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Design and Fabrication of Wear Test Rig

A Project Report

Submitted by

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BONAFIDE CERTIFICATE

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ABSTRACT

In any manufacturing process the important material property to be studied and noted is wear. Wear is the erosion of material from a solid surface by the action of another solid. Wear can also be defined as a process in which interaction of surface(s) or bounding face(s) of a solid with the working environment results in the dimensional loss of solid, with or without loss of material. Most of mechanical devices fail because of the wear in the moving parts due to friction. It is also responsible for the large sums of money spent on parts, spare, repairs and downtimes.

The correct tribological design will conserve both energy and raw material. Methods by which financial savings could be made through improved tribological practice. In UK industries the percentage proportions of the total annual saving, which was estimated at £515 million (at 1965 prices) from UK department of education & science, lubrication tribology.Some of the savings like Reduction in energy consumption from lower friction (5%),Savings in lubrication cost (2%),Savings in maintenance and replacements costs (45%), Reduction in manpower (2%) ,savings in losses resulting from breakdown (22%), savings in investment through greater availability and higher efficiency (4%).Therefore it is necessary to study the wear property.

In order to study and simulate real time wear we have designed and fabricated a wear testing rig. There are various types of testing rigs out of which we have selected pin on disc system. Using the above testing rig, we have measured wear rate of stainless steel material and in order to validate, our result has been compared with the standard result.

ACKNOWLEDGEMENT

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CHAPTER 1
INTRODUCTION

1.1. INTRODUCTION

Wear is the progressive loss of material from the surface of a body due to friction. It is damage to a surface as a result of relative motion with respect to another surface. It is a process in which wear is damage and it is not limited to loss of material from surface. However, loss of material is definitely one way in which a part can experience wear. The moving parts of mechanical devices fail after sometime because they are subject to wear. It is responsible for the large sums of money spent on parts, spare, repairs and downtimes. This also results in significant amount of investment to replace the worn out parts.

Therefore modern engineering education devotes a particular interest in the study of friction and wear. Tribologists study causes and mechanisms of wear in daily applications. To investigate the individual effects of varying conditions, tribologists use simulation. The factors such as load, speed, type of material, size of specimen, temperature, and humidity among others can thus be varied and the effects of each on wear observed individually.

Wear testing rigs are devices used to simulate wear in the laboratory. A wear testing rig is a simple apparatus designed to make two or more surfaces in contact move relative to each other under controlled conditions. Essentially, wear testing rigs enable the recreation of life conditions under which wear occurs and the observation of their effects on samples of commonly used or newly designed materials and lubricants.

There are various types of wear testing rigs available like Friction – force measuring rig, Fretting wear test rig, Reciprocated movement wear test rig, Impact wear test rig etc.., but the most effectively used testing rig is Pin -on- Disc system. It is a commonly used technique for investigating sliding wear, as the name implies such can be the test piece of interest. The contact surface of the pin may be flat, spherical, or indeed, if any convenient geometry including that of actual wear components. In this pin on disc system, a motor is coupled with disc holder, so that the disc is placed inside the holder and rotated. The pin is held by a pin holder mechanism and subsequently the weight is added at the other end of lever bar mechanism in order to increase the tension in the pin.

1.2. SCOPE OF THE PROJECT

The testing rig is mainly used to record function and wear in sliding contact in dry, lubricated surfaces. The testing rig can be used at testing centers, industrial areas and in colleges for research and experimental purposes. So that wear can be effectively simulated and studied using our testing rig.

1.3. APPLICATION

Fundamental wear studies, wear mapping can be incorporated from the data acquisition system. Friction and wear testing of metals, ceramics, soft and hard coatings, plastics, polymers and composites, lubricants, cutting fluids can be easily done.

CHAPTER 2
WEAR AND FRICTION

2.1. DEFINITION OF WEAR

Wear is damage to a surface as a result of relative motion with respect to another substance. It is significant to consider what is implied and excluded by this. One key point is that wear is damage and it is not limited to loss of material from surface. However, loss of material is definitely one way in which a part can experience wear. Another way included in this definition is by movement of material without loss of mass. There is also a third mode implied, which is damage to a surface that does not result in mass loss or in dimensional changes. This might be of significance in applications where maintaining optical transparency is a prime engineering concern.

In other definitions of wear, particular in older ones, additional terms and phrases would be used in conjunction with the phrase "as a result of relative motion with respect to another substance." Examples of these would be: "relative motion of two bodies in contact" and "sliding (rolling) between two surfaces." These types of definitions are more limiting than the given one; a common point is that there is relative motion involved. A surface of a body is damaged as a result of relative motion between this surface and some other substance. However the other substance is not limited to solid bodies nor is the relative motion limited to a sliding, rolling or impact actions that normally describe relative motion between such bodies. Surfaces can wear as a result of interactions with fluids, both impinging and streaming along a surface.

While one mode of wear by fluids is associated with damaged caused by solid particles entrained in the fluid, such as in the case of slurries and dust particles in air, it is not essential that this be the case. For example, cavitations phenomena in fluids can produce wear on solid surfaces. Engineering situations where these aspects are exhibited are pipe lines pumping slurries, surfaces of aircraft, and propeller blades of ships. Again, this more general or liberal definition of wear is appropriate for practical

At least in the context of engineering applications and design, these considerations essentially indicate the wear. A brief considerations as to what it is not is of importance as well. Engineers, designers and the layman frequently use the phrase "it's worn out". Basically, this means that is no longer works the way it should or it is broken.

2.2. EXAMPLES OF WEAR

The various examples of wear are

- Change in the geometry or dimension of a part as a result of plastic deformation(e.g., from repeated hammering)
- Development of a network of cracks in a surface.
- In order to maintain optical transparency, lens and aircraft windows are examples were this is an appropriate definition of wear.

2.3. FRICTION

In situations involving sliding or rolling contact a companion term with wear is friction. Friction can be defined as a force which opposes relative motion between the two surfaces. It acts parallel to the contacting surfaces and in a direction opposite to the motion or the incipient motion. Generally the magnitude of the friction force is described in terms of a coefficient of friction, μ , which is the ratio of the friction force, F, to the normal force, N, pressing the two bodies together.

$$\mu = F/N$$

Distinction is frequently made between the friction forces which must be overcome to initiate sliding to that which must be overcome to maintain a constant relative speed. The coefficient associated with the former is usually designated the static coefficient of friction, μ_s, and the latter the dynamic or kinetic coefficient of friction, μ_K.

A frequently encountered impression is that the two terms, wear and friction, are almost synonymous in the sense that high friction equates to a high wear rate are poor wear rate or good behavior. As a generality this is an erroneous concept. While there are common elements in wear and friction phenomena, as well as inter relation ship between the two, that simple type of correlation is frequently violated. This point will became clear as the mechanism for wear and friction, as well as design relationships are presented and discussed. However, the point can be illustrated by the following observation. Teflon is noted for its ability to provide a low coefficient of friction at a sliding interface (e.g.: a dry steel of Teflon system typically has a value of $\mu \leq 0.1$)/ however the wear of the system is generally higher than cab be achieved with a lubricated harden steel pair, where $\mu = 0.2$.

Another element that can be considered in differentiation between friction and wear is energy dissipation. Friction is associated with the total energy loss in a sliding system. The principal form of that energy loss is a heat. The energy associated with a movement or damage of the material at the surface, which is rear, is normally small in comparison to the heat energy.

CHAPTER 3
TYPES OF WEAR

3.1. WEAR CLASSIFICATIONS

There are three apparent ways in may be classified. One is in terms of the appearance of the wear scar. A second way is in terms of the physical mechanism which removes the material or causes the damage. The third one is in terms of the conditions surrounding the wear situation. Examples of terms in the first category are: pitted, spalled, scratched, polished, crashed, fretted, gouged and scuffed.

Terms like adhesion, abrasion, delamination, oxidative are examples of the second type of classification. Phrases are commonly used for the third method of classification Examples of this are: lubricated wear, unlubricated wear metal-to-metal sliding wear, rolling wear, high stress sliding wear and high temperature metallic wear. All three methods of classification are useful to the engineer but in different ways.

Classification in terms of appearance aids in the comparison of one wear situation with another. In this manner it helps the engineer extrapolate experience gained in one wear situation to a newer one. It also aids in recognizing change in the wear situation, such as differences in the wear situation at different locations on a part or at different portions of the operation cycle of a device. It is reasonable that if the wear looks different, different ways of controlling it or predicting it are required: if similar in appearance, the approaches used should also be similar.

The most common form of that damage is loss or displacement of material and volume can be used as a measure of wear volume of material removed or volume of material displaced. In many studies, particularly material investigations, mass loss is frequently the measure used for wear instead of volume. This is done because of the measure for wear when wear is equated with loss or displacement of material. This is the case most frequently encountered in engineering applications. However, in engineering applications, the concern is generally with the loss of a dimension, the increase in clearance or a change in a contour, not a volume loss .

These changes and the volume loss are related to each another through the geometry of the wear scar and therefore can be correlated in a given situation. As a result, they are essentially the same measure. The important aspect to recognize is that the relationship between wear volume and a wear dimension, such as depth or width, is not necessarily a linear one.

3.2 ADHESIVE WEAR

Before adhesive wear is considered directly some general concepts regarding the nature of the contact between two surfaces must be considered. The first aspect that will be considered is the area of contact.

In many engineering considerations the contact area is generally determined by considerations of the macro-geometry or contour of the bodies in contact. This is usually done by geometrical projection or by models with take into account the deformation, elastic or plastic, that materials exhibit. For example, the Hertz contact theory is frequently used not only to determine stress levels in the contact but the size of the contact region as well. In these approaches the surface are generally assumed to be smooth.

Actual surfaces, on the other hand, always exhibit some degree of roughness and as result the actual contact situation is different from that implied by those considerations. Figure 3.1 illustrates the actual situation. What this illustrates is that physical contact takes places at localized spots within the area that is defined by the macro-geometry. These points at which the actual contact occurs are referred to as junctions. The sum of the individual contact areas of these junctions is generally called the real area of contact.

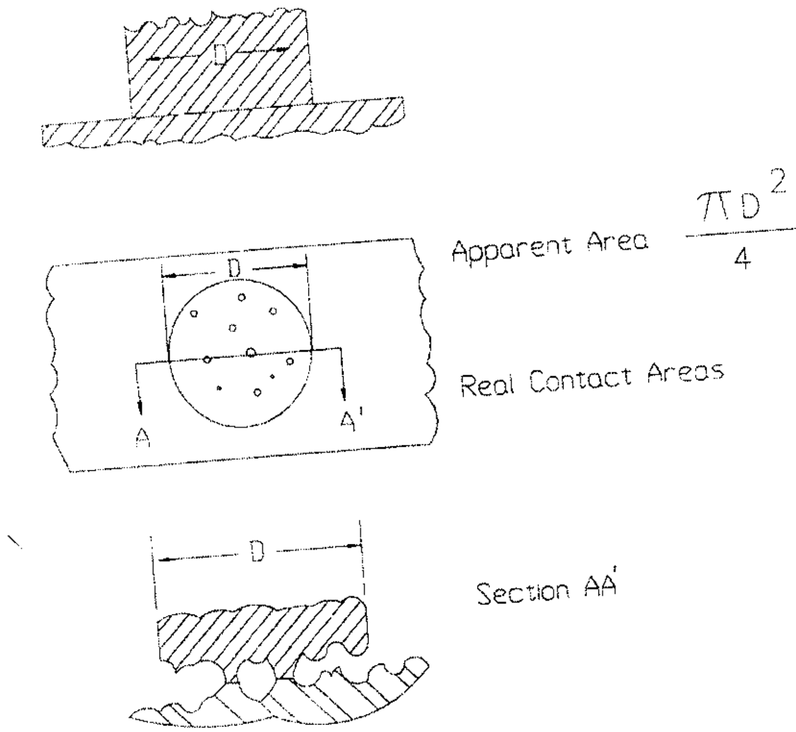


FIG 3.1 NATURE OF CONTACT BETWEEN REAL SURFACES

The area of contact that is determined through the macro-considerations is called the apparent area of contact. As will be seen, fundamental physical models regarding wear generally are developed in terms of real area considerations, while engineering formulations and models generally are related to the apparent area of contact.

The roughness characteristics of the surface have a significant influence on the number of junctions formed, as well as on the ratio of the real area of contact to the apparent area of contact. The degree to which one surface penetrates the other can also influence both these aspects. Figure 3.2 shows how the real area of contact changes, assuming one surface to be flat and smooth.

The real area in this illustration increases not only because the cross-sectional area of an asperity increase with penetration but also because the number of asperities encountered increase with penetration as well.

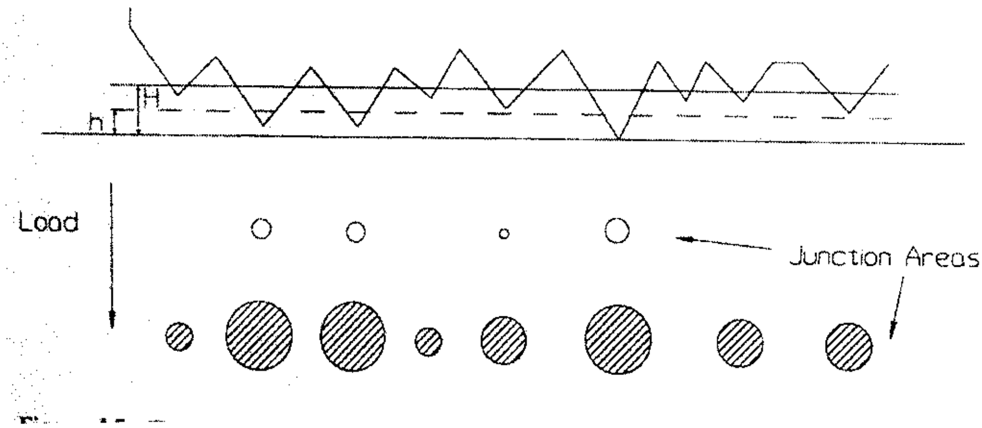


FIG 3.2 EFFECT OF INCREASED LOAD ON REAL AREA OF CONTACT

In practice, this occurs when the normal force pressing the two surfaces together is increased. The deformation properties of the materials involved and the loading conditions on the junctions also influence the real area of contact. Junction growth as a result of applied shear (i.e. friction), is a frequently observed phenomenon.

The size and number of these junctions and their relationship to the apparent area of contact have been investigated by both theoretical and experimental means. Because of the potential range of parameters involved, a wide range of contact conditional is possible: however, some generalization may be made. One is that the real area of contact is generally much less than the apparent area. The ration might be as small as 10^{-4} in practical situations.

A similar generalization can be made regarding individual junctions. It has been estimated that the diameter of typical junctions is in the range of 1 to 100 microns. The larger value would most likely occur for a very rough surface and high loads. Diameter of the order of 10 microns would be more typical for normal contact situation. For a stable contact it is often argued that there must be at least three junctions involved but generally the number is larger.

Estimates based on the yield point of materials and junction size generally indicate that the number ranges from the order of 10 to the order of 1000, with 10 to 100 being more likely.

In summary, the most significant points to be recognized about the contact between two bodies is that actual contact occurs at individual sites within an apparent area of contact and that the real area is generally only a fraction of the apparent area. The features observed in most micrographs of wear scars produced under sliding conditions support this view of the contact between two surfaces, as well as the generalizations regarding the ratio of real and apparent areas, junction size and number.

It is also important to understand the nature of the interaction that occurs at these junctions on both an asperity and as atomic level. At the asperity level the focus is on the type of deformation that occurs at these junctions. The deformation at the junction can be plastic as well as elastic. Just how much of each is involved depends on the number of junctions, their size, and the total load, as well as the properties of the materials involved.

While it is nit impossible to have only elastic deformation on all the junctions, it is not likely. Models based on typical surface profiles indicate that some plastic deformation general occurs at some of the junctions. This tends to be confirmed by the topography found on wear surfaces. Some evidence of local plastic deformation can usually be found on these. To understand the interactions on a atomic level, it is best to consider the nature of interatomic forces first.

The behavior of the force between two atoms is illustrated in Figure 3.3. For large separations between the atoms there is a weak attractive force. At separations comparable to interatomic spacing the attractive force increases rapidly.

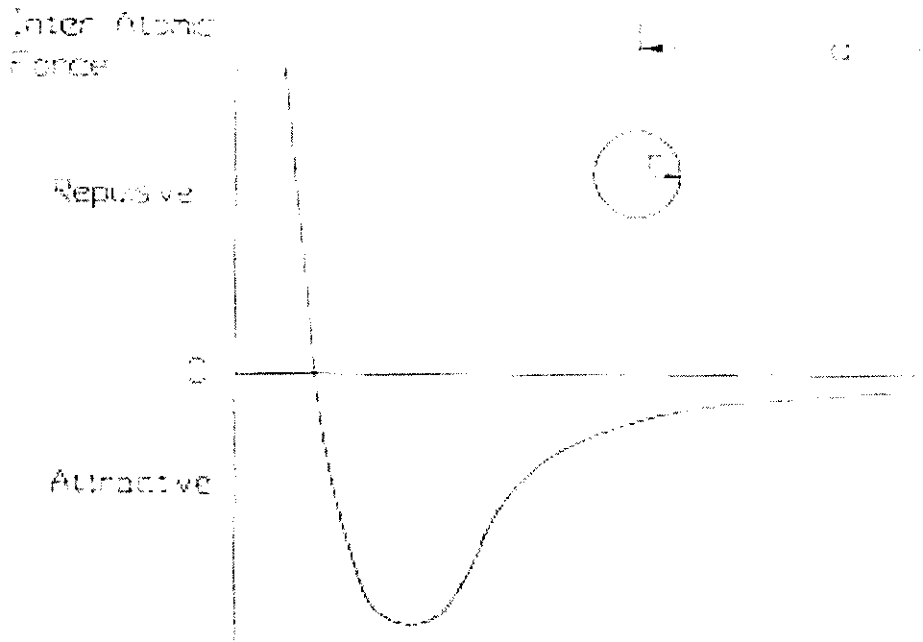


FIG 3.3 NATURE OF FORCE BETWEEN ATOMS

With still smaller separations, the attractive force begins to decrease and ultimately the force change to a repulsive one.

