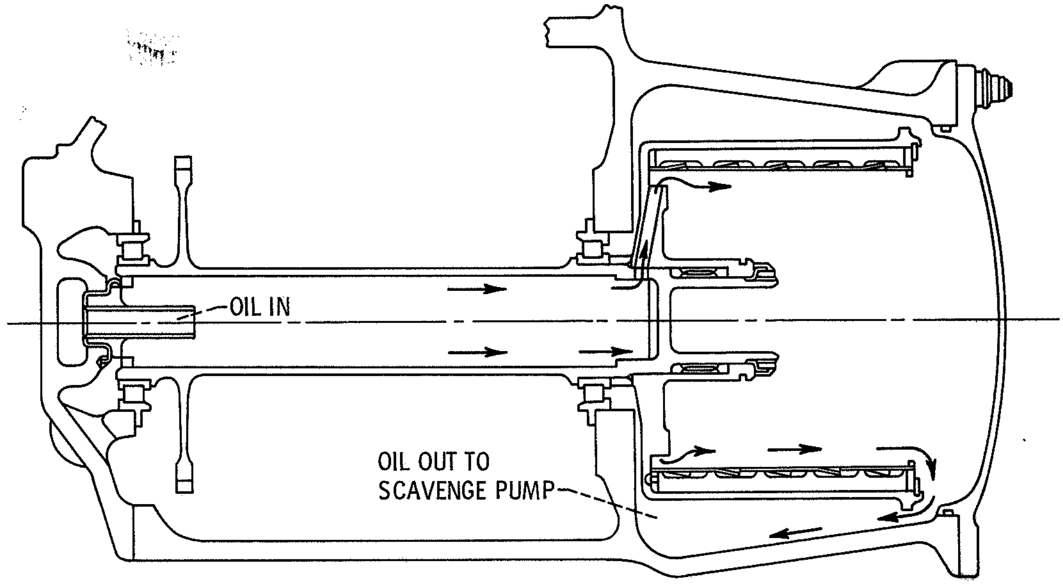


Figure 19. - Centrifugal filter.

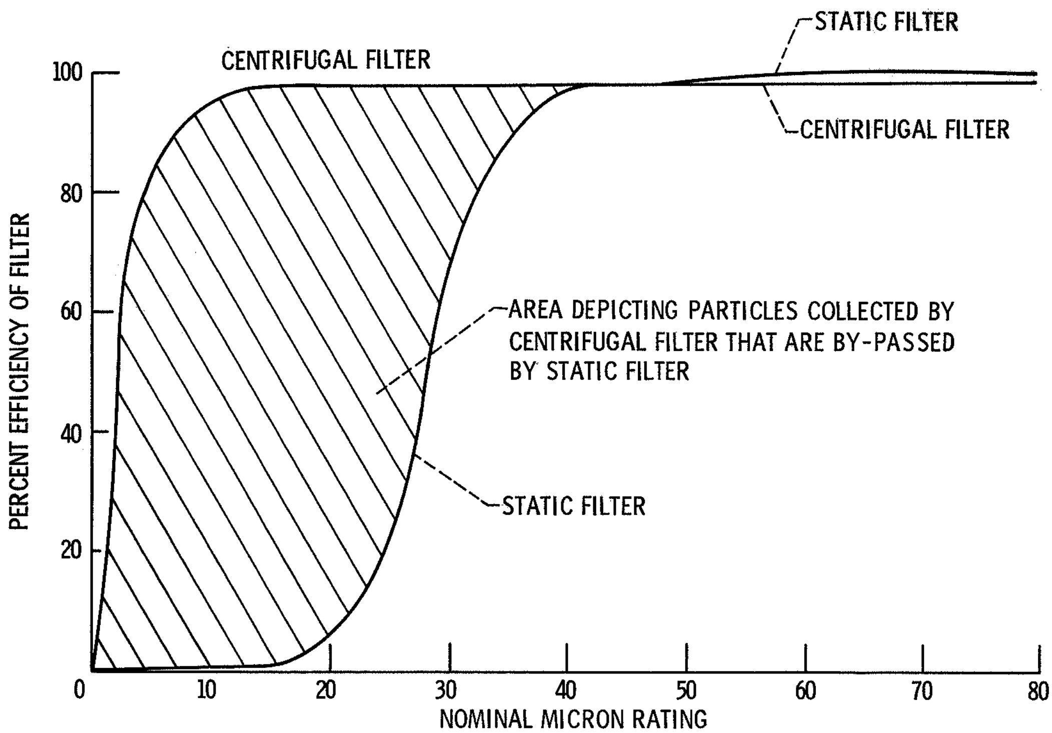


Figure 20. - Performance of centrifugal and static filters.

