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FEASIBILITY STUDY OF TECHNIQUES
TO PROTECT MECHANISMS OPERATING
IN SPACE FROM MALFUNCTION

PART I - SURVEY AND ANALYSIS

November 30, 1964

Mechanical Engineering Division
IIT Research Institute
Technology Center
Chicago, Illinois 60616

Technical Summary Report - K6055

FEASIBILITY STUDY OF TECHNIQUES TO PROTECT MECHANISMS OPERATING IN SPACE FROM MALFUNCTION

PART I - SURVEY AND ANALYSIS

28 June 1963 - 27 June 1964

by

W. E. Jamison

Prepared for the

National Aeronautics and Space Administration
George C. Marshall Space Flight Center
Huntsville, Alabama

November 30, 1964

FOREWORD

This technical summary report on IITRI Project No. K6055, "Feasibility Study of Techniques to Protect Mechanisms Operating in Space from Malfunction," covers the work performed during the period June 28, 1963 to June 27, 1964. The work was sponsored by the National Aeronautics and Space Administration, George C. Marshall Space Flight Center under Contract NAS8-11014, with Dr. W. R. Eulitz acting as the contracting officer's technical representative.

This work was performed by the Fluid Systems and Lubrication Section of the Mechanical Engineering Division, under the management of Mr. F. Iwatsuki. Principal investigator was Mr. W. E. Jamison. Support was provided by Drs. C. Riesz and A. Dravnieks and Mr. H. Weber of the Chemistry Research Division for the contact potential measurements and by Drs. P.R.V. Evans and R. Elliott and Mr. D. Warwick of the Metals Research Division in the preparation and analysis of the experimental materials.

FEASIBILITY STUDY OF TECHNIQUES TO PROTECT MECHANISMS OPERATING IN SPACE FROM MALFUNCTION

ABSTRACT

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A literature survey and analysis of currently available anti-friction techniques for space mechanisms has been conducted to determine their operational characteristics and limitations in terms of environmental parameters. These data are presented in a form useful to designers with direct hardware responsibilities.

The techniques are assessed for their underlying theoretical principles and predictions are made of their performance potentials.

It is concluded that significant improvements in performance of space mechanisms must be preceded by a better understanding and control of friction and wear properties of interfaces. Specific recommendations are made for research and development to accomplish this.

Author

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I. INTRODUCTION

A. Purpose of the Study

The purpose of this study is two-fold. First, an attempt has been made to catalogue existing anti-friction techniques and materials for space mechanisms and to define their operational characteristics and limitations in terms of environmental parameters. These data are presented in a form useful to designers and engineers with direct hardware responsibilities.

The second, and perhaps more important objective is a critical analysis of current space lubrication practices. Each anti-friction technique and each material is appraised for the fundamental principle by which friction and wear are reduced. The soundness of the theory and the extent to which it applies in practice are assessed. Predictions are made of the ultimate limitations of the various techniques and materials.

This analysis will be most useful to the long-range planners whose responsibility is to assure that low friction and wear techniques and materials will be available to meet future space vehicle requirements.

B. Scope

It is assumed that the reader is familiar with the space environmental parameters and their effect on mechanical components. These are thoroughly documented elsewhere[1-6], as are the techniques currently popular for lubricating space devices[7-11].

This report is an assessment of these techniques for providing effective, reliable operation of devices to meet current and future space hardware requirements. The evaluation is based on the theoretical principles underlying each technique, and predictions of utility are made from both environmental test data and from the theoretical limitations of the technique.

For the most part, it is assumed that encapsulating or otherwise shielding the mechanisms is impractical. Such obvious techniques as hermetic sealing are excluded from this study.

II. PERFORMANCE ANALYSIS OF EXISTING ANTI-FRICTION TECHNIQUES

A. Operational Requirements

In order to fulfill the objectives of present and future extraterrestrial flights, mechanical components must operate in the range of conditions listed in Table 1.

TABLE 1 - OPERATIONAL REQUIREMENTS FOR SPACE MECHANISMS

	Min. Expected Value	Max. Expected Value
Temperature	20°K	1650°K
Load Capacity	0	141 kg/mm^2 (200,000 psi)
Life	Few Hours	30,000 Hours
Environmental Pressure	760 Torr	10^{-13} Torr
Speed	Few cps (rpm)	1666 cps (100,000 rpm)
Reliability	98%	100%

In addition to those listed, other parameters which must be considered in the design of space mechanisms include:

1. Friction level
2. Type of motion
3. Load to weight ratio
4. Radiation resistance
5. Vibration resistance
6. Auxiliary equipment requirements
7. Power requirements
8. Pre-launch environment and handling

Of course, no single device will be subjected to all of the operational extremes listed in Table 1. However, requirements do exist for nuclear power generator turbine bearings which will operate at tens of thousands of rpm for years in a high radiation flux, and for airframe control hinges which may sit motionless at 10^{-13} torr and 20°K for months before they must operate in a searing 1600°K reentry.

B. Performance Levels of Existing Anti-Friction Techniques

The techniques by which designers may effect low friction operation of mechanisms are catalogued in the following categories:

1. Oil lubricated devices (Boundary lubrication)
2. Unlubricated devices
3. Solid film lubricated devices
4. Flexure devices
5. Electric and magnetic support
6. Fluid film lubricated devices (Hydrodynamic, self acting, etc.)

Each technique is discussed in detail in Section IV. This section discusses only the optimum values of each class to establish comparative performance levels.

1. <u>Temperature</u>

Figure 1 shows the operating temperature ranges for the various classes of anti-friction techniques. Ceramic and solid film lubricated bearings operate well at cryogenic temperatures; most test data have been obtained at liquid nitrogen temperatures (77°K) with little testing at liquid hydrogen temperatures (20°K). Gas and superconducting magnetic bearings offer potential in this area for light loads.

At the high temperatures, only ceramic and gas bearings can presently satisfy requirements for continuous rotary motion. The poor wear life and low reliability of ceramics and the high weight penalty associated with gas bearing systems points out the need for development progress in this area.

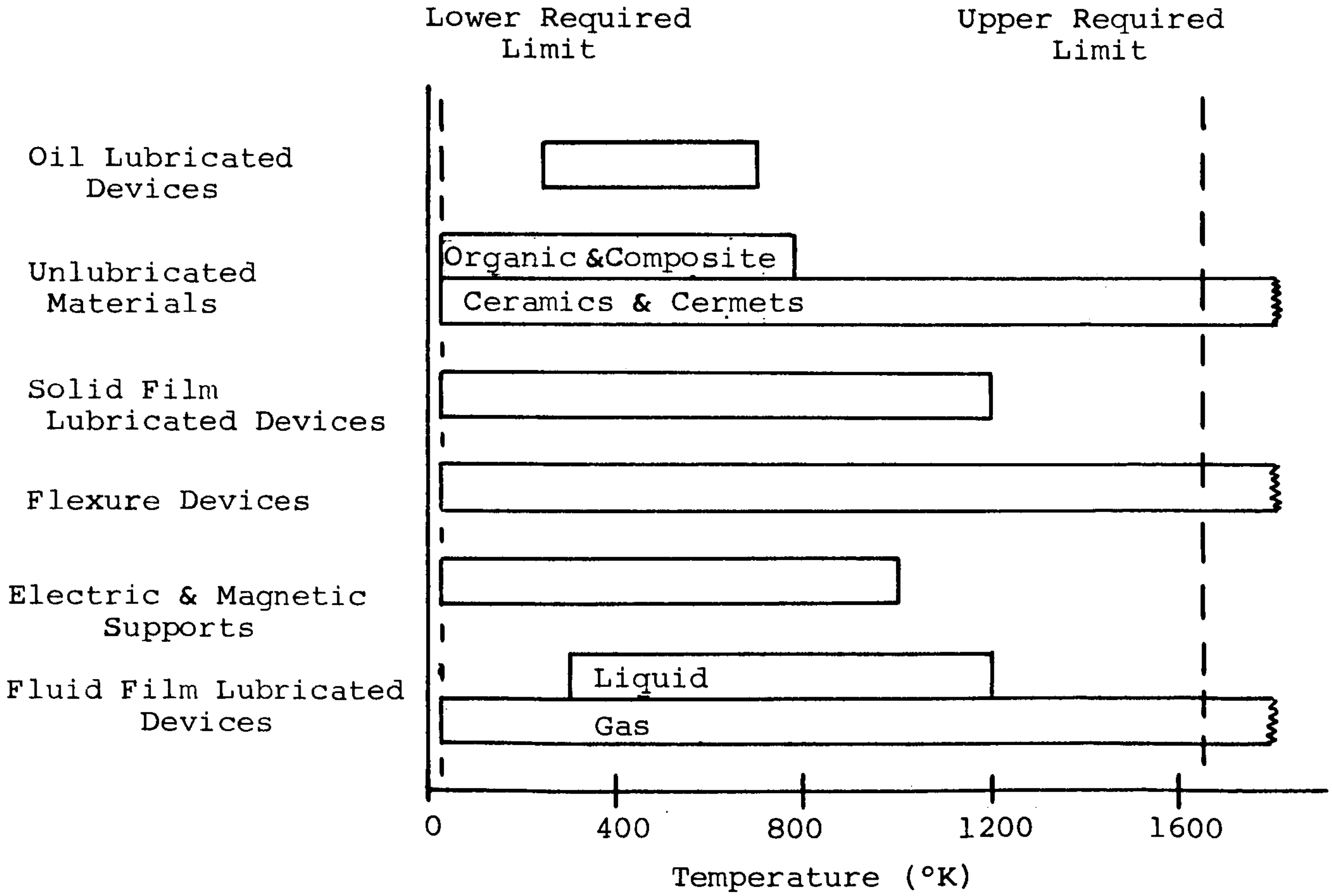

Fig. 1 TEMPERATURE CAPABILITIES OF ANTI-FRICTION DEVICES

2. Load Capacity

Figure 2 shows the maximum load capacity for the various anti-friction techniques. Although most mechanisms can be designed to operate within the load capability of existing techniques, the requirement for 200,000-psi bearings cannot be satisfied using existing materials. Since wear life is closely related to load capacity for most lubrication techniques, any improvements in maximum operating stress levels will probably reflect increased life and reliability.

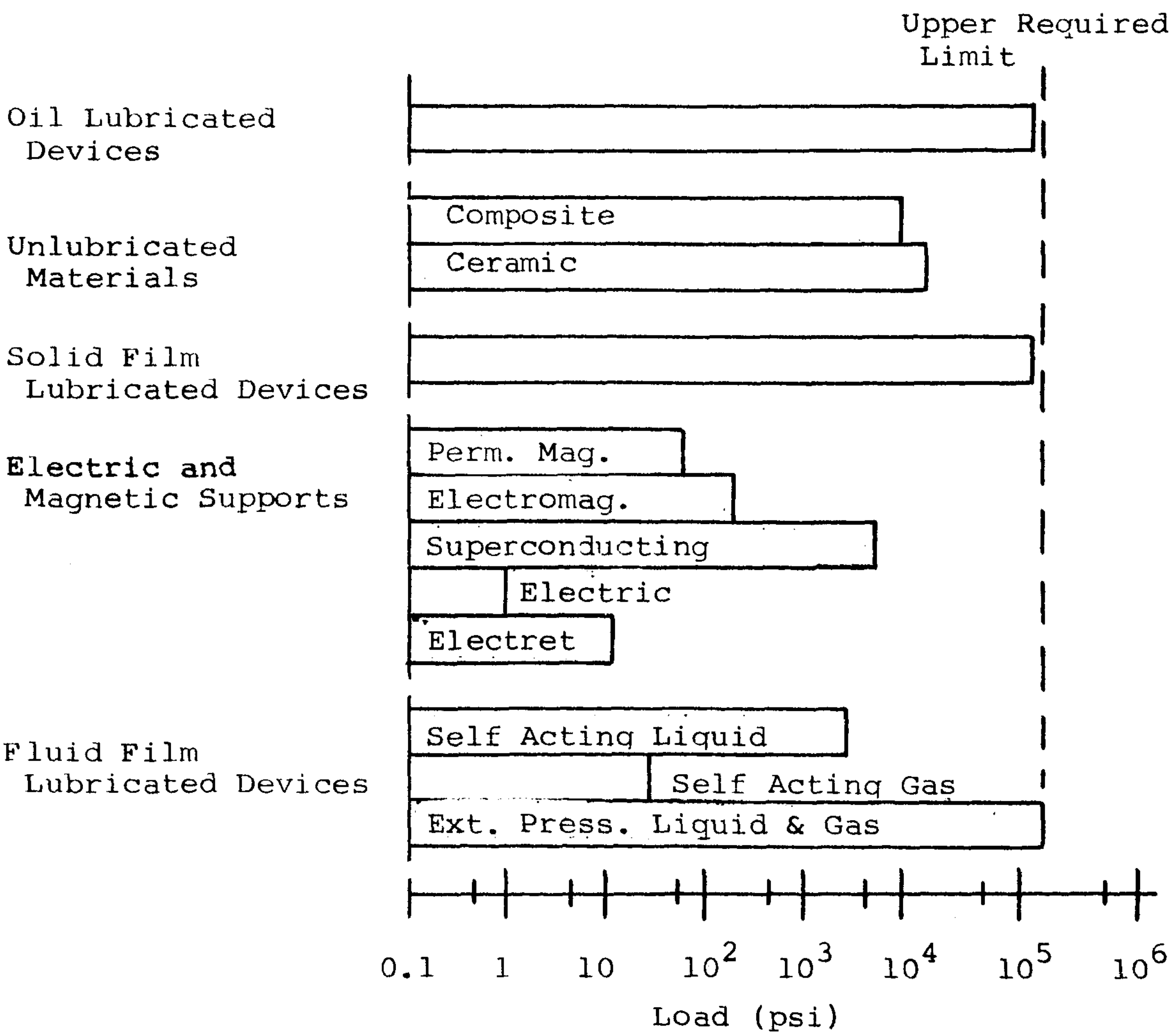

Fig. 2 LOAD CAPABILITIES OF ANTI-FRICTION DEVICES

3. Life

The life of mechanical components is usually a function of load, speed, environmental conditions and material properties. Since the operating conditions vary over extremes for different applications, only a qualitative assessment of the various techniques can be performed. The life of oil lubricated devices in space is related to the rate of loss of lubricant due to evaporation. Using reservoir techniques, life can be extended indefinitely. Ceramics show only moderate wear rates, but failure of ceramic bearings is usually due to catastrophic disintegration; an unpredictable event. Composites and solid film lubricants can be made to last thousands of hours in high vacuum chambers under mild operating conditions, but lives of 30,000 hours probably cannot be expected. Flexure devices offer infinite life, but their utility is limited. Complete separation of bearing elements by fluid film or electric and magnetic support also permits extended life.

