MECHANICAL WEAR AND LUBRICATION

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INTRODUCTION

The need for a better understanding of wear and lubrication of moving parts is greater today than ever before. Although much has been done to establish the laws and theories concerning wear and its prevention by proper lubrication, the ever changing and increasing requirements under which lubricant must work makes this field a continuous research endeavor. Engineers are constantly striving to obtain more power from engines and motors, longer life of moving mechanical parts, and new applications of standard mechanical equipment. All of these problems impose additional burdens on the lubrication engineers who must find new lubricants and lubrication techniques.

This presentation will consist of two parts. The first will be a discussion on the theory of wear, its causes, and so forth, and a brief introduction of the principles of lubrication. The second part will discuss the principles and three major types of lubrication as applied to practical uses in bearings and other mechanical devices. Examples of the types of lubrication commonly used are illustrated, and a brief discussion of current research for lubrication techniques for extreme environmental conditions is presented.

CONTACT BETWEEN TWO SOLIDS

Solid surfaces, as a rule, are extremely rough on the molecular scale. A mirror finished surface has a roughness of a few microinches. The intermolecular distances of most materials are only about 1/100 of a microinch. When two solids are brought into contact, the configuration is like that shown in figure 1 for smooth (in the common sense) surfaces. The contact is restricted to a number of asperities (peaks). Without load there would only be a few contact spots with a negligible area. When load is applied, the stress at the contact area far exceeds the elastic limit and plastic deformation takes place until the total actual area of contact A has grown such that:

$$A = \frac{W}{Pm}$$

where Pm is called the mean yield pressure if the weaker of the two solids and is approximately three times the elastic limit for most metals and W the applied load. This equilibrium contact area therefore, is proportional to the load, and is still much smaller than the apparent one. (This is the area of contact on a macroscopic scale and used only for calculating average load.) Friction between two solids, as we shall discuss later, depends on the actual area of contact. This explains the Amontons laws of friction which states that the friction between two solids is proportional to the applied load and is independent of the area (apparent) of contact.

The interaction between two moving solids is extremely complicated. The repeated deformation of the asperities results in considerable amounts of heat generation. Many hot spots are thus created at the interface. Since solid surfaces are non-uniform on a microscopic scale, the temperature distribution is also heterogeneous. Experimental evidence has shown that temperature rise is only limited by the melting points of the solids, and that the mechanical strength of the contacting area is greatly reduced.

There are two types of moving contacts, namely, sliding and rolling. Each type consists of a continually making and breaking contact area. The direction of surface stresses is very different. In rolling contacts, the stress is always very close to the normal direction and a good portion of the work done in elastic deformation is subsequently recovered. In sliding contacts, a very significant fraction of the stress is in the direction of travel and the absence of elastic recovery causes much higher friction. Furthermore, the high asperities are almost always in contact continuously in sliding contacts and therefore can have hot spots of much higher magnitude. Since metals are softened by heat, the actual contact area of a sliding pair is in general much larger than that of a rolling pair.

MECHANICAL WEAR

Many processes and their combinations may cause mechanical wear, the principle ones are discussed as follows:

1. Adhesion

This is undoubtly the chief source of heavy mechanical wear and failure, particularly in sliding contacts like sleeve or plain bearings. It has already been shown that the contact spots are very hot, thus welding between the two surfaces may take place. Furthermore, metals may diffuse into each other forming alloys even without welding. Subsequent shear-off of these joints may produce many irregular fractures. The damage frequently penetrates far beneath the contacting surface.

It is interesting to consider the sliding friction as the result of shearing-off the contact areas. The coefficient of friction:

$$\mu = \frac{F}{W} = \frac{sA}{Pm\,A} = \frac{s}{Pm}$$

Where s is the ultimate shear stress, usually s is of the same order of magnitude as the yield pressure Pm, and μ is therefore approximately unity for identical materials. Dissimilar materials do not weld readily and a smaller μ would be observed. Experimental results indeed show this is the case and usually mechanical wear can be reduced by using a pair of dissimilar materials.

Presence of a contaminant layer (mostly oxide) on the surface as a rule may drastically reduce the friction between the two metals by preventing welding. It is well known that in welding or soldering unless the surfaces are extremely clean, a flux must be used to remove the oxide film as well as other impurities. Large scale melting even without welding may cause adhesive wear. Such a case is illustrated by an inadequately cooled grinding wheel where molten metal penetrates into the pores and smears all over the wheel surface.

2. Abrasion

This form of wear is characterized by the fracture or plastic deformation caused by instantaneous high stress. This abnormally high stress may be caused by either impact or ploughing action. When a soft material is rubbed against a hard one, the resulting surface of the soft one should have a roughness comparable to that of the hard surface. The worn areas are much more uniform than those resulting from adhesive wear. In tool grinding or metal cutting a coolant is always used to prevent surface melting and adhesion in order to produce a good surface finish. During the running-in period of many moving parts, a specially thin lubricant is frequently used to facilitate heat transfer thus limiting the wear to the abrasive form.

The effect of high local surface temperature may be demonstrated by the polishing process. Zinc oxide, for instance, is considerably softer than quartz (M.H. 4 vs 7) yet quartz may be polished by zinc oxide. The fact is that zinc oxide has a melting point 3200°F as compared to 3100°F for quartz. The presence of a lubricant however is very essential to prevent adhesion.

The presence of dust particles and the like may cause excessive abrasive wear, particularly where close tolerance is needed. In a well lubricated moving assembly, the presence of undesirable materials become the main source of failure. Many ball and roller bearings are sealed for this purpose.

3. Pitting

This is the result of either defect in the material of localized high stress concentration under repeated load. Many failures in gear teeth and balls are attributed to this mechanism. In general, good design and selection of materials may minimize this form of wear.

4. Erosion

The presence of fast moving fluids leads to this type of wear. Examples are wind resistance in high speed vehicles or splashed oil droplets at high velocities in a crankcase. Except in special cases, the degree of damage is less severe than that caused by other processes.

5. Corrosion

Strictly speaking, corrosion is not a form of mechanical wear. However, the damage due to corrosion may be greatly enhanced because of mechanical motion. It is well known that corrosion rate increases rapidly with temperature and stress concentration. Oxidation or chemical reaction with other materials forms loosely adhered surface film that will fall off gradually.