Arrays of atoms also exhibit the same general behavior, which is a show in Figure A7 for the case of an A1 crystal and a Zn crystal. In this case, the figure shows the variation in the potential energy of such a contact as a function of the separation of the two crystals. This is the more common way of describing the interactions and is equivalent to the force representation. A negative potential energy indicates bonding. The force is represented by the curve. As a result a negative slope indicates a repulsive force and a positive slope indicates an attractive force.

Since junctions form as a result of two surfaces begin pressed together, the nature of interatomic forces indicates that bonding occurs at these junctions and that over some portion of the real area of contact the atoms of the two surfaces must have gone past the point of maximum bonding. This is the only way the forces can be balanced. This implied that some "adhesive" forces or bonds must be overcome to separate the two surfaces at these sites. This atomic view of the contact situation at the junctions

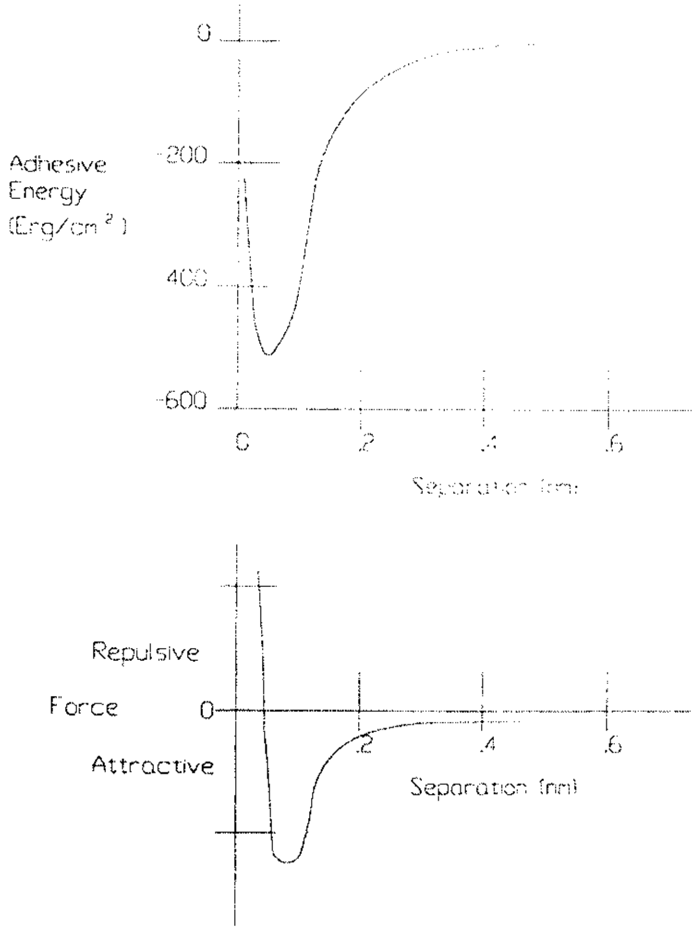
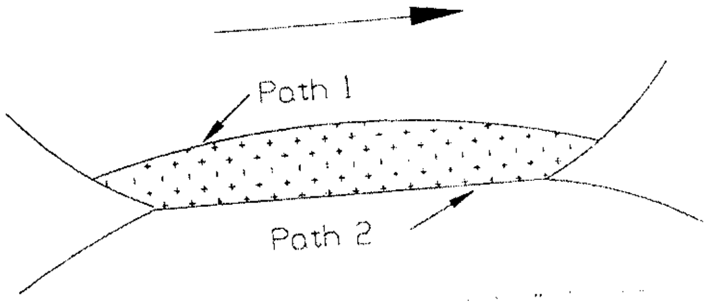


FIG 3.4 VARIATION IN ADHESIVE ENERGY

Consider the diagram shown in Figure 3.5. This depicts the situation at a junction at which bonding has occurred. As the two surfaces move relative to one another, rupture of the junction will eventually occur. If the rupture occurs along Path2, which is the original interface, no material will be lost from either surface, through some plastic deformation may have occurred. If on the other hand, the rupture occurs along some other path, illustrated by Path1 in the figure, the upper surface would have lost material. The removal of material from a surface in this manner is termed adhesive wear.

This series of micrographs follow the events associated with a single asperity on the counter face. In the lower right of the figure the result of asperity engagement and at the original interface, leaving only a plastically deformed groove in the wake of its motion.



FIGS 3.5 RUPTURE PATHS AT A JUNCTION

Some deformation of the asperity is likely as well during this period. At some point, failure no longer occurs at the original interface but at some depth within the asperity, leaving a portion of the asperity adhering to the flat surface. This is the event indicated in the middle of the figure by the adhered wear fragment. As sliding continued, the same series of events repeated (upper left in the figure) but with the asperity now modified both by plastic deformation and adhesive wear.

3.3 ABRASIVE WEAR

The contact situation that is generally considered for abrasive wear is different than that considered for adhesive wear. For adhesive wear, the view is the real area of contact is composed of junctions formed by the engagement of asperities on the two surfaces in contact. Three general situations for abrasive wear are identified and these abrasive wear situations are illustrated in Figure 3.6. One situation is when hard asperities of one surface are pressed in to a softer surface.

This abrasive wear situation is generally referred to as two-body abrasive wear, Filing, sanding, and grinding would be examples of two-body abrasion; a rough metal

FIG.3.6. ABRASIVE WEAR SITUATIONS

The second contact situation is one in which hard, loose particles are trapped between the two surface and the force between the two surface are transmitted thought these particles. This abrasive wear situation is referred to a three-body abrasion. Examples of this would be some polishing situations where a lap is used; sand trapped in a bearing, and hard wear debris trapped between two sliding surfaces.

The third contact situation is when hard particles directly impinge on a surface. In this case the particles are general entrained in a fluid, such as in slurry. Generally, this abrasive wear situation is referred to as erosion. The wear caused in pipe lines handling abrasive slurries would be an example; another would be the wearing action caused by sand and grit in air streams. Abrasive wear mechanisms are generally considered to be any mechanism by which the hard asperities or particles cause damage in a single action.

The damage, or wear, that they produce is of two general types, deformation or particle formation (material removal). The particular mode that occurs is a function of several parameters associated not only with abrading particles or asperities but with the abraded material as well. Figure 3.7 shows the model for abrasive wear.

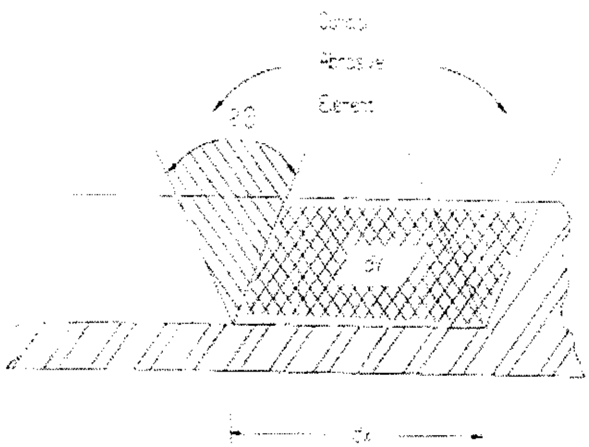


FIG 3.7 MODEL FOR ABRASIVE WEAR

Examples of the morphology that are associated with deformation and cutting abrasion are shown, as well as examples from sliding and particle impingement wear. There are considerable differences in the appearance of these wearing scars but all fall under the general heading of abrasion.

Many authors tend to use more specific terms to describe abrasive wear, instead of these more general terms. Frequently these more specific terms relate to specific conditions surrounding the wear but they often used to differentiate between specific abrasive wear mechanisms as well. For example, ploughing wear is used to connote the deformation mode. Cutting or scratching wear is used to describe specific material removal by these actions. Brittle fracture would be associated with particle generation by fracture. While these refinements are frequently significant in the treatment and understanding of abrasive wear, the general equations relating to abrasion can be developed from some very general concepts.

3.4 CORROSIVE WEAR

This one is not quite as neat as adhesive and abrasive wear. That's because any products produced by the corrosion of a surface are usually carried away when the two parts come together which breaks them loose and they are then swept off by the exhaust gas or air stream. If pits are formed on the worn surface it can very well be from corrosion unless they are simply indentations. Logically, we know corrosive wear must take place even though it can be difficult to prove. We can see the formation of surface compounds from the co rodents in non-worn areas and we know some of the properties of these compounds are high hardness and brittleness.

3.5 FATIGUE WEAR

The basic concept of fatigue wear is that with repeated sliding, rolling, or impacting, material in the vicinity of the surface experiences cyclic stress. As a result of this stress cycling, cracks (or damage) are initiated in these regions. With further cycling the cracks propagate, eventually intersecting with the surface and themselves. This crack network then produces free particles which are easily removed from the surface by a subsequent motion, thereby resulting in wear. This wear surface also experience stress cycling is illustrated in Figure 3.8.

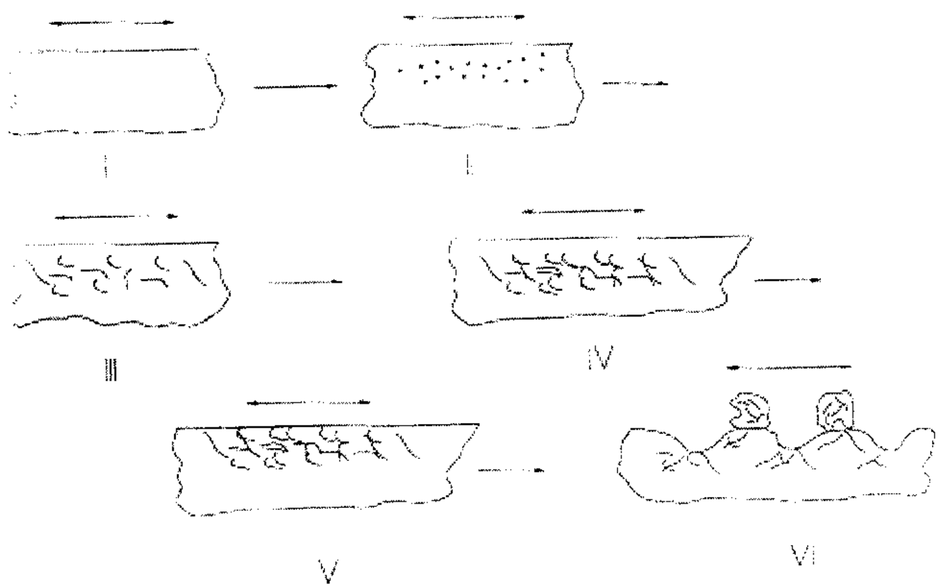


FIG 3.8 MODEL FOR FATIGUE WEAR

This wear mechanism is most evident in rolling and impact wear situations, where it is generally recognized as the principal mechanism. In the case of rolling and to a lesser degree in impact, the topological features of the wear scar are often quite suggestive of crack initiation and propagation. Under sliding conditions the topological feature are generally not as suggestive. Feature associated with adhesive mechanisms frequently confound the image. In addition, the crack network under rolling and impact tend to be more macroscopic (or coarse) than those encountered under sliding conditions and frequently result in larger particles being formed. Because of these aspects, often the only way to determine the existence of cracks under sliding conditions is by means of microscopic examination of cross-sections through the worn surface, such as those shown in Figure 3.9

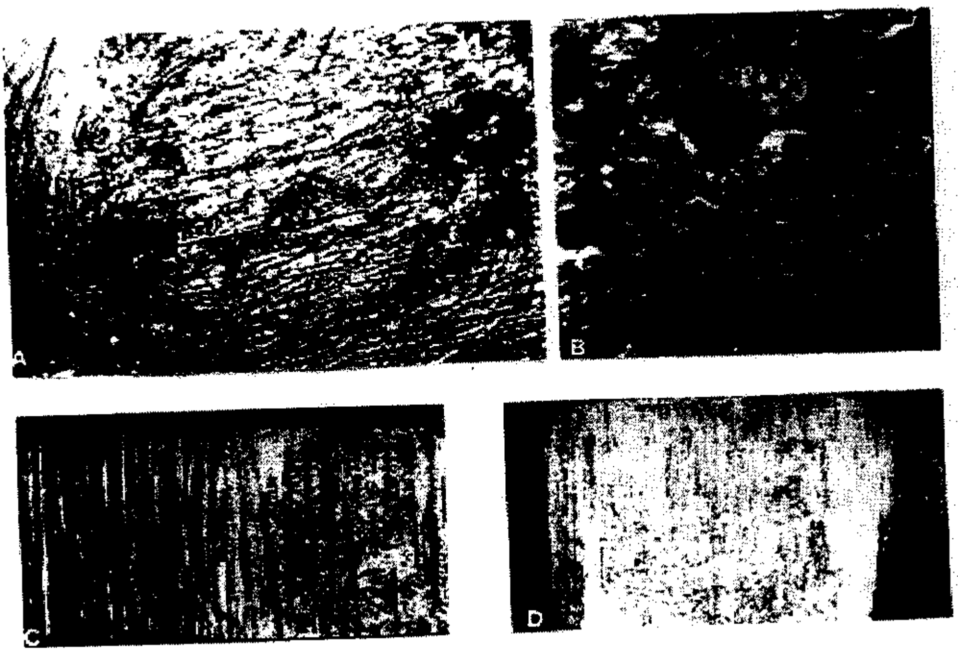
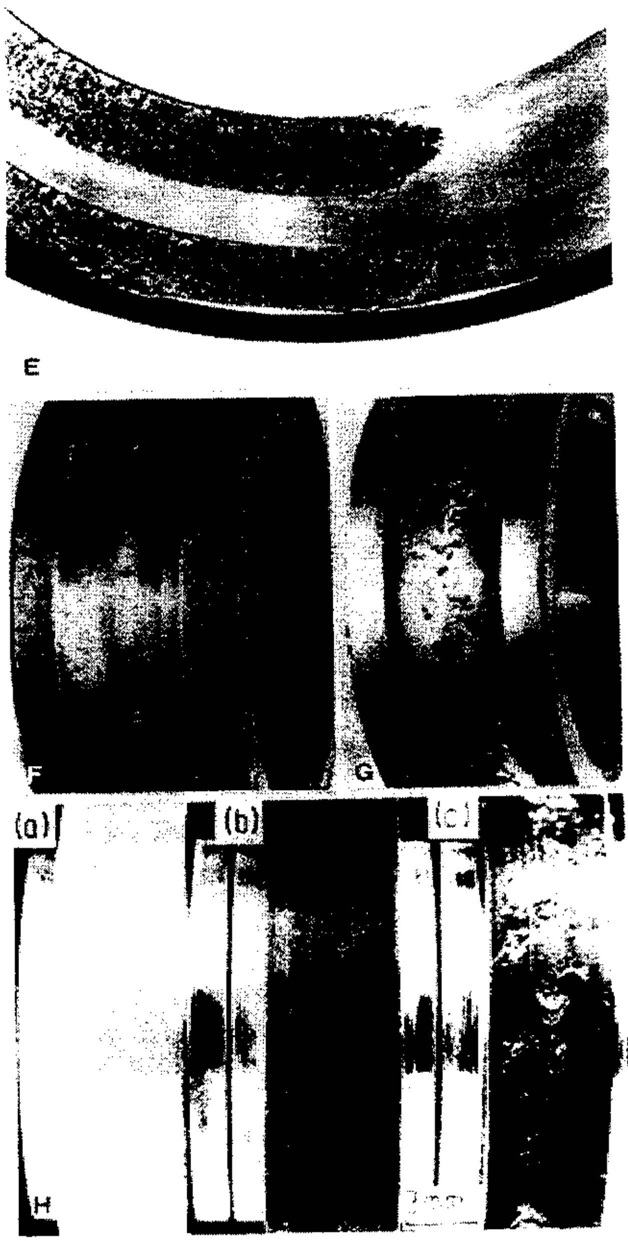


FIG 3.9 SURFACE FATIGUE WEAR IN METAL

A common feature of fatigue wear and normal fatigue is the existence of what is frequently called an incubation period. During this initial period, cracks are formed and propagate to the surface. Some topological changes might be evident during this period, including some evidence of plastic deformation. However, there is no loss of material from the surface or formation of free particles. In the case of conventional fatigue, this would be equivalent to the fracture of the part and the number of cycle to that point would be the fatigue life.

In the case of fatigue wear, the process would occur over and over again, resulting in a deeper and deeper wear scar. In addition to this is a further difference that should be noted. For conventional fatigue, most materials exhibit an endurance limit, that is, a stress level below which fracture will not occur. In the case of fatigue wear, there does not appear to be such a limit at least in terms of microscopic loads and stresses. For practical load conditions, no matter how small the load or stress, sufficient rolling, sliding or impact will result in the generation of fatigue wear. For rolling situation there is a generally accepted empirical relationship between load and the number of revolution defining the incubation period.



