

THE EFFECT OF THE SLIP AMPLITUDE  
AND LUBRICANT TYPE  
ON THE RATE OF  
FALSE BRINELLING  
IN ROLLER BEARINGS

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**SYNOPSIS**

False brinelling is a wear phenomenon that occurs in stationary rolling element bearings. Vibrations from adjacent running machinery are transmitted to the stationary bearing via the machinery foundations, and result in relative motion between the rollers and the races. This results in wear scars at the point of contact between the rollers and the race. The wear scars result in premature failure of the bearings.

There are a large number of parameters that affect the rate of false brinelling. Most of these parameters are not free to be altered by the user of the bearing. There are two parameters that can be altered, the type of lubricant, and the slip amplitude of the vibration. The slip amplitude can be altered by changing the stiffness of the foundation of the equipment. By varying these two parameters it should be possible to limit the amount of damage to the bearing.

In the work reported here, a shaker was used to axially vibrate a flat "outer race" in a lubricant bath, against a cylindrical roller. The relative motion was measured by means of a proximity probe.

Two lubricants were tested: SAE 30 oil and LGEP 2/1 grease.

Two different wear mechanisms, exhibiting vastly different wear rates, were discovered for the oil tests. At low slip amplitudes, smaller than 25um, the results suggested that elastic deformation of the contacting surfaces was the means of displacement accommodation. At larger slip amplitudes the results suggested relative motion between the two contacting surfaces, which was marked by a bifurcation in the wear rate versus slip amplitude curve. The one branch of the curve, which for ease of explanation is called wear mechanism one, exhibited a low wear rate, which increased in a linear fashion with respect to the slip amplitude. The second branch of the curve, wear mechanism two, approximated a step function, with a very rapid increase in the wear rate with respect to the slip amplitude. This rapid increase is followed by a slow linear increase in the wear rate with respect to the slip amplitude.

The onset of wear mechanism two for the oil tests was the result of a high level of silicon contamination in the oil sample. The silicon contaminants act as abrasive particles, which was

confirmed by the presence of ribbon-shaped wear particles in the lubricant sample. This leads to a high wear rate.

The tests using grease as the lubricant displayed three distinct contact conditions. The first region was characterized by elastic deformation of the contacting surfaces, as a means of accommodating the relative motion. This resulted in alternating stresses. The elastically accommodated region has a low wear rate, and the wear particles displayed fatigue wear characteristics, the result of the alternating stress.

The second region in the grease tests was the stick-slip region. This region was found to be present between 25um and 100um. This region has a high wear rate, in comparison to the elastically accommodated region.

In the final region the relative motion is accommodated by pure slip. The wear mechanisms present are indicative of relative motion, such as abrasion. The relative motion results in heating at the local micro-contacts, which was detected in the wear particle analysis. The wear rate with respect to the slip amplitude starts at a value below the wear rate for the stick-slip region, but increases gradually in a linear fashion.

From the results it was possible to determine guidelines with regards to lubricant selection under false brinelling conditions. It was found that oil is the better lubricant at both high and low slip amplitudes. The presence of silicon in the oil however results in wear rates very similar to the wear rates for grease.

The profile of the wear scar was also investigated. At right angles to the direction of the vibration the general shape of the profile was parabolic. In the direction of the vibration the scar was deepest at the point of maximum relative velocity. At high wear rates the scar had a very smooth profile, but at low wear rates the erratic profile shows signs of stick-slip.

The conclusions and recommendations that are presented include the following: Firstly that further research in the field of false brinelling be carried out. In particular the effect of different lubricants and additives must be investigated. Secondly, to reduce the amount of wear, oil be used as a lubricant in preference to grease. It is important, however to

ensure that the oil is free from contamination, as contaminants have an adverse effect on the wear rate. Thirdly, the slip amplitude must be restricted to as low a value as possible.

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GLOSSARYDefinitions

The following are definitions that are generally accepted in the literature, and that are used in this report.

- Slip Amplitude - The slip amplitude is the distance that the two surfaces move relative to each other in one cycle. The motion is sinusoidal, and the relative motion is measured from zero to peak.
- Polar Lubricant - A polar lubricant consists of fatty acids which are able to form a chemical attachment to a suitably reactive metal surface. The pole remains bonded to the metal surface, and thus ensures that a lubricant film is always present.
- Non-polar Lubricant - A non-polar lubricant is a lubricant that does not possess a pole that is attracted to the metal surface, and the lubricant film is determined by the relative motion of the two surfaces.
- Fretting - Fretting is the form of damage which arise when two surfaces in contact and nominally at rest with respect to each other, experience slight periodic relative motion.

Symbols

- a - acceleration ( $\text{ms}^{-2}$ )  
d - diameter (m)  
f - frequency (Hz)  
x - displacement (m)  
L - load (N)  
F - force (N)  
t - time (s)  
v - velocity (m/s)

Units

m	-	metres
mm	-	millimetres
um	-	micrometers ( $1\text{m} \times 10^{-6}$ )
N	-	newtons
kg	-	kilograms
m/s	-	metres per second (velocity)
m/s <sup>2</sup>	-	metres per second squared (acceleration)

## 1. INTRODUCTION

False brinelling is a form of wear that may be found in rolling element bearings. The resulting damage is very similar to the damage found in true brinelling, hence the name. The mechanics of damage are however very different. True brinelling is the result of plastic flow of the material due to high loads, but in false brinelling the load can be well below the limit for plastic flow. The wear in false brinelling is due to relative motion between the rollers and the race ways, when the bearing is nominally at rest.

False Brinelling can occur in the rolling element bearings of rotating equipment that is not running. This could mean that standby equipment, equipment stopped for maintenance [50] and equipment that is being transported are all susceptible to false brinelling. Operational equipment produces vibrations which causes relative motion between the rollers and the races of the stationary bearing. This relative motion sets up ideal conditions for fretting to take place [1,4], which results in damage to the bearing raceways.

The resultant damage **is** flutes in the bearing races at the points of contact between the races and the rollers. A coffee-coloured substance is found at the contact points. This substance is the oxidized wear debris [2,3,4]. The damage to the surface finish results in excessive running vibration [51], and thus premature bearing failure. There are documented cases of bearings failing as a result of false brinelling, with-in six days of installation [50].

There are a number of parameters which have an influence on false brinelling [1]. These may be classified as :

- (a) The bearing characteristics
- (b) the operating environment
- (c) the vibration dynamics.

The bearing characteristics which influence false brinelling can include the :

- 1) type of roller used
- 2) static load capacity
- 3) clearances within the bearing.

Parameters in the operating environment which affect the rate of false brinelling include the:

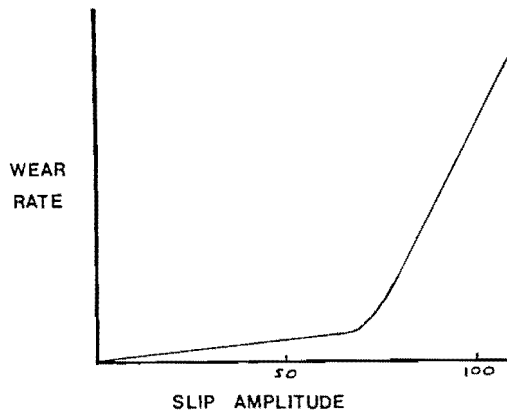
- 1) kind of lubricant (oil or grease, viscosity, additives etc.)
- 2) load on the bearing
- 3) ambient humidity
- 4) acidity.

The vibration parameters that affect the rate at which false brinelling occurs include the:

- 1) frequency
- 2) direction
- 3) slip amplitude.

For any given piece of equipment most of the parameters mentioned above are fixed and can not be changed by the user. Two parameters that can however be changed are the slip amplitude, and the lubricant. If the equipment user knows how these two parameters affect false brinelling, he may be in a position to alter them so as to reduce or eliminate the damage.