4. Environmental Pressure

The empirical derivation of our lubrication, friction and wear technology, coupled with our poor understanding of materials behavior in ultra high vacua, makes prediction of the effects of vacuum on mechanisms difficult. Such gross mechanisms as evaporation, dissociation, etc. have been studied for their influence on bearing operation. However, it is becoming clear that the interaction of bearing surfaces and lubricants with microquantities of contaminating species plays a profound role in promoting low friction and wear. Long term operation in the extreme vacuum environment may deplete these microconstituents in otherwise satisfactory systems. A further understanding of first principles of surface physics is essential before the effects of space vacuum on mechanisms can be predicted.

5. Speed

Figure 3 shows the speed ranges of the various techniques which can provide continuous rotary motion. No mechanical problems are foreseen in extending operation to 100,000 rpm, although frictional heat will be hard to dissipate without a convective atmosphere.

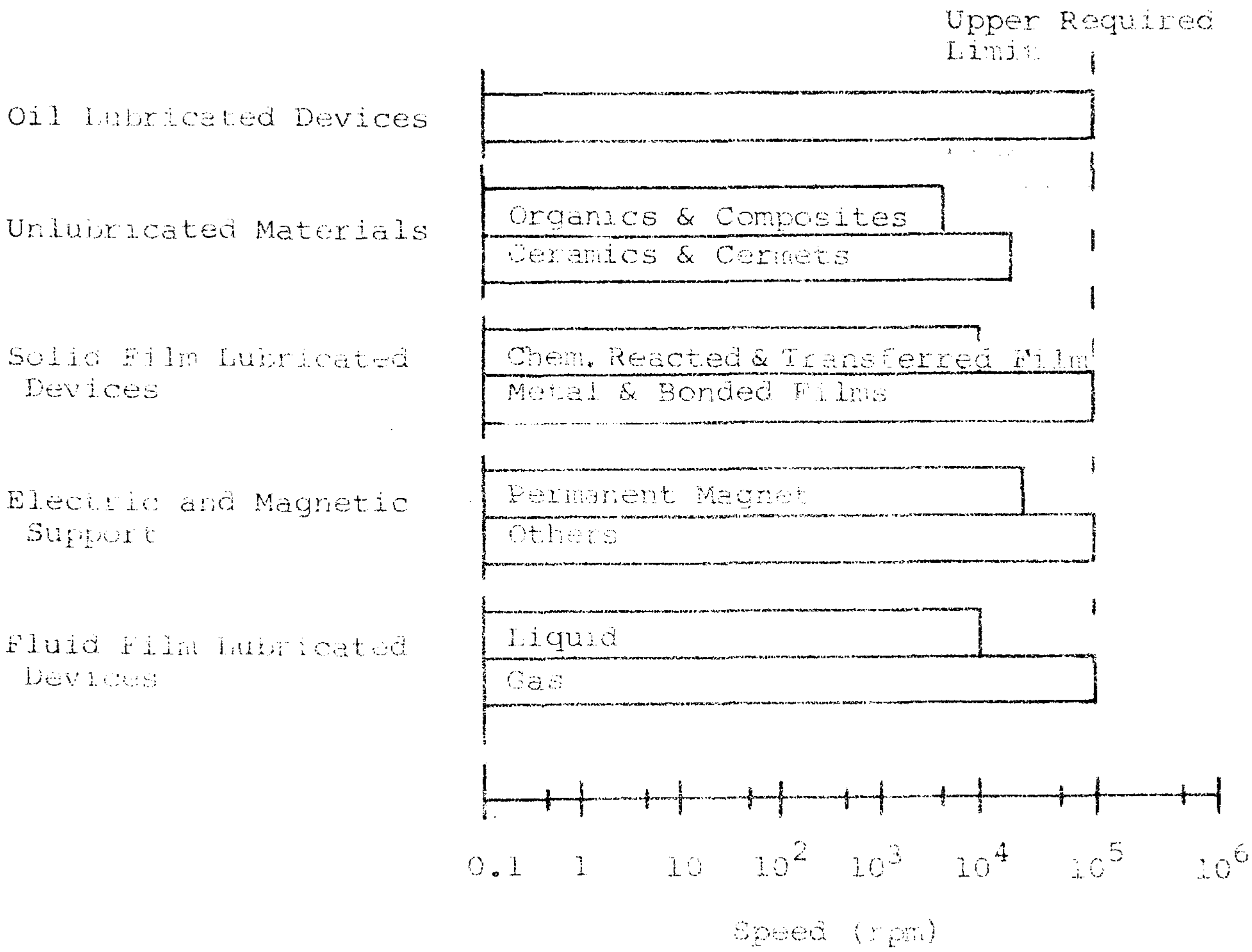

Fig. 3 SPEED CAPABILITIES OF ANTI-FRICTION DEVICES

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6. Reliability

The reliability of a mechanism is affected by the number of active components involved, the stress level and the number of modes of failure. It is usually predicted from statistical analysis of large numbers of test data. Since only meager test data are available on the particular space mechanisms studied here, only a qualitative assessment of reliability is stated here, as shown in Table 2.

TABLE 2 - RELIABILITY OF ANTI-FRICTION MECHANISMS

Mechanisms	Reliability
1. Oil Lubricated Devices	Excellent
2. Unlubricated Devices	
a. Organics & Composites	Good
b. Ceramics & Cermets	Poor
3. Solid Film Lubricated Devices	Fair to Good
4. Flexure Devices	Excellent
5. Electric & Magnetic Support	
a. Permanent Magnet	Excellent
b. Electromagnet	Good
c. Superconducting	Poor
d. Electric	Good
e. Electret	Excellent
6. Fluid Film Lubricated Devices	
a. Self-acting Liquid	Excellent
b. Externally Pressurized Liquid	Good
c. Self-acting Gas	Good
d. Externally Pressurized Gas	Good

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7. Friction

No operational requirements of friction level have been established. This is not because there is no need for low friction. On the contrary, friction in gyro spin bearings are a major source of instrument error and frictional losses in vehicle bearings consume precious power and generate undesirable heat. It is because friction is poorly understood that engineers will accept and work with whatever friction level exists in nature.

In cases of sliding mechanisms, friction coefficients below 0.1 are usually acceptable and can be provided by any of the lubrication techniques discussed above if temperature extremes are avoided. For higher temperatures, designers must accept friction coefficients of 0.2 to 0.5 (and higher) with currently available techniques. Rolling element bearings provide a tenfold reduction in friction, but also decrease reliability and load capacity. It is only by using the techniques which provide complete separation of bearing element that really low values of friction can be obtained.

C. Operational Requirements Not Fulfilled

It can be seen that the operational requirements listed in Table 1 are not completely satisfied using existing anti-friction techniques. The high temperature and load capacity deficiencies are related to bulk material properties. Materials currently exist which will withstand these extremes, but they have undesirable friction and wear characteristics. The life, speed and reliability deficiencies are caused by friction and wear effects. It would seem, then, that major improvements in space bearing performance must be preceded by better understanding and control of friction and wear properties of materials. This is discussed in the next section.

III. PREREQUISITE FOR FUTURE SPACE MECHANISM DEVELOPMENT- UNDERSTANDING THE NATURE OF FRICTION

A. Theoretical Considerations

The energy expended in overcoming the resistance to motion of two bodies in loaded contact is dissipated in three modes: (1) displacement of bulk materials, (2) the rupturing of adhesive bonds between the bodies, and (3) hysteresis losses in bulk elastic deformation. It is the second of these which is the prime source of friction in mechanical devices (viscous shear of liquid lubricants is seen to be a composite of all three modes; however, this discussion is limited to solid phase materials). The total friction developed in sliding is simply a summation of the strength of each bond in the direction of motion, summed over the total number of bonds. On a molecular level, these bonds are seen to arise from the interaction of the electrostatic and electromagnetic fields of the individual atoms. The fields establish both attractive and repulsive forces between the atoms. Since the attractive forces extend over much longer distance than the repulsive forces, the resultant force for any two bodies in close proximity is one of attraction. If the field interactions involve sharing of electrons, the bonds will be strong (metallic, ionic, covalent). If no electrons are shared, the bonds will be weak (van der Waal's, hydrogen bonding, etc.).

It seems logical, then, that friction performance of any two materials in contact could be predicted from a knowledge of the how their fields interact. At present, this can be done only in a qualitative manner. Some of the limitations on the development of this techniques are listed below:

1. The interactions are statistical in nature. The statistics describing the interactions of polycrystalline materials in non-equilibrium situations are not fully developed.

2. Many of the interactions are highly directional (as evidenced by the existence of crystalline states). The distortions of the fields produced at crystalline irregularities (such as surfaces) are not easily defined.

3. The interactions are highly sensitive to external influences such as temperature and stress.

4. The exact composition of surfaces in contact is seldom known. Even with chemically pure substances, a state of surface contamination almost always exists which markedly influences the interactions.

B. Practical Considerations

The foregoing discussion neglected a number of physical realities. Specifically, the contributions of the bulk material properties and environmental conditions were depreciated. In real situations, surfaces are monumentally rough on a molecular scale. In addition to modifying the directional effects of the bonding forces, this roughness emphasizes the effects of the bulk material properties. Although these properties may also be described in terms of atomic field interactions, the usual state of heterogeneity and crystalline imperfection of engineering materials makes the discussion simpler in terms of "engineering" properties.

It is generally recognized that, through contamination, the surface layers of solids exist in a chemical and physical state considerably different from that of the bulk material. In the absence of bulk material influences, the friction properties would be determined solely by the surface state. However, in the application of pressure and motion to irregular contacting bodies, considerable bulk deformation occurs. If the deformation is small, and the surface state remains essentially unchanged, little effect will be noted on the adhesive component of friction. If, on the other hand, the surface state is modified by bulk processes, friction may be altered considerably.

Examples of this may be seen in the sliding of two metals separated by a thin film of chemically different species (oxides or other). If the metals are hard and the film is soft and pliable, deformation may occur without damage to the film. Sliding usually occurs within the film or at the film-metal interface. Friction remains unchanged as long as the film is not worn away. If the film is hard and friable, contact will occur between the two metals. Friction will then be determined by the bonding between the metals as well as by the film. Covalent bonds thus formed are usually weak and friction and surface damage are low. If the bond is metallic, the friction will be high and separation may occur within the bulk metal. Thus surface damage occurs and the potential for further film penetration increases. If the metals are soft, the deformation will be increased, and thus the total number atoms in contact will increase. Not only will this increase the total friction of the film, but also metal displacement may take place within the bulk, thus destroying the geometry required for low friction sliding.

From this discussion, the general requirements for low friction sliding, and the necessary material properties may be established. These are listed in Table 3.

In practice, the differences between the requirements of the interface and those of the substrate are usually irreconcilable. Thus, current design practice is to use a two component system; the structural material fulfills the substrate requirements, and an artifical or natural surface film provides the necessary interfacial conditions. This imposes an additional requirement to describe the necessary film-substrate relationship; maintenance of a continuous film in the area of contact at all times. The methods currently used for supplying and maintaining surface films are listed in Table 4. The ranges of

TABLE 3 - REQUIREMENTS FOR GOOD FRICTION PERFORMANCE

Requirement	Necessary Interfacial Conditions	Necessary Interfacial Properties
1. Low ratio of attractive to repulsive forces between atoms in direction of motion	Low tangential bond strength	Low surface energy Low shear strength Low hardness
2. Minimum number of atoms taking part in friction	Low real area of contact	High hardness High elastic modulus
3. Maintenance of conditions throughout life of system	Low surface damage	High hardness High shear strength Low ductility

TABLE 4 - TECHNIQUES FOR SUPPLYING LOW FRICTION FILMS AT INTERFACES

	Type of Film	Film Materials	Supply Technique
1.	Chemically reacted	Oxides, sulfides, nitrides, chlorides	Initially applied, supplied from reservoir as gas or liquid
2.	Chemisorbed	Fatty acids, E.P. lubricants	Initially applied, supplied from reservoir as liquid
3.	Physically applied solids	MoS_2, graphite, PTFE	Initially applied, supplied from reservoir as solid (must physically contact wear track), supplied from composite structural element
4.	Physically applied liquid	Oil lubricants	Initially applied, supplied from reservoir as liquid or vapor
5.	Bonded solids	MoS_2 + binder graphite + binder metal films	Initially applied only

usefulness and design considerations of one component (without film) and two component systems (with film) are described in Section IV of this report.

The problems associated with development of low friction surfaces are many in number. They are briefly summarized below:

1. The molecular interactions leading to friction forces are complex and poorly understood. Thus, the required surface properties of materials are poorly defined.

2. The empirical process of finding and applying materials with good friction properties under certain environmental conditions does not assure good performance under all environmental conditions.

3. Low friction materials derived by empirical processes generally have undesirable secondary properties which limit their utility.

4. Empirical studies of low friction materials limits the ultimate performance to naturally occurring phenomena: i.e., barring accidental discovery, a lower limit is set on friction between surfaces by the naturally occurring force field interactions.

C. Techniques to Develop Desirable Friction Properties Of Materials

From the above discussion, development of materials with good friction properties must be preceded by a better understanding of the molecular interactions which give rise to friction. When this is accomplished, molecular properties can be adjusted to produce the desired effects. That this is within our present technical capability is evidenced by the development of semi-conducting electronic and superconducting devices and by advances in polymer synthesis.

Two approaches are envisioned. The first is to investigate those existing materials with low friction properties to determine the molecular interactions taking place. Then, the interactions may be extended in degree and/or artificially induced in other materials with more desirable secondary properties.

The second approach is to theoretically establish the desired interactions and to promote them by altering the molecular properties of existing materials. A more detailed discussion of these approaches is provided in Part II of this report.

The first approach is being investigated by several researchers[1-4]. The general feasibility of this approach was demonstrated by preliminary experiments conducted on this program. The results are also given in Part II.

No current programs involving the second approach were disclosed in the literature search conducted in this program. However, the general feasibility is established in Part II, and investigations of molecular interactions which apply to this technique are reported in the literature[5-8].

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IV. DESIGN PARAMETERS FOR EXISTING ANTI-FRICTION TECHNIQUES

A. Oil Lubricated Devices

1. Basic Design Considerations

The desire to use oil (and grease) lubricated devices in space needs no justification. The high level of reliability and development for earth-bound applications makes them the logical choices for extension to space hardware. However, the extrapolation is not straightforward, even when the limitations of high volatility and radiation degradation are overcome. Limited evidence exists[1,2] that oils will not lubricate under boundary conditions unless a contaminant film exists on the metal surface. Normally, sufficient oxygen and other reactant molecules are dissolved in the oil or supplied by outgassing to replenish the contaminant film as it is worn away. Even in vacuum tests, the effects of contaminant depletion may not become evident for realistic test periods. However, the increasing use of superrefined fluids and the long term operational requirements of space hardware demand consideration of this problem.