The progression of wear scar morphology for fatigue wear under sliding conditions was studied in Cu.The sliding system consisted of a hardened steel sphere sliding back and forth across the flat surface of Cu single crystals. Boundary lubrication was used and stress levels were maintained well under the yield point of the Cu. Three stages were identified and are shows in Figure 3.10. In the first stage grooves and striations in the direction of sliding were the predominated feature. There was no material loss and the topography would suggest a deformation mode of abrasive wear.

During this stage as sliding increase the density or number of these grooves increased. In the second stage, damage features perpendicular to the sliding direction appeared. Again, there was no loss of material. This feature, termed cross- hatching, suggests something other than a simple deformation mode of abrasive wear.

As sliding continues in this stage, the cross-hatching became more pronounced until ultimately spalling and flaking occurs. This is the start of the third and final stage. In this stage material loss occurs and, with continued sliding a wear groove of increasing depth is formed. The start of the third stage was considered to be the end of the incubation period.

The striations of the first stage are probably the result of local stress system associated with individual asperity contact, as was considered in abrasive wear. However, the cross-hatching feature occurs over many striations and is therefore probably associated with the overall stress system associated with the macro-geometry of the contact.

At the same time, this feature is also considered to be associated with the initiation and growth of subsurface cracks. Micrographs of cross- sections through the wear scar confirmed the existence of cracks in this situation.

3.6 OXIDATIVE WEAR

For dry sliding under light loads metallic wear scars tend to exhibit a smooth, glassy-like appearance, such as shown in Figure 3.11.Under these conditions the wear rate is generally low and fine wear particles of metallic oxides are observed. The glassy-like appearance in these cases has been shown to be associated with the formation of an oxide layer on the surface of the wear scar. An oxidative wear mechanism has been proposed that explains the wear under such circumstances.

FIG 3.11 SLIDING WEAR SURFACE

The basic concept for this mechanism is that wear occurs by the removal of the oxide layer as a result of sliding contact at the asperities. However, in between contacts, the oxide re-grows on these denuded areas of the surface and is again removed with subsequent asperity engagement.

A simple model can be used to describe the basic elements of this mechanism. The implicit assumption of the model is that the weakest point is at the interface between the metal and the oxide and that as the result of sliding engagement the oxide layer flakes off at the interface, much like a coating or plating with poor adhesion. The overall sequence is shown in Figure 3.12.

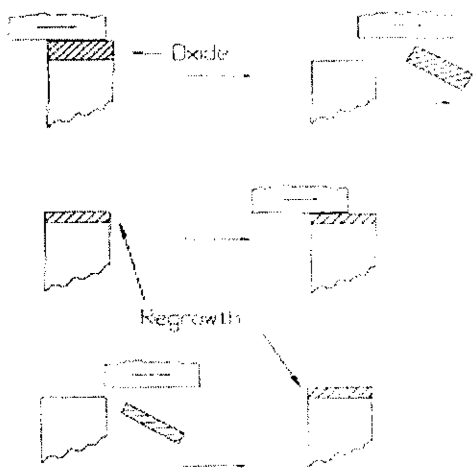


FIG 3.12 MODEL FOR OXIDATIVE WEAR

It is assumed that the real area of contact can be represented as a uniform array of circular junctions. The wear rate associated with a junction w_i, is given by

$$w_i = \frac{\pi a^2 d}{2a}$$

$$w_i = \pi a d$$

Where 2 a is the diameter of the circular junction and d is the thickness of the oxide film. The wear rate of the surface would then be

$$w = \frac{J I n a d}{2}$$

Where n is the number of junctions.

CHAPTER 4

WEAR PROBLEMS

4.1. PROBLEMS

The problems that are caused due to wear are:

- It causes damage to a surface as a result of relative motion.
- The moving parts of mechanical devices fail after sometime because they are subject to wear.
- It is also responsible for the large sums of money spent on parts, spare, repairs and downtimes.
- A significant amount of investment is needed to replace the worn out parts.
- It also results in initiation and propagation of cracks near the surface.

4.2. CORROSION AND FRACTURE

Corrosion is not considered to be wear because there is not the element of relative motion. Fracture, in the same sense referred to above, is not considered wear because it is more a body phenomenon rather than a surface phenomenon.

While corrosion and fracture are not included in the definition of wear, corrosion and fracture phenomenon are definitely elements in wear. This is because in a wearing situation. There can be corrosive (chemical) and fracture elements contributing to the damage that results from the relative motion. These interactions will become clearer for example a wear situation referred to earlier regarding, wear behavior associated with the pumping of slurries; this frequently involves corrosion as well as mechanical factors. Also, there are many situations in which a surface wears as a result of the initiation and propagation of cracks near the surface. Another aspect is that a part by propagation of a crack formed in the wearing process. The important point is to recognize in these considerations is that all failure devices or life- limiting aspects not the result of wear and wear process. To be considered wear, there generally has to be some surface, mechanical, and relative motion aspects involved. Wear mechanisms also involve a very large number of physical and chemical phenomena.

CHAPTER 5

WEAR MEASUREMENTS

5.1. WEAR MEASURES

The most common form of that damage is loss or displacement of material and volume can be used as a a measure of wear volume of material removed or volume of material displaced. For scientific purposes this is frequently the measure used to quantify wear. In many studies, particularly material investigations, mass loss is frequently the measure used for wear instead of volume. This is done because of the relative ease of performing a weight loss measurement. However, there are three problems in using mass as the primary or direct measure of wear. One is that direct comparison of materials can only be done if their densities are the same. For bulk materials this is not a major obstacle, since the density is either known or easily determined. In the case of coatings however, this can be a major problem, since their densities may not be known or as easily determined.

The other two problems are more intrinsic ones. A mass measurement does not be known or as easily determined. The other two problems are more intrinsic ones. A mass measurement does not measure displaced material. In addition it is sensitive to wear debris and transferred material that becomes attached to the surface and can not be removed. This material does not necessarily have to be from the same surface; it can be from the counter face as well. It is not an uncommon experience in wear tests, utilizing mass or weight loss technique, to have the specimen grow (i.e., indicate a mass increase as a result of transfer or debris accumulation).

From the above it can be seen that volume is the fundamental measure for wear when wear is equated with loss or displacement of material. This is the case most frequently encountered in engineering applications. However, in engineering applications, the concern is generally with the loss of a dimension, the increase in clearance or a change in a contour, not a volume loss per se. These changes and the therefore can be correlated in a given situation. As a result, they are essentially the same measure. The important aspect to recognize is that the relationship between wear volume and a wear dimension, such as depth or width, is not necessarily a linear one. This is an important aspect to keep in mid when dealing with engineering situations, since many models for wear mechanisms are formulated in term of volume.

Consider the situation where there is some wear experience with a pair of materials in a similar situation to the one currently under study. In the prior study it might have been concluded that wear is proportional to the load. In the current situation the wear is too large and there is the possibility to reduce the load by a factor of 2.The previous experiences suggests that this would result in decreasing the wear by a factor of 2 as well; however, when tried, only a 25% reduction in wear is found. The subtle difference that could explain the result is that the primary relationship between wear and load, in this particular case, is in terms of volume.

In the first situation, the part wearing could have had a uniform cross-section and, as a result, the wear volume, would have been proportional to the depth of wear, which was the measure used. Hence the result would imply that wear volume is proportional to load as the fundamental relationship. In the seconds case, the geometry of the wearing part was such that the volume of wear was proportional to $h^{1/3}$, where h is the depth of the wear and was again the engineering measure for wear. In this situation the proper relationship to be inferred between loads and wear depth is a cubic one.

Therefore, for one half the loads, only a 25% reduction in wear should be expected. This is not a very profound point but is one that is frequently overlooked or not initially recognized in design work.

5.2 TESTING METHODS

Many different experimental arrangements have been used to study sliding wear. Laboratory investigations of wear are usually carried out either to examine the mechanisms by which wear occurs or to simulate practical applications and provide useful design data on wear rates and coefficients of friction. For both purposes, control and measurement of all the variables, which may influence wear, are very important. It is vital to appreciate that wear rate and friction are often critically dependent on the sliding conditions; apparently minor changes in conditions can lead to radical changes in the dominant mechanism and associated rate of wear. Close control and monitoring are essential if the results of a test are to be useful either as a simulation of practical application, or for wider scientific purposes."Figure4.1" shows the geometrical arrangements employed in several common types of wear testing apparatus. The word tribometer, first used in 1774 for an instrument intended to measure friction, is sometimes used for such apparatus: more recently the inelegant term tribotester and its associated verb have been coined.

The methods shown in "Figure. 5.1" may be divided into two types those where the sliding surfaces are symmetrically disposed, in which the wear rates or two surfaces of identical materials should be the same, and the more common arrangement where the system is inherently asymmetric, in which the two sliding bodies, even of the same material, will almost certainly experience different rates of wear for reasons discussed below. Symmetrical arrangements are not often used to study wear: examples are the ring-on-ring (or two discs) devices, with contact either along a line or face to face. Such devices are only truly symmetrical if both components are rotated.

The most common asymmetric test rigs employ a pin pressed against a disc, either on the flat face or on the rim, a block loaded against a ring or a pin on a flat. In these cases the contact may initially be over an extended nominal contact area (e.g. with a flat-ended pin on a flat disc, or a conforming block-on-ring), or only at a point or line (e.g. a round-ended pin on a disc, or a plane block-on-ring). In asymmetric usually treated as the specimen, and is the component for which the war rate is measured, while the other; often the disc, flat or ring is called the counter face.

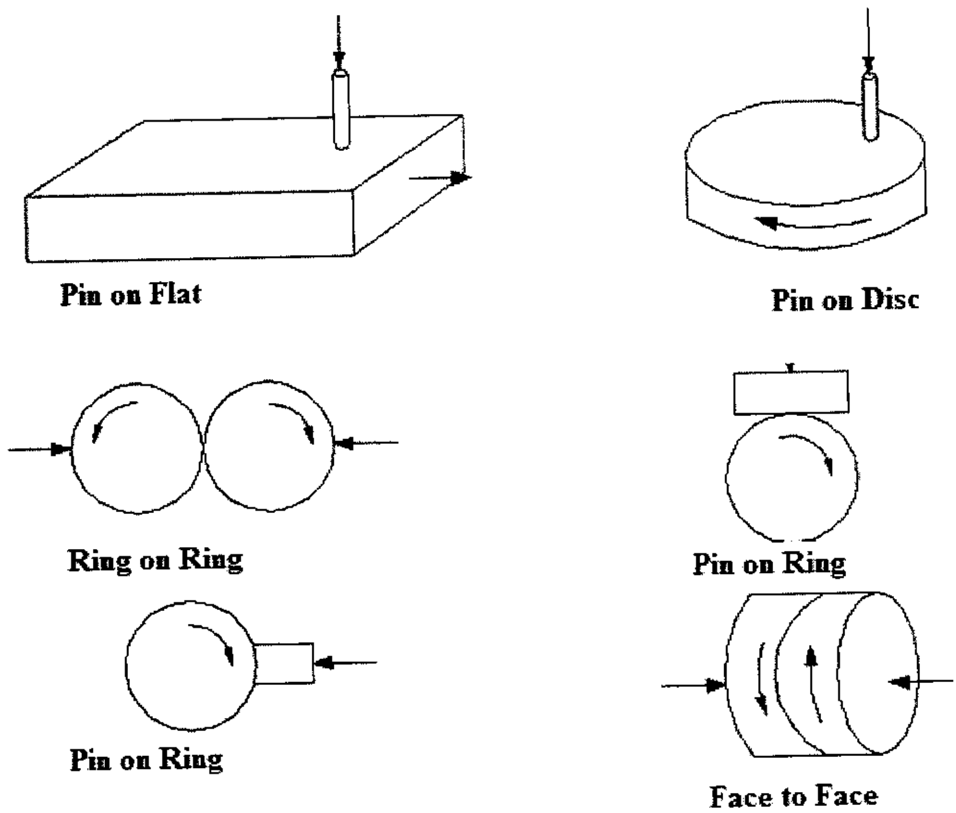


FIGURE 5.1 GEOMETRIES EMPLOYED IN SLIDING WEAR TEST.

5.3 BLOCK –ON- RING WEAR TEST

The basic configuration of this is shown in Figure5.2. It is one of the more used tests configurations to study sliding wear and to rank materials in terms of resistance to sliding wear. While both the block and ring can wear in this test, the test is primarily used to evaluate the wear of the block material. This same test configuration has been used to evaluate lubricants. The method of conducting the test, the data obtained, and

However, many of the aspects associated with control are the same. The test itself can be conducted under a variety of conditions of load, speed, lubrication, and even environments. When this test is used to rank material, the ring material is fixed and the block material is varied, but it must be recognized that the wear of the block, which tends to experience the most pronounced wear in the test, can be influenced by the material of the ring. As a consequence, when used to rank individual materials for an intended application, the ring material should be one of the materials used in the application.

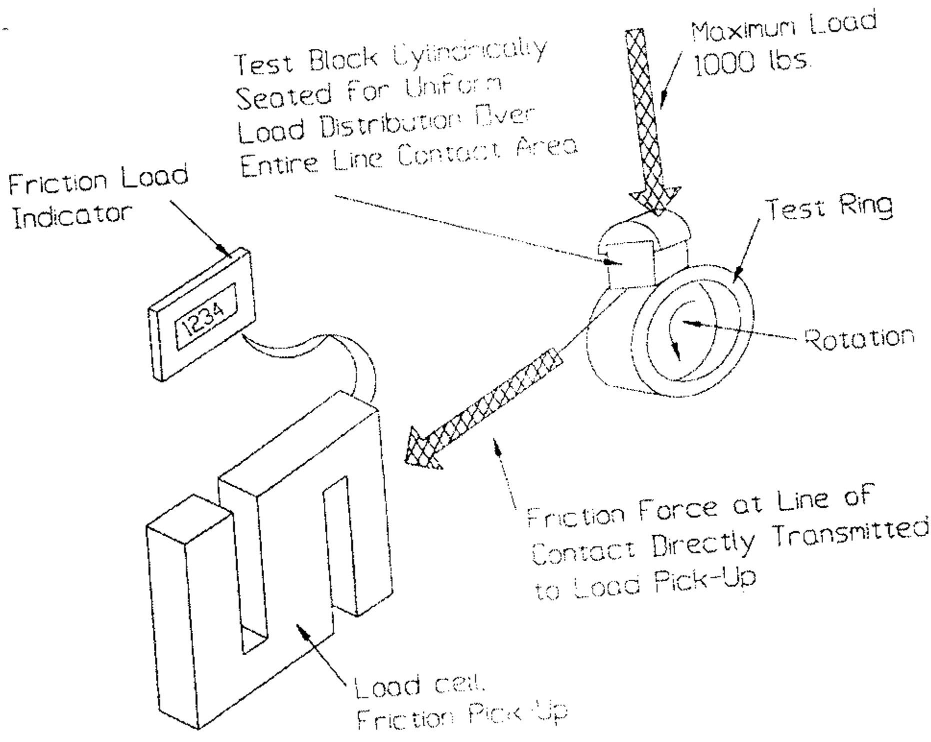


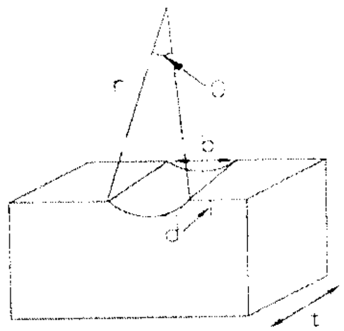
FIG 5.2 BLOCK ON RING WEAR TEST

If not, the correlation between test rankings and field performance is likely to be poor. Also, in relating wear behavior in the test to wear behavior in an application, it is necessary to consider the wear on the block and the ring, not just the wera on the block. When this is done and the test conditions have provided good simulation of an application, material rankings obtained with this test have been found to correlate

An ASTM standard for wear testing using this configuration has been developed. (ASTM G77)It provides guidelines for conducting the test and analyzing and reporting data and a recommended test procedure for the evaluation of plastics. Round-robin test programs using the procedures of ASTM G77 have indicated that the intralaboratory coefficient of variations for the block wear volumes are typically 20% for metals and 40% fro plastics. The interlaboratory variations are large, 30% and 60%, respectively. The coefficients for ring volume tend to be significantly higher than those obtained for the block (e.g., 2 times higher). The large variation associated with this is partially the result of measurement accuracy and partially the result of the sensitivity of this type of wear to a large number of parameters.

The coefficient of variation for the width of the wear scar on the block, which is directly measured in the test and used to compute the volume, is significantly less(e.g., they are in the range of 5-20%). However, for the geometries of the test, wear volume is related to the square of the width, which result in large coefficients for this measure. Fro the ring, wear volume is determined by measure a small change in a large mass. Because of the large variation associated with wear volumes in this test it is generally recommended that several replicates (e.g., three or four tests) be done when using it is to rank material pairs.