It has been accepted by most researchers [1] that false brinelling is a fretting corrosion phenomenon. It is thus possible to apply much of the information on fretting to false brinelling. Fretting corrosion is "the forms of damage which arise when two surfaces in contact and nominally at rest with respect to each other, experience slight relative periodic motion" [4].



**Figure 1** Wear Rate vs Slip Amplitude

The effect the slip amplitude plays in fretting has been investigated previously [1]. It has been found that a plot of the wear rate versus slip amplitude (fig.1) approximates a step function. This characteristic slip amplitude is dependant on the load, material, and the frequency [2].

In the present study tests were run by vibrating a cylindrical roller axially against a flat specimen under lubricated conditions. The slip amplitude and the type of lubricant were varied. The load was kept constant for all the tests.

Damage was quantified by means of surface profile and by lubricant debris analysis.

Certain samples were tested to determine if there was a relationship between the slip amplitude and the contact temperature.

The scope of this project was to investigate the rate of false brinelling as a function of the slip amplitude and the lubricant.

The objectives of this project were four-fold, as follows:

- 1) determine the relationship between the slip amplitude and the wear rate under lubricated conditions

- 2) determine whether false brinelling can be detected prior to it leading to bearing failure
- 3) determine the wear mechanisms present in false brinelling
- 4) draw conclusions as to how to reduce false brinelling.

This report begins with a literature review which discusses the present understanding of false brinelling, the basic wear mechanisms and techniques of lubricant debris analysis. The experimental plan is then discussed, high-lighting the assumptions that were made in the planning of the project. This is followed by a discussion of the test apparatus and procedures. The test results are reported, discussed and conclusions are drawn. From the conclusions a list of recommendations is developed.

## 2.LITERATURE REVIEW

The literature review deals firstly with false brinelling. False brinelling is however a fretting corrosion phenomenon, thus there is much to be learnt about false brinelling from the existing knowledge of fretting corrosion.

Fretting corrosion is made up of a combination of the basic wear mechanisms, which include adhesion, delamination, oxidation wear and abrasion. A brief discussion of the theories of these mechanisms is thus included.

Finally the relevant literature on lubricant debris analysis is presented and discussed.

### 2.1.FALSE BRINELLING

The term false brinelling was first used by Almen in a report for General Motors in 1937 [28]. Almen investigated the failure of front axle bearings of motor cars that were being transported by rail from Detroit to California. He found that the combination of the high stresses on the bearings due to overtight tie-downs and the small vibrational movements at the contact points between the balls and the raceways led to fretting corrosion.

Paland [28] outlined the parameters that affect the rate of false brinelling, but did not go into any in-depth studies of any of these parameters.

A study was made by Jordan [16] into the effect of the slip amplitude on the rate of false brinelling. Little was learnt from this report as apparatus limitations led to a limited range of slip amplitudes being investigated.

The present understanding of false brinelling is that it is a fretting corrosion phenomenon [28,16,42,49,41]. Scars called flutes develop at the contact points between the rolling elements and the bearing race-ways. At present the methods known for preventing or limiting false brinelling are to keep the equipment rotating with a barring motor, to reduce the load on the bearing as much as possible, or to limit the vibration.

## 2.2.FRETTING CORROSION

### 2.2.1.Characteristics

Fretting corrosion can be distinguished from other forms of wear, such as spalling and flaking, in a number of ways. The first of these is that fretting corrosion is the result of small oscillatory movements, normally smaller than 300um. Fretting corrosion of steel results in a debris which is normally iron oxide [49] of the form Alpha  $Fe_2O_3$ . This debris takes on a coffee-like appearance due to the dark colouring of the iron oxide.

### 2.2.2.Mechanisms of Fretting Corrosion

It is generally agreed that fretting corrosion develops in a number of stages. The number of stages, the length, and what happens in each stage is not however agreed upon, although the differences in the theories are minor.

Even though there is no agreement on the number, and how long the stages last, it is important to have some idea as to final stage has been reached. This in the decision as to the length that each test should be run. The results would be unrepresentative if the test was stopped before the final stage was reached.

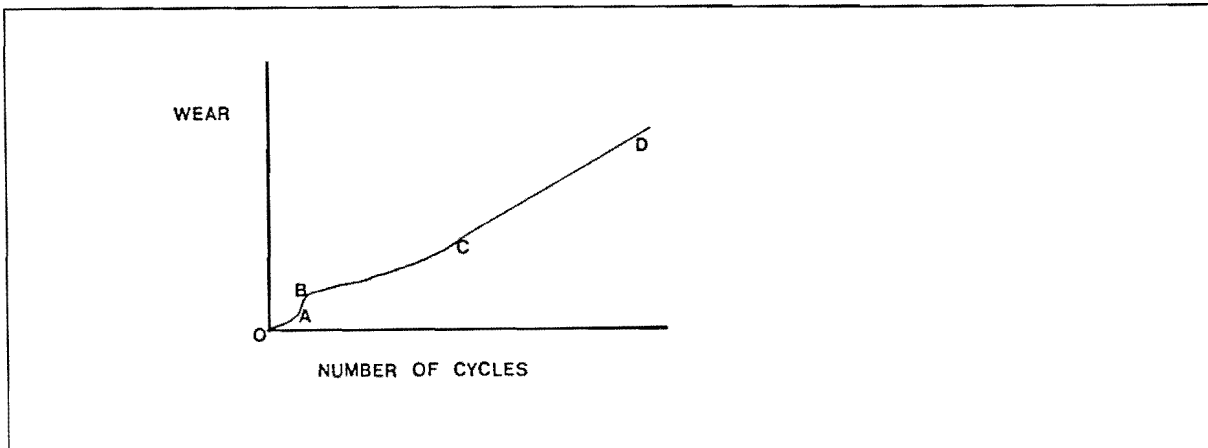
Stowers et al [43] give a very simple theory for the stages of fretting development. The authors forward a theory of two stages. The wear rate and coefficient of friction both start very high, but drop to a steady value. The time for the wear rate and the coefficient of friction to reach steady-state are consistent with each other.

Pugh [32] also forwards a very simplistic model. Fretting commences as adhesion, and if the debris is not removed from the contact zone, abrasion becomes the predominate wear mechanism. The amount of abrasion will be greater if the wear debris oxidizes, as it will then be harder than the parent material.

Aldham et al [1] believe there are three stages to fretting wear. The initial stage, which lasts for approximately 3000 cycles, involves plastic deformation, adhesive contact and material transfer, but does not result in significant metal loss or oxide formation. The intermediate stage, is complete after approximately 300 000 cycles. This stage is characterised by a fairly constant coefficient of friction, which is lower than in the initial stages. During this stage the authors believe that the metal is converted to oxide. The final or terminal stage is characterised by extensive beds of compacted oxide debris. The rate of wear, and the coefficient of friction both attain a constant value during this final stage, which are at their lowest for the test.

Colombie et al [6] found three stages of development. The initial stage begins with adhesion, and the surfaces are disturbed by plastic deformation. During the second stage the contact area becomes work hardened, brittle and prone to fragmentation. In the final stages the fragmented particles become detached, crushed and migrate over the contact zone. The particle migration is dependant on the particle size, the frequency, and the slip amplitude. Eventually a bed of oxidized wear particles is formed. The system is now at a steady state, with the wear rate being determined solely by the rate of particle ejection from the contact zone.

According to Sarkar [40] there are four stages to fretting (Fig. 2). During the first stage (OA) material is transferred from one surface to the other due to the adhesion of surface asperities. The work hardened debris abrades the metal surfaces in the second stage (A-B). This is followed by a third stage (B-C), during which the surfaces work harden, and thus the wear rate decreases. The final stage (C-D) is steady state, with debris generated at



**Figure 2.** Wear Rate vs Number of Cycles

a constant rate.