The vacuum evaporation rates from bulk fluids and from bearings have been adequately investigated and reported. Similarly, radiation and thermal stability limits have been established. These data for the more promising lubricants are summarized in Table 5. Additional data may be found in References 3 through 11.

Various techniques have been developed for minimizing the loss of lubricants through volatilization, and for replenishing the lubricants as they evaporate. These are summarized in Table 6.

TABLE 5 - PROPERTIES OF LIQUID LUBRICANTS

Lubricant	Useful Temperature Range (°K)	(°F)	Maximum Radiation Dose (ergs/g°C)	(rads)
Mineral Oils	240 to 650	-65 to 700	5×10^{10}	10^{8}
Dibasic Acid Esters	250 to 590	-70 to 600	5×10^{10}	10^{7}
Phosphate Esters	250 to 750	-70 to 900		10^{8}
Silicones	240 to 650	-85 to 700	$10^{8} - 10^{10}$	10^{9}
Silicate Esters	260 to 700	-60 to 800	10^{10}	10^{8}
Polyglycol Ethers	260 to 590	-60 to 600		
Halocarbons	250 to 560	-70 to 550		
Polyphenyl Ethers	270 to 700	+30 to 800	10^{11}	10^{9}
Silanes	250 to 650	-70 to 700		

Lubricant	Lubricity	Volatility (As Compared with Mineral Oils)	Viscosity-Temperature Characteristics
Mineral Oils	Good	---	Fair to Good
Dibasic Acid Esters	Fair to Good	Lower	Good to Excellent
Phosphate Esters	Good to Excellent	Much Lower	Fair to Good
Silicones	Poor to Fair	Much Lower	Excellent
Silicate Esters	Fair to Good	Lower	Excellent
Polyglycol Ethers	Good	Same	Good to Excellent
Halocarbons	Poor to Excellent	Same	Poor
Polyphenyl Ethers	Fair to Good	Much Lower	Poor
Silanes	Fair to Good	Much Lower	Fair to Good

TABLE 6 - OIL LUBRICATION TECHNIQUES

A. Loss Limiting Techniques

1. Hermetic Encapsulation
2. Rubbing Seals
3. Labyrinth Seals
4. Molecular Pump Type Seals
5. Liquid Metal Seals

B. Loss Replenishing Techniques

1. Grease Packing
2. Oil Impregnated Structural Members
3. Wicking from Remote Reservoirs
4. Controlled Evaporation Condensation from Remote Reservoirs

Two of the most promising replenishment techniques are wicking from a sealed reservoir and controlled evaporation-condensation. These techniques are shown schematically in Figs. 4 and 5.

The wick feeds lubricant at a constant rate by capillarity (Fig. 4). The evaporation-condensation technique (Fig. 5) feeds lubricant by evaporation from the surface of a porous membrane and by subsequent condensation on the bearing surface. The membrane surface is kept wet by diffusion from the reservoir.

Problems arise with all reservoir techniques from included air at atmospheric pressure which must escape as the external pressure diminishes.

In addition to evaporation, lubricants can be lost through creep along surfaces which have been denuded by evaporation. Low surface energy creep barriers (such as teflon) can be incorporated on bearing housings and shafts to control this loss.

Greases have long been used as lubricants, However, the thickeners contribute nothing towards lubrication and merely

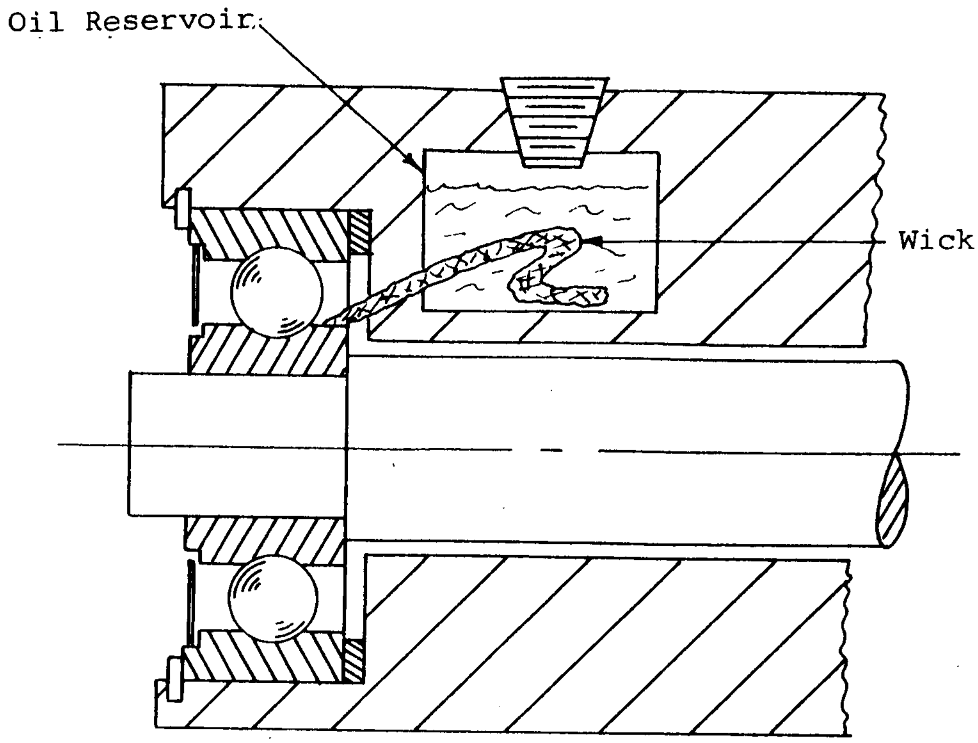

Fig. 4 WICK FEED LUBRICANT SUPPLY

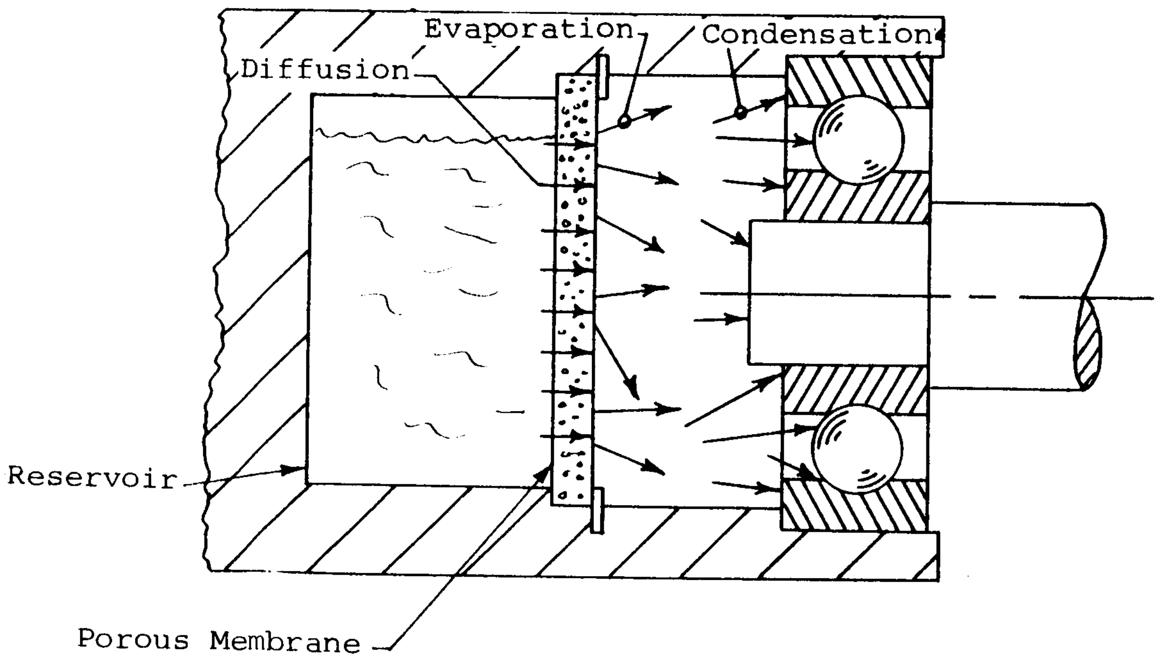

Fig. 5 EVAPORATION-CONDENSATION LUBRICANT SUPPLY

act as a reservoir for the oil. Considering their problems of radiation and thermal instability and the limited reservoir capacity, their utility for space applications is limited.

2. State of Development and Performance Potential

Liquid lubricants have positive advantages for producing low friction in space mechanisms. However, their use is restricted to a narrow range of temperatures. Good design practices can minimize their loss through evaporation, and replenishment techniques can extend the useful life almost indefinitely. The interfacial chemical reactions occurring between the lubricant, the bearing surfaces and the gaseous environment which are necessary to insure long term operation are obscure. These must be defined before reliability in space can be assured.

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B. Unlubricated Devices

1. Basic Design Considerations

An unlubricated device is one whose friction and wear characteristics are determined by the properties of the materials of construction and not by the presence of any artificially introduced surface films. The need for this clarification arises from hardware design considerations, and not from any inherent differences in the mechanism of friction.

In some applications, it is possible to employ materials with low inherent friction and wear as structural elements (i.e., cams, latches, bushings). In others, constructional problems dictate the manufacture of structural elements from materials whose friction and wear characteristics must be modified to obtain acceptable performance (i.e., ball bearings, gears, sliding electrical contacts). The following discussion pertains to the former class of devices.

a. Solid Organic Materials

Many organic materials have been used in unlubricated sliding, both as structural materials and as thin films applied to metal substrates. The advantages of their use are:

1. Low inherent friction
2. Good performance under marginal lubrication
3. Good damping characteristics

The major disadvantages are:

1. Poor thermal conductivity
2. Poor dimensional stability
3. Large differences in thermal expansion coefficients from metals

The disadvantages have led to blending of the substances with inorganic materials to improve performance characteristics. Complete listing of the properties and performance characteristics is beyond the scope of this report. Summary data on materials for space applications are listed in Table 7. These data are representative values and should not be used for design purposes. More comprehensive design data can be obtained from References 1 through 4, and from manufacturers' literature. Information on materials properties in space are found in References 5 through 11.

The lowest friction of any plastic is provided by Teflon. The major limitations in its use are thermal degradation from frictional heating and its poor dimensional stability under load (cold flow). Addition of metal fillers improves its mechanical properties and increases the thermal conductivity (thereby decreasing interfacial temperatures), but also increases friction. Optimum performance is provided by a thin film of Teflon on a metal substrate The recent development of polyimides represents an improvement in mechanical properties New fluorocarbons are under development[12], although no design data are currently available.

b. Inorganic Materials

Ceramics and cermets offer advantages for use in space due to their high hardness, high temperature stability and minimum cold welding tendencies Disadvantages arise from their high friction and brittleness. Best use is made of these materials as high temperature rolling elements for bearings, although their fatigue life is low and failure is almost always catastrophic. Typical performance characteristics are listed in Table 8.

c. Composites

The more desirable properties of two or more materials may be obtained by combining the materials in a composite structure. The structures may have the following forms:

1. Metals incorporating microinclusions
2. Sintered powdered metals impregnated with:
   a. oils

b. solid organic lubricants
c. solid inorganic lubricants
3. Porous ceramics impregnated with:
a. solid organic lubricants
b. solid inorganic lubricants
4. Inorganic fiber structures incorporating:
a. solid organic lubricants
b. solid inorganic lubricants
5. Sintered organic powders impregnated with:
a. oils
b. solid organic lubricants
c. solid inorganic lubricants
6. Compressed powder mixtures of:
a. metals
b. solid organic lubricants
c. solid inorganic lubricants

The performance of typical composites are summarized in Table 9. The basic scheme is to take a satisfactory structural material and to combine with it a sufficient amount of material with low friction properties to assure satisfactory performance. In general, it is desirable to incorporate a substance which will form a continuous film over the interface under the action of sliding, so that the structural materials do not contact. The number of possible combinations is infinite and the designer is referred to References 1-5, and 7-10 for specific applications.

Performance characteristics of composite structures are usually a compromise between those of the structural material and those of the filler material. Temperature and radiation tolerance limits are usually established by the weaker of the two.

2. <u>State of Development and Performance Potential</u>

The use of organic materials as structural bearing elements offers no great potential for space mechanisms. It does not appear that new polymer formulations can overcome the thermal problems associated with vacuum operation. The use of teflon in solid film lubricant composites has some advantage, as is discussed in Section C, Solid Film Lubricated Devices.

TABLE 7 - PROPERTIES OF ORGANIC BEARING MATERIALS

	Maximum Load Capacity (kg/mm^2)	(psi)	Dry Friction Coefficient	Relative Wear in Sliding	Maximum Useable Temp. (°K)
Fluorocarbon (TFE + Filled Teflon)	4.2	6,000	0.05 - 0.4	High	530
TEF with Glass Filler	8.4	12,000	0.05 - 0.4	Low	530
TFE with Metal Filler	8.4	12,000	0.05 - 0.4	Low	500
Polyamide (Nylon)	8.4	12,000	0.15 - 0.7	High	425
Acetal (Delrin)	7.0	10,000	0.15 - 0.35	Moderate	350
Polyimide	9.2	13,000	0.15 - 0.55	Very low to Moderate	775
Polyimide with Graphite Filler	6.3	9,000	0.25	High	775

References 1,2,3,4,5,6,7,8,9

TABLE 7 - PROPERTIES OF ORGANIC BEARING MATERIALS (CONT'D)

	Vacuum Evaporation ($g/cm^2 sec$) (Room Temp.)	Radiation Resistance	Thermal Expansion (per °F)	Thermal Conductivity ($Btu/°F/hr/ft^2/in.$)
Fluorocarbon (TFE + Filled Teflon)	10^{-9}	Good	10^{-9}	1.7
TEF with Glass Filler	----	Good	9×10^{-5}	2.1
TFE with Metal Filler	----	Good	----	---
Polyamide (Nylon)	10^{-7}	Good	6×10^{-5}	1.7
Acetal (Delrin)	10^{-7}	----	5×10^{-5}	1.6
Polyimide	----	Excellent	3×10^{-5}	2.2
Polyimide with Graphite Filler	----	Excellent	2×10^{-5}	---

TABLE 8 - FRICTION PERFORMANCE OF CERAMICS AND CERMETS

Material	Composition	Sliding Against	Atmosphere	Friction Coefficient at -140°F	90°F	500°F	1200°F	1300°F	1500°F	1700°F	1900°F
LT-2	W-25Cr-15 Al_2O_3	LT-1B	Air	----	0.48	----	----	0.033	0.30	0.30	0.31
LT-1B	Cr-19Al_2O_3-20Mo-2TiO_2	LT-1B	Air	----	0.6	----	----	0.035	0.38	0.45	0.50
LT-2		LT-1B	Air	----	0.48	----	----	0.035	0.35	0.39	0.31
LT-2		LT-2	Vacuum	0.51	0.95	----	----	-----	3.93	----	----
Rene 41	Ni-19Cr-11Ca-10Mo-3Ti-1.5Al-0.09Cr-0.005B	LT-1B	Vacuum	----	0.61	----	----	-----	0.75	----	0.83
Boron Nitride		Boron Nitride	Vacuum	----	0.72	0.67	0.35	-----	----	----	----

Material	Wear ($in.^3/ft$) at 3000 psi	7500 psi	12,000 psi	15,000 psi	Source
LT-2	1.2×10^{-7}	2×10^{-7}	---	---	17
LT-1B	1×10^{-7}	1.8×10^{-7}	---	---	17
LT-2	---	2×10^{-7}	---	5×10^{-7}	17
LT-2	---	---	3.5×10^{-5}	---	15
Rene 41	---	---	1×10^{-6}	---	15
Boron Nitride	---	---	---	---	14

Ceramics and cermets seem to offer great potential for future space bearings. The atomic bonding is primarily ionic or covalent, which gives them their high stability and hardness. Slight adjustments in the structure of these materials could markedly improve their friction and reduce their brittleness.