The basic test method is to press the block against the rotating ring and the wear on both the block and ring is measure after a specified number of revolutions. On the block a cylindrical groove is generated as a result of the wear. The volume of the wear is determined by first measuring the width of the groove and to use this to calculate the volume.



$$D = 2r$$

Scar Width $= b = D \sin \frac{\theta}{2}$

Scar Volume $= \frac{D^2 t}{8} (\theta - \sin \theta)$

where $\theta = 2 \sin^{-1} \frac{b}{D}$

Scar Volume $= \frac{D^2 t}{8} \left[2 \sin^{-1} \frac{b}{D} - \sin\left(2 \sin^{-1} \frac{b}{D}\right)\right]$

FIG 5.3 DETERMINATION OF BLOCK RING VOLUME

The geometrical relationship and the equation is shown in Figure 5.3. The volume of wear for the ring is determined by mass loss and converted to volume loss by means of the density of the ring. The standard test method specifies that either of two number of revolutions be used. The number of revolutions used is to be reported with the wear volume measurement which in turn is used to rank materials in terms of wear resistance.

5.4 CROSSED- CYLINDER WEAR TEST

This is a test that has been used to rank material pairs in terms of their resistance to sliding wear. It has been used for a number of years in industry, principally to evaluate tool steels and hard- surfacing alloys. However, procedure and parameters used by the different laboratories tended to vary although recently a standard practice has been developed and issued as ASTM G83. The basic configuration of the test is shown in Figure 5.4. One cylinder is held stationary and the other is pressed against it

The basic concept is to rank materials in terms of wear produced after a fixed number of revolutions. Wear is directly measured by mass loss techniques but converted to volume loss for comparison. Studies which used the standard practice have show that the coefficient of variation for intralaboratory test is within 15%, and for interlaboratory test, 30%.

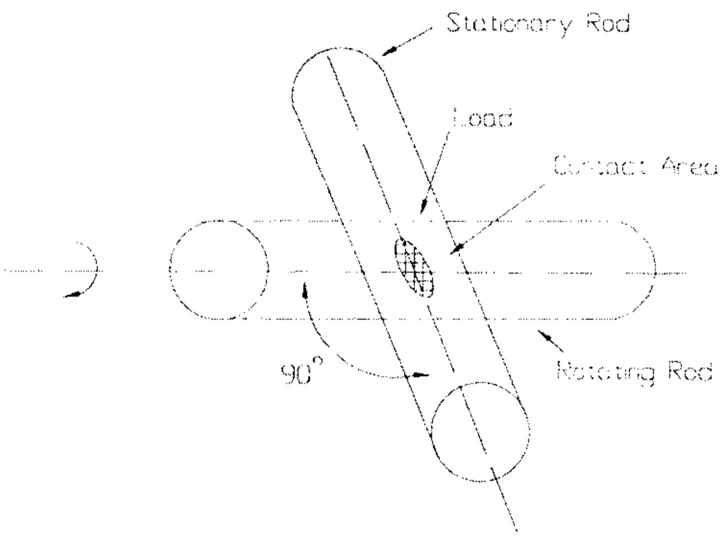


FIG 5.4 CROSSED – CYLINDER TEST

Unlike the block-on-ring procedure, which allows considerable flexibility in terms of test parameter and materials evaluated, the standard test method for the crossed-cylinder test is specific in terms of test parameters and limited in terms of the materials which it can be used. The test is designed for unlubricates evaluations of metals but other materials are allowed if they are sufficiently strong and stiff so that the specimens do not deform, fracture, or significantly bend under the load conditions specified. In generally this would exclude polymers. The standard test method has three procedures, which differ in terms of speed and duration to address different levels of wear behavior. Procedure A is the most severe test and is recommended for the most wear resistant materials. Procedure B is a shorter version of A that can be for less wear resistant materials that exhibit sufficient wear in the shorter period of time.

Procedure C is a milder test (i.e., lower speed), that is run for a fewer number of revolutions. This was developed for the evaluation of materials that exhibited such severe wear under the conditions of A or B that valid or useful comparisons could not be made.

This could be because of excessive heating under the more severe conditions, extensive galling or adhesion, which would influence the accuracy of the measurement technique, or complete wear- through of surface treatment layers. The selection of which procedure to use therefore depends considerably on the nature of the materials and associated wear behavior. It is possible that more than one procedure might be acceptable; however, the ranking and comparison of materials should be confined to within one test procedure. Inferring the relative behavior of materials in one test procedure, based on relative behavior in another, should not be done for several reasons.

For example, using result from Procedure C with results from either of the other procedures to establish a ranking should not be done, since the test conditions are different and their effect on material behavior is generally not known. Also, since the nature of the wear curves in this type of test tends to be of a variable, nonlinear nature, cross- comparison between procedures A and B may not be valid, even though the test conditions are the same

This test method doest not provide much flexibility in terms of providing simulation since there is only a high and low speed version of the test. Basically the test simulates high speed, dry sliding wear. The used of the test has to decide first of all whether or not the application can be described in those same general terms. If the answer is yes, the test provides first-order simulation. Second-order simulation would depend on the similarity of the standard test parameters and those of the application and the sensitivity of the materials to the differences in those parameters.

Because of the unique geometry of the test, which is complicated by the wear of both cylinder, it is generally not possible to make an a priori judgment regarding second-order simulation. Therefore comparison of wear scar morphology from the test with that from the application is one way of deciding on the degree of simulation and the likelihood of good correlation between he test and the application. Similarity in the appearances generally implies that there should be correlation, even though there might be difference in specific values of the parameters

The primary intention of the test is to characterize the wear resistance of self-mated pairs. In this case the total volume of wear (i.e., the sum of the wear volumes obtained from the stationary and rotating member) is used as the measure of wear resistance. The test method allows testing with dissimilar metals as well; in this case the wear volume for both specimens should be reported separately.

5.5. ROLLING WEAR TEST

A configuration that has been successfully used for sometime to address rolling wear is illustrated in Figure 5.5. Basically, it consists of a pair of driven rollers presses against one another. The typical procedure is to visually monitor the condition of the rollers surface and determine the number of cycles of a selected level of surface damage to occur. This could be the appearance of cracks, surface texture change, or spalls. Figure 5.6illustrates such conditions. These test are usually quite long, extending for days or weeks and inspections are done on a periodic or scheduled basis. This is another example of the use of appearance criteria in a wear test. The longer the number of cycles, the more resistance the pair is to rolling wear.

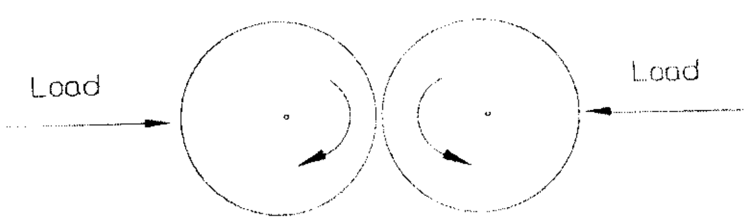


FIG 5.5 ROLLING WEAR TESTS

Critical element of this test is control of the surface velocities of the two rollers, alignment of the rollers, and geometrical tolerance of the rollers. With respect to the last, particular attention has to be paid to the edge conditions of the rollers so that a significant stress concentration condition does not occur. This means that the edges of rollers should be well rounded. Use of rollers of the same length can help to minimize this exposure, as well. Another approach that has been used is to use slightly curved rollers.

This type of test has been used to address conditions of pure rolling, in which case the surface velocities of the two rollers must be identical. In addition, the test has also been used to address conditions of mixed rolling and sliding. In this case, the relative velocities must be controlled so that the proper ration of sliding to rolling is achieved and maintained.

FIG 5.6 WEAR PRODUCED IN ROLLING WEAR TESTS

There are two elements to controlling the velocity; one element is the rotational speeds of the two rollers and the other is the radii of the rollers. In addition control of the preparation and cleaning of the rollers are important, as well as the uniformity of the material, and lubrication if used.

This test has been used for the evaluation of material pairs for rolling applications such as gears, cams, roller bearings, and ball bearings. In these types of applications, additional forms of wear might also be present. Fro example, different regions of gear line, while sliding predominates at other locations along the tooth profile. Nonetheless, generally good correlation has been found between this test and actual performance for those regions where rolling is the major characteristic. One way in which this type of test has been used is to develop data in conjunction with a model for rolling wear that has been used for a number of years. The basic concept of that model is that there is a power law relationship between the number of cycle to failure and the stress level under which rolling takes place.

5.6. OSCILLATING BALL- PLANE TEST

Generically this test is very similar to the pin-on-disk test and either can be considered a variation of the other. The basic features of the ball- plane test is shown in Figure 5.7.One different between the two tests is the shape of the flat member of the contact. In the pin-on-disk it is a disk to accommodate rotation, while in the ball-plane test it is normally a rectangular block or flat specimen. The fundamental different between the two tests is with the type of motion that each provides. The motion is unidirectional at a constant speed in the pin-on-disk test. In the ball-plane test there is a reversing of the direction of sliding and the speed may very throughout the cycle.

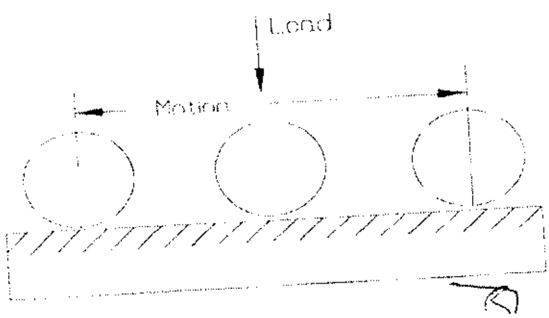


FIG 5.7 OSCILLATING BALL – PLANE TEST

One of the consequences of the change in directions is that each cycle contains an disk test. The velocity profile tends to vary with different ball-plane apparati and depends on the nature of the drive mechanism used. Fro example, the profile is sinusoidal if a rotating eccentric is used. If a rotating eccentric is used. If a linear stepper motor is used, it would have a square wave profile. These differences in the motion can influence wear behavior for a variety of reasons including the influence of debris, build-up of transfer and third-body films and fatigue wear mechanisms (which can be influenced by stress reversals).

Consequently, one of these two types of test could provide better simulation to an application than the other. While this potential exists and must be recognized, it generally does not appear to be a major factor. Both tests have been used effectively to address wear concerns in both unidirectional and oscillatory applications

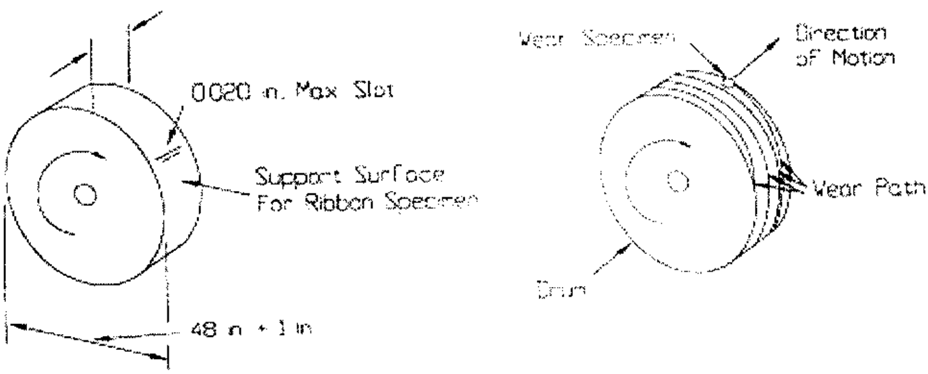
5.7 DRUM WEAR TEST

The test apparatus for this test is shown in Figure5.8. This test was developed to address wear problems associated with such materials as papers, printer ribbons and tapes. These materials tend to be abrasive and can wear hard, wear-resistant materials (e.g., hardened steel, tungsten carbide and diamond). At the same, time the wear resistance of these materials is very low in comparison to that of the counter face materials used in most applications. The use of more conventional test configurations (such as pin-on-disk or block-on-ring, in which one of the members could accommodate the mounting of paper, ,tape, or ribbon samples) generally result in little wear to the wear specimen but significant wear to the tape, ribbon, or paper specimen.

In addition to this, the abrasively of these materials tend to decrease with wear and, as a result, it is generally not possible with these types tests to either determine the wear resistance of the counter face or to get an accurate measure of the abrasivity of the paper, tape, or ribbon. Furthermore, in many of the applications it is the counterface which experiences significant wear and the paper, ribbon or tape provide good simulation. The drum test apparatus was designed and developed to provide a large amount of surface area of the paper, ribbon, or tape, against a relatively small amount of wear area for the wear specimen and to provide simulation in terms of loads, speeds, and relative wear.

While this apparatus was developed to specifically address wear between magnetic heads and paper imprinted with magnetic characters and bar code, it can be used with any web-like materials. This apparatus, like the slurry abrasivity apparatus, can be used either to determine the abrasivity o materials or to determine the wear resistance of materials to this type of wear. Several example of its use are discussed in the literature and test results have been found to correlate with a variety of applications subject to this type of wear (e.g., with the wear of magnetic heads, type surface in printers, punches, and other guiding surface for papers, ribbons and tapes).

A standard test procedure (ASTM G56) has also been established with this apparatus to characterize the abrasivity of printer ribbons. While details of the test procedures associated with these applications do vary, there are some common features and elements.



Ribbon Support Surface   Test Motion

FIGS 5.8 DRUM WEAR TEST

In this type of wear test the ribbon or other web material is wrapped around the periphery of the drum and the wear specimen is loaded against the wrapped surface of the drum. As the drum rotates, the wear specimen moves across the surface of the drum in an axial direction. The resultant wear path on the surface of the drum is a helix. The values of the load, rotational speed, and cross- feed speed of the specimen, as well as the shape of the wear specimen, can be varied to provide simulation. These parameters also influence the wear behavior in the test. For the standard test to determine ribbon abrasivity studies were performed to investigate the influence of these parameters on the wear and specific values were selected for the standard.

CHAPTER 6
WEAR TESTING RIGS

6.1. WEAR TESTING RIG

Wear testing rigs are devices used to simulate wear in the laboratory. A wear testing rig is a simple apparatus designed to make two or more surfaces in contact move relative to each other under controlled conditions. Essentially, wear testing rigs enable the recreation of real life conditions under which wear occurs and the observation of their effects on samples of commonly used or newly designed materials and lubricants.

There are various types of testing rigs available like

- Friction- force measuring rig
- Fretting wear test rig
- Reciprocated movement wear test rig
- Impact wear test rig
- Roller – block rig etc.,

6.2. FRICTION – FORCE MEASURING RIG

It is used to determine static and dynamic coefficients of sliding friction.

Construction: Rig is built from a base, which has a moving table on it, and loading lever. Movement of a table with flat specimen is realized with help of the load that is connected with table through the flexible connection (cable). Speed of the movement is regulated with DC motor, which is used as brakes. Mating specimen, fixed on the lever, is being in contact with flat specimen (fixed on a moving table), and friction force is determined with load cell (fixed on a lever).

Construction of the rig makes it possible to determine also friction path and normal

TECHNICAL SPECIFICATIONS

Contact Mode: Flat on Flat, Ball on Flat.

Friction Mode: Dry / Lubricated Sliding Friction.

Velocity: 0.5 - 10 mm/sec.

Load: 0.01 - 15N

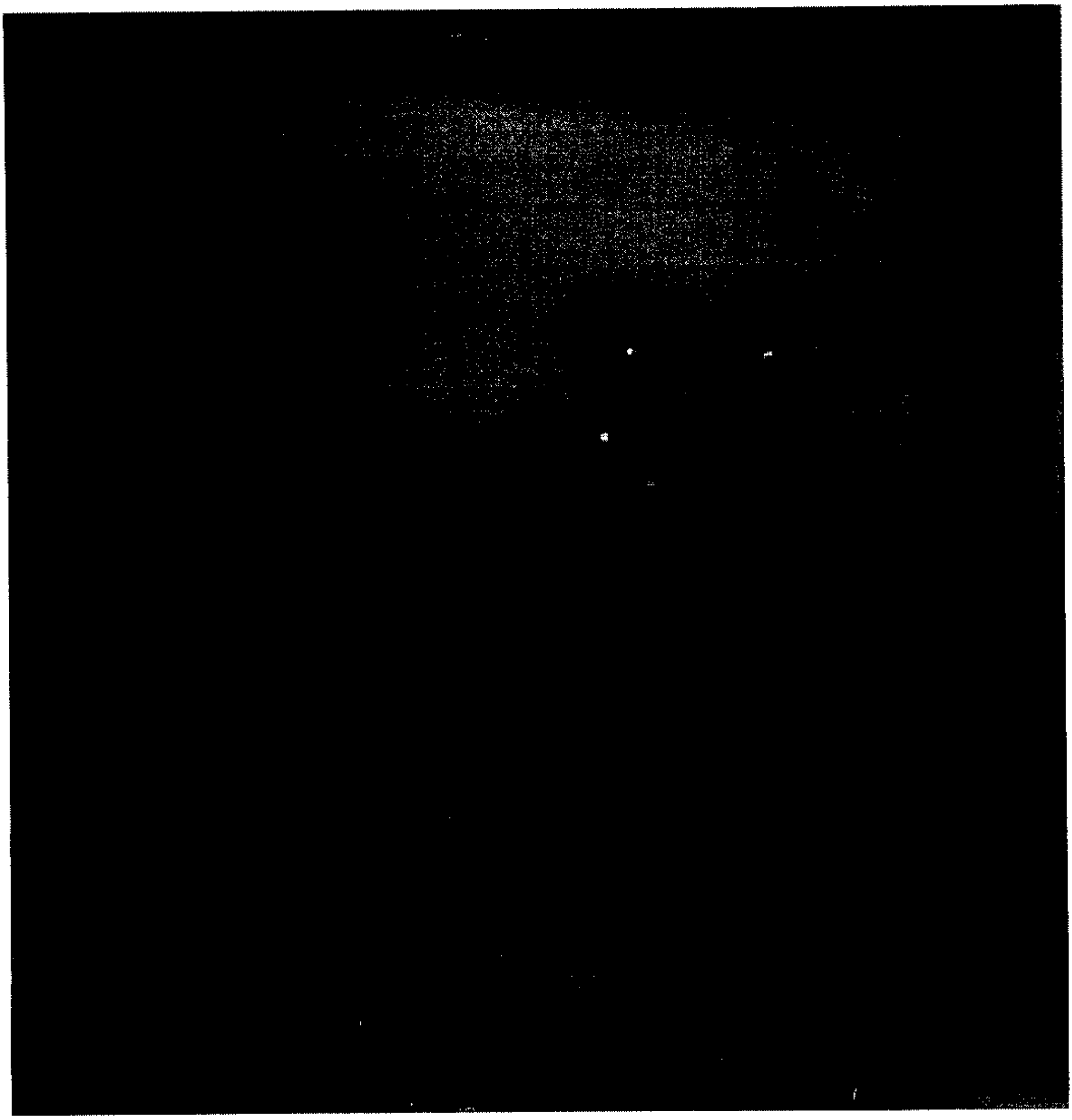


FIG 6.1 FRICTION FORCE MEASURING RIG

6.3. ROLLER – BLOCK WEAR MEASURING RIG

It is used for Adhesive Wear Researches.