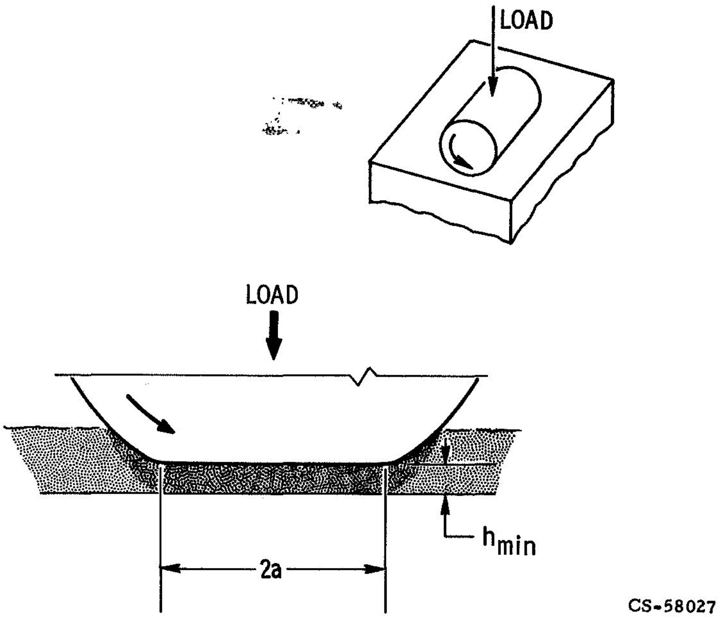


Figure 21. - Grubins model for an EHD contact.

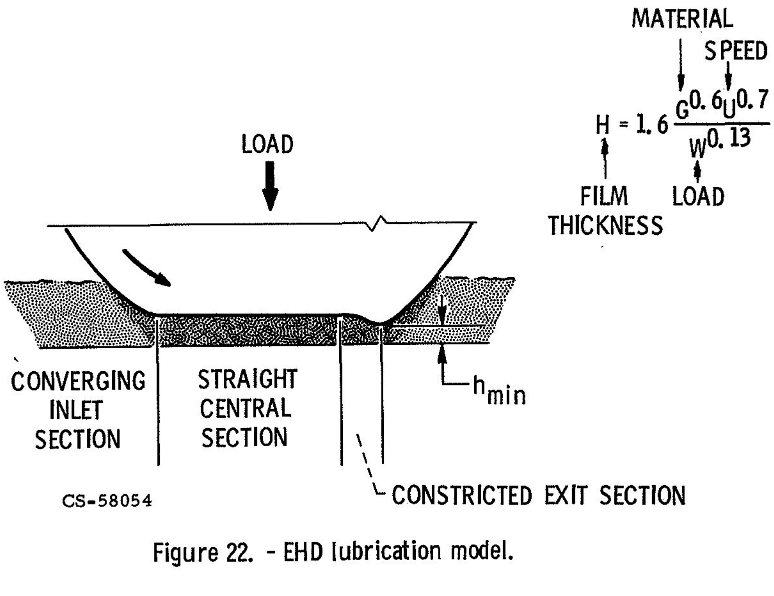


Figure 22. - EHD lubrication model.

F-6313

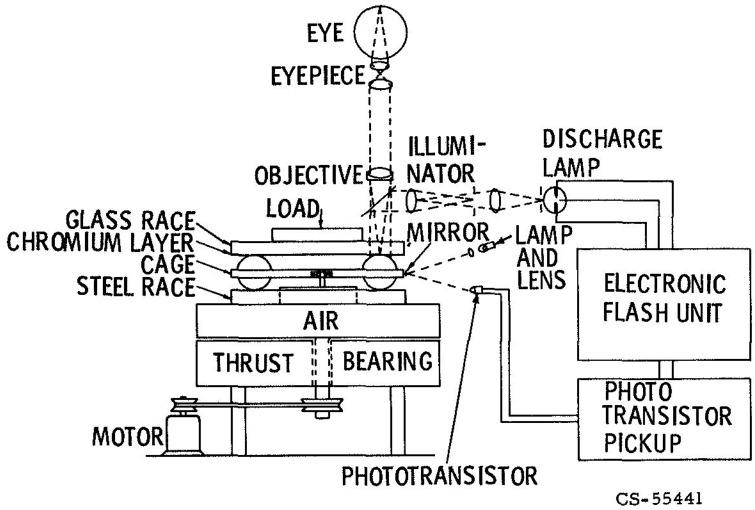


Figure 23. - Schematic diagram of apparatus.