Table 10 summarizes the operational characteristics of unlubricated devices.

TABLE 9 - PERFORMANCE CHARACTERISTICS OF COMPOSITE MATERIALS

	Sliding Against	Atmosphere	Temperature (°K)	Temperature (°F)	Load (kg)	Load (lb)
Sintered Cu/Teflon	Steel	Air	288-523	59-482	---	---
Sintered Nylon/Oil	---	---	---	---	13.7 (kg/cm^2)	195 (psi)
Sintered Nylon/Oil	---	---	---	---	9.5 (kg/cm^2)	135 (psi)
Sintered Cu/MoS_2	Steel	Air	---	---	4.0	8.8
85% Ag, 5% Cu 10%MoS_2 (Hot Pressed)	Steel	Air	---	---	1.0	2.2
Nylon/40% MoS_2	M-10 Steel	N_2	303-344	86-160	1.36	3.0
Nylon/20% C	M-10 Steel	N_2	303-344	86-160	1.36	3.0
Glass Fiber/ Teflon-MoS_2	M-10 Steel	N_2	303-344	86-100	1.36	3.0
Glass Fiber/ Teflon	M-10 Steel	N_2	303-344	86-100	1.36	3.0
Carbon/Teflon	M-10 Steel	N_2	303-344	86-100	1.36	3.0
20% Ni, 80% C (Pressed)	M-10 Steel	N_2	---	---	1.36	3.0
70% Ag, 20% Teflon 10% WSe_2 (Pressed)	440 Stainless Steel	Vacuum	Room		1-4	2.2-8.8
Teflon/7% WSe_2 (Pressed)	440 C Stain-less Steel	Vacuum	Room		1-4	2.2-8.8
Teflon/3% WSe_2 (Pressed)	440 C Stain-less Steel	Vacuum	Room		1-4	2.2-8.8
20% Ni, 80% C (Pressed)	M-10 Steel	N_2	1360	1000	1.36	3.0
Co/Sb_2S_3	M-10 Steel	N_2	1360	1000	1.36	3.0
40% Fe, 60% C (Heat Treated)	M-10 Steel	N_2	1360	1000	1.36	3.0
Ni, 2% NiO	---	10^{-8} - 10^{-9}	291	75	1.0	2.2
Ni, 20% Sn	---	10^{-9} torr	291	75	1.0	2.2
Fe, 0.825% FeS	---	10^{-9} torr	291	75	1.0	2.2

TABLE 9 - PERFORMANCE CHARACTERISTICS OF COMPOSITE MATERIALS (CONT'D)

	Sliding Speed (m/sec)	Sliding Speed (fps)	Friction Coefficient	Wear Rate (cm^3/cm^2hr)	Reference
Sintered Cu/Teflon	---	---	0.05	----	10
Sintered Nylon/Oil	73	240	0.10	0.11	10
Sintered Nylon/Oil	73	240	0.12	0.08	10
Sintered Cu/MoS_2	---	---	0.13-0.2	----	10
85% Ag, 5% Cu 10% MoS_2 (Hot Pressed)	2440	8000	0.21	----	10
Nylon/40% MoS_2	70-140	230-460	.17-0.20	0.09	13
Nylon/20% C	70-140	230-460	0.08-0.09	0.09	13
Glass Fiber/ Teflon-MoS_2	70-140	230-460	0.02-0.03	0.19	13
Glass Fiber/ Teflon	70-140	230-460	0.39-0.50	0.36-0.44	13
Carbon/Teflon	70-140	230-460	0.02-0.03	0.09-0.16	13
20% Ni, 80% C (Pressed)	70-140	230-460	0.05-0.09	0.75	13
70% Ag, 20% Teflon 10% WSe_2 (Pressed)	0.265	0.87	0.09-0.12	0.90	13
Teflon/7% WSe_2 (Pressed)	0.265	0.87	0.20	0.70	13
Teflon/3% WSe_2 (Pressed)	0.265	0.87	0.04-0.06	0.013	13
20% Ni, 80% C (Pressed)	140	460	0.05	0.65	13
Co/Sb_2S_3	70	230	0.15	0.25	13
40% Fe, 60% C (Heat Treated)	140	460	0.1	0.44	13
Ni, 2% NiO	20	65	0.76	0.020 (cm^3/hr)	16
Ni, 20% Sn	20	65	0.17	0.0066 (cm^3/hr)	16
Fe, 0.825% FeS	20	65	0.25	0.041 (cm^3/hr)	16

TABLE 10 - SUMMARY OF OPERATIONAL CHARACTERISTICS OF UNLUBRICATED DEVICES

	Organics	Ceramics & Cermets	Composites
Max. Load Capacity (kg/mm^2)	9.2	14	7
Max. Load Capacity ($\#/in.^2$)	13,000	20,000	10,000
Type of Device	Sliding Bearings Static Devices	Rolling and Sliding Bearings	Sliding Bearings Static Devices
Speed Range (rpm)	0 - 5000	0 - 20,000	0 - 5000
Wear Life	Poor to Fair	Excellent	Good
Reliability	Good	Poor	Good
Operating Temperature Range (°K)	20 - 775	20 - 2500	20 - 775
Environmental Pressure	Unlimited	Unlimited	Unlimited
Radiation Resistance	Fair to Good	Excellent	Fair to Good
Vibration Resistance	Good	Poor	Good

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C. Solid Film Lubricated Devices

1. Basic Design Considerations

It was stated in Section II that by providing a film of solid material with low friction properties on a substrate of high hardness, acceptable friction performance can be effected. This technique has received considerable attention for space devices and countless numbers of materials and application methods have been investigated. The major problems have been short wear lives and lack of reproducibility of performance.

The various techniques for lubricating with solid films are listed in Table 11. The materials which show the best performance characteristics for each technique are also listed. The principles underlying each technique and their limitations are discussed below.

a. Chemically Reacted Films

In this technique, lubricant films are formed by chemical reaction of the surface layer of the bearing material with a vapor or liquid. Such films are thin and easily worn away, thus requiring continuous or periodic replenishment for severe duty.

Materials such as vapors of carbamates and amines show good potential with low flow rates[2,3]. Light weight reservoir systems can probably be developed which will make this technique useable, but the state of development is low. Inclusion of MoS_2 yielding compounds in oils offers a method of improving the reliability of oil lubricated devices, but does not solve the problems of volatility and radiation damage associated with oil lubrication.

b. Self-Adhering Films

Of the various techniques for using self-adhering films, only soft metal platings and transfer film techniques have

TABLE 11 - SOLID FILM LUBRICATION TECHNIQUES

Type of Film	Application Technique	Most Suitable Materials	Performance	Limitations	Source
1. Chemically Reacted	Reaction with Vapors	Sulfides Chlorides	Poor to Fair	Requires Continuous Supply	Ref. 1
		Carbamates Amines	Good	Requires Continuous Supply	Ref. 2,3
		Halogenated Hydrocarbons	Fair	Requires Continuous Supply	Ref. 4
	Reaction with Liquids	Phosphates Sulfides	Fair	Requires Continuous Supply	Ref. 5
		MoS_2 Yielding Compounds	Good	Short Wear Life	Ref. 6,7,8
2. Self-Adhering	Burnishing	MoS_2 WS_2	Fair	Short Wear Life	Ref. 9
	Metal Plating	Gold Silver Indium	Good	High Friction	Ref. 10,11
		Gallium Barium	Fair to Good	Highly Reactive	Ref. 12,13
		Lead Bismuth	Good		Ref. 14
	Gas Entrained Powder	Graphite + CdO, MoS_2 + Metal-Free Phthalocyanine	Fair	Requires Continuous Supply	Ref. 15,16
	Transfer Film	MoS_2 + WS_2 + WSe_2 + PTFE	Excellent	Reservoir Sometimes Structurally Unstable	Ref. 4,17,18, 19,20
3. Bonded Solid	Spray	MoS_2 + Graphite + Gold with Silicate Binders	Excellent		Ref. 21,22,23, 24,25,26, 27
	Vacuum Deposition	MoS_2	Good		Ref. 28

any real application to space mechanisms. Thin films (1 micron thickness) of low shear strength metal, applied to hard bearing materials can effect good friction performance in rolling bearings for many thousands of hours. The use of gold[10] probably compromises low friction (low shear strength) for good wear life (high hardness) and chemical inertness. In many applications, superior friction performance could be effected by using metals (lead, bismuth[14], barium[13] or gallium[12]).

Films of solid lubricant applied by transfer during sliding provide excellent long term operation of bearings and gears. The films are transferred to load bearing elements by rubbing against a lightly loaded secondary member (bearing retainer, idler gear) made from the solid lubricant material. Generally, two classes of material are used together; a layer lamellar powder (MoS_2, graphite, etc.) and a film former (teflon, nylon) to hold the powder in place. Such materials are available commercially[29] and from research labs operating under government contract[17,18]. Members made from these materials are generally weaker than other parts and may suffer structural damage if loading is high.

c. Bonded Solid Films

A widely used and generally satisfactory technique for applying solid lubricant films is to bond them to the load bearing surfaces with an organic or inorganic binder. Excellent results have been achieved with a mixture of MoS_2, graphite and gold flakes bound with a sodium silicate binder[23,25,30]. Other bonded coatings are available commercially[26-28]. Special surface preparation is required for optimum performance with bonded coatings[23,24,26].

2. State of Development and Performance Potential

Four techniques appear promising for solid film lubrication of space mechanisms; reaction films, thin metal films, transfer films, and bonded solid films. The first two have recieved insufficient attention to make them generally practical. Both offer potential for meeting high temperature requirements. The bonded film has been developed to the point where good performance and reliability are readily achieved. The particular mixture discussed above which gives best results cannot be rationalized on theoretical grounds; therefore, its ultimate performance potential cannot be established.

Ranges of performance and operational characteristics for these techniques are listed in Table 12.

TABLE 12 - SUMMARY OF OPERATIONAL CHARACTERISTICS FOR BEST SOLID FILM LUBRICATION TECHNIQUES

	Chemical Reaction Films	Metal Films	Transfer Films	Bonded Solid Films
Max. Load Capacity (kg/mm^2)	105	105	105	105
Max. Load Capacity ($\#/in.^2$)	150,000	150,000	150,000	150,000
Load to Weight Ratio	10 - 100	100 - 1000	100 - 1000	100 - 1000
Friction Coefficient	0.05 - 0.2	0.05 - 0.5	0.05 - 0.1	0.05 - 0.1
Wear Life (hrs)	10*	2000	10,000	10,000
Speed Range (rpm)	0 - 10,000	0 - 100,000	0 - 10,000	0 - 100,000
Reliability	Fair	Good	Good	Good
Operating Temperature Range (°K)	300 - 800	100 - 800	20 - 1200	20 - 1100
Environmental Pressure	Unlimited	Unlimited	Unlimited	Unlimited
Radiation Resistance	Good	Excellent	Fair to Good	Excellent
Auxiliary Equipment Requirements	Lubricant Replenishment Equipment	None	Lubricant Reservoir	None

* Without Continuous Replenishment

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D. Flexure Devices

1. Basic Design Considerations

Many of the problems of space operation of mechanical components can be circumvented by avoiding sliding entirely. For mechanisms requiring only limited rotation or translation, flexure devices can often be used to great advantage. These devices utilize the elastic properties of structural members to provide constrained relative motion. The advantages of this technique for space applications are the high bearing load to weight ratios for simple configurations, the relative insensitivity to the space environment, and the inherent reliability and unlimited life of the devices. The major disadvantages are limitations on angular rotation and translation, and (in the simplest configurations) the presence of a restoring force acting against the motion.

The simplest configurations of flexure devices are shown in Figs. 6 through 13.

a. Rotation

1. Emery Fulcrum

The simplest form of flexure pivot is the Emery Fulcrum (Fig. 6). Under the action of a transverse load, P_t, the beam will deflect giving a rotation of the axis of the normal load, P_n. The major disadvantages of this support are:

a. The bending stresses in the material produce a restoring moment proportional to the angle of rotation.

b. The center of rotation changes position as deflection takes place.

c. Stiffness in torsion about the normal load P_n is small.