Four stage development is supported by Berthier et al [2]. During the first stage it appears that the first bodies (parent plates) are protected by a natural surface layer. This stage lasts for a few hundred cycles. Surface and subsurface damage begins in the second stage. The coefficient of friction also increases. The third stage of development sees the third-body debris developing. The third-body is the detached wear debris, which remains trapped in the contact area. The final stage is governed by third-body dynamics. Steady-state conditions are reached when an equilibrium between generation and elimination of third-body debris is reached.

Once fretting corrosion has proceeded through all its stages, as discussed above, it reaches a steady state condition. There are various theories as to what occurs during this "steady state" or terminal phase. Most researchers are agreed that the behaviour of the third body (wear particles) is important in determining the final wear rate.

Berthier et al [2] suggested that fretting wear is a source and sink problem. The parent material is the source, and the area into which the wear debris is thrown is the sink. Berthier et al suggest that the rate of wear is governed by the sink, and not the source. They believe that the smaller the sink the less the wear. Thus if the third-bodies stay in contact with the first body, wear will be reduced. The third body staying in contact with the first body achieves two objectives. Firstly, the third body acts as a velocity accommodation site. The first body is thus not exposed to the stresses that would be present if the third body was absent. Thus the third body particles act as a lubricant by separating the two surfaces. The second objective that the third body fulfils if it remains in the contact zone is that it fills all the available sinks. Thus if the wear rate is determined by the sinks, and not the source, then the lack of sinks will mean that the wear rate will drop accordingly.

Godet [13] theorises that the wear rate is determined by the dynamics of the third-body or debris. The wear particles get trapped in the contact zone, and the time that they stay trapped determines the amount of wear. The trapped particles separate the rubbing surfaces, and alter the interactions between the original bodies. It is obviously very difficult to predict the wear and friction, as an extensive knowledge of the rheological behaviour is needed. To complicate matters further the particles can adhere, roll or slip, plus they can change in size, shape and morphology. According to this theory the presence of wear particles in the contact zone of dry fretting will reduce the amount of wear, but if there is a lubricant present the debris will have a detrimental effect.

Pendlebury [30] also believes that the wear particles play a vital role in fretting wear. He found that the particles ejected after 10000 cycles were smaller than when they were originally artificially introduced at the beginning of the experiment. The particles were not however oxidized. After 100000 cycles however, most of the debris ejected from the contact zone was a red-brown oxide. Some of the particles were also magnetized due to the load and the motion.

Berthier et al [3] believe that most of the velocity accommodation is taken up by the third-bodies in the

interface, but that the relative displacements can also be accommodated in other ways. Firstly, the parent material can deform elastically. Secondly, the surface can develop little cracks perpendicular to the surface as a result of the high tension. These little columns, which are more flexible than the bulk material then deform elastically. A third option is for the surfaces to deform plastically, without any particles actually becoming detached. This final option is normally found in soft materials.

Lofficial et al [22] further extend the third-body theory for lubricated conditions. Tests confirmed the existence of a fluid load carrying mechanism. It was found in these test that the lubricant goes through three stages. Initially there is a thick viscous film which ensures minimal surface rubbing. The lubricant then starts to thicken with debris, and can not re-enter the contact zone. Finally the lubricant becomes a thick paste, and exhibits practically no lubricating properties.

The third-body plays a significant role in the rate of wear, and it will be necessary to discern what role the third body plays in false brinelling, in an effort to better understand the mechanism of false brinelling. It is also evident that wear of the parent material is not the only consequence of false brinelling. The surface finish can also be destroyed, by the formation of cracks, thus reducing the functionability, and increasing the running vibration.

### 2.2.3.The Effect of the Slip Amplitude

The slip amplitude can be considered as the most important parameter involved in determining the rate of wear in fretting corrosion according to Kayaba and Iwabuchi [17]. The rate of wear is the total amount of wear, however it be measured, divided by the amount of time over which the fretting occurred. The total distance that the first bodies move relative to each other is directly proportional to the amount of time if the motion is periodic. The wear rates in this investigation are all relative to the amount of time that the tests were run, unless otherwise stated.

Waterhouse [49] states that for an unlubricated steel cylinder against a steel flat, the damage is very slight for slip amplitudes below 75 $\mu$ m. When the slip amplitude reaches 100 $\mu$ m the volume of the wear increases dramatically. Waterhouse suggests that for movements below 50 $\mu$ m the velocity accommodation is taken up by elastic deformation of the parent metal. He does however, in a later publication [50] state that damage can be observed for slip amplitudes as low as 0.5 $\mu$ m.

Zhang et al [51] ran experiments using carbon steels (unlubricated), with loads of 100N. The test rig fretted a cylinder against a flat surface. The results that Zhang et al published were very similar to Waterhouse's. Wear was very slight below 70 $\mu$ m, but increased rapidly for slip amplitudes above 70 $\mu$ m. Zhang et al also found that the wear rate settled down to a constant rate for slip amplitudes above 300 $\mu$ m.

Kusner [19] found that the wear rate was not linear in relation to the slip amplitude. In tests conducted with magnesium against steel it was found that for slip amplitudes of less than 50 $\mu$ m the resultant debris was a white oxide, but for slip amplitudes of larger than 100 $\mu$ m the debris is a dark oxide. Kusner surmised that this could be due to a different wear mechanism being prominent at the different slip amplitudes. It is not known if the change in the wear rate with respect to the slip amplitude in carbon steels is the result of a change in the wear mechanism.

Kayaba and Iwabuchi [17] believe that the adhesive wear is most prominent at 70 $\mu$ m, and that it decreases with increasing slip amplitude. Below 70 $\mu$ m the authors believe that mild oxidation plays a significant role. Above 70 $\mu$ m the predominant wear mechanism is the abrasive action of the wear debris.

It was found by El-Sherbiny [8] that the slip amplitude at which the wear rate changes, the transition amplitude, is dependant on the load. It was found that the higher the load the smaller the transition amplitude and the more severe the increase in the wear rate.

Farrahi et al [11] explain the influence that slip amplitude has on the wear rate by using the third-body theory of wear. The authors explain that for low slip amplitudes the wear debris gets trapped in the contact zone. This results in an even load distribution, and limited wear. At higher slip amplitudes the debris gets ejected from the contact zone, and a situation very similar to continuous sliding arises.

Transition slip amplitudes have been reported by Bryggman and Soderberg [5]. Bryggman and Soderberg discovered two transition amplitudes, and thus three different slip regimes. The first regime is stick contact. The bulk displacement is taken up by elastic\plastic shear, and there is no relative slip at the interface. The second regime involves plastic deformation of the individual contact points at the centre of the contact zone. There are sliding wear marks at the outside of the wear scar. The final regime is that of gross slip over the entire wear scar. Gross slip is accompanied by a sudden drop in the tangential friction force.

Vingsbo and Soderberg [48] investigated fretting maps. They too discovered three wear regimes, and developed this theory further. The major addition made was in the "stick/slip" regime. It was found that the centre of the scar was the stick region, and that slip occurred at the extremities of the contact zone. At the stick-slip boundary extensive crack formation occurs. When the surface cracks intersect particles are released at the interface. The detached particles normally remain trapped between the contacting surfaces.

In the gross slip regime it was found that plastic surface shear resulted in the development of scales, which delaminated and became detached from the surface. These flakes are often oxidized, and crushed into smaller fragments.

Vincent et al [47] further support the two critical amplitude\three regime theory. According to the theory forwarded, the transition from one regime to the next regime is determined by the load, slip amplitude, frequency, the temperature and the environment.

There is not much agreement in the literature as to why there is a change in the wear rate with regards to the slip amplitude, or at what amplitude this occurs. It is however agreed that there is a change in the wear rate. The amplitude at which this occurs is dependant on a number of factors, but at present can not be predicted. The reason for the change of wear rate is not understood, but it is surmised by various authors [5, 11, 17, 19, 49] that the reason is either a change of wear mechanism or the result of a change in the dynamics of the wear debris.