TECHNICAL SPECIFICATIONS

Friction Mode: Lubricated / Dry.

Load: < 150N.

Velocity: 100 - 600 r.p.m.

Other Features: On-Line measurements

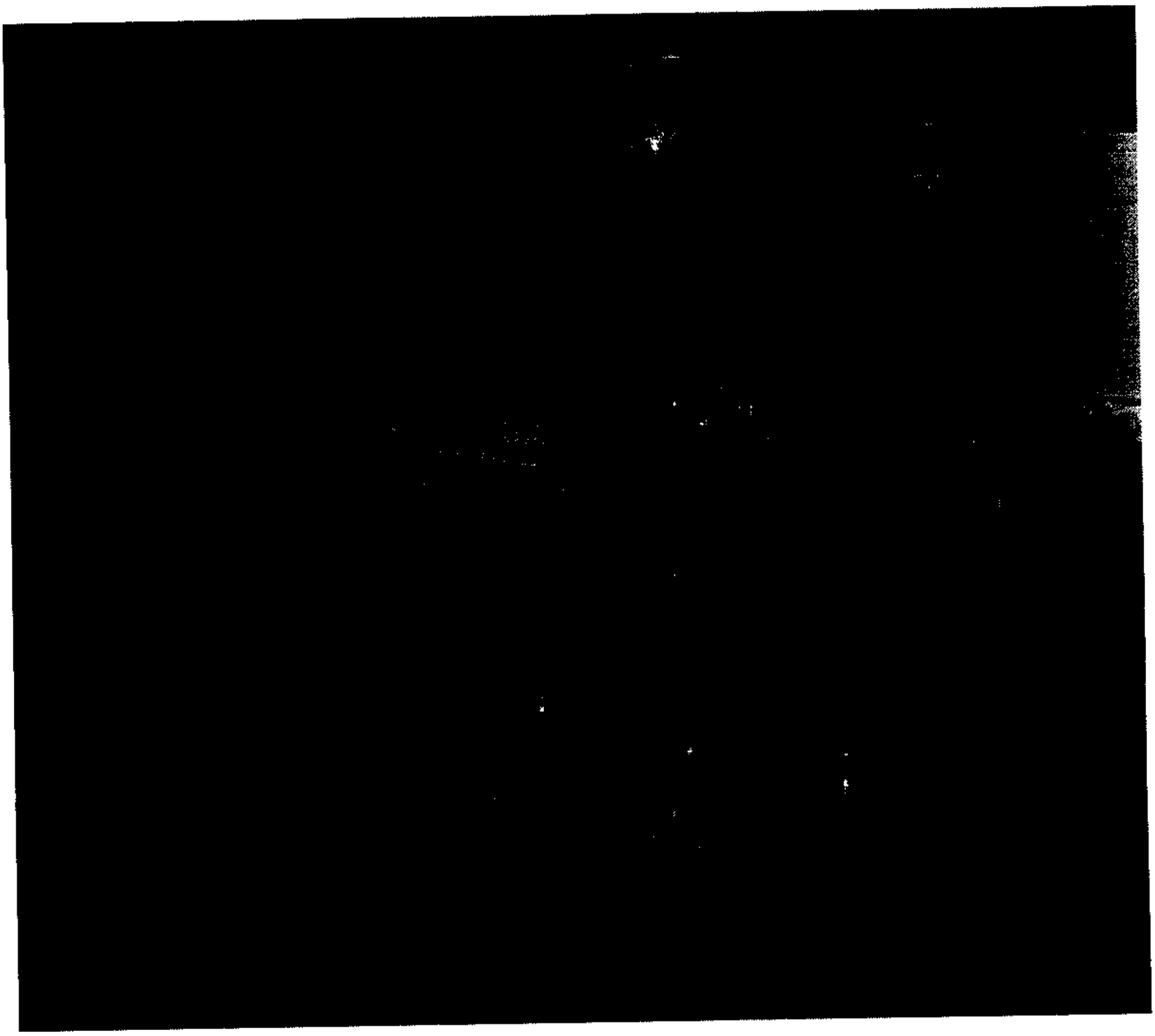


FIG 6.2 ROLLER – BLOCK WEAR MEASURING RIG

6.4. IMPACT WEAR TEST RIG

It is used for Impact Wear Tests in 2 modes:

- Without Specimen's movement.
- With Reciprocated movement

TECHNICAL SPECIFICATIONS

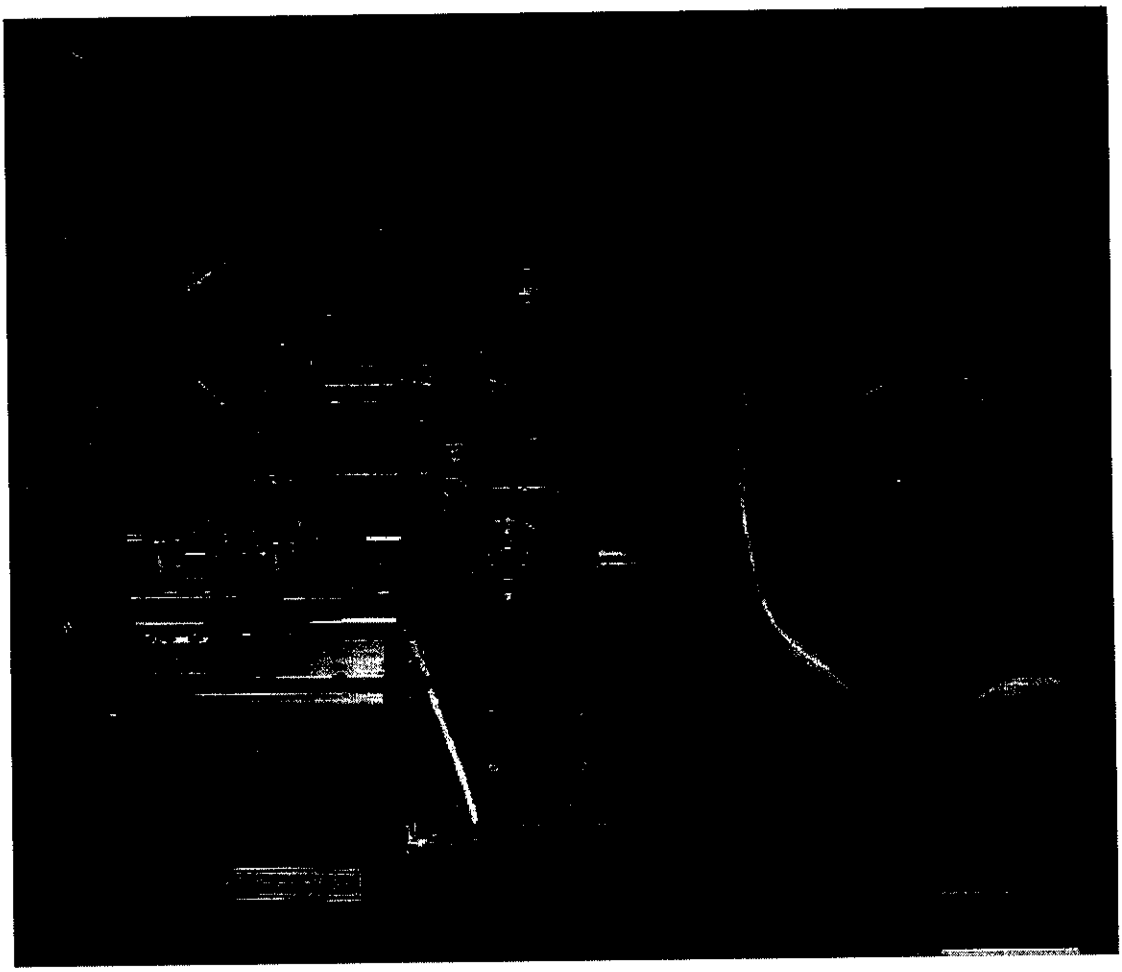
Contact Mode: 1) Pane on Plane. 2) Ball on Plane

Friction Mode: Dry / Lubricated Friction.

Impact Force: 10 Mpa (max).

Impact Velocity: 1 - 2 m/sec.

When Used In Mode 2, Table Displacement: 0.046m



6.5. FRETTING WEAR TEST RIG

It is used for research of wear and friction in conditions of vibrational contact.

TECHNICAL SPECIFICATIONS

Contact Mode: Plane on Plane, Ball on Plane.

Friction Mode: Dry/Lubricated Friction.

Load: 0.5 - 300 N.

Displacement Amplitude: 0.01 - 1mm.

Frequency: 2 - 20 Hz.

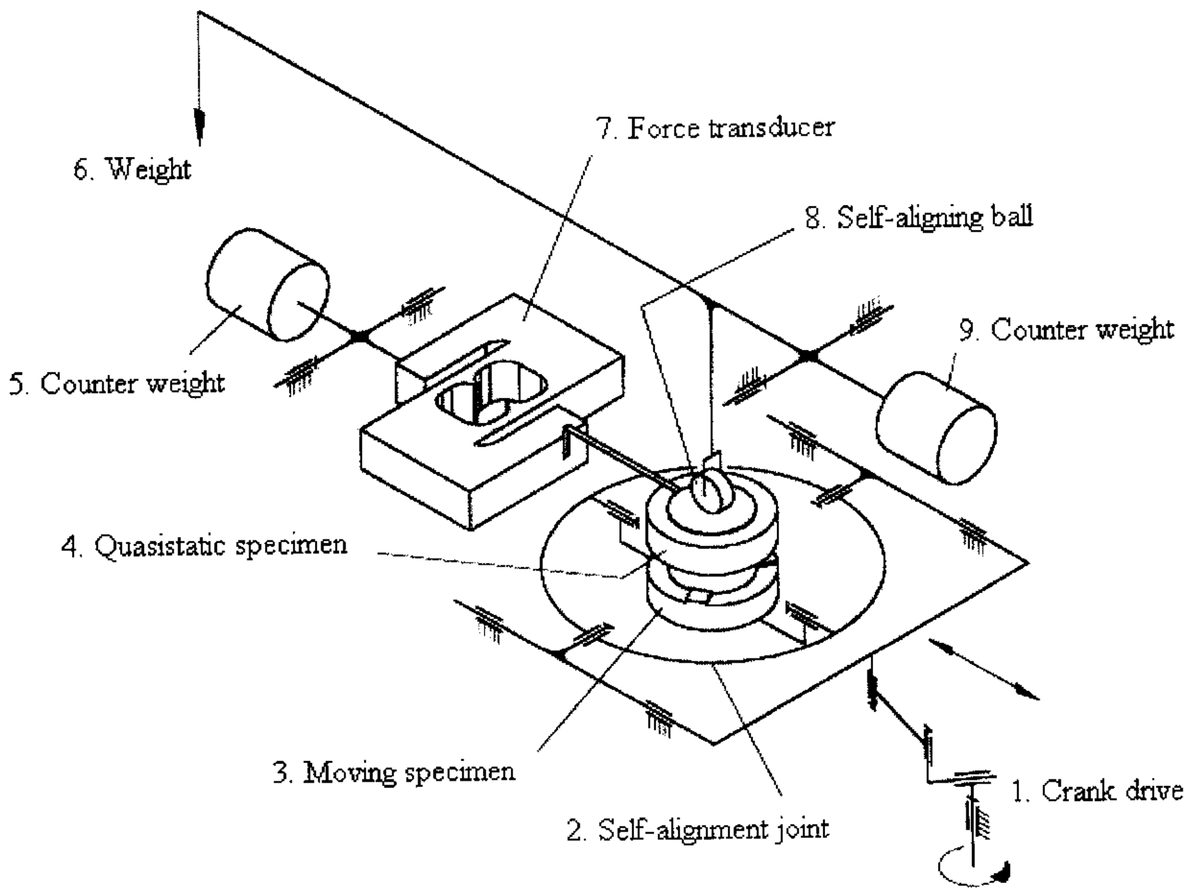


FIG 6.4 FRETTING WEAR TEST RIG

A schematic diagram of the wear test rig constituting the flat-on-flat scheme is eccentricity and connecting rod, transforms the rotary motion of the motor shaft into a linear cyclic motion of a moving table. Two rotary degrees of freedom self-aligning joint (2) carrying a moving specimen (3) is located on the moving table. The conditions required for fretting in the contact zone consist of loading a quasi-static specimen (4) against the moving one by the weight (6). The self-aligning joint axes are coplanar with the friction plane of the two specimens.

Hence, the friction forces acting in this plane do not produce moments about the joint axes and full contact is always retained between the friction surfaces. All the system's joints are specially designed to eliminate unwanted clearances. The amplitude of vibration is defined by the variable eccentricity, but is also affected by the normal load (through the friction force) and by the system's stiffness. The tangential (friction) force between the specimens is measured by a force transducer (7). This force transducer that carries the quasi-static specimen is mounted on a hinged arm, which allows lifting and lowering of the quasi-static specimen.

The loading is transmitted to the quasi-static specimen via a self-aligning ball bearing (8) thus preventing any tangential load component. The loading arm is balanced by a counterweight (9). An eddy current proximity probe mounted on the quasi-static specimen holder is used to measure the relative displacement of the moving specimen. The electrical resistance of the contact, R, which is affected by the presence of oxide wear debris, is recorded on-line. This is accomplished by transmitting a constant current, I, through the specimens, and measuring the voltage drop, V, between them at two points near the contact zone.

Converting of the experimental apparatus into a ball-on-flat scheme is achieved by replacing the quasi-static ring holder with a ball holder, taking the self-aligning joint out and fastening the moving specimen holder directly on the moving table. In this case a ring without the legs is used as the lower specimen. Both the ball and the ring specimen can be used for a number of tests by turning them in their holders. Adjustment for the different mass of the hinged arm is made by the counterweight (5).

6.6. RECIPROCATED MOVEMENT WEAR TEST RIGS

It is used for measuring wear in reciprocated movements

TECHNICAL SPECIFICATIONS

Contact Mode: Plane on Plane, Ball on Plane.

Friction Mode: Dry / Lubricated.

Load :< 50 MPa .

Velocity: 15 - 250 r.p.m.

Sensitivity: Can Measure Wear Changes with 0.1 nm precision.

Table Displacement at Work: 0.046 m.

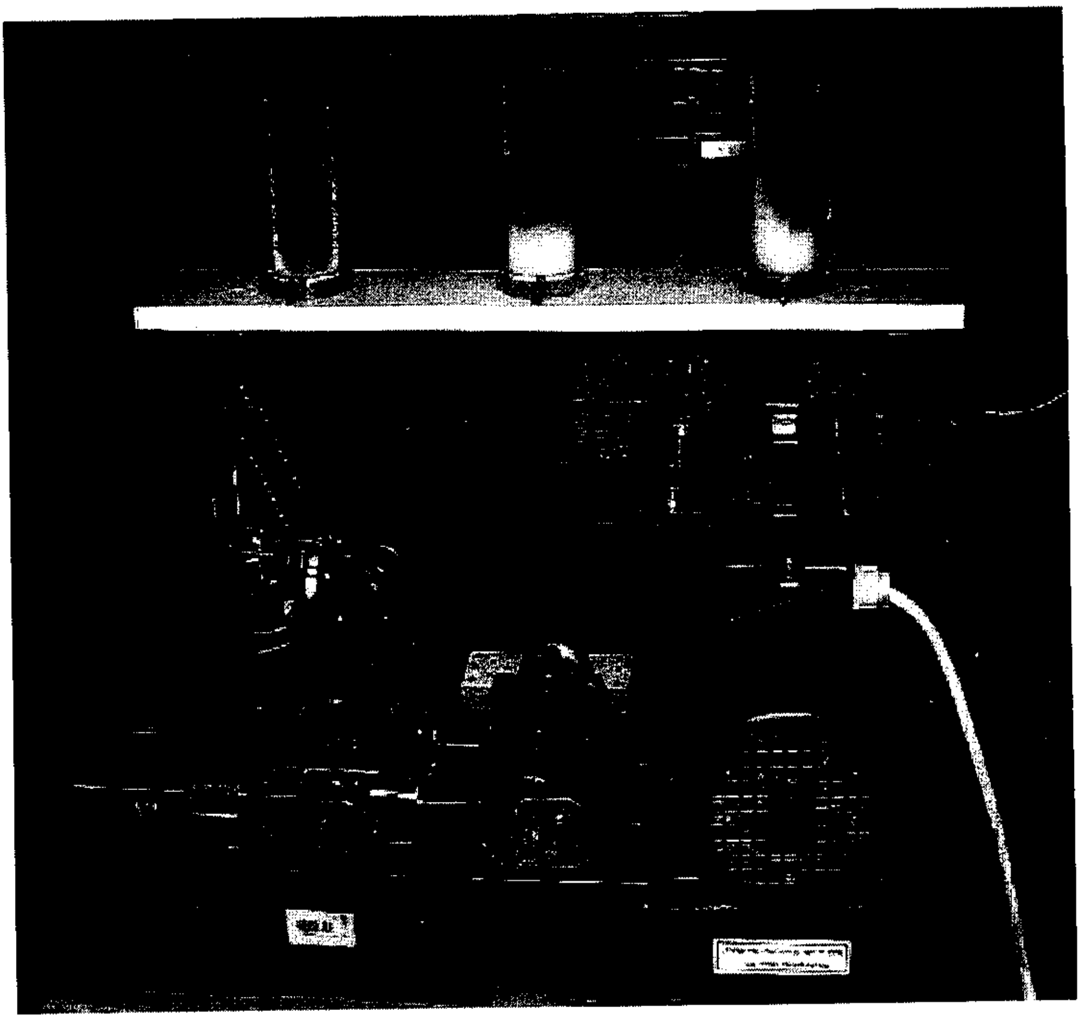


FIG 6.5 RECIPROCATED MOVEMENT WEAR TEST RIGS

6.7. PISTON RING – SLEEVE COUPLE SIMULATING RIG

It is used for Friction force measurement with reciprocated movement and Wear measurement.