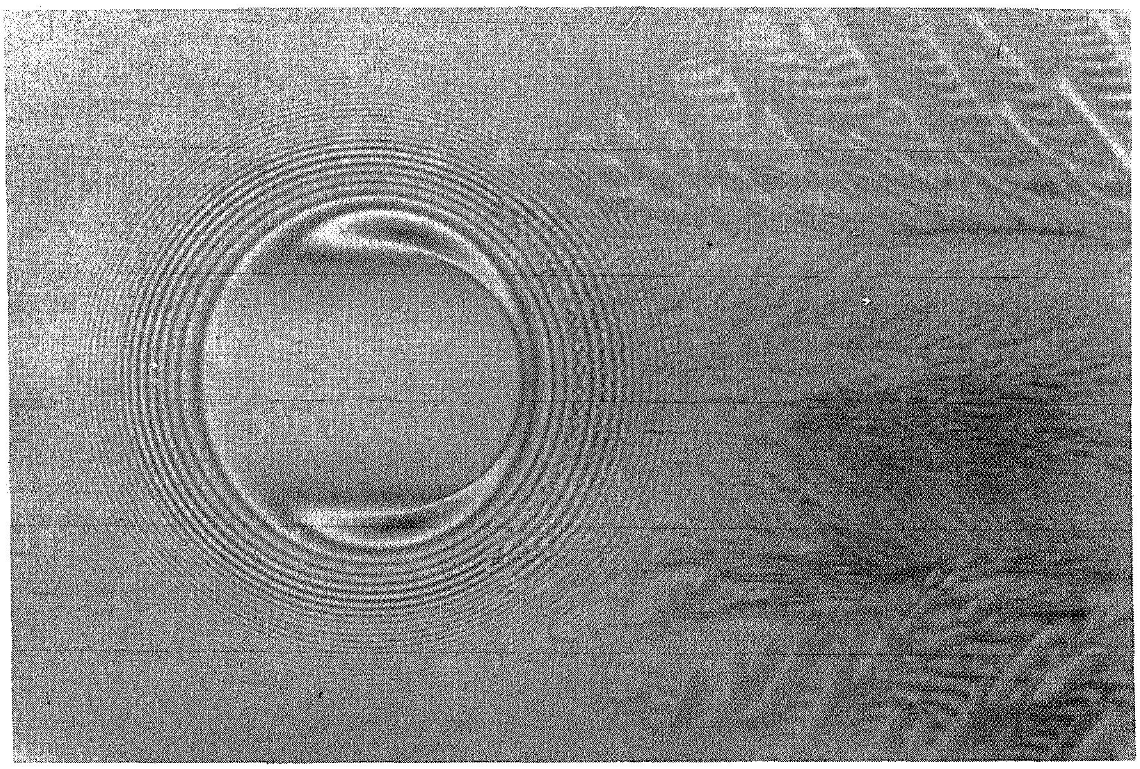


Figure 24. - Elastohydrodynamic contact.

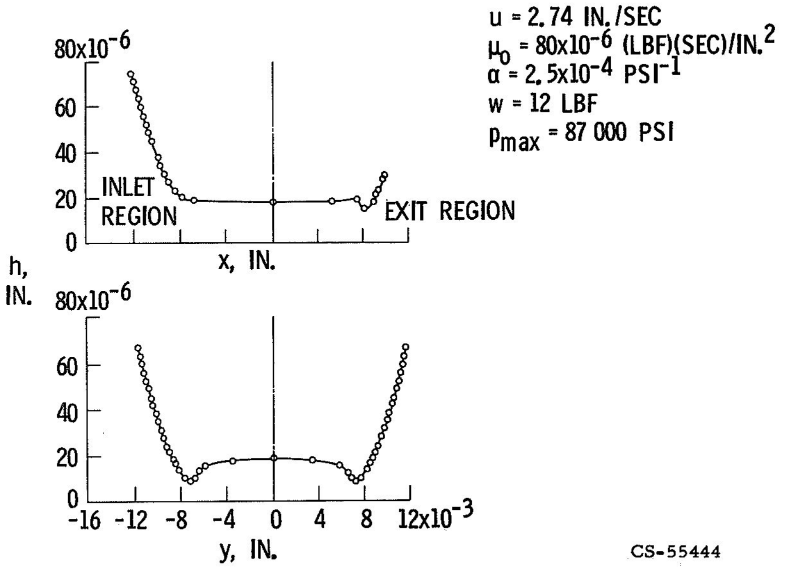


Figure 25. - Measured film shape (from ref. 57).