### 2.3. Basic Wear Mechanisms

It has been surmised by certain researchers [17,19] that the change in the rate of false brinelling with respect to the slip amplitude is due to a change in the predominant wear mechanism. To be able to discern whether or not this is the case it is essential to be able to identify the different basic wear mechanisms. For this reason a literature review of the basic wear mechanisms is presented. A description of each of the four basic wear mechanisms follows, the theory of what occurs, and how the mechanism can be identified.

#### 2.3.1. Adhesion

The basic theory behind adhesive wear is that when two metal surfaces come into contact, asperity to asperity contact occurs. These junctions may weld together, but then due to the relative motion between the two surfaces these welds break and generate wear particles. The wear particles are spherical due to the nature of the asperity contacts [32].

According to Rabinowicz [36] there are three types of adhesive wear. The first is galling, which occurs between clean metal surfaces that are highly metallurgically compatible. The wear particles are normally between 20-200um in size.

The second type of adhesive wear is moderate wear. This occurs between less compatible materials that are lubricated. The size of the wear particles normally varies between 2um and 20um.

The least severe form of adhesion is burnishing. This is normally characterised by low loads and very good lubrication. There are no detectable debris particles.

An in-depth study of adhesion was made by Sargent [39]. It was found that surface contamination was the major barrier to adhesion. Adhesion was found to correlate well with properties such as hardness, elastic modulus, melting point, atomic size and surface energy. The correlation with these properties led to the assumption that adhesion is a function of the primary bonding energies of the atoms. If adhesion is a function of the atomic structure, then the electronic configuration of the material should dictate the amount of adhesion that occurs. This would account for the susceptibility of metals to wear in an adhesive manner, and metal oxides to be relatively immune.

Adhesive wear is very difficult to limit, as it has been found that it can not be eliminated by lubrication [38]. McClintock et al [23] agree that lubrication can not eradicate adhesive wear, but do concede that lubrication does reduce the amount of wear. McClintock estimates that the wear rate can be reduced by a factor of as much as 27000. This figure seems to be unrealistically high, and makes no allowances for the relative velocity, which would surely play a significant role in determining the thickness of the lubricant layer.

It is apparent that adhesive wear is present whenever two metal surfaces move relative to each other. The amount of adhesion that occurs is determined by the electronic configuration of the atomic structure, and can not be eliminated by lubricating the metals. The wear debris is normally spherical in shape, and ranges in size from being as small as the lower detection limits of lubricant analysis techniques, to particles in excess of 200um.

### 2.3.2. Delamination Wear

Delamination wear originates in the sub-surface layers of the material. The traction of the one surface over the other deforms the subsurface layer, generating dislocations and vacancies. This continued deformation leads to the formation of voids in the subsurface layer. Inclusions and large precipitate particles enhance void formation. The voids eventually coalesce, either by growth or shearing. Eventually a crack develops parallel to the surface, which shears at a critical length [37]. It is assumed that the metal wears layer by layer. The wear particles are sheet-like in morphology [7].

The delamination rate is controlled by the microstructure. For low wear the material must have few inclusions or voids, and be able to accommodate high flow stress.

Lim and Ashby [21] believe that the voids are found at a characteristic depth within the material. The depth of the voids is determined by the manufacturing processes that are used. The depth of the voids is typically of the same order of size as the asperities.

Jahanmir [15] also theorises that the surface and subsurface deformities influence the wear mechanism. The author theorises that delamination can lead to scuffing. Scuffing is normally found when two heavily loaded surfaces which are poorly lubricated have a high degree of relative sliding. This results in local welds, which are then torn apart by the relative motion [48]. This is because delamination causes contact conditions similar to those found in unlubricated wear.

To summarise, delamination wear is characterised by plate-like wear particles. It is initiated by subsurface defects, and the size of the wear particles is determined by the material being used.

### 2.3.3.Oxidative Wear

The oxidative theory of wear assumes that oxidation occurs at every contact between the two interacting metal surfaces in the presence of oxygen. The thickness of the oxidized layer builds up until it reaches a critical thickness, at which stage it becomes detached. Models have been forwarded that suggest that the oxide layer is removed continuously, but this is not supported in practise [18].

The way in which the oxide layer grows has been queried. One theory is that contacts have very little effect on the growth of the oxide. This theory has however been rejected by Quinn [33] and other researchers.

Quinn, Rawson and Sullivan [34] found that the wear rate for oxidational wear was proportional to the load, as long as the type of oxide produced did not change. The wear rate was found to be strongly related to the surface temperature. It was also found that the thickness of the wear particles was fairly constant in all the experiments.

In unlubricated wear the oxides are produced at the "hot-spot" temperature. In lubricated wear, results correlate with "hot-spot" temperatures at low oxidation temperatures, implying that the role of lubrication in fretting corrosion is to reduce the oxygen partial pressure [33].

Quinn et al [35] in further investigations found that fretting corrosion does not occur under lubricated conditions unless oxygen is present in the lubricant. There is however enough oxygen present in most oils to produce a significant reaction with the surface at the temperatures normally attained in the contact zone between rollers and raceways of rolling element bearings.

The effect of the oxygen content of oils and its effect on oxidation wear has been investigated by Newby et al [27]. Evidence was found that confirmed that reactions do take place between the oxidized lubricant and the metal. Some of the products of oil oxidation were found to have anti-wear properties especially when non-polar lubricants were used. Non-polar lubricants are lubricants that do not consist of

a chain with an active head that clings to one of surfaces [49]. For metals used in rolling element bearings a correlation was found between the amount of corrosion and the concentration of peroxide, but little correlation between the corrosion and the acidity. It was confirmed that the wear rate is proportional to the oxygen concentration in the lubricant. It was concluded that to reduce the amount of oxidation wear it is essential to restrict the amount of atmospheric oxygen, and to increase the oxidation resistance, add oxidation and peroxide decomposers to the lubricant.

Oxidation wear will have wear debris that is composed of oxidized platelets, all of a fairly similar thickness. A lubricant will reduce the amount of oxidation wear, but will definitely not prevent it. It is thus very likely that oxidative wear will be found to some degree or other in almost any situation where metal to metal contact occurs.

#### 2.3.4. Abrasive Wear

Abrasive wear is the wear that results when a particle ploughs into a surface. There are two types of abrasive wear, two body and three body abrasion. Two body abrasion occurs when particles rub against a surface. Three body abrasion involves hard particles between two mating surfaces.

The amount of abrasion has been found to be dependant on the difference in hardness of the parent material and the abrasive particles. It has however been shown that work hardening of a material does not aid in wear resistance. During the abrasion process the material is in its maximum work-hardened state, and thus preliminary work hardening has little or no effect [32].

It has been found that material can be removed by either a fracture process, or by plastic cutting and deformation [20]. The ductile-brittle transition is important in the abrasion of brittle and semi-brittle surfaces. The rate of removal of the material is higher for brittle scratching than for ductile cutting or scratching. A knowledge of the brittleness index is thus useful in predicting wear rate. The initiation of the scratches is also dependant on the geometry of the abrasive tip. A transition in the wear rate has also been discovered with respect to the scratching depth.

It has been found by Misra et al [24] that wear can occur even when the abrasive particles are softer than the metal. It was confirmed by Misra et al that work hardening had no effect on the wear resistance.

For lubricated contacts it has been found by Eve and Williams [10] that the wear rate is at a maximum when the lubricant film thickness is of the same order as the size of the abrasive particles. This means that to reduce the amount of wear it is advantageous to promote grooving in the surface, and thus increase the thickness of the lubricant film.

The wear particles produced in abrasive wear are spiral or ribbon shaped, similar in appearance to machining swarf [12]. These spiral shaped wear particles are normally the result of abrasive particles that are harder than the abraded surface.