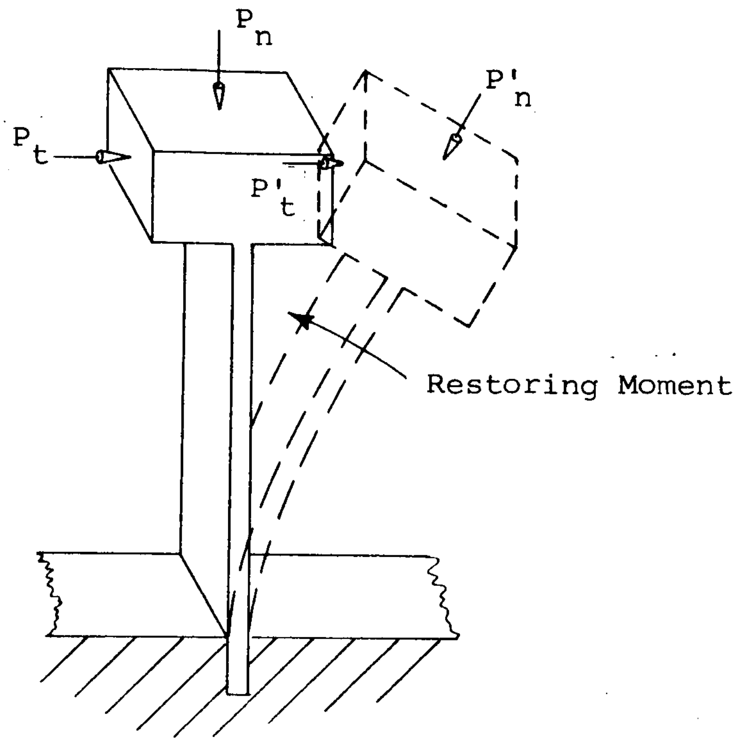

Fig. 6 EMERY FULCRUM

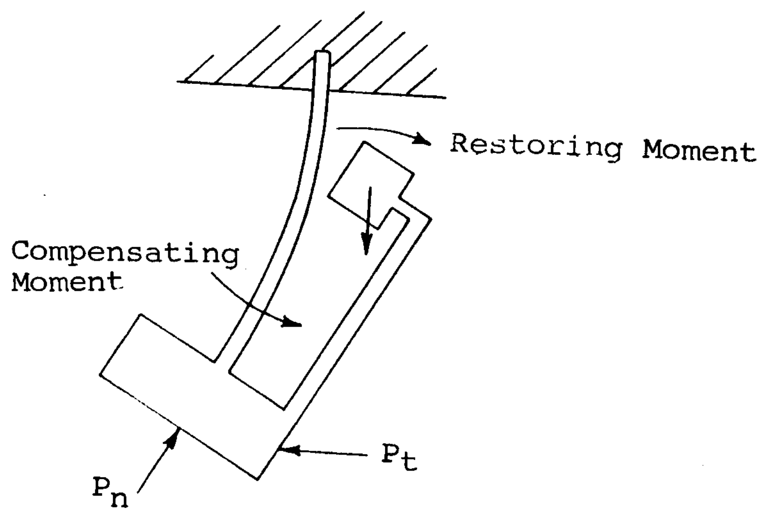

Fig. 7 WEIGHT-COMPENSATED EMERY FULCRUM

The Emery Fulcrum has found wide application in torsion balances[1] and other applications where small angles of rotation are required. Further design information may be found in References 1 through 3.

2. Compensated Emery Fulcrum

The first disadvantage to the Emery Fulcrum may be remedied by adding a compensating moment which cancels the restoring moment. Figure 7 shows how this may be accomplished using counterweights. For zero gravity applications, additional flexure pivots are required to provide the compensating moment.

3. Cross Spring Pivot

A single axis flexure pivot with fixed center of rotation (remedy for disadvantage "b") is illustrated in Fig. 8. Such devices are useful for up to 90° rotation. Applications for cross spring pivots include universal joints, mechanical linkage for instruments, gimbal joints for rocket engines and gyro tables, torsion balance fulcrums and hinge pins. Small units for limited load and rotation are available commercially[7]. Their useful design characteristics are summarized in Table 13. Complete design data for other load and deflection ranges may be found in References 1,4, and 5.

Disadvantage "c" may be overcome by applying three (or more) strips to the pivot as in Fig. 9, thus increasing the torsional stiffness about P_n.

4. Spoked Pivot

Three axis rotation with fixed center may be provided with a "bicycle wheel" suspension as illustrated in Fig. 10. Wire spokes are arranged in two cones with a common vertex. The suspension acts as a spherical bearing, with very little restoring moment.

TABLE 13 - LOADS AND SPRING RATES FOR STANDARD PIVOTS (FIG. 9)*

Diameter (mm)	Diameter (in.)	Maximum Deflection (± Degrees)	Maximum Load (kg)	Maximum Load (lb)	SPRING RATE (kg-cm/radian)	SPRING RATE (lb-in/radian)
3.175	1/8	7-1/2	11.341	25.0	0.922	0.800
3.175	1/8	15	5.671	12.5	0.155	0.100
3.175	1/8	30	1.588	3.5	0.013	0.011
4.763	1/4	7-1/2	25.409	100.0	3.120	6.540
4.763	1/4	15	12.702	50.0	0.376	0.817
4.763	1/4	30	3.085	14.0	0.047	0.102
12.700	1/2	7-1/2	181.457	400.0	25.400	52.000
12.700	1/2	15	90.729	200.0	3.170	6.500
12.700	1/2	30	25.540	56.3	0.381	0.813
19.050	3/4	7-1.2	408.279	900.0	210.000	182.000
19.056	3/4	15	204.140	450.0	26.300	22.800
19.056	3/4	30	57.613	127.0	3.280	2.850
25.400	1	7-1/2	725.829	1600.0	497.000	431.000
25.400	1	15	362.915	800.0	62.000	53.800
25.400	1	30	102.070	225.0	7.760	6.730

*From Reference 7

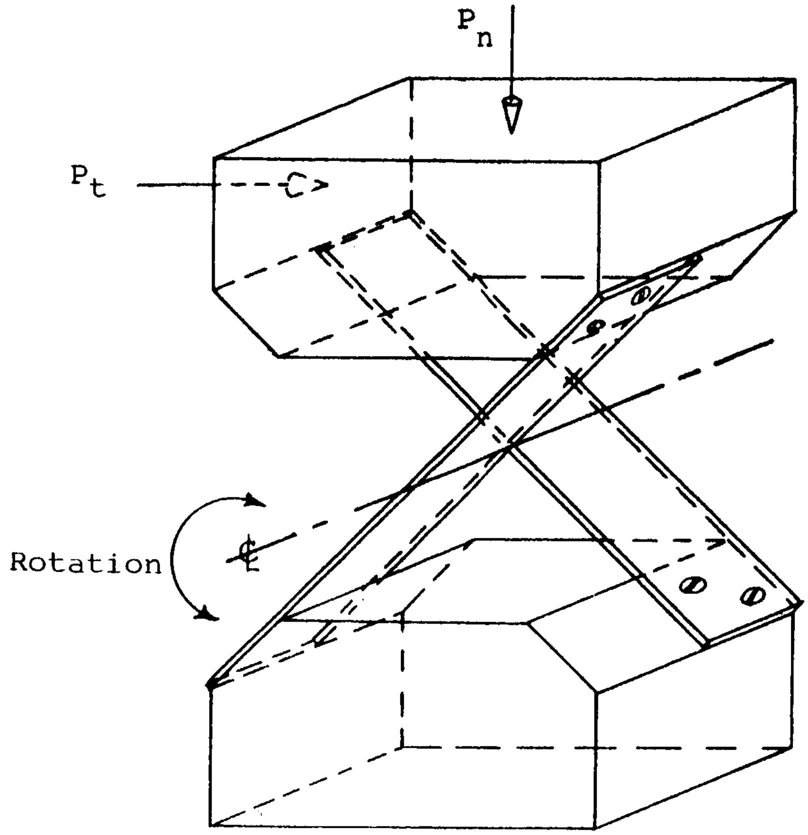

Fig. 8 CROSS SPRING PIVOT (TWO STRIP)

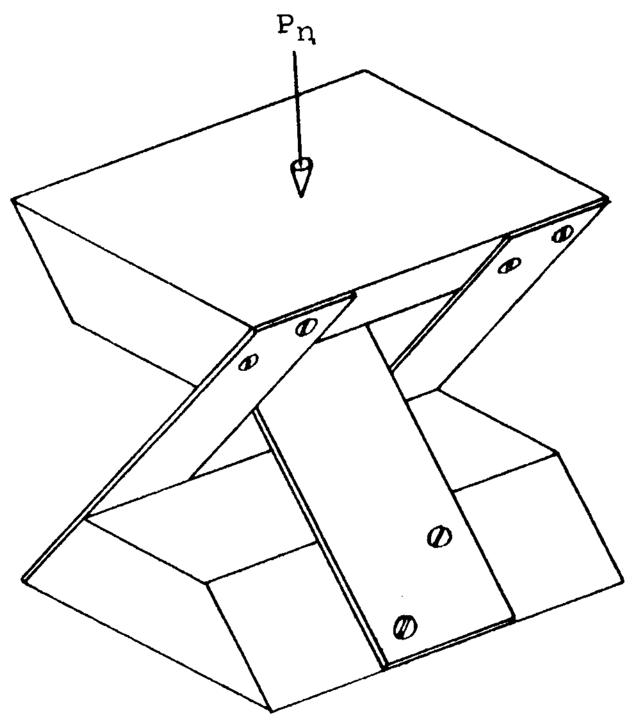

Fig. 9 CROSS SPRING PIVOT (THREE STRIP)