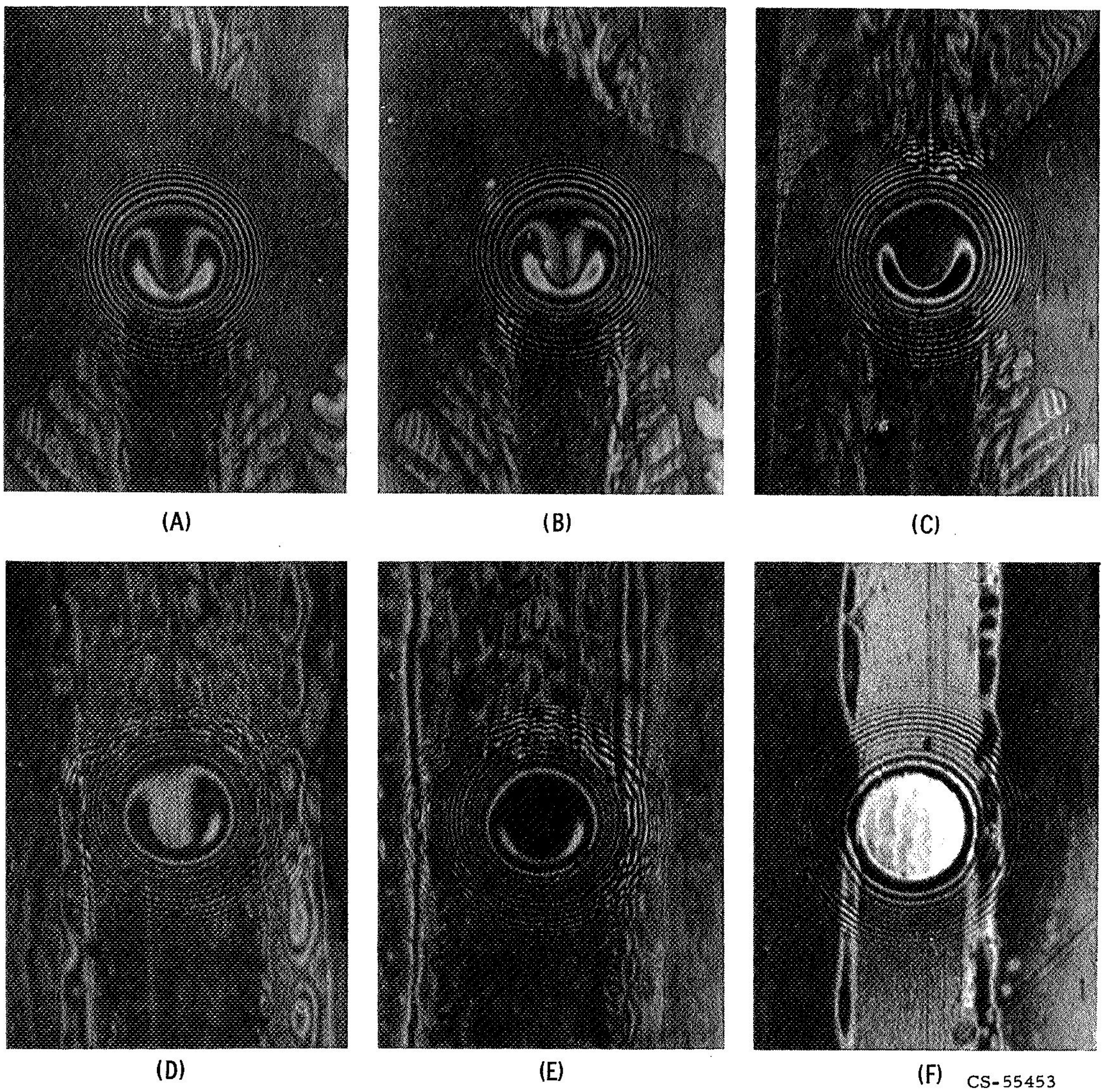
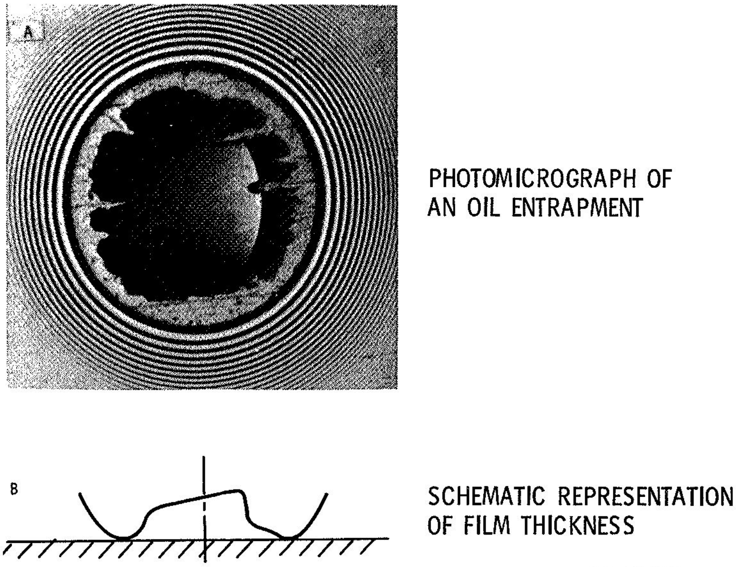


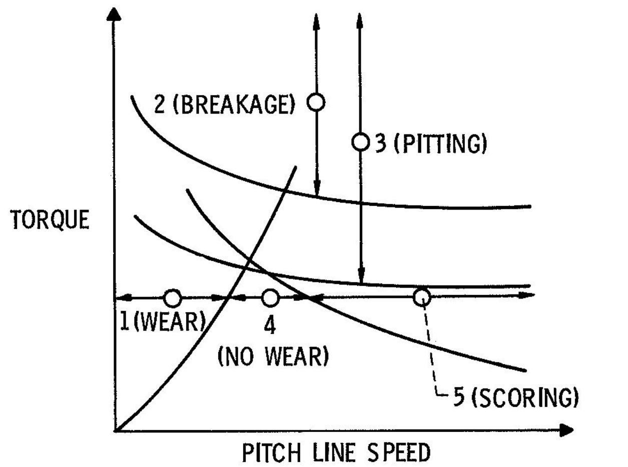
Figure 26. - Photomicrographs showing progressive starvation (from ref. 57).