## 2.4.Wear Debris Analysis

Wear debris analysis was used in this project as a part of the experimental procedure. Wear debris analysis can indicate much about the formation of the wear particles. It has been accepted that the study of wear can be greatly enhanced by studying the role of wear debris [43].

There are various methods of wear debris analysis, which differ in the method of collection of the debris, as well as the method of analysis. The three major methods of debris analysis are:

- (1) Ferrography
- (2) Spectrographic Oil Analysis Procedure (SOAP)
- (3) Filtergrams.

### 2.4.1.Ferrography

Ferrography uses the different settling times of the different sized particles to size the wear particles [44]. The downward velocity of the wear particle is inversely proportional to the size of the wear particle squared. Thus the smaller particles descend at much higher velocities than the larger particles. There is a chance that very small particles will be "missed" during the counting of the particles.

A disadvantage of ferrography is that an experienced technician is required, and the cost of the analysis is high. Even though there are a number of disadvantages involved in ferrography it has been proven to yield reliable results [45].

### 2.4.2.Spectrographic Oil Analysis Procedure (SOAP)

SOAP gives the elements present in the wear debris, but it can not detect if the wear debris is in the elemental form or a compound. This is a particular disadvantage as it does not allow for the oxidized wear particles to be differentiated from the parent material. SOAP does not give the particle size or shape, and can not detect particles larger than 10um. These limitations severely limit the amount of data that can be gathered for a machine that is deteriorating [14].

### 2.4.3.Filtergram

The oil sample is filtered through a membrane filter, the ferrous wear is then quantified by means of a debris analyzer. The debris analyzer merely gives a count of the number of particles in the sample, normally by passing the sample across a beam of light and counting the number of times the beam is broken, or by means of measuring the change in an induction field. The debris is then visually analyzed under a reflected light microscope normally at 100x magnification [12]. By using this method it is possible to find the total number of particles, the type of particle, the average particle size, the maximum particle size, and the contamination levels.

The procedure does not require a great level of expertise, and is fairly inexpensive. The greatest advantage of this system is the amount of information that it produces. The particle morphology, maximum size and the average size can tell an investigator much about the wear mechanism present.

### 3. EXPERIMENT DESIGN

Very little work has been done on the phenomenon of false brinelling in the past, and there is thus no accepted test apparatus or procedure. It was thus necessary to make a number of planning decisions, which strongly affected the design of the apparatus. Each of these decisions will be explained, and justified in this section.

#### 3.1. Roller

It was decided to simulate a cylindrical roller rather than a ball or taper roller. The reason for simulating a roller in preference to a ball is that the roller has fewer degrees of freedom, and is thus a simpler system to model. The roller is free to roll in one direction only, where as a ball is free to roll in any direction.

Taper rollers have only one degree of freedom, as do cylindrical rollers. The load pattern is however not as simple for a taper roller as for a cylindrical roller.

Cylindrical rollers are very similar to needle rollers, in both load distribution and in the number of degrees of freedom, thus the false brinelling behaviour of both systems would be very similar.

### 3.2.Direction of Vibration

For all experimentation the vibration was in the axial direction. It was felt that under these conditions that the greatest amount of wear would occur, and that the relative slip amplitude between the two surfaces would be the greatest [55].

It was felt that if the vibration was in the transverse direction the relative motion would not be as large. The roller could actually roll, and not slide. This would not result in a large amount of wear.

As the loads found in false brinelling are normally well below the static load rating of the bearing it was felt that the amount of wear that would result from vibrations in the vertical direction would be minimal. When the vibration is in this direction it would be expected that the loading would promote fatigue in the bearing, but this was not within the scope of this investigation, and was not investigated.

### 3.3.Outer Race

A flat outer race was used in the test rig. The reason this was done was to facilitate the use of a talysurf to measure the amount of wear. It was necessary that this be done to ensure the accuracy of the wear measurements, which is not possible for a curved surface.

The use of an outer race with an infinite radius has an effect on the contact stress. The roller used in the test rig is used in an N203 bearing. A comparison between the contact stresses of the N203 bearing and the test rig "bearing", with an infinite radius, reveals that the test rig bearing experiences 91% of the load that the N203 bearing experiences. This slight reduction in the load can be rectified by increasing the load applied. The use of a race with an infinite radius thus does not detract from the results obtained.

### 3.4.Measurement of Scar Profile

Scar profile measurements were made on the "outer race" only. No profile measurements were made of the rollers.

The first reason for this is the difficulty of taking accurate wear profiles from such a short surface. This was further complicated by the small radius of the roller which makes it very difficult to get a true reading from the surface.

Secondly, it was found in preliminary tests that the roller experiences very little wear, in comparison to the outer race. An attempt was made to measure the damage by means of mass loss. It was found that the amount of material removed was well below the resolution of any of the available mass meters, and thus insignificant in comparison to the amount of wear detected on the "outer race".

### 3.5.Load

The load that was applied to the roller simulated the most highly loaded roller in a complete bearing. The load that was decided upon was chosen to simulate a N203 bearing that had been sized according to dynamic considerations. The reason that the maximum loaded roller was simulated was to ensure that the worst case be investigated.

False brinelling is a totally different wear mechanism to true brinelling. True brinelling involves the plastic flow of the material. This plastic flow is due to loads that exceed the load carrying capacity of the material. False brinelling does not involve any flow of the material and is solely a wear mechanism. For this reason it was necessary to choose a load for the tests that was well below the plastic limit for the material. The load that was eventually chosen was a dead weight of 5,8 kg. This results in a contact stress of 285.01 MPa, which is well below the static load rating of a roller bearing, which is 4000MPa. As the load is so far below the value for plastic flow no plastic flow would be expected, and all damage would thus be the result of material removal.

#### 4. APPARATUS

It was necessary to build a test rig as there was not an existing rig that could be used to perform the necessary tests.

##### 4.1. Requirements of Test Rig

The rig was required to simulate a single cylindrical roller vibrating against the outer race of a rolling element bearing. The roller must be loaded, as would be the case in a stationary bearing. It must be possible to vary the load, frequency and slip amplitude.

As lubricant debris analysis is to be used, and a comparison is to be made of the effectiveness of different lubricants, the rig must have the capability to be lubricated by different lubricants. It must be possible to collect the used lubricant at the end of the test.

In order to quantify the damage a talysurf was to be used to measure the surface profile. For this reason it has to be possible to access the wear scar with a talysurf.

#### 4.2. Test Rig

The final test rig complied with all the requirements listed above.

A single roller was placed in a fixture, and the roller was then lowered against the "outer race". The outer race was placed in a bath that contained lubricant. The load was applied by means of mass pieces. To alter the load on the roller, mass pieces can be added or subtracted to the fixture holding the roller.

The vibration was generated by a shaker. The shaker was attached to the "outer race", and the roller was held stationary in the fixture, by two columns. The shaker is driven by an amplifier, which allows for the frequency and the slip amplitude to be altered.

In practice the vibration would be transmitted via the equipment's foundations, into the equipment's casing. The outer race would thus be vibrated via the casing. The inner race, which is on the shaft, would be held almost stationary by the inertia of the shaft. This is closely simulated in the rig.

The slip amplitude is measured by means of a proximity probe. The proximity probe is attached to the "outer race", and measures the slip amplitude relative to the fixture holding the roller. The signal from the proximity probe is fed into an oscilloscope, and the slip amplitude as well as the frequency can be read from the output of the oscilloscope.