Test Rig Description:

The test rig was designed to measure friction force, wear, and surface temperature. It has the following maximum capabilities:

Average speed of reciprocating motion - 5 m/s

Rotational speed of crankshaft - 1500 r.p.m.

Stroke - 0.10 m

Contact pressure between sliding surfaces - 0.5 MPa

TECHNICAL SPECIFICATIONS

Contact Mode: 1) Plane on Plane.

2) Ring on Sleeve.

3) Piston on Sleeve.

4) Ball on plane

Friction Mode: Lubricated Friction.

Load: < 2 MPa.

Velocity: 300 - 1500 r.p.m.

Temperature Changes of Contact Surfaces: 80 °C (Max).

Sensitivity: Can Measure Wear Changes with 0.1 nm precision.

Table Displacement at Work: 0.08 - 0.1 m.

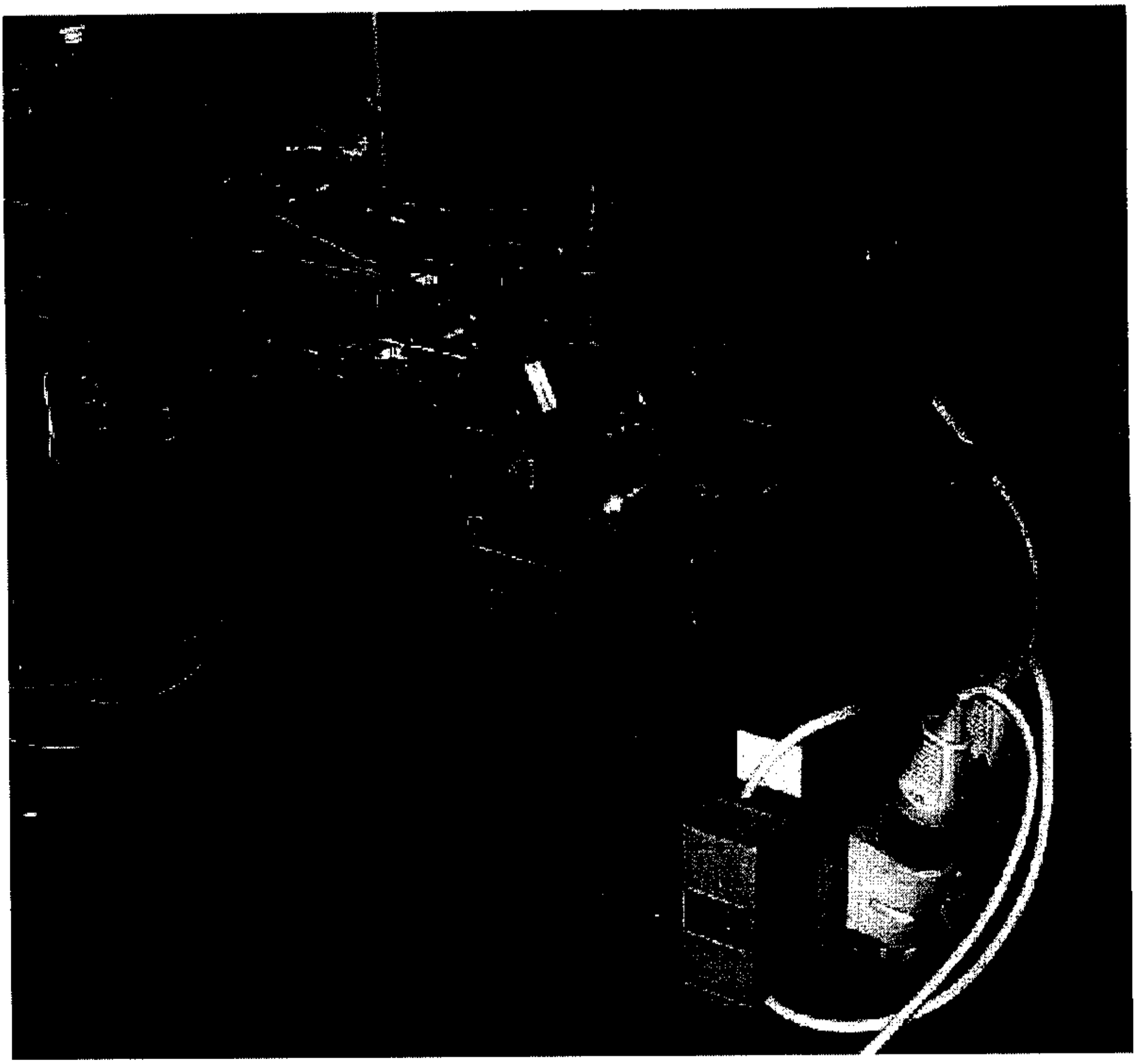


FIG 6.6 PISTON RING – SLEEVE COUPLE SIMULATING RIG

The following parameters can be measured during tests:

· Friction force;

· Motion speed;

· Normal force applied to the test specimen;

· Specimen's surface temperature in the friction zone;

· Wear of sliding surfaces;

· Flow rate of lubricant, supplied to the friction zone.

6.8 THRUST SEAL TESTING RIG

It is used for Friction tests of the mechanical seals and thrust bearings.

TECHNICAL SPECIFICATIONS

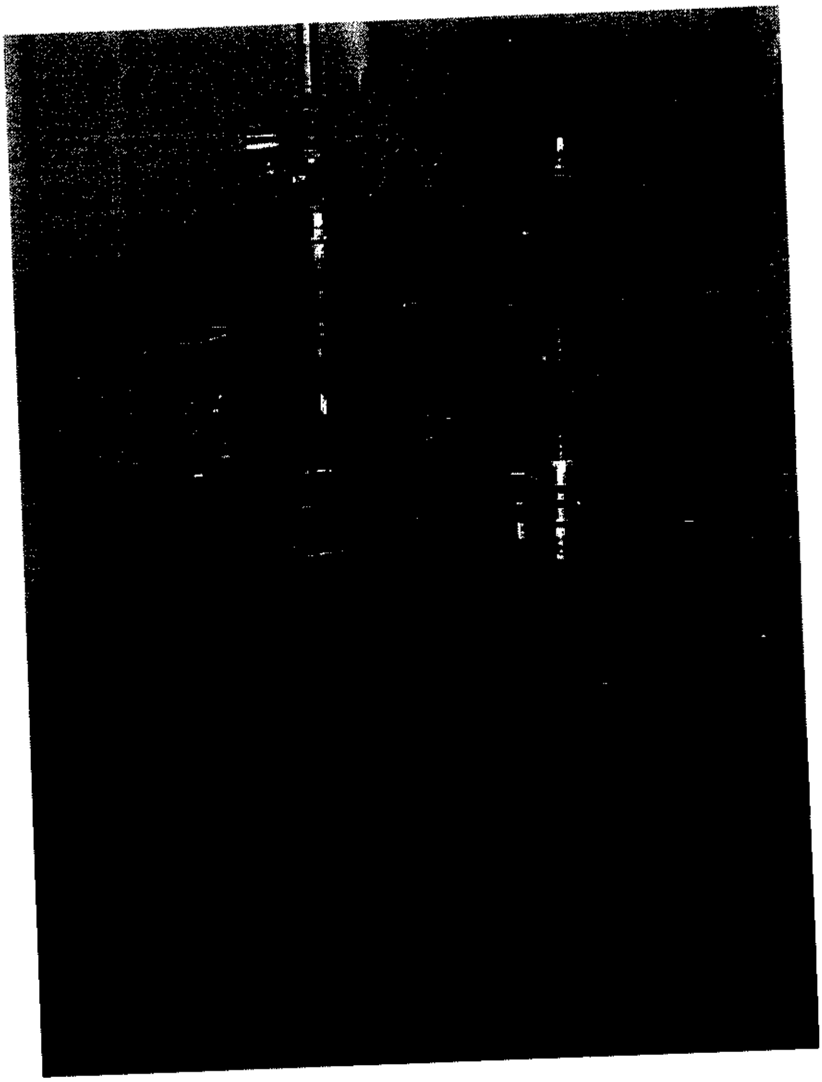
Contact Mode: Plane on Plane

Friction Mode: Lubricated Sliding Friction

Load: 2 to 460 N

Velocity: 750 - 6000 r.p.m

Pressure: 1 to 25 atm



6.9. FRICTION AND WEAR TESTING OF BRAKE MATERIALS

It is used for

- Tests of the break materials according to SAE-J661 standard
- Friction and wear tests of clutch lining materials.
-

Design of the Test Rig:

The rig is built up of the following units:

- Main pillar (on a massive base) supporting all other units;
- Drive, comprising electric motor, timing belt transmission and high-accuracy spindle;
- Self-aligning rotor;
- Three self-aligning holders for specimen to be tested;
- Moving table, comprising a device for heating, cooling and temperature control;
- Table fixture which enables vertical motion (in a guide) of the table and enables it freedom to turn (for measurement of friction force) ;
- Unit for friction force measurement, with a calibration device built into it;
- Screw jack for fine-tuning of the table position relative to the rotor.

In addition to the mechanical units mentioned above, there are a number of means for the control of drive motor, load-unload stepper motor, heating element and air cooling system (for counter-part temperature control), and measurement of friction force and wear value. Moreover the rig has a means to polish the working surfaces before every test.

Depending on needs, the machine can operate in one of two arrangements: common (with pins stationary and disk rotating), or inverse (pins rotating).

In the common arrangement, the counterpart, having the shape of a ring, is affixed to the rotor, and the specimen holders are affixed to the stationary table. The self aligning property of the rotor assures the same load to all three specimens, and compensates for non uniform wear and misalignment in assembly.

The original construction of self-orienting specimen holders eliminates, in principle, the tipping-moment which the (tangential) friction force applies onto specimens in simple holders. That is because the intersection of two free axes of motion (of the specimen) is in the work surface. It is possible to use the holders in a mode that has no degree of freedom.

The self aligning action of the holders and the self aligning action of the rotor together with the polishing of the counterpart make break-in of the specimens practically redundant. This way the test results depend less on the accuracy of the specimens and scatter of test is down to a minimum.

In the inverse arrangement it is possible to control the interface temperature from the attached computer. Every step of the test procedure can be given any specified temperature within the operating range, and the machine will maintain this temperature within the operating hysteresis, regardless of momentary load. To accomplish that, the specimen holders are affixed to the rotor, while a special counterpart of a toroidal shape is affixed to the table. The toroid is hollow, and a heating element is built into it. The channel in the toroid is also used for cooling by forced air. On top of the toroid the counterpart is attached. The counterpart has a thermocouple built into it, for the temperature control circuit.

Main characteristics of the machine are summarized:

Axial load 10 to 500 N
Torque 0 to 25 Nm
Interface temperature ambient to 400 C
Heating rate 22 C/min
Cooling rate 25 C/min
Angular Velocity 6 to 120 r/sec
Linear Velocity 0.3 to 6 m/sec
On-line wear 0 to 1.5 mm
(Resolution is 1 micrometer for on-line wear monitoring).

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TECHNICAL SPECIFICATIONS

Contact Mode: Plane on Plane

Friction Mode: Dry / Lubricated Sliding Friction.

Axial Load: 10 to 500 N

Angular Velocity: 6 to 120 rad/sec

Linear Velocity: 0.3 to 6 m/sec

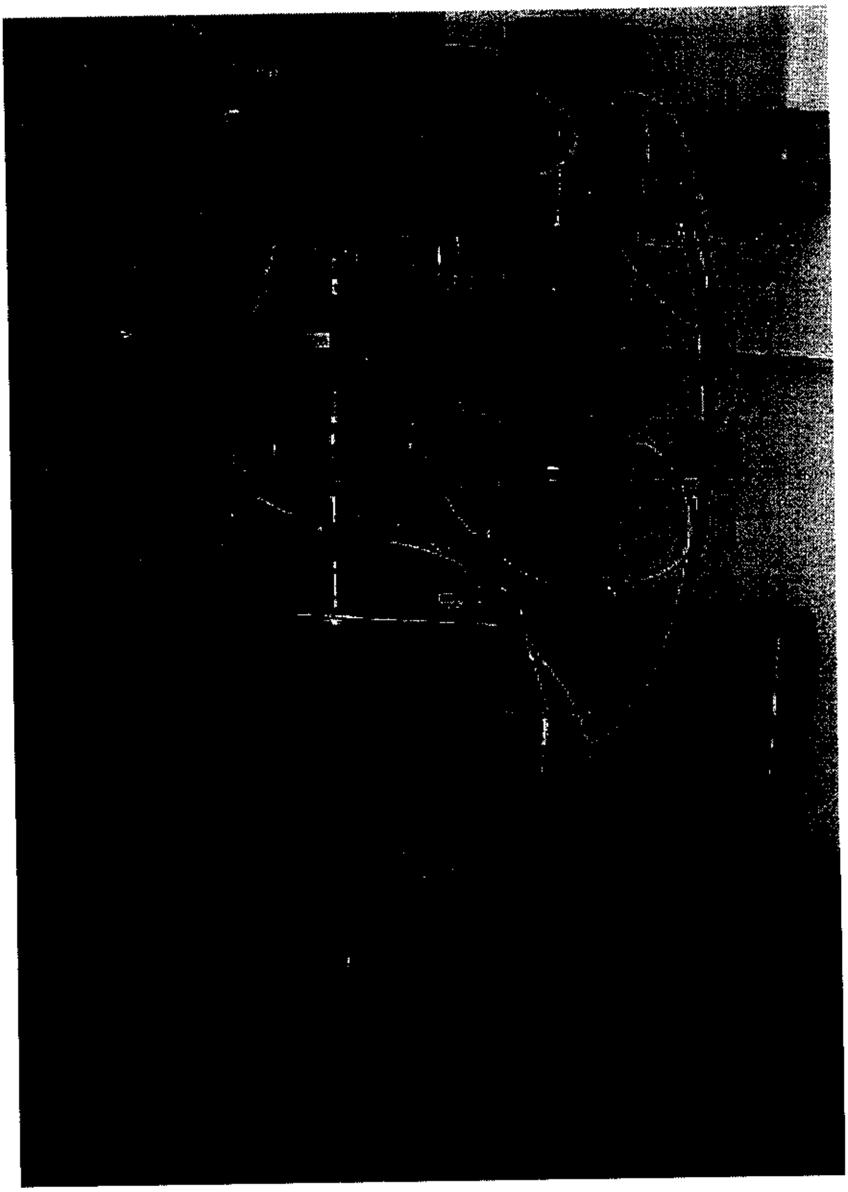


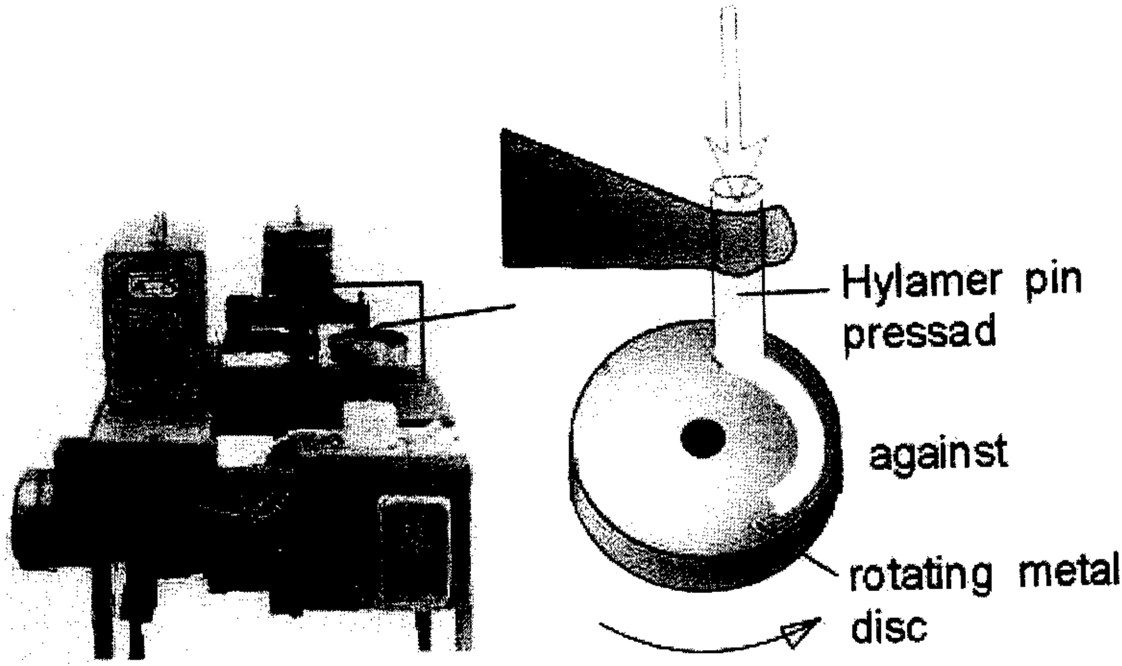
FIG 6.8 FRICTION AND WEAR TESTING OF BRAKE MATERIAL

CHAPTER 7

PIN-ON-DISK SYSTEM

7.1. PIN- ON –DISK WEAR TEST

This is another configuration that has been used extensively to study wear and to rank materials. it is viewed as a general test that can be used to evaluate the sliding wear behavior of material pairs and its correlation with an application depends on the degree of simulation that the test parameters have with those of the application. The basic configuration is shown in Figure 7.1.