E-6313



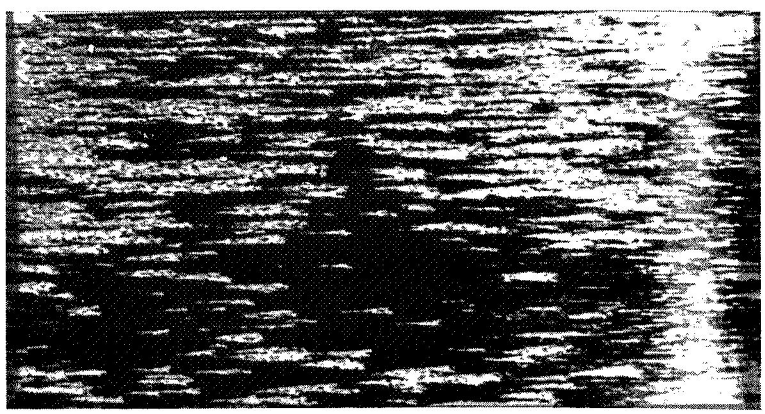
CS-55450

Figure 27. - Lubricant entrapment (from ref. 57).



CS-58028

Figure 28. - Gear failure regions.



CS-58037

Figure 29. - Score streaks on roller used to simulate gear rolling and sliding action.

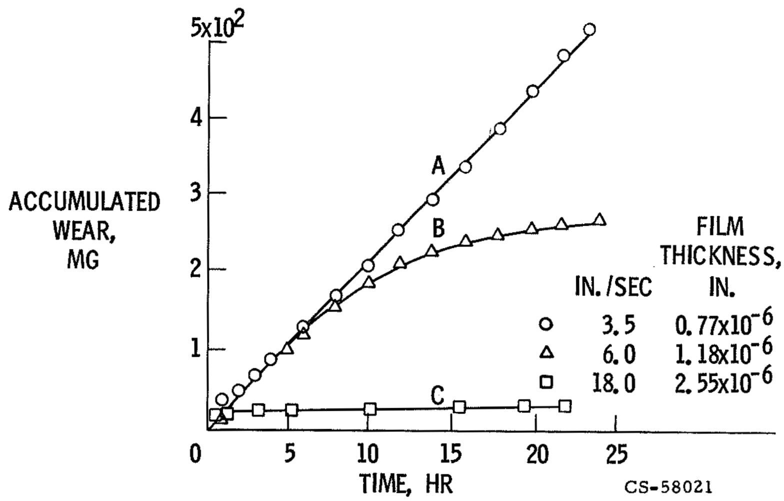
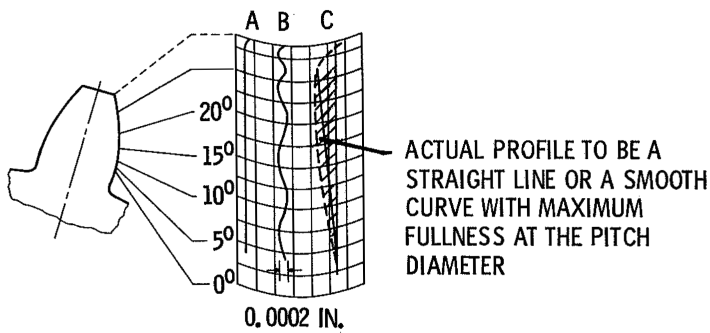
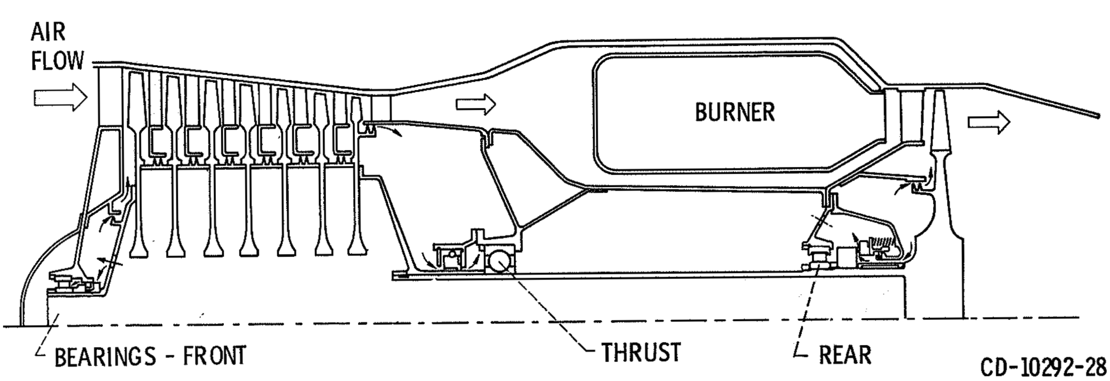


Figure 30. - Slow speed roller wear rate.



CS-58053

Figure 31. - Gear tooth profile errors.