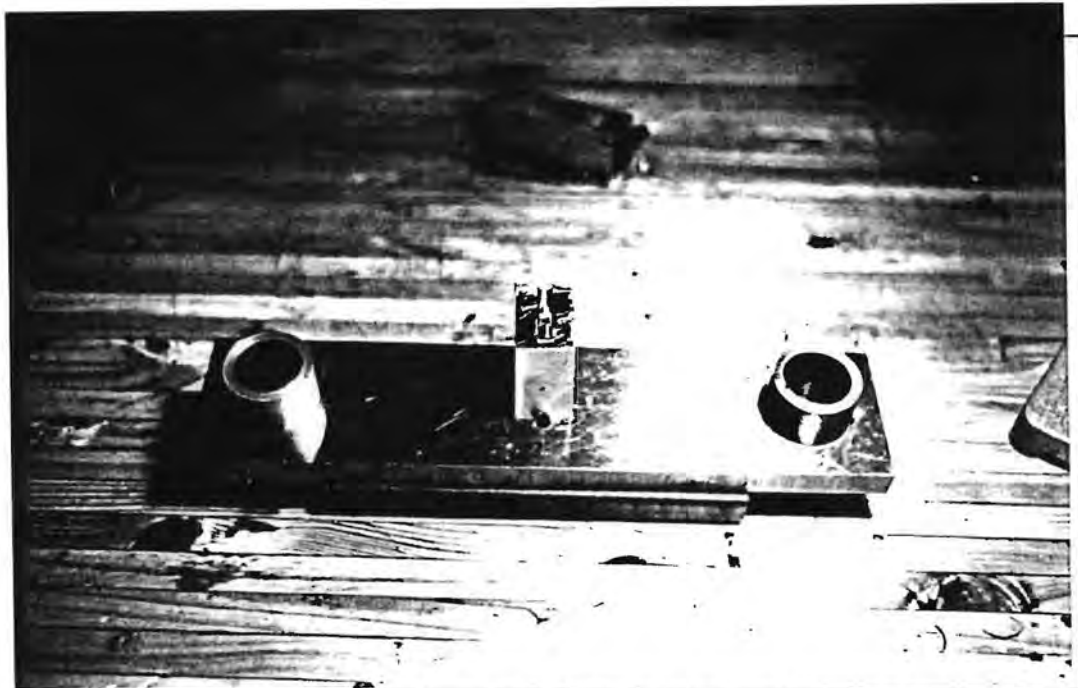


Figure 1 Roller Fixture with Mass Piece Attached

To make it possible to take lubricant samples the "outer race" was placed in a lubricant bath. To gain access to the bath the fixture which holds the roller can be removed. It is then possible to take the lubricant samples with a syringe. The "outer race" is clamped into the lubricant bath so as to hold it

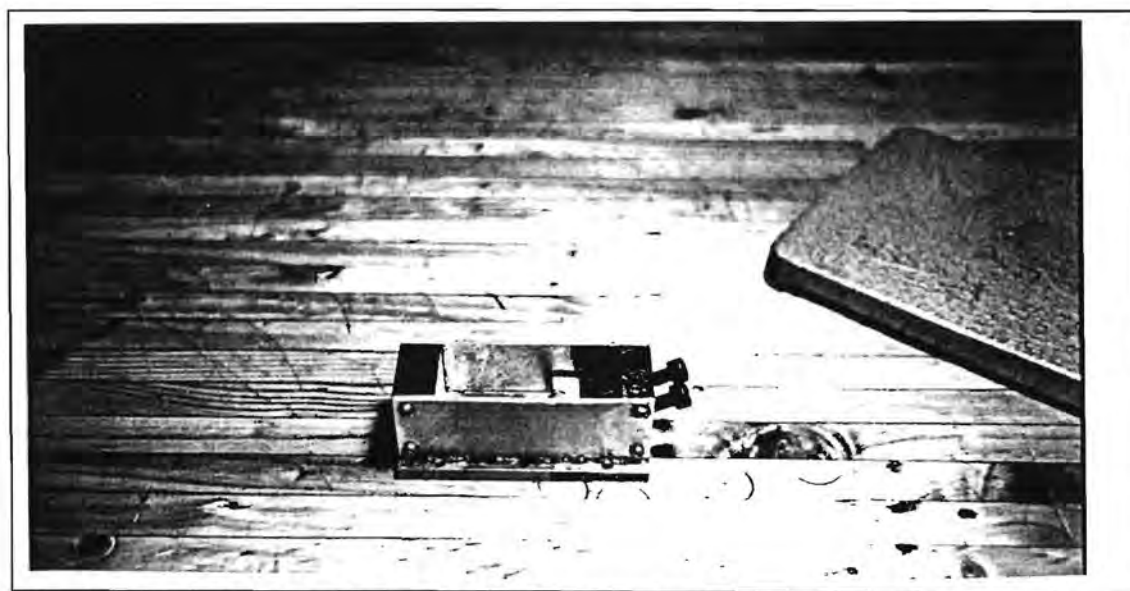


Fig 2 The Lubricant bath with the "outer race" in position

firmly in position (Fig 2).

Specially machined specimens were made to ensure accurate measurements could be made on the talysurf. The specimens were manufactured from the carbon chromium steel (1% carbon, 1.5% chromium) used in bearings, and then given the same heat treatment. It was thus possible to have flat "outer races" that only differed from the actual article in shape. The shape should have very little effect on the final results. The contact stresses using a flat outer race were found to be 91% of a bearing with a curved race. This is because the radius of curvature of the outer race is very much larger than that of the roller, and thus the Hertzian contact stresses will not be very different to those found in actual bearings.

#### 4.3.Rig Variables

It was possible to vary a number of the test parameters with the test rig. This is obviously a great advantage, and does not limit the test rig to this investigation alone. It will thus be possible to use this test rig in further studies of false brinelling.

The following variables can be altered:

- 1)Load- It is possible to alter the load by adding or subtracting mass pieces from the fixture that holds the roller.
- 2)Slip Amplitude- The slip amplitude can be altered by changing the gain setting on the shaker's amplifier.
- 3)Frequency -The shaker's amplifier has variable frequency settings, which can be set at different levels.
- 4)Time -The length of each test can be set on an electric timer that automatically switches off the shaker at the end of the test.
- 5)Lubricant -The lubricant in the bath can be changed, and it is thus possible to use

this rig for investigating the effect of different lubricant on false brinelling.

- 6) Specimens -Both the roller and the flat specimen can be removed, and it is thus possible to use the rig to investigate the resistance of different materials to false brinelling.

The test rig met all the prerequisites for a false brinelling rig. The assembled rig can be seen in figure 3. The drawings that were used for the construction of the test rig are displayed in Appendix 8.

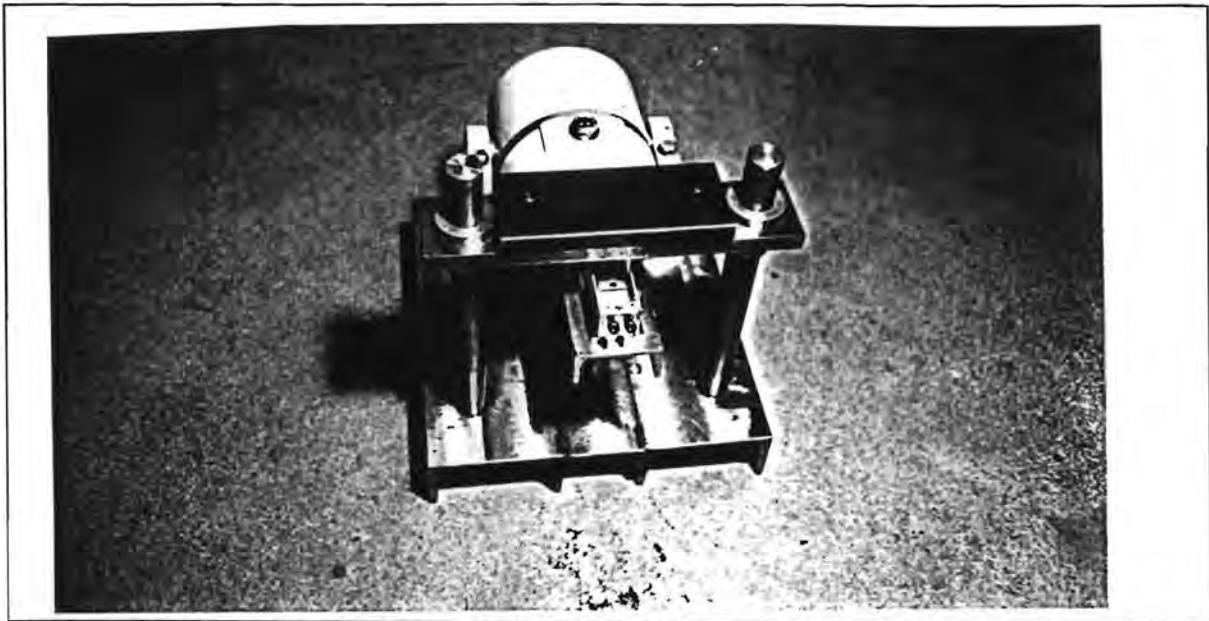


Figure 3 Assembled Test Rig

## 5. TEST PROCEDURE

A number of different tests were run to investigate the effect of oil and grease on the rate of false brinelling at low and high slip amplitudes. The temperatures in the contact zone were also measured, which involved a different procedure to that used in the other tests.