We must emphasize that more frequently failures are caused by the combination of the above than other processes. In this presentation, attention has been given to adhesion and abrasion, as usually these are the two predominant processes. The prevention of wear by lubrication will be considered in the next section.

PRINCIPLES OF LUBRICATION

When two solids are in moving contact, wear invariably results. The most effective method of lubrication is to keep the solids completely separated. A journal bearing is chosen as an example for this discussion because the principle is identical for other forms of moving assemblies. Figure 2 shows schematically the action of a journal bearing. Consider such an assembly as at rest, completely filled with lubricating oil, the shaft naturally rests on the middle of the sleeve. As the shaft starts to rotate clockwise, the oil layer next to the shaft is carried by the viscous force. A differential oil pressure is developed from between the right and left sides of the shaft. This pressure shifts the shaft to the left and in the mean time provides a lifting action as oil is essentially pumped into the contacting area. As the rotating speed increases the displacement and the lift of the shaft is increased. Eventually an equilibrium configuration is reached as shown in figure 2 together with the oil pressure profile.

This most popular technique of lubrication is known as hydrodynamic lubrication. Since the separation between the shaft and the sleeve is caused by the pumping action of the shaft, the performance of journal bearings is largely determined by the properties of the lubricating oil, particularly its viscosity. The minimum film thickness (or minimum separation) h_o depends on various factors such as load, speed, viscosity of the oil, and so forth, and is usually 0.1 to 5 mils. This layer of oil although thin, is enough to separate the shaft completely from the sleeve. The force between the two bodies is completely transmitted hydraulically. The effect of surface roughness (small compared to h_o) is insignificant other than causing a small change in h_o.

The advantage of hydrodynamic lubrication is not only limited to the low wear, but lies in its low friction. In well designed bearings, friction coefficient as low as 0.001 is frequently achieved. Due to the complete separation of moving parts, wear becomes very low which is mainly due to corrosion and erosion.

In many assemblies, complete fluid layers between moving parts cannot be created by their own motion yet a hydrodynamic type lubrication is necessary. This can be achieved by using an external pump to maintain pressure and a flowing layer of lubricant. This technique is known as hydrostatic lubrication. The motion again is restricted to the lubricant layer. The friction and wear characteristics resemble those of hydrodynamic lubrication. By varying the pumping rate, the film thickness may be varied within limits, this enables one to minimize friction, the reason for this will be given later.

Frequently the separation of moving parts or bodies by a fluid cannot be maintained, the less desirable boundary lubrication technique must be used. The principle involved is to introduce a contaminant layer at the solid interface such that slipping within this layer is easy and that adhesion is eliminated. To fulfill these requirements, the lubricant must be able to attach firmly to the solids and be very soft in itself. Many organic acids can react with metal surface to form a firmly attached film of soap and are therefore very good boundary lubricants. Oxide films of metals as mentioned before frequently can act as boundary lubricants for limited durations. Contaminants can be beneficial by acting as boundary lubricants or detrimental by acting as abrasives, depending on their nature. In boundary lubrication, the separation between the moving bodies becomes very small (frequently only a few molecules thick). Force is directly transmitted between the shaft and the sleeve. The two bodies are in direct contact mechanically.

Deformation of the moving bodies therefore do take place. Due to the presence of contaminant layers, damage is reduced by minimizing adhesion. Wear of the surface is much higher than that in hydrodynamic lubrication.

The coefficient of friction is usually in the neighborhood of 0.15 which although much higher than the figure for hydrodynamic friction is still a very small fraction of the value in the absence of lubrication.

In rolling contact bearings, such as ball and roller bearings, at any instant, the load is supported by a small number of rollers (or balls). The average pressure at the contact area of these rollers is very high. As a result the races and the rollers have very intimate contact and the lubricant film needed for hydrodynamic lubrication cannot be maintained. Since the friction of rolling contact is intrinsically low, boundary lubrication is sufficient to reduce the coefficient of friction close to .001 comparable to that of hydrodynamically lubricated journal bearings. Unlike the latter, a rolling contact bearing does not require a starting period to establish the equilibrium condition. Consequently its starting friction is only slightly higher than its running friction, while the starting friction in a sleeve bearing is much higher (in the boundary region). However, from a wear aspect, hydrodynamically lubricated journal bearings are much superior. The rolling contact bearings are, therefore, mostly used for light duty applications.

In the discussion of hydrodynamic lubrication, we pointed out that the minimum film thickness may change depending on various factors. With gradually increased load and reduced speed, the film thickness may be reduced continuously to practically zero (boundary lubrication). The coefficient of friction may be plotted versus the factors $\frac{Z\,N}{P}$ as shown in figure 3. Where P is the average pressure on the bearing surface, Z is the viscosity of the oil and N is the rotation speed in rpm. In numerous practical cases, bearings actually operate in the mixed range (thin film lubrication) where the shaft and sleeve are in partial mechanical contact. Figure 4 shows schematically the different types of lubrication as characterized by the thickness of the lubricant layer. Hydrostatic lubrication can be viewed as using an external pump to increase the film thickness from the boundary or mixed region to the point where friction is at the minimum.

LUBRICATION IN PRACTICE

The theory of wear and some of its causes have just been discussed. Basically, we saw that when two surfaces move past each other in close proximity, as in a bearing, high friction and wear occurs because of the asperities of the surfaces coming into contact and being torn away by the shearing force. We also saw that to prevent this friction and wear, lubrication must be employed. The basic theories of lubrication were discussed and listed in three main categories: hydrodynamic, hydrostatic, and boundary. The purpose of this part of the presentation is to discuss the application of these theories to actural practice, to discuss the various techniques used as well as to illustrate examples of each type in use today. Some of the advance techniques which are currently being developed to provide lubrication in unusual applications and under extreme environmental conditions will be discussed. Most of these techniques are not known to the general lubrication industry and may hold potential for use in various industrial applications.

HYDRODYNAMIC LUBRICATION

This type of lubrication is probably utilized more in industrial applications than all other types combined. Almost any machine which requires lubrication will depend on this hydrodynamic action in one part or another. In the discussion of hydrodynamic theory this action depended on the creation of a load carrying fluid film. This film can be obtained with almost any material which can be considered fluid. Bearings have been run successfully on water, oil, alcohol, mercury, acids, molten metal, gasoline, and even air.