FIGS 7.1 PIN ON DISK WEAR TEST

A radius tipped pin is pressed against a flat disk. The relative motion between the two is such that a circumferential wear path on the disk surface is generated. Either the pin or the disk can be moving. The test parameters that have been used with this test vary. The ASTM standard for this test, ASTM G99, does not specify specific values for the parameters, but allows those to be selected by the user to provide simulation of an speed and material pairs. The test can also be done in a controlled atmosphere and with lubrication.

Like the block-on-ring and crossed-cylinder tests stress levels change during the course of the test, as a result of the wear, and the relationships between wear and duration or amount of sliding is often nonlinear. For the material ranking and comparison the ASTM standard recommends measuring the wear on both members after a fixed number of revolutions. It is also recommended that with dissimilar pairs of materials that two tests be performed with the materials changing positions in the test.

The standard allows the use of wear curves for comparison. This is particularly useful if nonlinear behavior is to be taken into account. When this approach is used it specifies that new specimens are to be used for each data point on the curve. The test should not be stopped for intermediate wear measurements and restarted. Because of the end of the pin has a radius, useful wear data can be obtained after small amounts of sliding and thereby provide a continuous curve. If the pin was flat on the end, this would not be possible since the initial portion of the wear curve would be strongly influenced by the misalignment between the pin and the disk. In such a case it is necessary to allow the specimens to wear-in (so that uniform contact is achieved) before useful data can be obtained. This is illustrated in Figure7.2. The block-on-ring test has a similar problem associated with alignment in the axial direction of the disk.

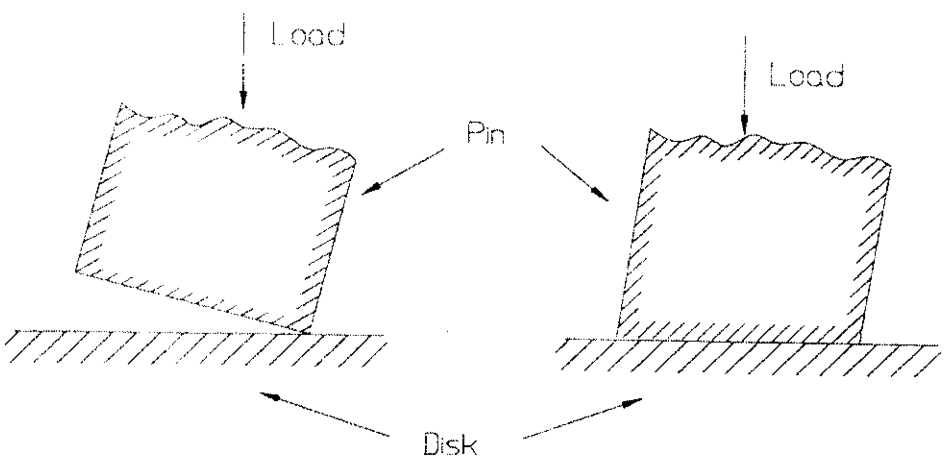


FIG 7.2 MISALIGNMENT IN THE PIN ON DISK TEST

The test method allows both geometrical and mass loss methods for determining wear but in either case the measurement should be converted to volume loss for reporting. With mass loss, this is to be done by dividing the mass loss by the density. With the geometrical approach, this is done by converting a measured linear dimension, to a volume using the appropriate relationship for the geometry of the wear scar. For example, in the case of negligible wear on either member or a spherical ended pin, the width of the wear scar can be used to compute the volume by means of the following equations,

$$V = \pi W^4/64R \quad \text{(pin wear)}$$

$$V = \pi D W^3/6R \quad \text{(disk wear)}$$

Where,

V is the volume of wear;

W is the width of the wear track on the disk or width of the flat on the pin;

D is the radius of the wear track;

R is the spherical radius of the pin;

In both cases the wear scar is either the volume of a spherical cap of cord W (pin wear) or a groove whose profile is a circular section of cord W (disk wear).

7.2. PIN ON DISK WEAR TEST RIG

The Pin-On-Disc machine is a versatile unit designed to evaluate the wear and friction characteristics on a variety of materials exposed to sliding contacts in dry or lubricated environments. The sliding friction test occurs between a stationary pin stylus and a rotating disk. Normal load, rotational speed, and wear track diameter can be varied.

Electronic sensors monitor wear and the tangential force of friction as a function of load, speed, lubrication, or environmental condition. These parameters as well as the acoustic emissions at the contact are measured and displayed graphically utilizing the

Construction: This rig is constructed using a classic scheme, including rotating disk (250 mm diameter), and mechanism for holding an abrasive paper. Specimen, that needs to be tested, is fixed on a special lever, which press specimen against the disk with certain normal load. To guarantee wear in permanent conditions, specimen with a lever is moving in the radial direction relative to disk. When specimen reaches edge points, rig stops automatically. Wear of the specimen is determined using a weight method.

TECHNICAL SPECIFICATIONS

Contact Mode: Plane on Plane.

Friction Mode: Dry Sliding.

Disk Velocity: 70 r.p.m

Specimen Radial Velocity: 2 mm / revolution.

Load: < 20N.

7.3. DIFFERENT TYPES OF PIN ON DISK RIGS

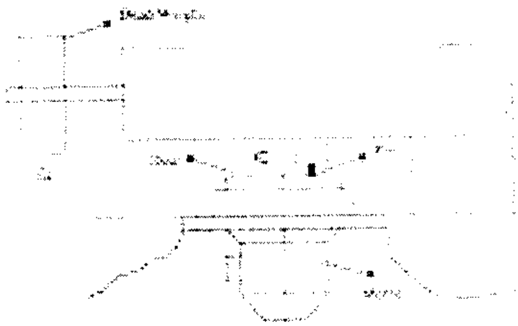
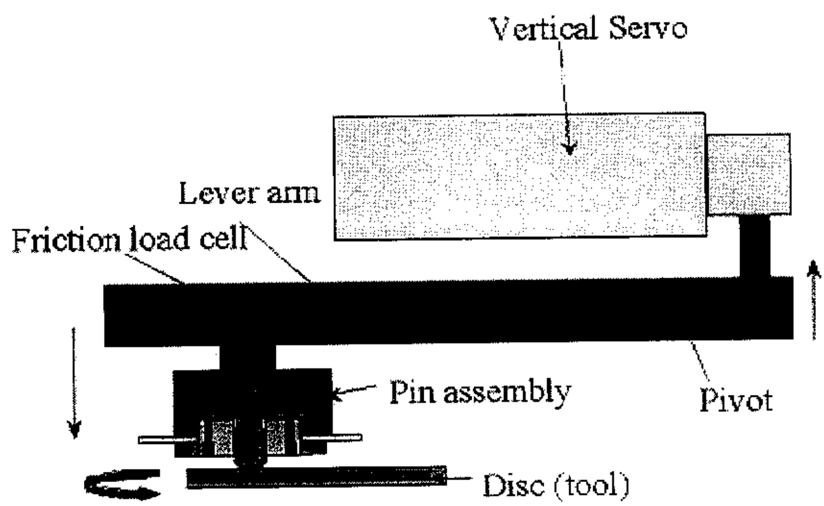


FIG 7.4.a HORIZONTALLY MOUNTED PIN TYPE RIG



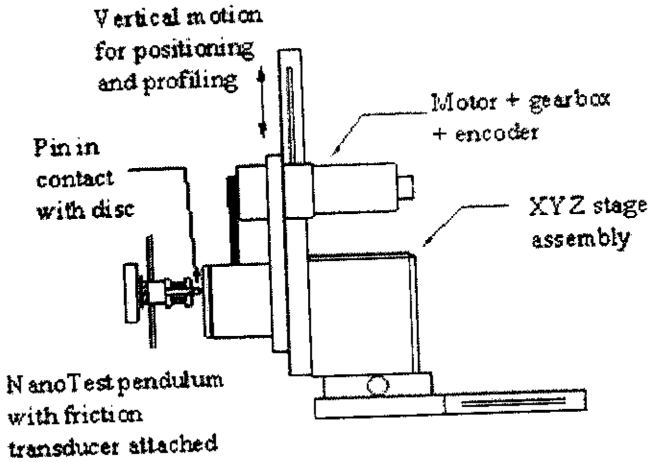


FIG 7.5 VERTICALLY MOUNTED PIN TYPE RIG

CHAPTER 8
DESIGN AND FABRICATION

8.1 DESIGN

In order to design our test rig, we have utilized the PRO-E modeling. Each and every part of our test rig was designed separately according to the specifications and they were assembled finally. The various parts of our test rig are:

- Bed and the stand
- Disc system
- Pin holder
- Weight acting lever set up
- Sliding lever mechanism
- Weight acting mechanism – pulley setup

8.2 BED AND THE STAND

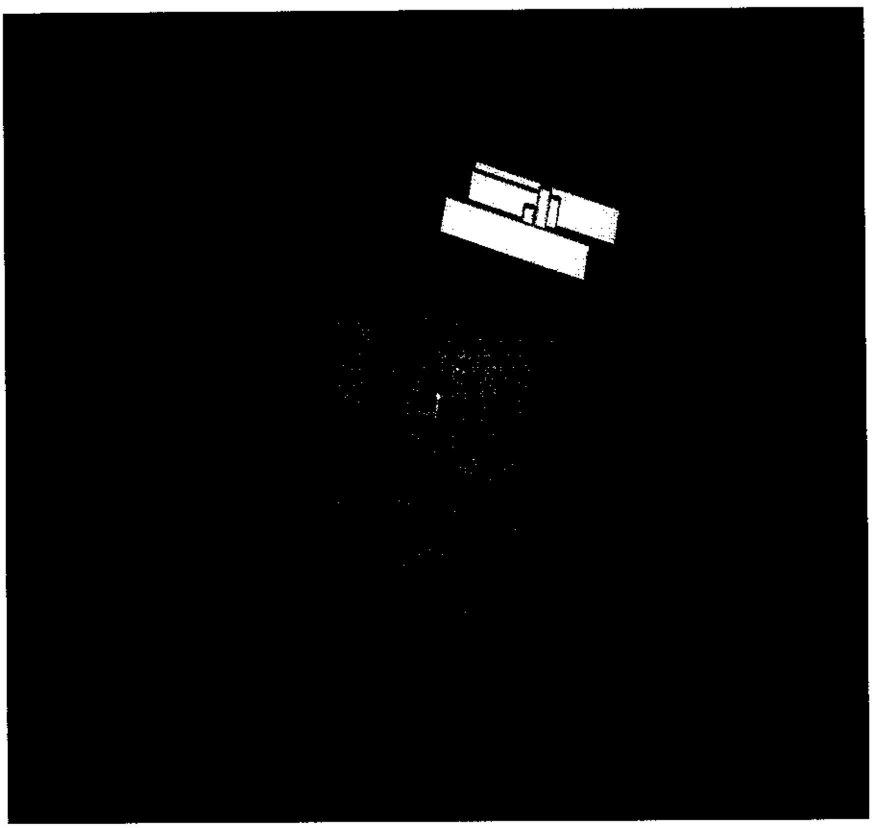


FIG 8.1 BED AND THE STAND

The bed of the test rig is supported by a four leg L-angle bars. The bed is the base for all mechanisms and setup to function, under the bed a motor is fitted in order to run the disc. It is also provided with guide ways to support the sliding lever mechanism. The above figure 8.1 illustrates the design of the bed and the stand of the test rig. The bed is made up of mild steel and the stand is made up of four L-angle bars.

8.3 DISC SYSTEM

The disc system is used to hold the disc rigidly, and the disc is coupled with the motor and it is fitted vertically below the disc. The disc holder is also made up of mild steel The following figure 8.2 illustrates the design of disc system

FIG 8.2 DISC SYSTEM

8.4 PIN HOLDER

The pin used for the testing purpose is held by this pin holder. It is a adjustable type, where the pin diameter can vary from 3 to 16mm.The pin holder is screwed with the weight acting lever mechanism and it is also made up of mild steel. The following figure 8.3 illustrates the design of pin holder.

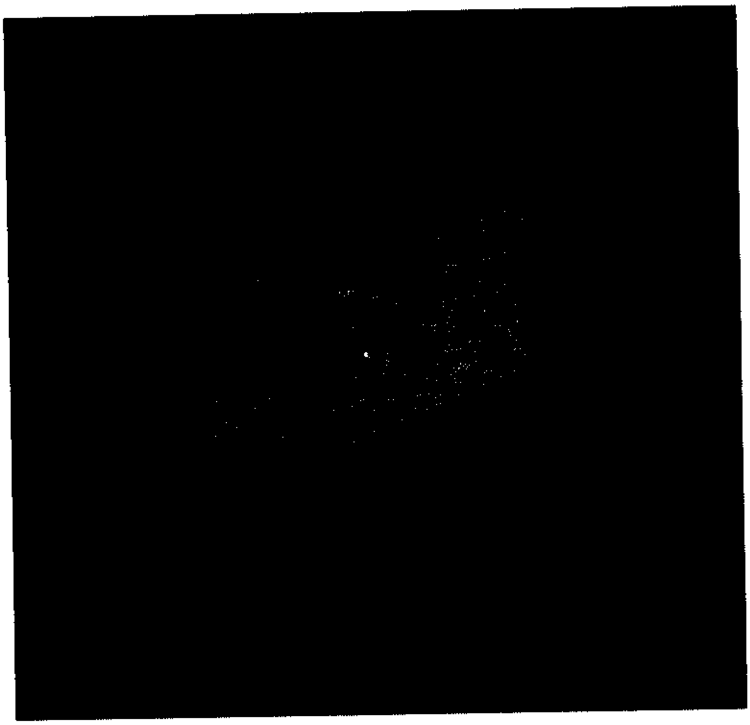


FIG 8.3 PIN HOLDER

8.5 WEIGHT ACTING LEVER SETUP

Only by this weight acting lever, the weights are added the through the pulley mechanism and the tension that is felt by the pin is rested on the disc. A whole is drilled at the end of the lever and a string is passed through it to the pulley setup to add weights. This weight acting lever is also made up of mild steel. The following figure 8.4 illustrates the design of weight acting lever setup.

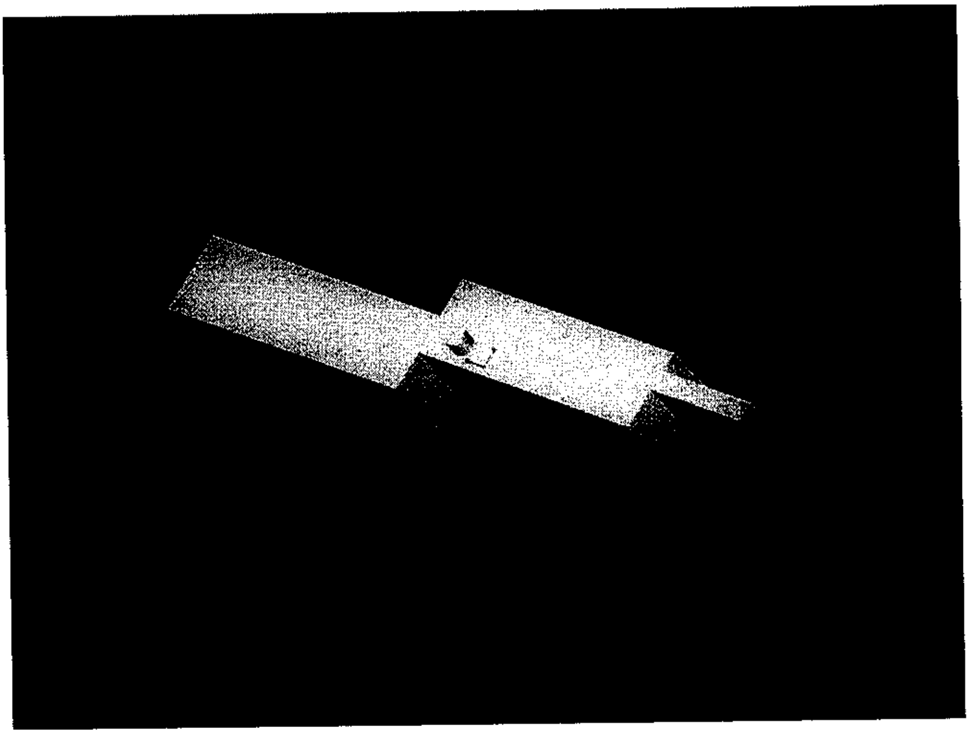


FIG 8.4 WEIGHT ACTING LEVER SETUP

8.6 SLIDING LEVER MECHANISM

This sliding lever mechanism is mainly used for changing the wear track diameter. By moving this setup, the weight acting lever mechanism which is hinged in the sliding lever is also moved to a certain distance. The sliding lever mechanism is supported by two guide ways for the movement purpose. The following figure 8.5 illustrates the design of sliding lever mechanism.