### 5.1. Wear Tests

The wear tests involved fretting the "outer race" against the roller. The following procedure was followed:-

- 1) The flat "outer race" was clamped into the lubricant bath.
- 2) The "outer race" was then submerged either in oil or grease.
- 3) The roller, in its fixture, was then lowered into the lubricant bath. The load was applied by attaching the mass pieces to the roller fixture.
- 4) The shaker was then turned on, and the slip amplitude was set. The "outer race" vibrated relative to the roller.
- 5) Every half hour the slip amplitude was recorded to ensure that it has not changed considerably.
- 6) Each test was run for six hours.

## 5.2. Surface Profile

The surface profile was used to quantify the amount of wear that had occurred. The profile was measured as follows:-

- 1) The "outer race" was removed from the lubricant bath.
- 2) The "outer race" was then washed with paraffin, to remove the lubricant.
- 3) The surface profile at right angles to the direction of the vibration, was recorded with the talysurf.
- 4) The surface profile was also recorded in the direction of the vibration. The trace that shows the most material removed was retained, and was used in conjunction with the trace from step 3, to find the total amount of material removed.

## 5.3. Lubricant Debris Analysis

The lubricant debris was studied extensively in this investigation. The analysis was performed by C.M.S. International, a company involved with condition monitoring which uses lubricant debris analysis amongst other methods. The advantage of using a company that is involved in lubricant analysis is that the tests that are conducted on the lubricant samples are standardised, and are thus comparable with results that would be found in industry.

The wear tests were run with a lubricant, either oil or grease, in the lubricant bath. The lubricant was placed in the bath at the beginning of the test, and was not topped up and no samples were taken until the end of the six hour test. At the end of the six hour test the entire contents of the lubricant bath were transferred to a sample jar. The sample jar was then sent to C.M.S. for analysis.

C.M.S. then conducted the following tests on the lubricant sample (42):

- 1) A measured amount of the sample was placed in a clean beaker, and was then diluted with a solvent.
- 2) The sample was drawn through a membrane filter.
- 3) The amount of ferrous wear was quantified by means of a debris analyzer such as the PQ2000.
- 4) The debris was then analyzed visually with a reflected light microscope at 100x magnification. The type of particle (relating to the mechanism of removal), the average size and the maximum size of the particle were quantified. The contamination index, a measure of the level of contamination, and the density index, a measure of the number of wear particles bigger than 5µm in a set volume of the sample, were also reported.

#### 5.4. Temperature Measurement

The effect that the different slip amplitudes had on the temperature of the "outer race" was investigated. The temperature probe was placed as close to the contact zone as possible. At first this was attempted by gluing the temperature probe on to the top surface of the "outer race", but this was found to be insensitive to the contact temperature, and more dependant on the ambient lubricant temperature. A hole was then drilled into the "outer race", and the probe then approached with-in 5mm of the contact zone from beneath the surface. This method was found to be more sensitive to the changes of temperature of the "outer race". The temperature tests were run in the following manner:-

- 1) The "outer race" was clamped into the lubricant bath, with the temperature probe in position.
- 2) The lubricant was then introduced into the bath.
- 3) The roller, in its fixture, was then lowered into the bath.
- 4) The shaker was started.
- 5) The slip amplitude was set at a particular level, and kept constant for 30 minutes.
- 6) At the end of the 30 minutes the temperature was recorded.
- 7) Steps 5 and 6 were repeated until the full range of slip amplitudes had been investigated.

## 6. RESULT

The results of the tests run in this investigation are reported in this section.

Firstly it must be pointed out that "no systematic study of wear particle size distribution versus slip amplitude appears to have been reported," (43). Thus there was no preliminary work on which to build, and the work done here is looking for basic trends, on which further work can be based.

### 6.1. Wear Rate vs Slip Amplitude

Graphs of the wear rate (total mass of the material removed divided by the time that the test ran) versus the slip amplitude are shown below. The units for the wear rate are milligrams per hour. The slip amplitude is the zero-to-peak value of the relative displacements. The slip is a sinusoidal function of time.

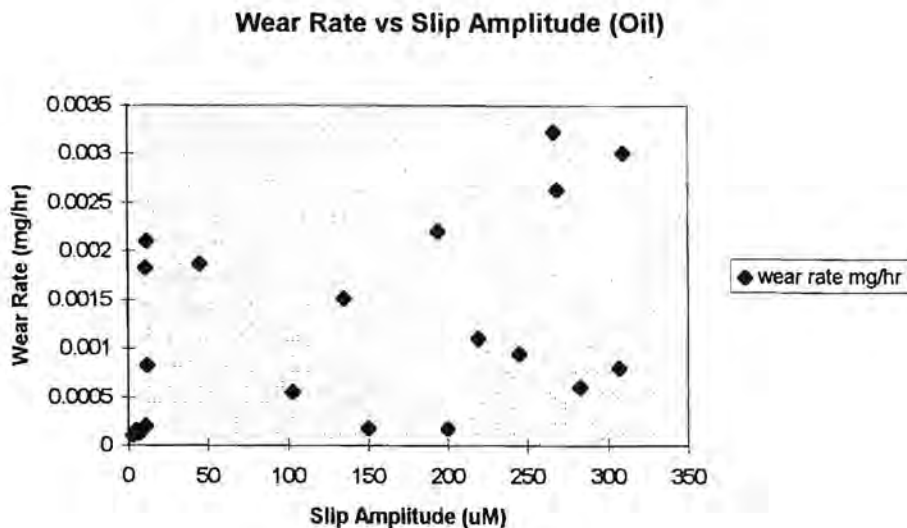


Figure 6 Wear Rate vs Slip Amplitude (Oil)

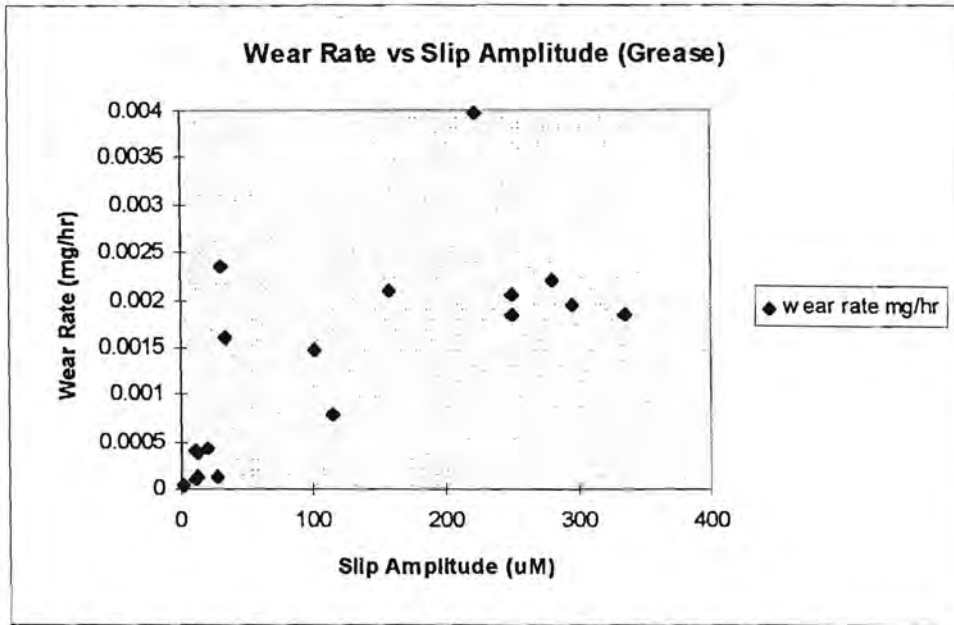


Fig. 7 Wear Rate vs Slip Amplitude (Grease)