The discussion here will be centered around the most familiar and probably the most useful of all bearings, -- the journal bearing. This type of bearing consists of a sleeve of bearing material wrapped partially or completely around a rotating shaft or journal and is designed to support a radial load.

Going back to figure 2 in the first part of the presentation, we see a schematic view of a journal bearing, with a greatly exaggerated clearance, running at some speed under a given load. At rest, metallic contact probably exists under certain heavy load conditions; and as the shaft starts to rotate, it will start to run up the right side of the bearing. In doing so it will enter a region where oil comes between the bearing and journal. Then it begins to slip. As the journal comes up to speed and revolves faster and faster, the wedge shape film is built up and the journal is in the position shown in figure 2. For heavy loads and low speeds this position is common in bearing operation but at high speed and light loads this eccentricity approaches zero. Thus, it is evident that in the design of a bearing, one of the most important items to be considered is the minimum film thickness. If the film thickness becomes too thin, metal-to-metal contact will be initiated and wear or failure results. Some of the things to be considered in establishing the minimum acceptable film thickness are surface finish of the sliding surface, thermal distortion, and the size of any contaminating particles in the oil stream. For very finely finished surfaces such as found in automotive or aviation engine bearings with clearance of .0001 to .0002 in., it is obvious that the oil must be filtered to take out all particles larger than the minimum film thickness or serious wear and ultimate failure may take place.

Since these bearings must be supplied with the fluid to form this film, bearing designers must take into account methods for distributing the fluid to the rubbing surfaces of the bearing. The simplest configuration is that of a hole in the bearing, however, in some cases insufficient fluid is supplied and poor distribution over the sliding surfaces is obtained. Bearing designers for years have used grooves in the bearing to distribute the fluid. The most important consideration is the location of the grooves. Figure 5 shows the effect of a groove on the hydrodynamic pressure. This groove bleeds off the hydrodynamic film at that point and the pressure falls to the value of the supply pressure. The destruction of this film by the groove may well cause the bearing to fail.

Bearing seals are an important consideration in certain types of bearings. These seals essentially serve two purposes, to keep the lubricant in and to keep dirt and contamination out.

Supply of fluid to the bearing must be in relation to the needs of the bearing and depends on the type as well as the conditions under which it must run. Some bearings are supplied by fluid pumped under pressure. This type is used when the bearing requires a considerable amount of fluid due to leakage and so forth. Other bearings that require low rates of supply can be adequately lubricated by felt wicks or pads. Some can be supplied by use of waste packing such as railroad journal bearings and the electric-motor armature bearings. Here the waste presses against some part of the rotating shaft and transfers the oil to its surface by capillary action.

To this point, most of the discussion has centered around the bearing and its design. Something should be said about the lubricant or fluid and the properties which are considered when choosing a lubricant. Probably the most important property of any fluid is viscosity. In bearings of all kinds, in gear boxes, hydraulic systems, engines, or whenever a fluid lubricant is used, all other things being equal, it is the viscosity that determines the friction loss, heat generation, thermal efficiency, load carrying capacity, film thickness, lubricant flow, and in many cases wear. Viscosity is truly a measure of the physical ability of a fluid to maintain lubrication under the specified conditions of speed, temperature, and pressure. To illustrate the different viscosities of different fluids, water has a value of 1 at 70°F, olive oil - 100, 30 wt. motor oil - 300. So it can be seen that the ability to pump a fluid, or the fluids ability to flow, carry a load, and so forth, depends on its viscosity. A lubricant, for any particular use, must be a suitable viscosity at the operating temperature of the bearing because of the inverse relationship of viscosity and temperature. The viscosity must not be too high at start up and not become too low as the bearing goes to temperature. If the viscosity drops too low the load carrying ability drops, leakage may occur and bearing failure results.

Thermal and oxidative stability of a fluid must be considered when operation will be above ambient temperature. At high temperatures, organic fluids tend to breakdown, due to the temperature and oxidation, change in viscosity, and generally lose their lubricating ability, which of course, ultimately causes failure.

The fluids must be chemically compatible with the bearing materials and other materials with which they come in contact. It is known that certain metals such as pure copper can cause catalytic reactions between the metal and fluid which could produce undesirable products and breakdown the fluid. Also, the fluid can attack some metals and cause severe corrosion and erosion on vital parts.

Under some extreme load conditions, such as found in hypoid gears, and so forth, the use of extreme pressure additives in the fluid is warranted to increase the load carrying ability and to prevent scuffing and wear. These additives, although carried in the fluid serve more as boundary lubricants which will be discussed later.

These properties discussed above are some of the general characteristics considered when selecting a lubricating fluid. Other physical and chemical properties must be considered when selecting fluids for specified applications. These general characteristics apply to fluids used in hydrodynamic lubrication as well as hydrostatic lubrication which will be discussed next.

HYDROSTATIC LUBRICATION

This type of lubrication is similar to that of hydrodynamic in that both rely on a film of lubricating fluid to separate the moving parts. The primary difference lies in the fact that the pressure which forms the load carrying film comes from an external source such as a pump rather than generated within the bearing itself. Figure 6 shows a generalized schematic illustrating how the hydrostatic pressure supports a load. The bearings using this type of lubrication have long been known as oil pads. One of the most notable uses of this type of lubrication is with the big 200-inch Hale Telescope. This telescope weighs nearly one million pounds, and the problem in moving this mass in accurately tracking stars, and so forth, was not an easy one. Ball and roller bearings had been considered but even their relatively low coefficient of friction (.001) introduced serious complication due to the friction torque, which would have resulted in greater twisting of the structure.

This great weight was supported at three points by oil pads. Each pad is loaded with a force of about 164,000 pounds. They are 28 inches square and contain four recesses each 7 inches square. A high pressure pump supplies the oil to each of the recesses which in turn supports the load. The primary reason for the use of oil pads on the Hale Telescope is the low friction that results. This great bulky structure turns smoothly on its pads, following the path of stars with exertion of only about 50-foot-pounds of torque. This means that with the one million pounds of weight, the coefficient of friction is less than 0.000004 for the entire supporting system. The power required is extremely small and a 1/12-hp clock motor used to move it provides considerably more power than is actually required.