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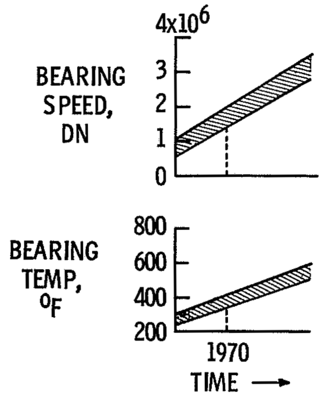
CS-49806

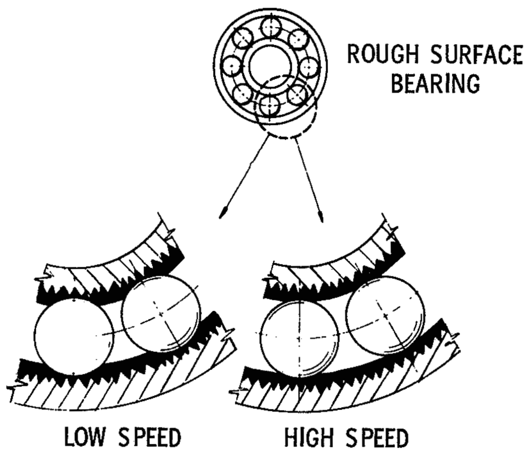
Figure 32. - Engine schematic showing bearing locations.

E-6313



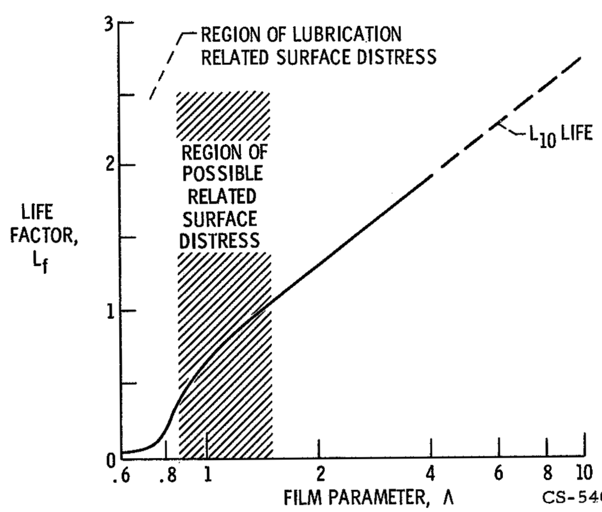
CS-56858

Figure 33. - Advanced engine bearing requirements.



CS-58023

Figure 34. - Influence of speed on EHD film thickness (ref. 74).



CS-54096

Figure 35. - Effect of EHD film parameter on ball bearing life (ref. 23).

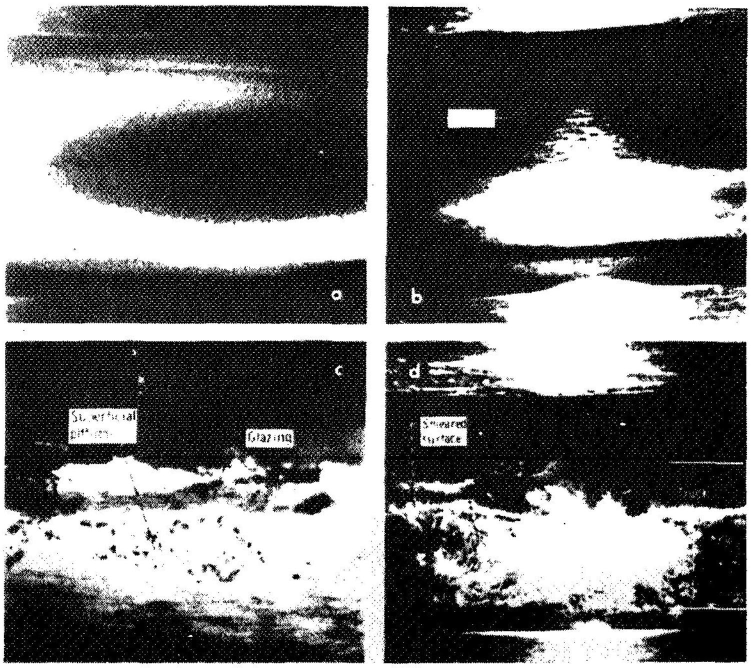
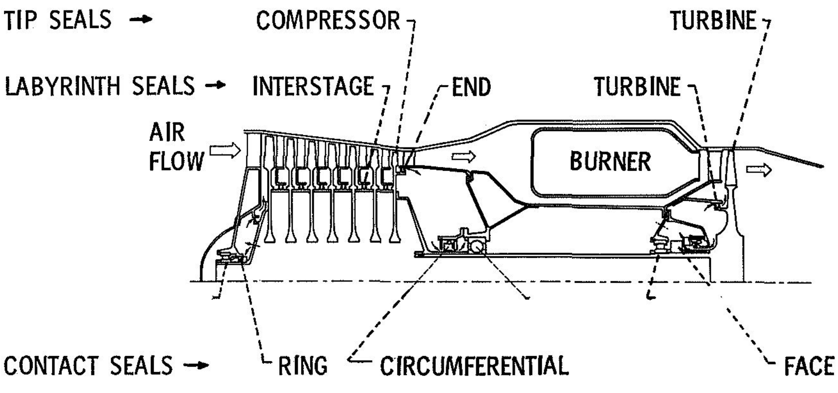


Figure 36. - Effect of EHD lubrication on surface damage to bearing races. Full EHD film results in normal race appearance, a. Marginal EHD film results in race glazing, b; glazing and superficial pitting, c; and gross plastic deformation or smearing, d.



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CS-56848

Figure 37. - Engine schematic showing seal locations.