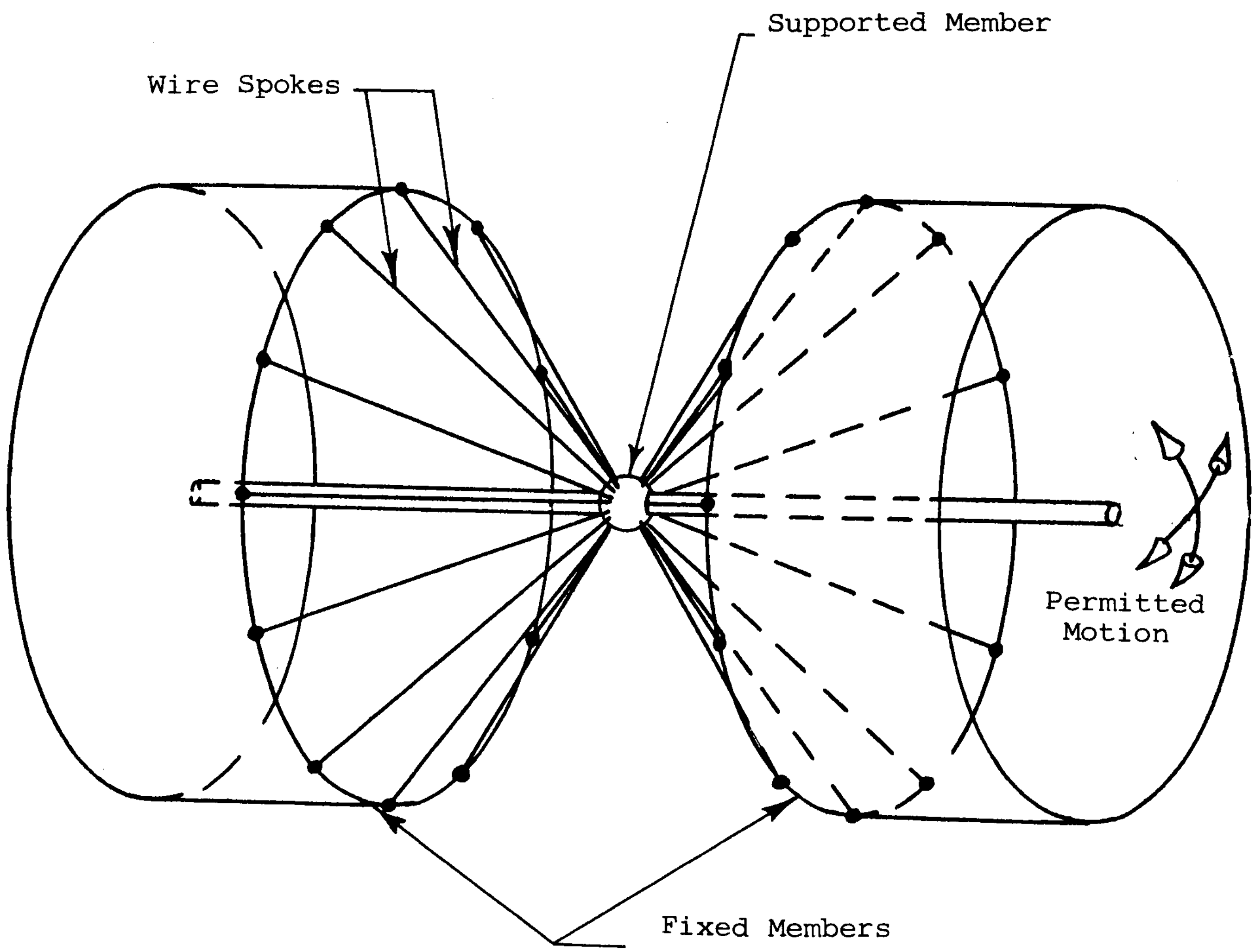

Fig. 10 SPOKED PIVOT

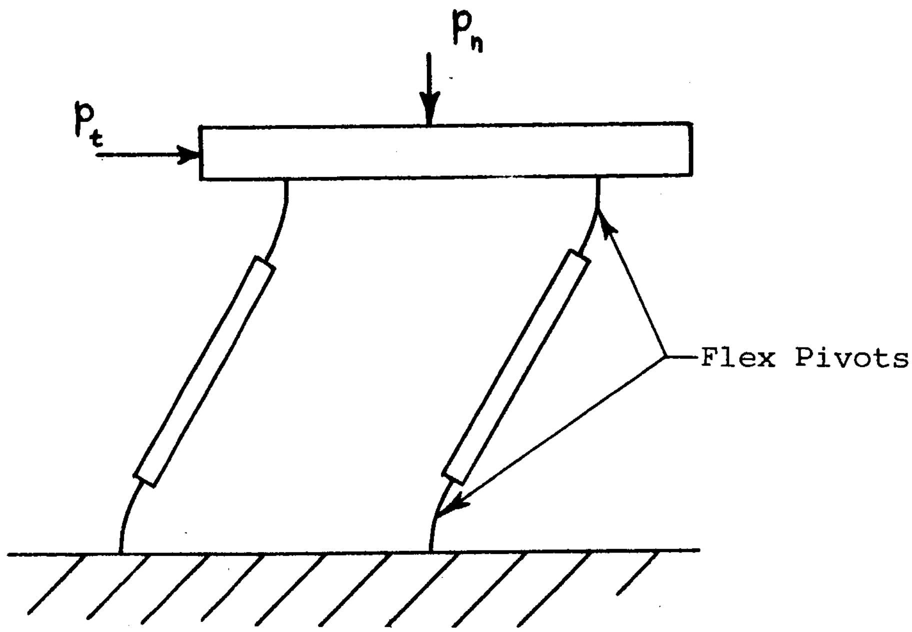

Fig. 11 PARALLEL MOTION DEVICE

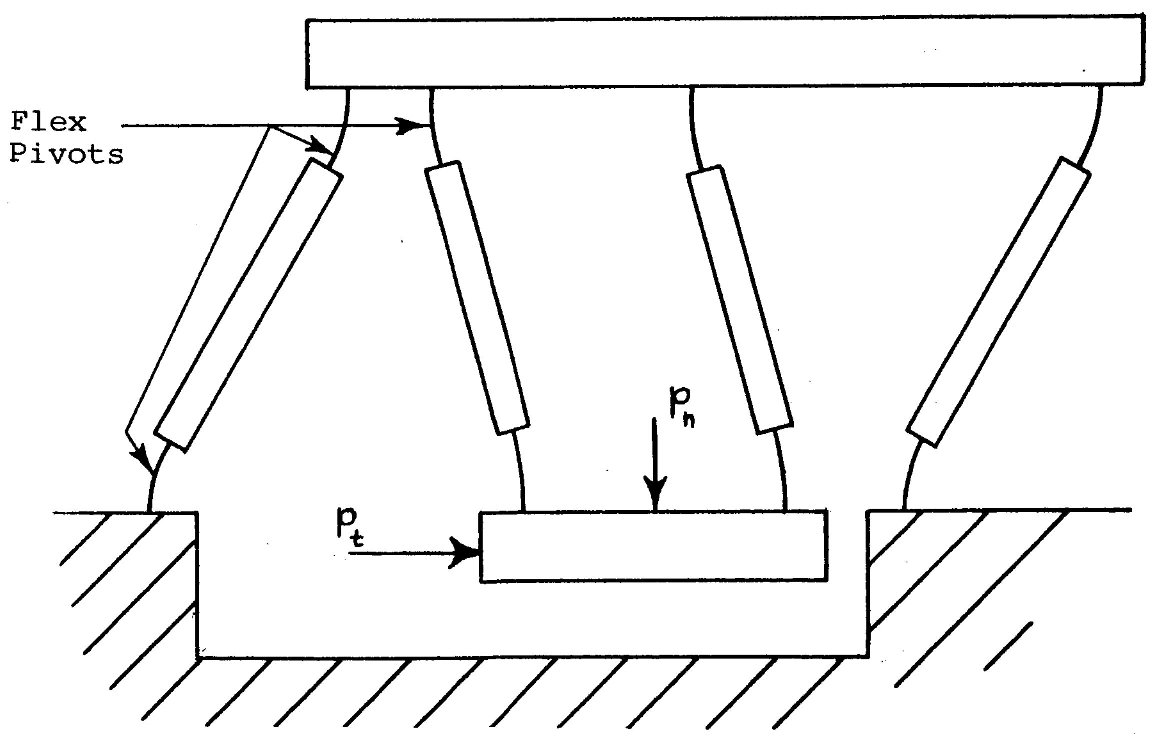

Fig. 12 RECTILINEAR MOTION DEVICE

A device of this type was used as the gimbal bearing in the 200 inch Mt. Palomar Telescope.[8]

b. Translation

Limited parallel translation may be obtained with a simple flexure device as shown in Fig. 11. However, in this case, the line of action of P_t must move closer to the base as translation occurs.

More nearly rectilinear motion is achieved with the apparatus shown in Fig. 12. In both of these apparatuses, stiffness perpendicular to the applied load is poor. Applications for these mechanisms include wind tunnel model constraint and force measuring apparatus[1,3].

Dunk[6] proposes the apparatus shown in Fig. 13, which gives good stiffness in all transverse planes, but adds slight rotation to the moving element.

Design data for these devices may be found in References 1,2,3, and 6.

2. State of Development and Performance Potential

Flexure devices have long been used to support light loads in instrument applications. The theory behind their use is well defined, and the load ranges can be readily extended upwards. The only foreseeable problem concerns their use at high temperatures. Most materials undergo stress relaxation at approximately 1/2 their melting point.

Columbium alloys appear most promising; they retain elastic moduli of 4.4×10^8 kg/mm^2 (1.5×10^6 psi) at temperatures up to 1650°K.

A wide range of space hardware applications exists in which flexure devices could be used effectively. The low utilization of these devices to date can probably be attributed to lack of communication, rather than to any inherent limitations in the mechanisms. Table 14 summarizes the range of conditions and operational characteristics of the various mechanisms.

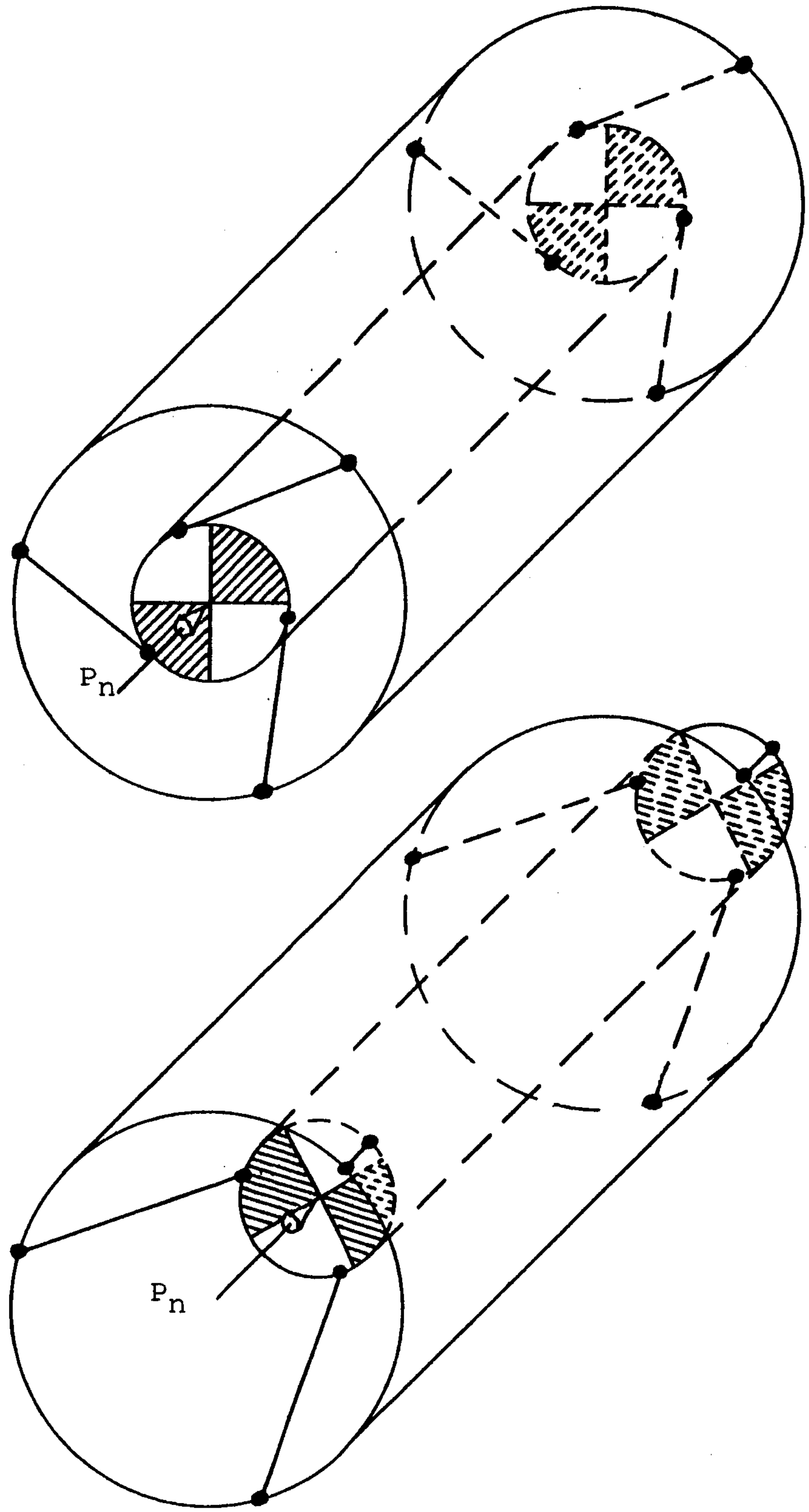

Fig. 13 DUNK'S TRANSLATION DEVICE

TABLE 14 - SUMMARY OF FLEXURE DEVICE OPERATIONAL CHARACTERISTICS

Type of Support	Max. Load Capacity (kg)	Max. Load Capacity (lb)	Load-to-Weight Ratio	Type of Motion	Range of Motion	Life (Cycles)	Reliability
Emery Fulcrum	5×10^5	10^6	10^2 to 10^4	Curvilinear	$\pm 45°$	Indefinite	Excellent
Cross Spring Pivot	5×10^3	10^4	10^4	Rotary (Single Axis)	$\pm 60°$	Indefinite	Excellent
Spoked Pivot	5×10^5	10^6	10^6	Rotary (3 Axis)	$\pm 45°$	Indefinite	Excellent
Parallel Motion Device	5×10^5	10^6	10^2 to 10^3	Linear	10 cm	Indefinite	Excellent
Rectilinear Motion Device	5×10^5	10^6	10 to 100	Linear	Few cm	Indefinite	Excellent
Dunk's Device	50	100	1 to 100	Rotary and Linear	Few cm	Indefinite	Excellent

TABLE 14 - SUMMARY OF FLEXURE DEVICE OPERATIONAL CHARACTERISTICS (CONT'D)

Type of Support	Operating Temperature Range (°K)	Environmental Pressure	Radiation Resistance	Vibration Resistance	Auxiliary Equipment Requirements
Emery Fulcrum	20 to 1650	Unrestricted	Excellent	Good, if designed properly	*
Cross Spring Pivot	20 to 1650	Unrestricted	Excellent	Good, if designed properly	None
Spoked Pivot	20 to 1650	Unrestricted	Excellent	Good, if designed properly	*
Parallel Motion Device	20 to 1650	Unrestricted	Excellent	Good, if designed properly	*
Rectilinear Motion Device	20 to 1650	Unrestricted	Excellent	Good, if designed properly	*
Dunk's Device	20 to 1650	Unrestricted	Excellent	Good, if designed properly	None

*External constraints to restrict undesired motion.

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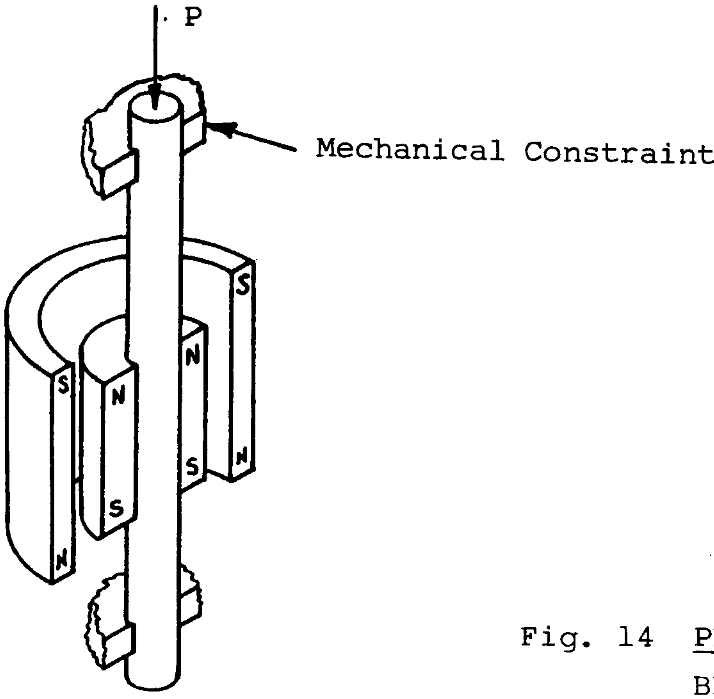

Fig. 14 PERMANENT MAGNET THRUST BEARING - ATTRACTIVE (Ref. 1)

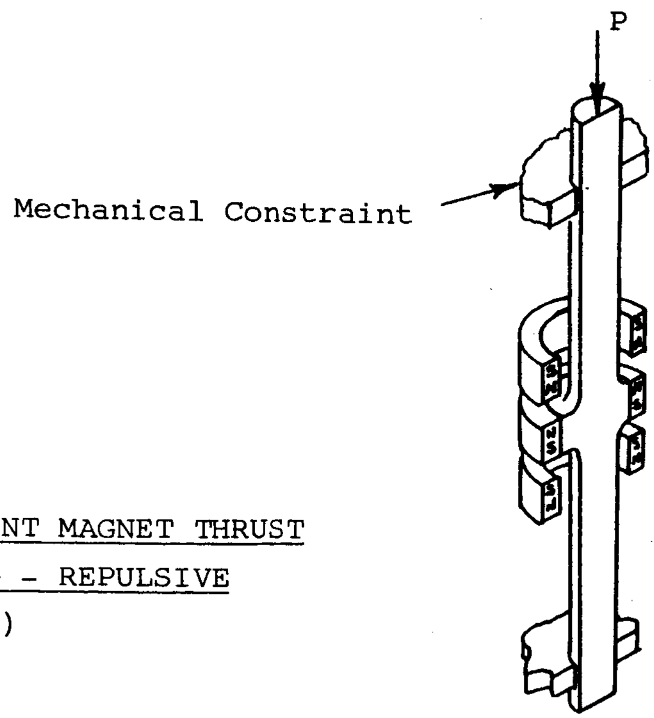

Fig. 15 PERMANENT MAGNET THRUST BEARING - REPULSIVE (Ref. 1)

E. Electric and Magnetic Support

1. Basic Design Considerations

The use of forces established by magnetic and electrical fields offers a method for providing bearing support with almost no friction. However, the utility of this technique is limited for space applications. The most formidable restriction is that a system utilizing electrostatic or magnetostatic support cannot be stabilized along three axes if the dielectric constant and relative permeability of the materials of construction are greater than one. Since virtually all engineering materials fall into this category, other types of bearing support must be provided along at least one axis. Other restrictions stem from power requirements for electric and electromagnetic supports, weight penalties for permanent magnets, and the inherently low bearing stresses which these devices can tolerate.