This same principle is used in step bearings taking thrusts at the ends of shafts; in oil lifts to reduce starting friction with heavy machinery such as motor generators or turbines; to prevent metal-to-metal contact on sleeve bearings where rotational motion is too slow to produce the necessary hydrodynamic film and in rolling-mill bearings and in many other applications where the principle can be applied. Figure 7 shows the application of this principle to a step or thrust bearing in vertical turbogenerator. Figure 8 is an illustration of using this principle as an oil lift to support the load in cases where the load is too great for the hydrodynamic film or the rotational motion is too slow to produce the hydrodynamic film. This prevents metal-to-metal contact and increased friction and wear in starting, stopping, reversing, and so forth.

BOUNDARY LUBRICATION

In almost all cases, the exception being very stringent laboratory conditions, surfaces are covered with some kind of surface film. These films are water, vapor, oxides, oily films, dust, corrosion, and so forth. In fact, it is exceedingly difficult to obtain a surface with a truly clean surface in the chemical sense. Some of the early work conducted in this area has shown that the friction is high for clean surfaces but drops sharply when surfaces are contaminated. This discussion then centers around applications in which surfaces rub together without the benefit of the fluid film discussed earlier. These are the conditions which are known as boundary lubrication conditions. In order to establish an understanding of the relationship of the boundary lubrication to that of fluid film in so far as friction is concerned, we refer to figure 3. We see that under boundary conditions the friction is much higher than with fluid lubrication. The zone separating the two is called mixed and in this region both boundary and fluid film lubrication occurs. In the $\frac{ZN}{P}$ relationship, Z is the viscosity in centipoises, N the journal speed in rpm, and P the pressure in psi.

Boundary lubrication is recognized to exist in almost every kind of bearing in one form or another. In rolling-contact bearings, the load is carried by balls or rollers which are in intimate contact with the race and therefore the lubrication is considered essentially boundary lubrication. Although these bearings are sometimes lubricated with fluid lubricants, the loading due to point or line contact is too great to permit the formation of the hydrodynamic fluid. While the low friction of ball and roller bearings is due mainly to this rolling contact, sliding does occur, especially between the cage or retainer and the balls and rollers. The condition here is primary boundary and accounts for the cages and retainers showing the greatest wear in these types of bearings.

In journal bearings, during the starting, stopping, reversing, where the shaft passes through zero speed, boundary conditions may develop. The rubbing surfaces of pistons, piston rings, and so forth, frequently operate in the boundary region. These conditions are found between gear teeth, especially hypoid gears. Boundary lubrication is present during the running-in process of most lubricated moving surfaces.

Much of the earlier work with boundary lubrication was conducted with fatty acids and other organic compounds. These compounds possessed the characteristics of adhering so tightly to the surface that they resisted being torn off when the two surfaces rubbed over each other. Further work resulted in the use of active chemicals which when reacted with the metal surface produced an inorganic film of low shear strength. Chemicals containing chlorine, sulphur, and phosphorus are used for this purpose and produce films which are much more thermally stable than the older organic films. These chemicals are used to produce extreme pressure additives in greases and oils for use in gears and other highly loaded applications. The hypoid gear in the rear axle of an automobile is a good example of this application.

Many research laboratories have developed various test machines from the study of variables of boundary lubrication. Of these, probably the best known are the Falex and Timken Testers. Schematics of the test specimen and loadings of the two testers are shown in figures 9 and 10. The loads in these tests are usually such that the hydrodynamic film cannot develop, thus, the boundary lubrication properties can be studied.

The best known types of boundary lubrication today which have been tested on the machines above are known as solid film or dry film lubricants. These consist of laminar or layer-lattice structure solids, such as molybdenum disulphide or graphite, which exhibits low shear strength within the crystal structure. A coating of the solid material is bonded to the bearing surface with some type of an adhesive. The resulting film will carry extreme loads and sliding friction is on the order .1 to .2. These dry films are beginning to find many applications in lubrication of moving parts which have been problem areas, and are being investigated for various uses in the Air Force.

From this discussion, it can be seen that boundary lubrication which does not completely separate the two sliding surfaces as in hydrodynamic or hydrostatic lubrication, can still effectively reduce friction and wear in applications too strenuous for the load carrying fluid films. In keeping with our general theme of failure it is readily apparent that with this type of lubrication, wear and possibly bearing, failure is more likely to occur unless extreme care is taken to choose the correct lubricants for the operating conditions. This type of lubrication is not new, however, it is not understood as well as hydrodynamic. Much work is currently being conducted in this area and new applications are being found.

NEW TECHNIQUES FOR FUTURE APPLICATIONS

We have just discussed the theories and practices of lubrication which have been with us since the turn of the century. What is currently being done to expand these practices and to find new applications in lubrication ? The Air Force is currently conducting programs to meet existing and future requirements which also have potential applications for industry in the more distant future.

One of these programs is concerned with the use of gas bearings. In this case, a gas rather than a fluid is used to separate the moving parts and provide lubrication. In some instances air is used as the lubricating medium. Some of the advantages of using air are (1) lubricant is always available, at least for earth bound applications, (2) absence of contamination, (3) low friction, (4) high temperature applications. As is typical with gases, the viscosity of air increases with temperature in the regions of 1000°F and higher, consideration of air as a lubricant seems promising. The disadvantages include low load carrying capacity, and extremely precise geometry and surface finish of the parts. While using air or gases as lubricants is a new technique, the principles involved are the same as discussed earlier. The air can form a hydrodynamic film and support a load or it can be applied to the bearing under pressure to work as hydrostatic lubrication.