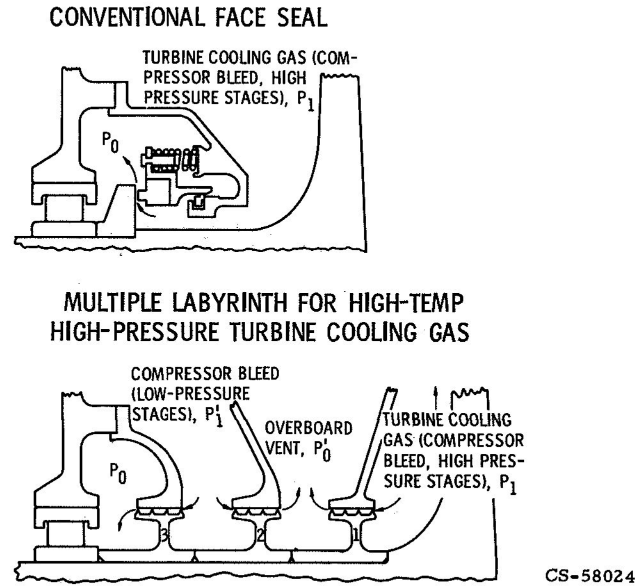

Figure 38. - Seal systems.

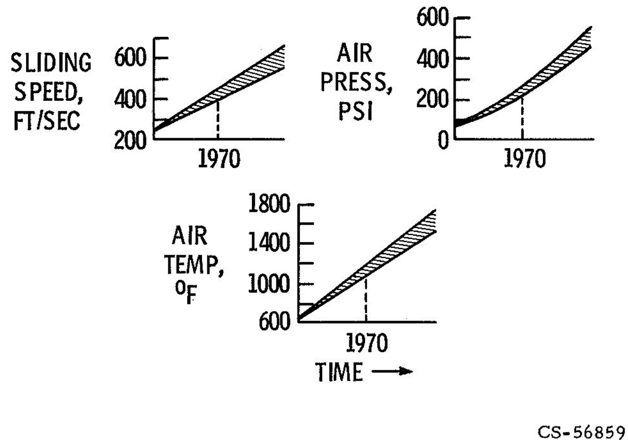

Figure 39. - Mainshaft seal environment.

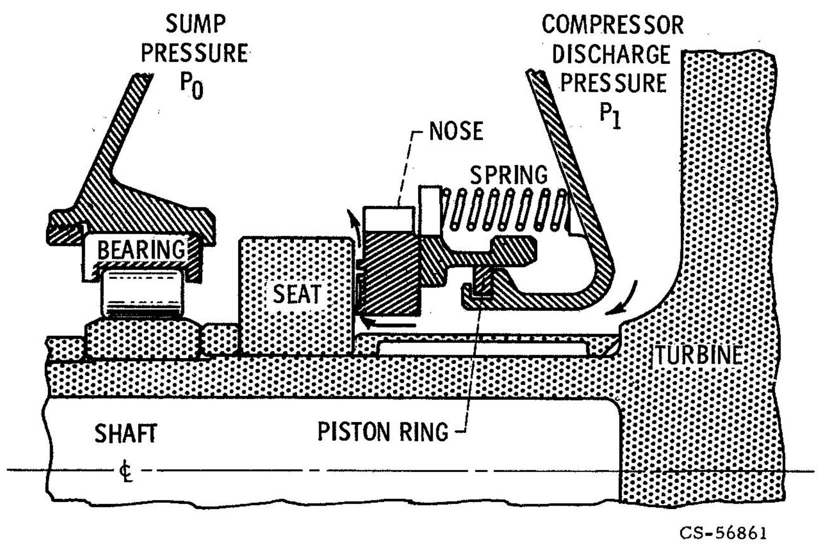

Figure 40. - Self-acting face seal.

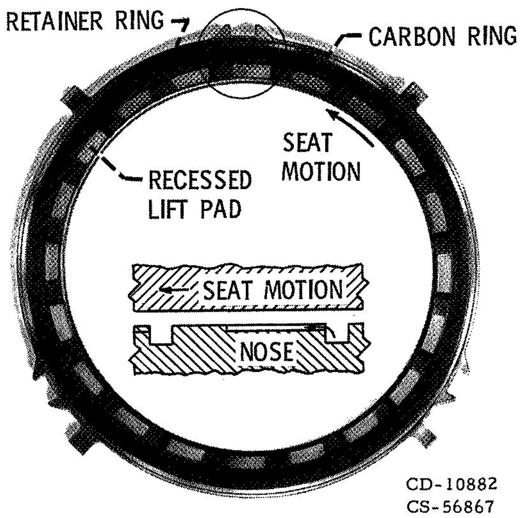

Figure 41. - Primary ring assembly.