Electric and magnetic fields can be oriented to provide either attractive or repulsive forces, the choice for a particular application depends on the geometry of the support and the materials used. Out of the many configurations which are possible, only a few offer any engineering feasibility. These are illustrated schematically in Figs. 14 through 18. Only the major support is shown; the additional constraints indicated must be provided either mechanically or by additional servo-controlled fields. Table 15 lists the maximum bearing stress predicted by theoretical considerations using existing materials. References for further design and analysis are also listed in Table 15

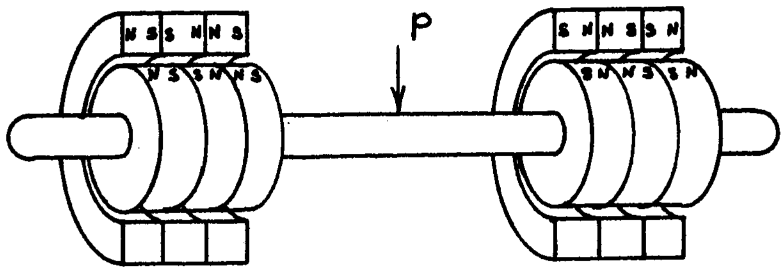

Fig. 16 PERMANENT MAGNET RADIAL BEARING - REPULSIVE
(Ref. 4)

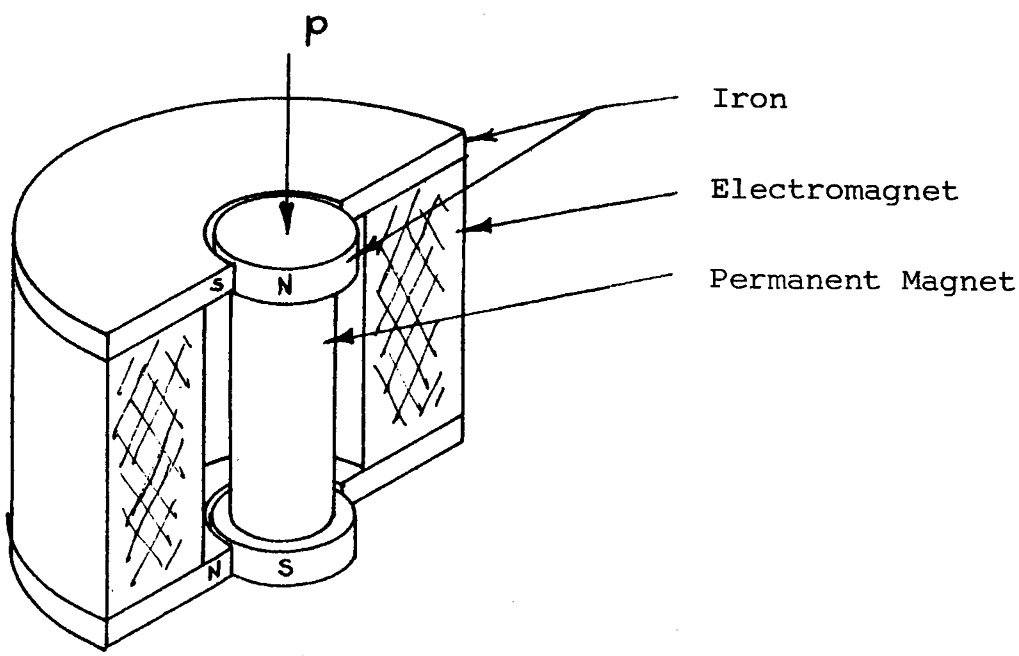

Fig. 17 ELECTROMAGNET THRUST BEARING - ATTRACTIVE
(Ref. 1)

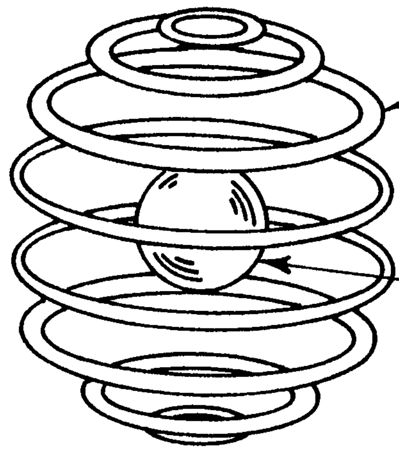

Fig. 18 SUPERCONDUCTING SPHERICAL BEARING
(Ref. 8)

TABLE 15 - MAXIMUM THEORETICAL BEARING STRESSES PROVIDED BY ELECTRIC AND MAGNETIC SUPPORT

Type of Support	Type of Bearing	Figure	Max. Bearing Stress (kg/mm^2)	Max. Bearing Stress (psi)	Design Reference
Permanent Magnet, Attractive	Thrust	1	0.042	60	1,2,3,4
	Radial	1	0.042	60	1,2,3,4
Permament Magnet, Repulsive	Thrust	3	0.042	60	1,2,3,4
	Radial	4	0.042	60	1,2,3,4
Electromagnetic	Thrust	5	0.17	240	1,5,6,7
	Radial	6	0.17	240	1,5,6,7
Superconducting	Spherical	7	4.2	6000	1,8,9
Electric	Spherical	8	7.8×10^{-4}	1.1	1,10,11
Electret	Radial Thrust		5.1×10^{-4}	12	12

The maximum bearing stress possible for magnetic support is limited by the maximum flux density, B, which can be provided at the bearing surfaces according to the formula:

$$\sigma = 4.1 \times 10^{-10} B^2 \text{ (kg/mm}^2\text{)}$$

$$\sigma = 5.77 \times 10^{-7} B^2 \text{ (\#/in.}^2\text{)}$$

where B is given in gausses.

Currently available permanent magnetic materials have a maximum residual flux density of about 12,000 gausses. Geometric and other factors limit the useful flux density to about 10,000 gausses, limiting the bearing stress to 0.042 kg/mm^2 (60 psi). Electromagnets can provide flux densities approaching the saturation point of the core material, approximately 20,000 gausses, giving maximum bearing stresses of 0.17 kg/mm^2 (240 psi). The penalty for the increased bearing load capacity is the additional weight of the magnet windings and power supplies. The use of superconducting elements reduces the weight of the bearing support elements and provides an inherently stable bearing (since the permeability is essentially zero). Recently developed superconducting materials (Niobium Stannide) permit flux densities in excess of 100,000 gausses at temperatures up to 18°K(13), producing bearing stresses of 4.2 kg/mm^2 (6000 psi) or greater.

Electric support is limited in air by the dielectric strength of air (3×10^6 volts/m) and in vacuum by field emission (c.a. 3×10^{-7} volts/m). The bearing stress for a given field strength, E, is:

$$\sigma = 9 \times 10^{-19} E^2 \text{ (kg/mm}^2\text{)}$$

$$\sigma = 1.28 \times 10^{-15} E^2 \text{ (\#/in.}^2\text{)}$$

where E is given in volts/m. Thus the maximum attainable bearing stress is 7.8×10^{-4} kg/mm^2 (1.1 psi). The power supplies for maintaining voltages greater than 10,000 volts is an added weight penalty. A novel (and as yet untried) method of electric support without power supplies is the use of electrets, or persistent polarization in dielectrics. Permanent surface charge densities of 1.5×10^{-4} coulombs/m^2 have been achieved on certain ceramics with this technique. These charge densities, when induced on both bearing surfaces, could provide bearing support stresses 5.1×10^{-4} kg/mm^2 (0.73 psi).

2. State of Development and Performance Potential

The use of magnetic and electric bearings has thus far been limited to specific instrument applications where the high cost of development and low load capabilities are outweighed by the almost zero friction which these devices can provide. One commercial magnetic bearing[14] which uses permanent magnets to support radial loads and servo-controlled electromagnets for axial loading, and several experimental magnetic bearings have been analyzed for load carrying performance characteristics. These are summarized in Table 16. It is noted that, at least in one case, the magnet design was inoptimum. The analysis showed that the performance of this bearing could be improved considerably by altering the magnet design and that large bearing loads can be supported more efficiently with large numbers of small magnets than with large magnets. Also included in the table are the performance characteristics of an electrically supported gyroscope rotor.

At their present level of development, electric and magnetic support systems have limited utility in space. Table 17 summarizes the characteristics, limitations and useful ranges of bearing systems employing these techniques. Recent

TABLE 16 - SUMMARY OF LOAD CARRYING PERFORMANCE OF EXISTING MAGNETIC AND ELECTRIC BEARINGS

	Type of Support	Type of Bearing	Diameter (mm)	Length (mm)	No. of Rotor Magnets	Rotor Weight (g)
Philips Lab (Ref. 15)	Perm. Mag.	Radial	60	30	20	240
IITRI (Ref. 4)	Perm. Mag.	Radial		43	9	216
General Electric (Ref. 1)	Perm. Mag.	Axial	35	30	1	35
Optimized, G.E. (Theoretical)	Perm. Mag.	Axial	35	70	1	82
High Load Optimized General Electric (Theoretical)	Perm. Mag.	Axial	35	210	5	350
Cambridge Thermionic (Ref. 14) (2 bearings each end)	Perm. Mag. Radial-Electro-Mag. Axial	Radial, Axial	38	82 1/2	2	18
University of Illinois (Ref. 11)	Electric	Spherical	50			25

TABLE 17 - SUMMARY OF OPERATING CHARACTERISTICS OF ELECTRIC AND MAGNETIC SUPPORT SYSTEMS

Type of Support	Maximum Load Capacity (kg/mm^2)	(psi)	Load to Weight Ratio*	Type of Bearing	Speed Range (rpm)	Wear Life
Permanent Magnet	0.042	60	1-20	Axial, Radial	0-25,000+	Unlimited
Electro-magnet	0.17	240	0.1-1.0	Axial, Radial	100,000	Unlimited
Super-conducting	4.2	60,000	10-100	Axial, Spherical	100,000	Unlimited
Electric	7.8×10^{-4}	1.1	0.1-1.0	Axial, Spherical	100,000	Unlimited
Electret	5.1×10^{-4}	0.73	100	Axial, Radial, Spherical	100,000	Unlimited

*Including Auxiliary Equipment

TABLE 16 SUMMARY OF LOAD CARRING PERFORMANCE OF EXISTING MAGNETIC AND ELECTRIC BEARINGS (CONT'D)

	No. of Stator Magnets	Total (g) Weight	Maximum Load* (g)	Maximum Load* (lb)	Bearing Stress (kg/mm^2)	Bearing Stress (psi)	Load/ Weight
Philips Lab (Ref. 15)	20	765	450	0.99	4.4×10^{-4}	0.625	00.59
IITRI (Ref. 4)	9	1475	1395	3.1	8.5×10^{-4}	1.2	0.95
General Electric (Ref. 1)	2	105	650	1.4	9.2×10^{-4}	1.31	6.2
Optimized, G.E. (Theoretical)	2	246	3000	6.6	4.3×10^{-3}	6.05	12.2
High Load Optimized General Electric (Theoretical)	6	770	8000	17.6	2.2×10^{-5}	3.23	10.4
Cambridge Thermionic (Ref. 14) (2 bearings each end)	2	235**	(ax.) 33 (rad.) 68	0.073 0.15			0.14 0.29
University of Illinois (Ref. 11)		200***	100		4.9×10^{-5}	0.07	0.5

* Including Rotor Weight
** Not Including External Power Supplies
*** Estimated, Not Including External Power Supplies

TABLE 17 - SUMMARY OF OPERATING CHARACTERISTICS OF ELECTRIC AND MAGNETIC SUPPORT SYSTEMS (CONT'D)

Type of Support	Reliability	Operating Temperature Range (°K)	Environmental Pressure (torr)	Radiation Resistance	Vibration Resistance	Auxiliary Equipment Requirements
Permanent Magnet	Excellent	0-700	Unlimited	Excellent	Depends on Design	None
Electro-magnet	Good	250-500	Unlimited	Poor	Depends on Design	Power Supply
Super-conducting	Poor	0-18	Vacuum Only	Good	Depends on Design	Power Supply plus Refrigeration
Electric	Good	0-1000	Vacuum Only	Poor	Depends on Design	Power Supply
Electret	Excellent	0-500	Unlimited	Good	Depends on Design	None

developments in permanent magnet materials have yielded only modest increases in energy content and maximum flux densities. Prospects are dim for major improvements.

Advances in superconducting magnets are more significant. However, the requirement that the entire rotor mass be kept at cryogenic temperatures in any realistic superconducting bearing configuration limits the utility of this technique. The low load capability of electric supports, combined with the weight penalties associated with maintenance of high potentials limits the number of applications for these devices. The use of electrets offers an interesting possibility for further developments in electric support.

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F. Fluid Film Lubricated Devices

1. Basic Design Considerations

The environment of space places a severe limitation on the use of fluid bearings as known on earth. The loss of lubricant due to vaporization and the extremes in temperature, both low and high, leave only a few special applications where fluid bearings are feasible or advantageous over other types of bearings and lubrication systems.

Fluid bearings can be calssified according to the state of the fluid used, i.e., liquid or gas, and these in turn can be classified into self acting (hydrodynamic and pneumo-dynamic) and externally actuated (hydrostatic and pneumo-static). Self acting fluid bearings do not require external power to operate except if circulation of the lubricant is required (an extremely small power requirement in most cases). Externally actuated bearings, on the other hand, require an external energy source in order for operation. The energy source can be electrical (pressurized gas tank), mechanical (spring or flywheel), or chemical (burning of a gas-producing propellant).

Since the availability of power in a space vehicle is rather limited, a large, long lived externally actuated bearing does not seem feasible, even in a closed loop where the fluid is used over and over again. However, it may be possible to design practical bearings for many hours of operation if they are small enough and if the weight and space penalty can be tolerated.

The temperature range in which fluid bearings can operate successfully ranges from cryogenic temperatures up to the highest temperatures required. However, no single fluid (with the exception of Helium) can operate throughout the whole range. Gases are by far the fluids with the widest temperature operating range. Liquids are much more restricted because of phase changes and decomposition.

Table 18 gives a summary of the operating temperature ranges for several fluids that can be used in fluid bearings.

TABLE 18 - TEMPERATURE RANGES FOR VARIOUS FLUIDS

Fluid	Min. Temp.	Max. Temp.
Gases (Helium)	20°K	1650°+K
Cryogenic Fluids	20°K	150°K
Lubricants (synthetic, natural)	220°K	770°K
Liquid Metals	700°K	1100°K

a. Self Acting Liquid Bearings

The limitations of a hydrodynamic bearing are governed by speed, temperature and pressure. Excessive speed will result in turbulence with a considerable increase in heat generation and the consequent power loss. The temperature limitations are determined by the fluid used, there existing both a lower and upper limit, which will depend on the viscosity-temperature properties of the fluid, thermal degradation characteristics, chemical activity, and possible phase changes. Very low ambient pressures can also limit the use of a liquid lubricant due to vaporization loss and cavitation.

The inherently low ambient pressure of a space environment limits the use of liquid bearings to either a temporary application (i.e., until the lubricant is exhausted), or to the inside of a hermetically sealed container. The operation of a liquid bearing inside a sealed container in space would be no different from one on earth, except for the temperature extremes, lack of gravitational forces, and high particulate radiation environment. The absence of gravitational forces would hardly affect the bearing itself but proper care should be taken in the design of the fluid reservoir. If the lubricating fluid is carefully selected, it will not be affected by particulate radiation.

b. Externally Pressurized Liquid Bearing

Externally pressurized liquid bearings, commonly referred to as hydrostatic bearings, have basically the same limitations with respect to speed, temperature and pressure as self acting liquid bearings. They have the advantage of a load carrying capacity independent of speed, and also allow more freedom of design with respect to configuration. Their main disadvantage is the requirement of an external power source to provide the pressurized fluid needed to support the load.