We briefly touched on solid film or dry film lubricants under the discussion of boundary lubrication. This is an area that is receiving considerable interest for many varied types of application. As stated before much of the earlier work has been conducted with MoS_2 or graphite and phenolic or epoxy resins as binders. These resulting films have shown excellent wear life in the temperature range of ambient to 450°F. Above this temperature the organic resins suffer thermal degradation and cannot be used. The interest in this type of lubrication is of course at much higher temperatures and low pressures found in space and much more stable pigments and bonding agents are needed. The upper temperature limit of the lubricating pigments, MoS_2 or graphite, was reached at about 750°F. At this temperature MoS_2 begins oxidizing to MoO_3, and abrasive, and graphite begins to lose its lubricating ability on the theory that the adsorbed gases or water vapor is being driven off. The researchers have now turned to search for more thermally and oxidatively stable pigments and bonding agents. Pigments being studied at the present time include lead sulphide (PbS) which shows good lubricating characteristics at 1000-1200°F. The bonding agent used is a ceramic type material, boric oxide (B_2O_3). Other work is being conducted in the temperature range of 1500°F to 1800°F. Calcium fluoride (CaF_2) has shown good lubricating qualities in this temperature range. Other methods of bonding pigments to the bearing surface are being explored. These include vacuum deposition, flame spraying, plasma arc, and the use of ceramic adhesives developed for metal-to-metal bonding. In the temperature range of 1800°F to 2000°F the feasibility of operating bearings without a lubricating medium has been demonstrated. This type of approach has shown promise by careful selection of bearing and shaft material which will not seize or gall at these temperatures and which have fairly low friction and wear. Many of these approaches are only in the study stage and much work remains to be done, but all hopefully point toward one thing -- the successful operation of moving parts without excessive friction and wear under extreme environmental conditions which was once considered impossible.

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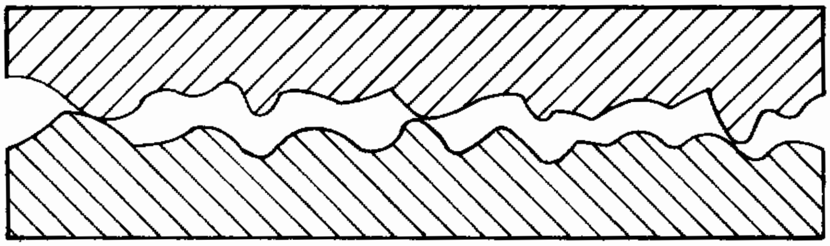

Figure 1

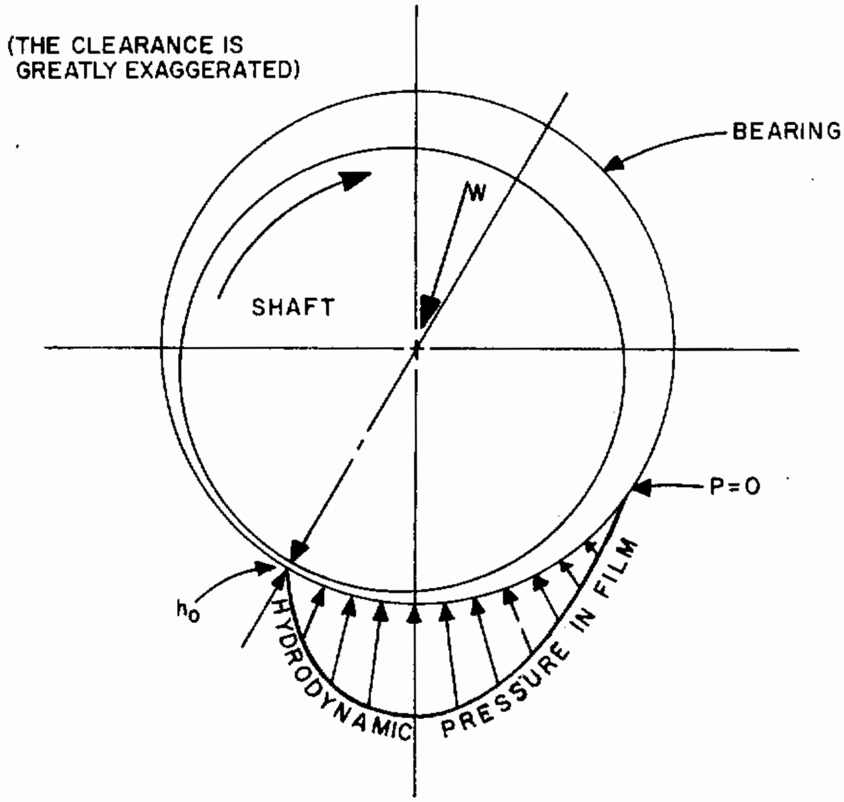
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Figure 2

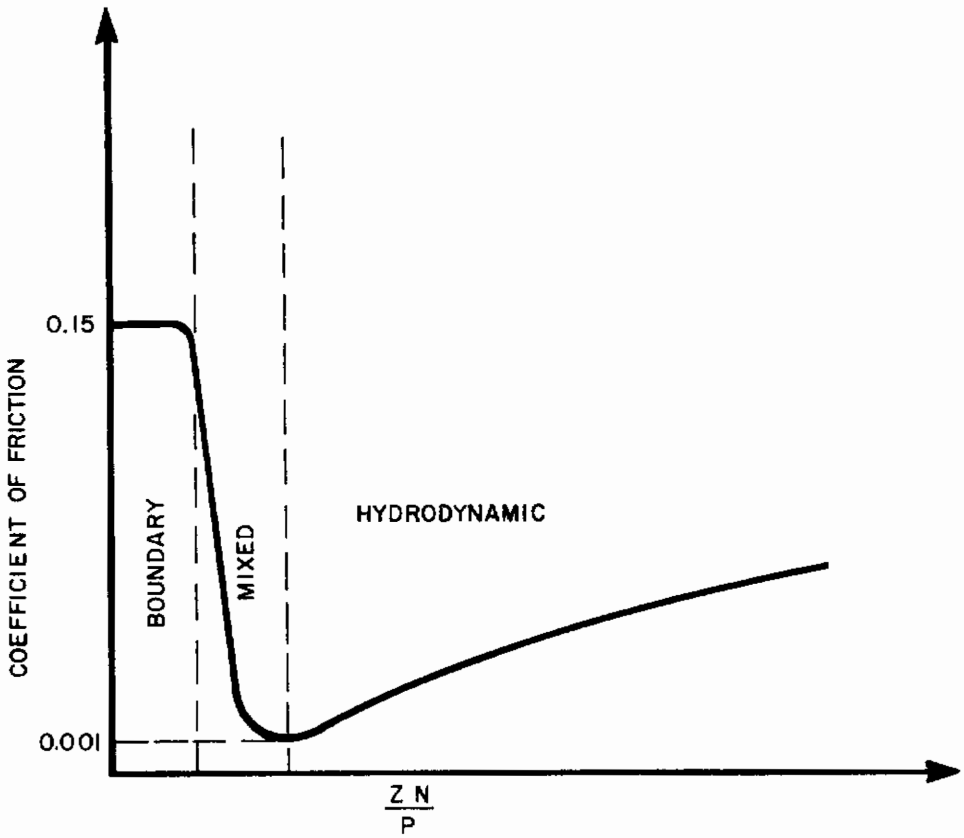
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Figure 3