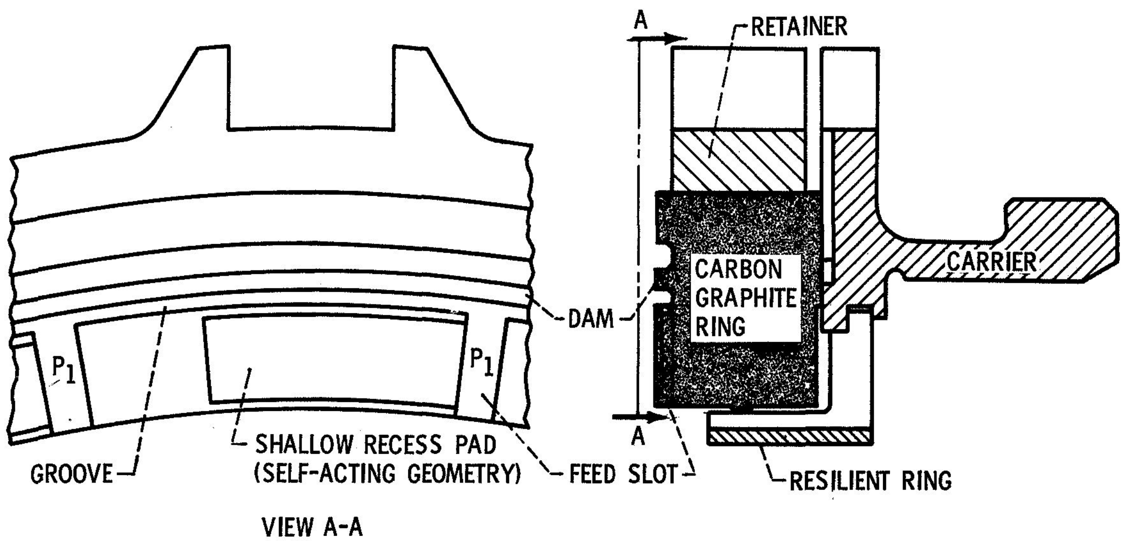

Figure 42. - Self-acting seal with pads on primary ring.

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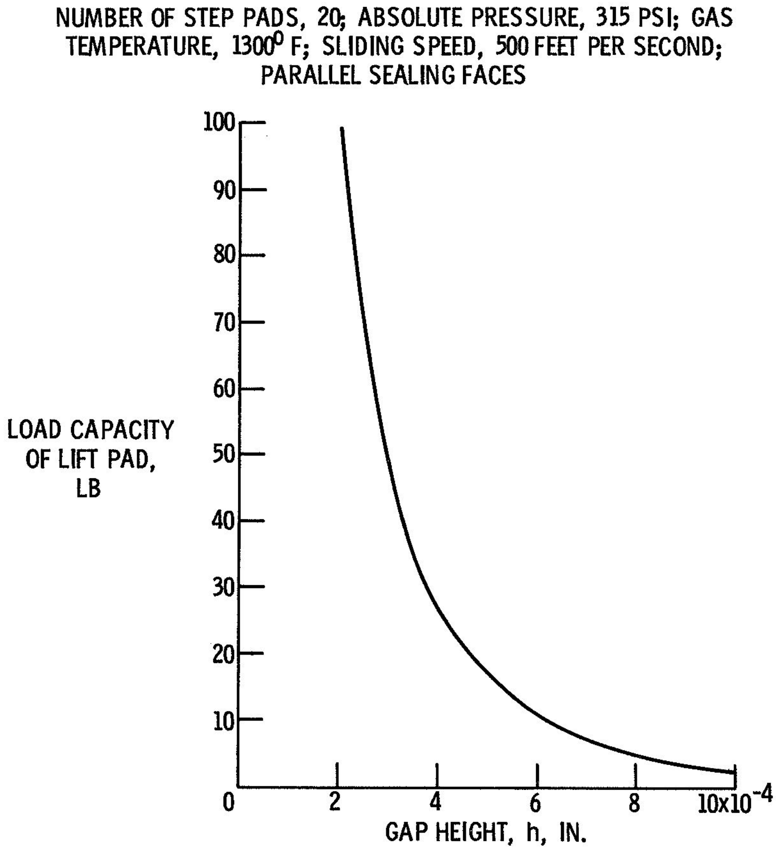

Figure 43. - Calculated load capacity of lift pad portion of seal.

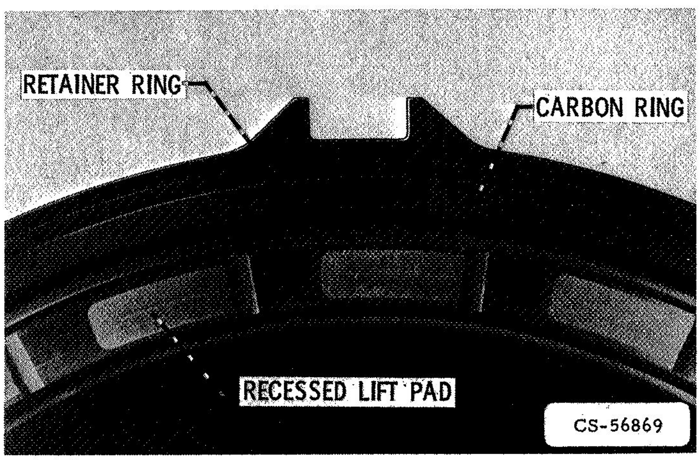

Figure 44. - Section of primary ring assembly.