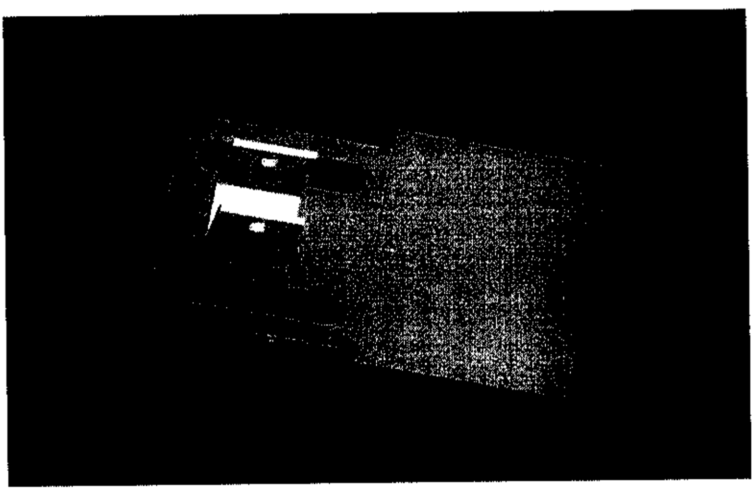


FIG 8.5 SLIDING LEVER MECHANISM

8.7 WEIGHT ACTING MECHANISM – PULLEY SETUP

From this weight acting lever mechanism the string is passed through the pulley set up in order to add weights and apply tension in the pin holder. This pulley set up is also made up of mild steel. The following figure 8.6 illustrates the design of pulley setup.

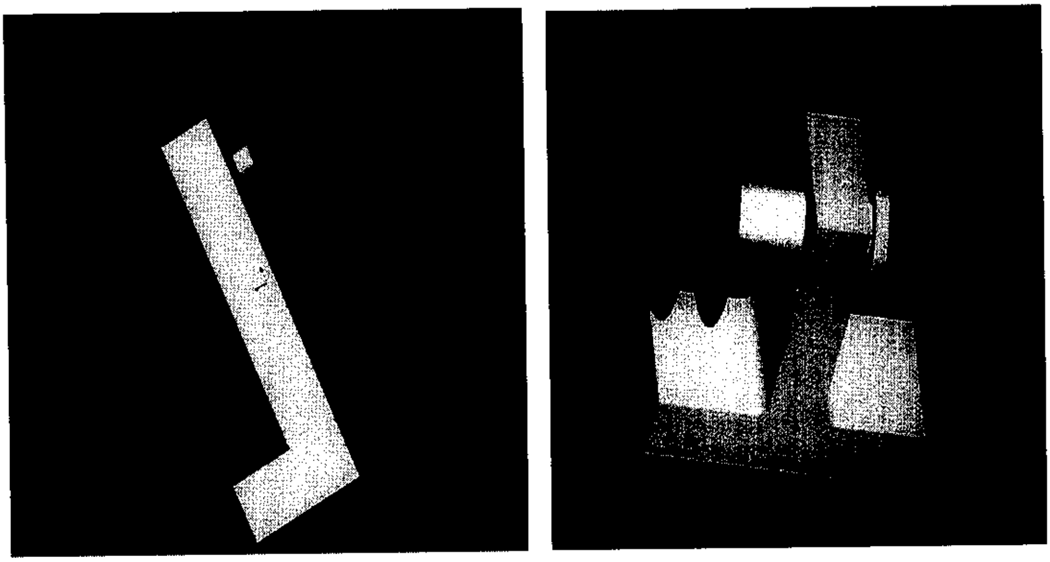
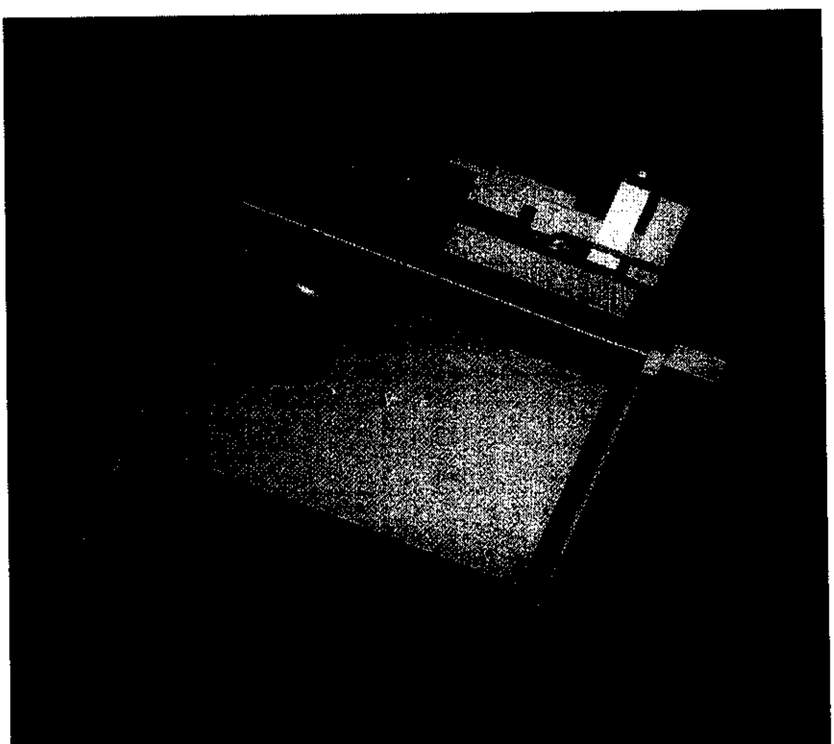


FIG 8.6 WEIGHT ACTING MECHANISM – PULLEY SETUP

8.8 FINAL DESIGN

The assembled and the final design of our test rig is shown below in fig 8.7.



8.9 SPECIFICATIONS OF THE TEST RIG

The various specifications of the test rig are:

Sliding Speed Range: 0.26-10 m/sec

Motor used: A.C (0.25 HP)

Maximum Normal Load: 100 N

Frictional Force: 0-100 N

Wear Measurement Range: 4 mm

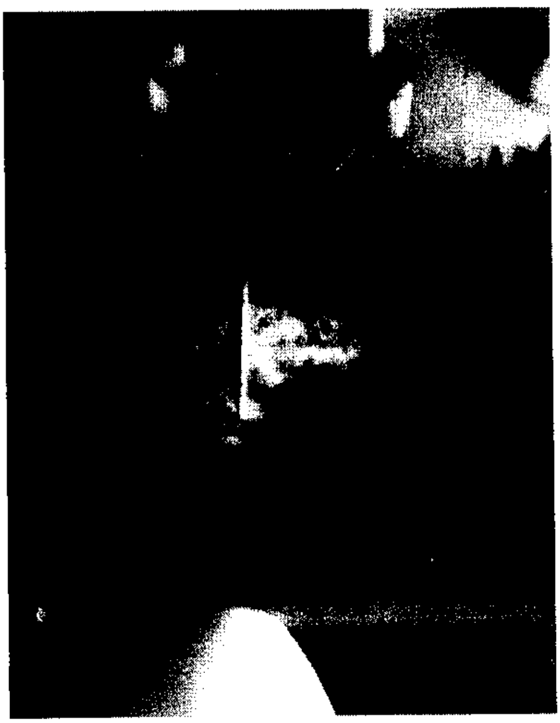
Pin Size: 3-16 mm diagonal/diameter

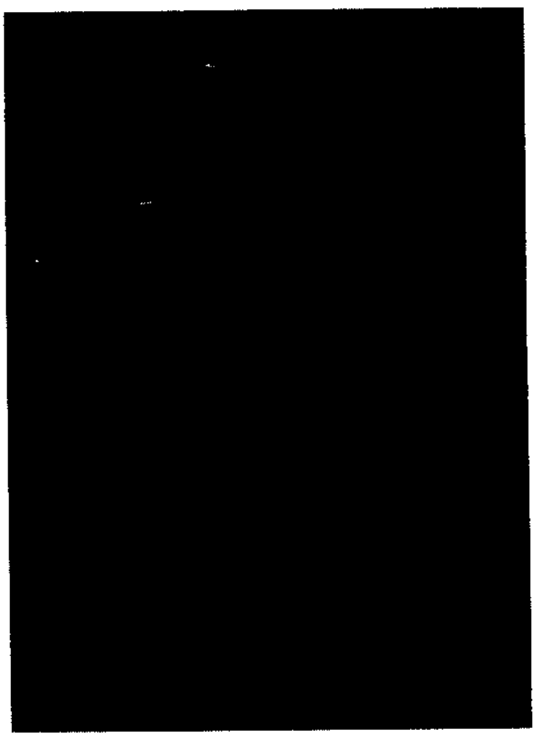
Disc Size: 165 mm x 10 mm thick

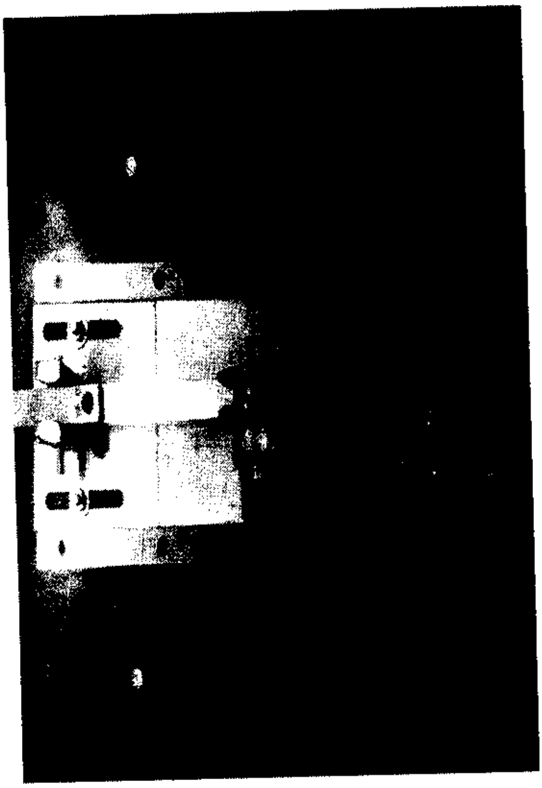
Wear Track Diameter: 10-60 mm

8.10 FABRICATION OF TEST RIG

The various parts of test rig were fabricated separately and assembled finally. The following figures 8.8 will show the fabrication of our wear test rig.







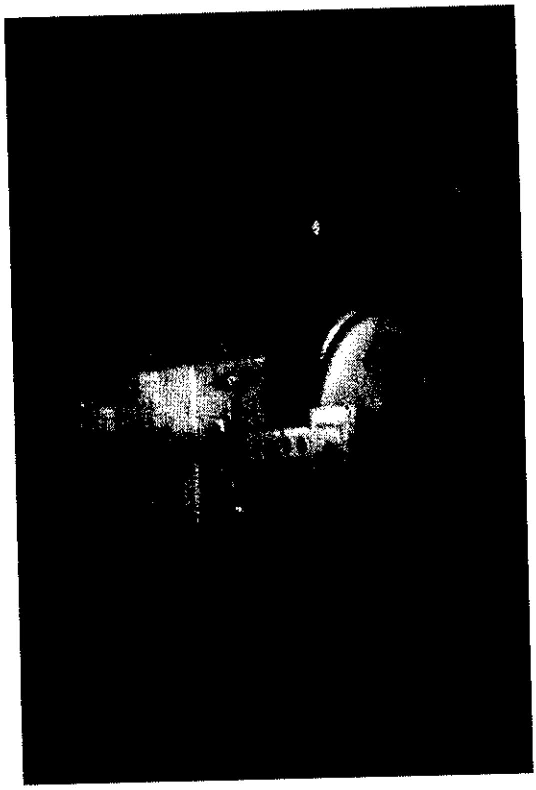


FIG 8.8 FABRICATION OF WEAR TEST RIG

CHAPTER 9

LIST OF MATERIALS

TABLE 9.1 BILL OF MATERIALS

Sl.No	Name of the Parts	Quantity	Materials
01.	Bed (250*125mm)	1	M.S
02.	Washer	4	C.I
03.	Pulley	2	C.I
04.	Square Bars	10	M.S
05.	L-angle	4	C.I
06.	Sliding Bar	1	M.S
07.	Round Plate	1	M.S
08.	Bush	1	M.S
09.	Nuts & Bolts	20	C.I
10.	A.C.Motor	1	Electrical

Thus the above parts were machined, drilled, grinded and welded according to the design and fabricated finally.

CHAPTER 10

COST ESTIMATION

10.1. MATERIAL COST:

TABLE 10.1 MATERIAL LIST

Sl.No	Name of the Parts	Quantity	Materials
01.	Bed (250*125mm)	1	M.S
02.	Washer	4	C.I
03.	Pulley	2	C.I
04.	Square Bars	10	M.S
05.	L-angle	4	C.I
06.	Sliding Bar	1	M.S
07.	Round Plate	1	M.S
08.	Bush	1	M.S
09.	Nuts & Bolts	20	C.I
10.	A.C.Motor	1	Electrical

Material Cost = Rs.7200 /-

10.2. LABOUR COST:

LATHE, DRILLING, WELDING, GRINDING, GAS CUTTING AND ALL OTHER ASSEMBLY WORKS:

Labour Cost = Rs. 3000/-

10.3. OVERHEAD CHARGES:

The overhead charges are arrived by "Manufacturing Cost"

Manufacturing Cost	=	Material Cost + Labour cost
	=	7200+3000
	=	Rs.10, 200/-
Overhead Charges	=	10% of the manufacturing cost
	=	Rs.1020/-

10.4. TOTAL COST :

Total cost	=	Material Cost+ Labour Cost + Overhead Charges
	=	7200+3000+1020
	=	Rs. 11,220/-

Total Cost for this project = Rs. 11,220/-

CHAPTER 11

EXPERIMENTATION

11.1 EXPERIMENTATION

As a part of our fabrication it is necessary to validate our test rig for that an experiment is carried out to calculate the wear rate of stainless steel material at different load conditions and it is compared with the ASTM standard result. For this experiment the pin material is made up of stainless steel material and disk is covered with an abrasive emery paper. The various test conditions for the experiment are:

Material specimen	a) Pin diameter	: 10mm
	b) Pin length	: 15mm
	c) Sliding distance	: 1000m

From the above mentioned conditions, the volumetric loss of the material was calculated using the weight loss of the material from which wear rate is calculated by using the below formula:

$$\text{Wear rate } K = \text{Weight loss} / \text{Load} * \text{Sliding distance}$$

$$K = 1{*}e^{-5}\ mm^3/\ Nm$$

The following table will show the weight loss at different load conditions:

TABLE 11.1 WEIGHT LOSS AT DIFFERENT LOAD CONDITIONS

LOAD (Mpa)	0.35	0.7	1.05
Weight loss mm^3 at 0.5 m/s sliding velocity	0.001	0.0003	0.0251
Weight loss mm^3 at 1m/s sliding velocity	0.004	0.0100	0.0290
Weight loss mm^3 at 2 m/s sliding velocity	0.0012	0.0303	0.0341

The calculated weight loss values were compared with ASTM standard result and it was found that there is a deviation of about 5-7% from the standard result. Thus the fabricated test rig is validated with a small deviation from the standard result.

CHAPTER 12
CONCLUSION

In our project, the test rig was designed and fabricated successfully in order to study the wear rate at different testing conditions. By changing testing conditions such as sliding distance, sliding velocity, wear track diameter, test load and other various factors the wear rate can be calculated effectively from the test rig. In order to validate our test rig by ASTM standard a combination of stainless steel material i.e. pin material and disc material is made up of abrasive emery paper undergone a test at certain sliding speed and the wear rate that is found was compared with the ASTM standard result .From the comparison it was found that there is some 5-7 % of deviation from the standard result. Thus our test rig can be used for finding wear rate.

Since coimbatore is a pool of many manufacturing and pump industries it involves a large number of machining processes in such processes wear plays a vital role. Therefore it is necessary to simulate and study the wear property, for such purpose our test rig will be more helpful to calculate the wear rate through volumetric loss of the material.

The test rig we have fabricated can be used in colleges, testing centres and also in various industrial places. The test rig can be used more effectively with a data acquisition system which is more helpful in record the readings and values. With more financial assistance the test rig can be designed with more additional features. The test rig can be used in our college for experimental and research purposes so that the students and as well as institution can be benefited.

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