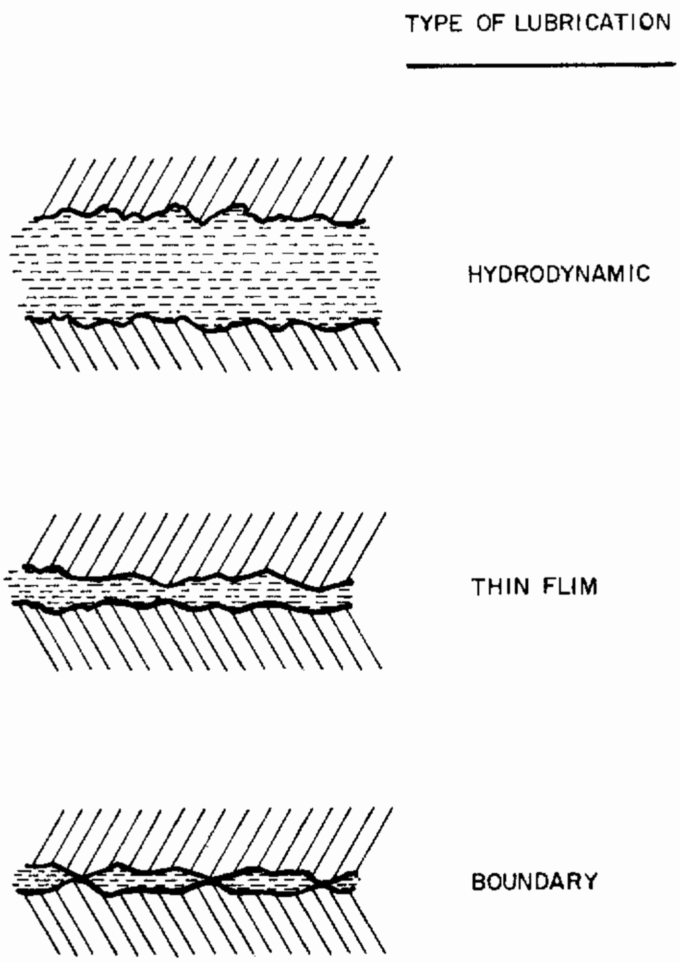

Figure 4

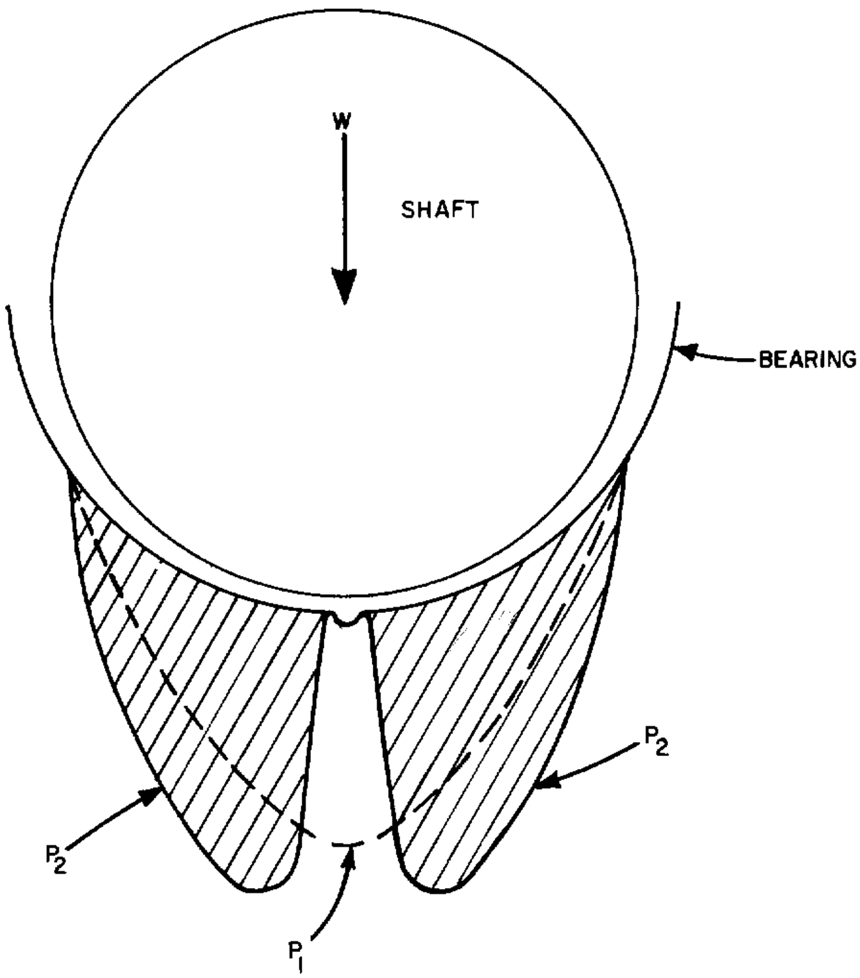

Figure 5

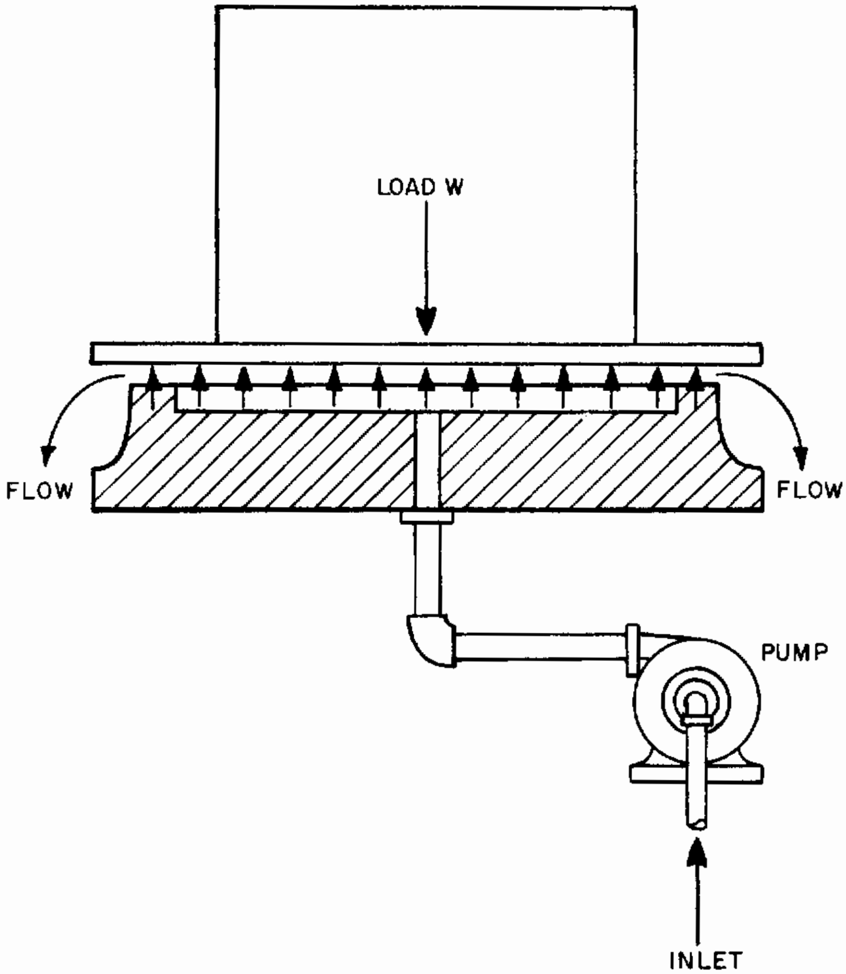

Figure 6