As in the case of hydrodynamic bearings, two possibilities exist with regard to the installation of a hydrostatic bearing: (1) sealed inside the vehicle, and (2) exposed to the vacuum of space. Since the flow of lubricant through a hydrostatic bearing is much higher than that through a hydrodynamic one, it is reasonable to expect a higher rate of lubricant loss by evaporation with this type of bearing. If the fluid is not recirculated, i.e., "once through" lubrication is used, then the rate of lubricant loss equals the total flow through the bearing, which can be considerable. Since the lubricant supply aboard a space vehicle is necessarily limited, the application of this type of bearing will be limited to either very small loads or to very short operating periods.

Sealed hydrostatic bearings will not experience lubricant losses, but will still require external power for operation.

c. Self Acting Gas Bearings

Gas bearings offer the widest operating temperature ranges of all known bearings. If a gas with a low boiling temperature such as helium is used, the bearing could operate from temperatures as low as 15°K up to the highest temperature the bearing material could withstand without limitations being imposed by the fluid.

Self acting gas bearings do not require an external power source to maintain a load carrying capacity. However, their load carrying capacity is inherently low and not comparable to that of liquid bearings. The practical limit unit load of 15 psi has been advanced by some authors [1], and the practical rule of "one (1.0) psi load capacity per 100 rpm per inch of diameter per 1000°F" has been suggested by Mack[2]. For spiral groove bearings, the load capacity is about 0.4 psi instead of 1.0 psi.

Self acting gas bearings are used in applications where speeds are high and loads are small. Under these conditions liquid bearings are not suitable because of the large viscous drag and high power dissipations. Further problems are introduced by the turbulence characteristics of high rotational speeds.*

As in the case of liquid bearings, gas bearings require a fluid media which is not available in space, and must therefore carry their own. Again, two methods of operation can be conceived: hermetically sealed and vented to space. The first case would not differ materially from the systems used

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* The viscous drag of a gas bearing is much less than of a comparable liquid bearing at the same speed. However, if the drag is compared for the same load carrying capacity, then the coefficient of friction (i.e., the ratio of friction or viscous drag force to load) will be larger for the gas bearing.

on normal earth environments, while the second would have to cope with the loss of gas through the seals, and which will have to be replenished from a reservoir to maintain adequate operating pressure.

Besides the small load carrying capacity inherent with gas bearings, there are other serious limitations that impose further restrictions in their use. The damping characteristics of gas bearings are rather poor, and this in turn coupled with the compressibility of the fluid media may give rise to rotor instability. It is also a characteristic of gas bearings that the clearances between rotor and bearing be small, of the order of less than .001 in. Small clearances not only introduce production and assembly difficulties, but they also increase the contamination sensitivity of the bearing and require careful filtering of the gas. A good surface finish, of the order of a few microinches is also necessary. If the bearing is to last a relatively long time, it is also necessary to insure that the bearing materials are dimensionally stable, so that the dimensions do not change appreciably during use or storage. Finally, extreme care should be excersized during design of the bearing to avoid differential thermal expansion of the shaft and sleeve when the temperature varies over a wide range.

Since a high relative speed is required for self acting bearing operation, practical configurations are restricted to journal, radial, conical or spherical bearings.

d. Externally Pressurized Gas Bearings

Externally pressurized gas bearings or pneumostatic bearings, have many of the characteristics common to self acting bearings. Clearances are small, damping is poor, and instability is often present; temperature range is wide, and the viscous drag can be very low, depending on the speed of the bearing.

Unlike self acting bearings, their load carrying capacity is independent of speed. However, an external power source or storage is required for operation.

Two modes of operation can be conceived for pneumostatic bearings, "once through" flow (vented to space), and flow with recirculation. A bearing with "once through" flow would necessarily be either very small or very short lived, since the storage capacity of gases in a space vehicle is very limited. If the gas is used over and over again, then it is necessary to have a power supply available to compress it to the required operating pressure. This requirement will limit the use of pneumostatic bearings to rather small or extremely important applications, since again the power available in a space vehicle is limited.

Since the load carrying capacity of pneumostatic bearings is independent of speed and dependent only on size and supply pressure, more freedom of design is possible and many different bearing configurations are practical. Pneumostatic bearings can be made in the form of journals, radial, conical, spherical, rectangular pads, bearings. etc. Configurations are limited only by the skill and imagination of the designer.

If the relative velocity between the elements of the bearings is low, viscous drag will be negligible, thus offering a system with the minimum possible friction. This is the main advantage of pneumostatic bearings, indicating its use in applications where friction must be minimized but the speed is relatively low, or where a gas must be used as lubricant, but the speed is too low to operate a self acting bearing effectively.

It seems rather unlikely that the need for heavily loaded bearings will arise in the gravity free environment of space. The type of loads encountered in space will be of a dynamic nature, such as centrifugal forces, jet reactions, etc. In many cases the bearings will act more as guiding devices, i.e., maintaining concentricity or alignment of lightly loaded components, rather than load carrying members.

Taking a single-axis gyro as a typical example of space application, it is possible to make some simple calculations about power, weight and volume required for different bearings and its peripheral equipment, if any.

A typical hydrostatic gas bearing, to support the gimbal axis of a single-axis gyro, will consume approximately .7 cfm of standard air at 32 psig plenum pressure when orifice regulation is used, or .15 cfm if slit regulation is used instead(3). A gas floated spinning sphere may consume approximately .05 cfm when supported by four or six orifice compensated pads(3).

The pressurized gas required for operation can be either carried in a bottle and flow through the bearing once and be vented, or could be recovered and repressurized in a compressor. In the first case, adequate supply must be carried to maintain operation for the duration of the mission, and in the second, sufficient internal power must be available to run the compressor.

For the higher temperatures and large flow rates, the power requirements are exceedingly high. Even for an intermediate temperature such as 300°K, and an average flow rate, say .3 cfm, the power needed to operate the bearing of just one gyro would be around 50 watts. This is probably more than can be spared to run the bearings from the meager vehicle power supply.

A considerably reduction in pumping or compressor power demand is obtained when a condensable vapor is used as working fluid in the bearing. This technique has been proposed(3) for use aboard vehicles designed for missions of long duration.

The scheme operates in a cycle resembling the Rankine power cycle, except that the bearing is substituted for the turbine or engine normally used in the Rankine cycle. The working fluid must be readily vaporized at a reasonable temperature, and have a low specific heat and heat of vaporization. Different grades of Freons have been suggested as working fluids. They vaporize at a relatively low temperature, yet not so low that condensation becomes difficult. They also have favorable chemical properties, i.e., non-toxic, non-flammable, compatible with most materials, etc., and are readily available at reasonable cost.

The use of such a cycle would reduce the internal power requirements by about two orders of magnitude (less than 0.1 watt for comparable bearings), but requires a fair amount of heat exchange with the environment in order to evaporate and condense the working fluid. The cycle, shown schematically in Fig. 19, consists of a condenser, a pump, an evaporator, a pressure regulator, and the bearing.

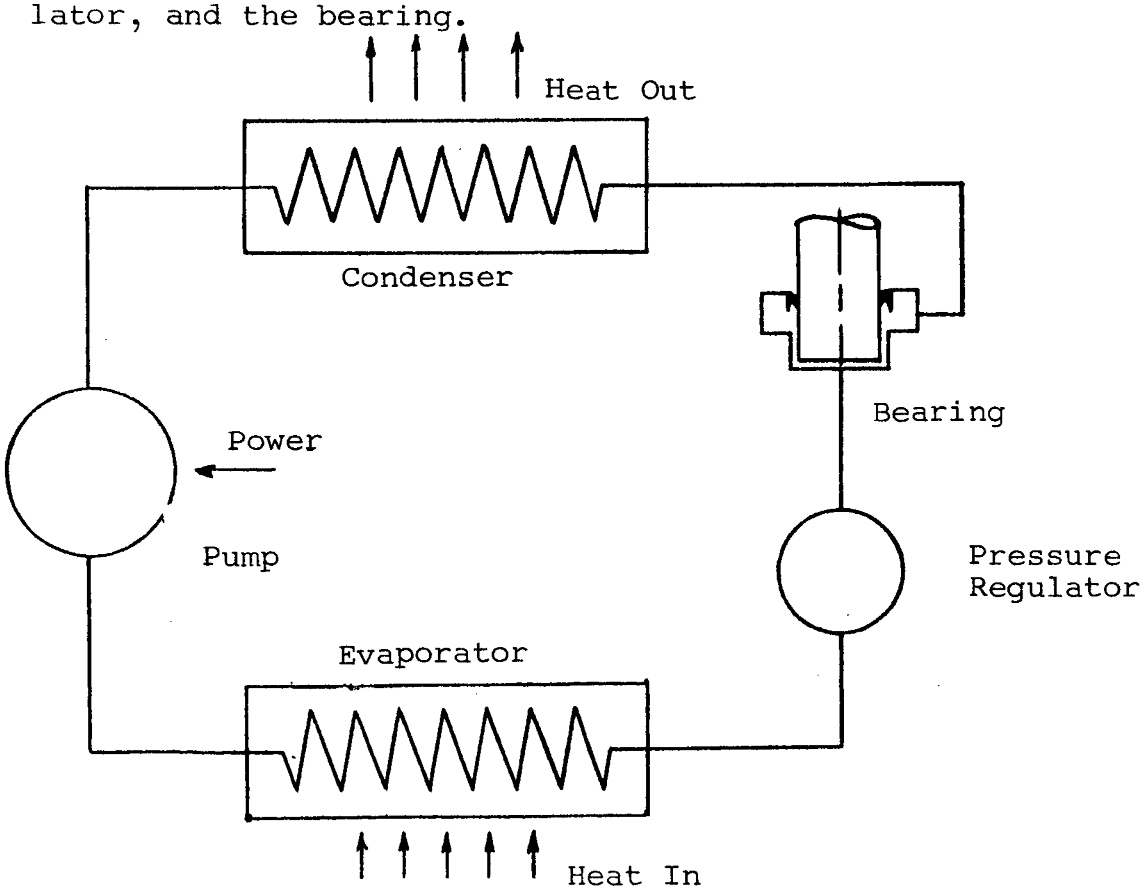

Fig. 19 PNEUMOSTATIC BEARING WITH VAPORIZABLE FLUID

The condenser is simply a heat exchanger exposed to the dark side of the vehicle where the working fluid loses heat at constant pressure until it becomes a saturated liquid. The pump transfers the liquid to the evaporator or boiler and increases itspressure to the level required by the bearing. The evaporator is also a heat exchanger, this time located on the illuminated side of the vehicle, and where the fluid will gain heat by radiation at a constant but higher pressure until it becomes saturated, or even superheated vapor. Going through the pressure regulator, whose function is to maintain correct operating pressure, the pressurized vapor is throttled through the bearing, thus loosing its pressure at constant enthalpy and finally enters the condenser completing the cycle.

A variation of this cycle would be a scheme where the mechanical pump is substituted by a capillary tube to force the liquid into the evaporator at constant pressure. Once the evaporator is charged, a valve closes and the phase change proceeds at constant volume until the required operating pressure is reached. Then a valve cracks open releasing vapor at the required rate. This scheme is not a true thermodynamic cycle and by necessity is of intermittent operation. Its main advantage is that no internal power whatsoever is needed for operation. All the energy requirements can be supplied by the environment.

The thermal exchange requirements are not excessive. In order to supply .2 cfm of 32 psig vapor at 300°K, it is necessary to add heat to the system at a rate of less than 100 watts and reject at about the same rate.

The reason why the vapor-liquid scheme is so conservative of power is that the compression process is performed on a liquid rather than a gas, with the consequent minimum of energy storage in the fluid. One of its main advantages is that the power required for operation can be obtained from space, since thermal energy can be utilized directly. Its main disadvantage is the added complexity and weight of the heat exchangers and other peripheral equipment.

Liquid or hydrostatic bearings, require much less power than pneumostatic bearings to support the same load. For instance, a pneumostatic thrust bearing 2 in. in diameter, environment pressure of 14.7 psi and pressure ratio of 6, can support a load of about 93 lbs with a gas flow rate of $.3 \times 10^{-3}$ lb/sec at 300°K if a clearance of around h = .0008 in. is maintained. The same load can be carried by a bearing of the same size and clearance, operating with a fluid comparable to SAE 5 light oil at 300°F (= 4.65×10^{-6} reyns, s.g. = .8) with a flow rate of 3.58×10^{-3} lb/sec. The ideal pumping work required to operate the gas bearing is around 15.2 watts, but the liquid bearing requires only about .9 watts to support the same load.

2. State of Development and Performance Potential

The theory, utility and limitations of fluid film bearings are well defined. The disadvantages of high power consumption and fluid reservoir weight penalties associated with externally pressurized bearings, limits their utility for space mechanisms. The fluid film bearings offer little advantage over equivalent antifriction bearings. The place of hydrodynamic bearings in space seems to lie in special applications where a fluid is already available, and a compatible antifriction bearing cannot be found, or when the good damping characteristics, or small space requirements of a fluid bearing are necessary.

A summary of fluid film operating characteristics is presented in Table 19.

TABLE 19 - SUMMARY OF OPERATIONAL CHARACTERISTICS OF FLUID FILM BEARINGS

	Liquid Bearings		Gas Bearings	
	Self Acting	Externally Pressurized	Self Acting	Externally Pressurized
Max. Load Capacity (kg/mm^2)	2.3	No Limit	0.021	No Limit
Max. Load Capacity (#/in.^{2})	3000	No Limit	30	No Limit
Load to Weight Ratio	1000 - 10,000	100 - 1000	50 - 500	100 - 1000
Type of Bearing	Rotary, Linear	No Limit	Rotary	No Limit
Speed Range (rpm)	10 - 10,000	0 - 10,000	0 - 100,000+	0 - 100,000+
Life	Unlimited	Unlimited	Unlimited	Unlimited
Reliability	Excellent	Good	Good	Good
Operating Temperature Range (°K)	240 - 1200	240 - 1200	20 - 1650	20 - 1650
Environmental Pressure (torr)	1 - 760	1 - 760	760	10^{-13} - 760
Radiation Resistance	Good	Good	Excellent	Excellent
Vibration Resistance	Excellent	Excellent	Fair to Good	Fair to Good
External Equipment Requirements	None	Pumps, Fluid Reservoir	None	High Pressure Gas Reservoir or Pumps

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3. A Closed-Cycle System for Gas Bearings, J. H. Laub and H. D. McGinness, JPL Technical Release No. 34-174, January 19, 1961.