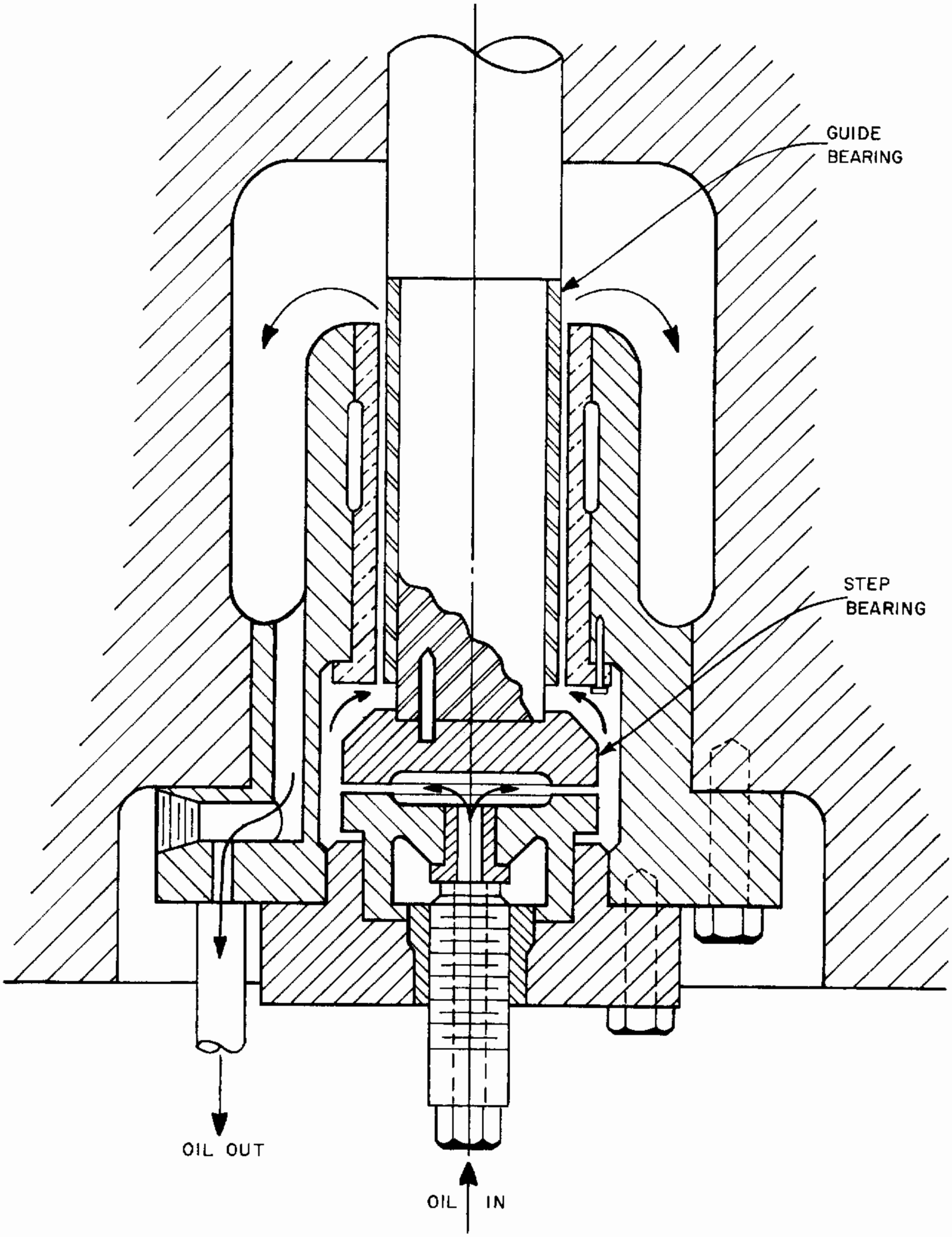

Figure 7

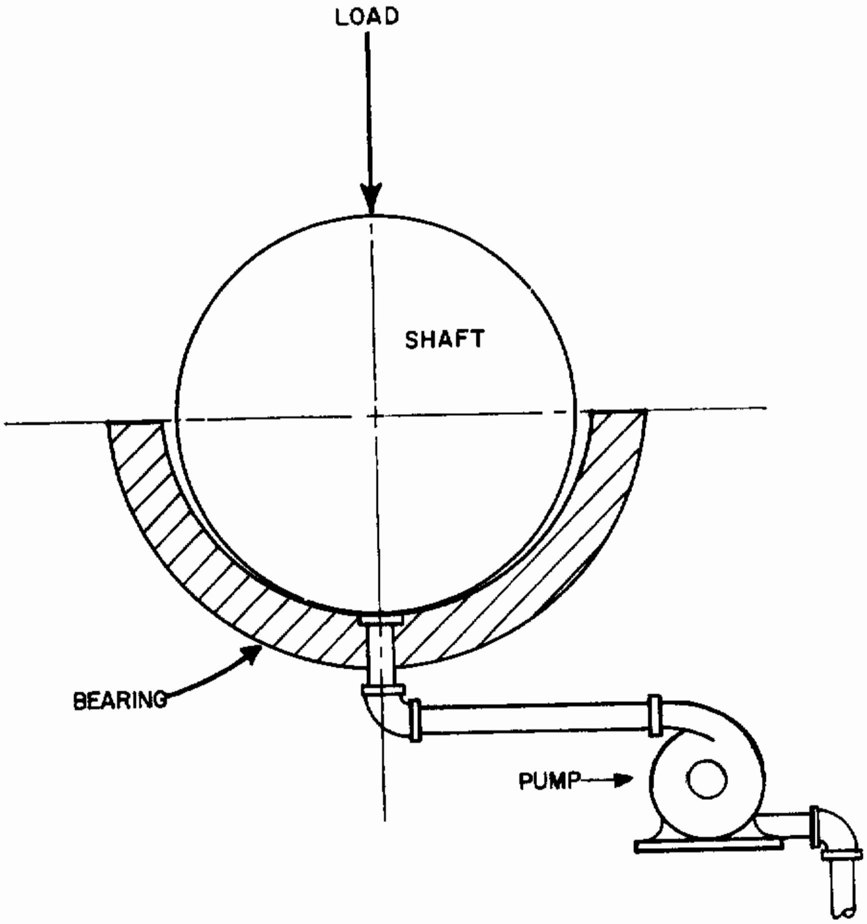

Figure 8

PRINCIPLE OF LOADING AND OPERATION FALEX LUBRICANT TESTER

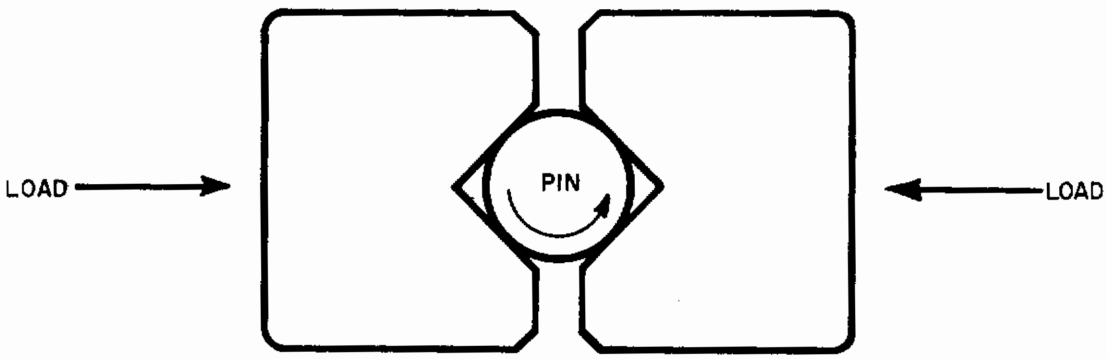

Figure 9

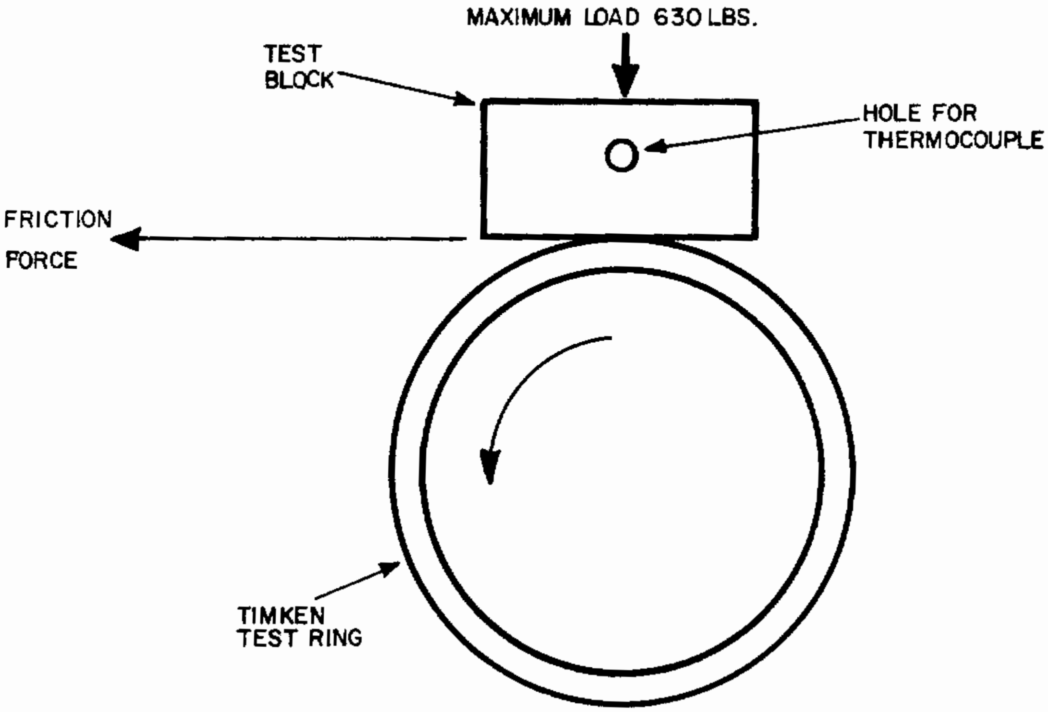
MAXIMUM LOAD 630 LBS.
TEST BLOCK
HOLE FOR THERMOCOUPLE
FRICTION FORCE
TIMKEN TEST RING

Figure 10