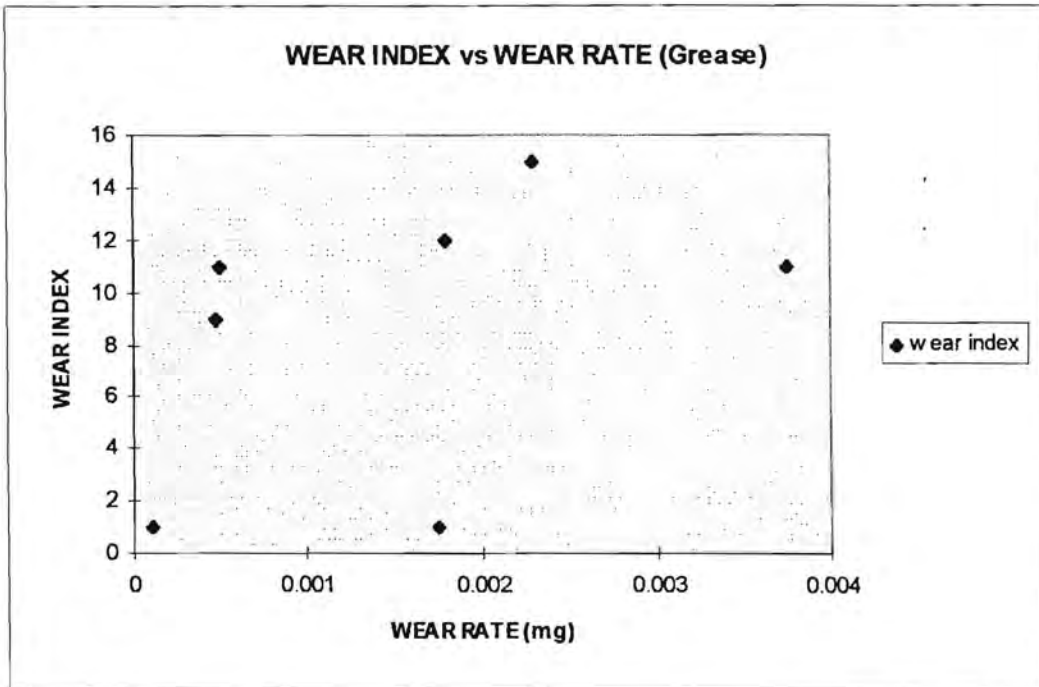


Fig 9 Wear Index vs Wear Rate (Grease)

### 6.3. Average Particle Size vs Wear Rate

The average size of the particle was represented by a code

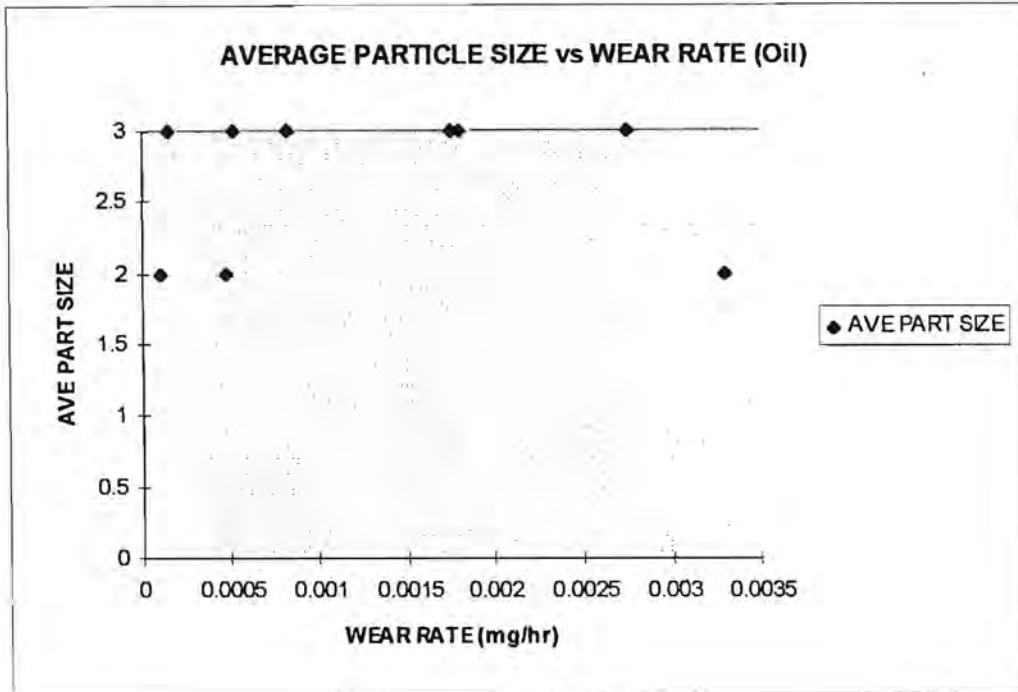


Fig 10 Average Size of Wear Particles vs Wear Rate (Oil)

number. The code number was proportional to the actual size

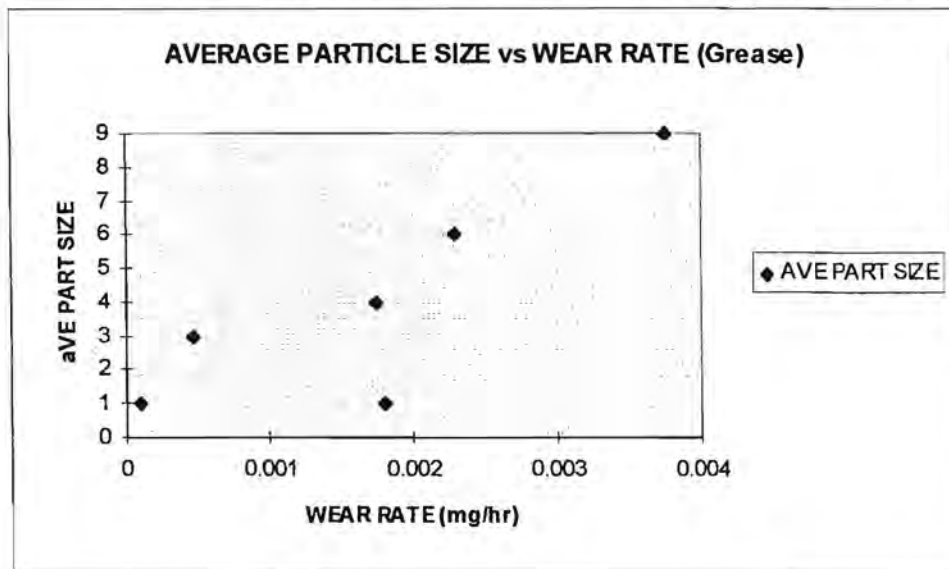


Fig 11 Average Size of Wear Particle vs Wear Rate (Grease)

of the particle (for real size of particle see Appendix 7).

#### 6.4. Maximum Particle Size vs Wear Rate

The maximum size of the wear particles was represented by codes. The code number was proportional to the maximum size

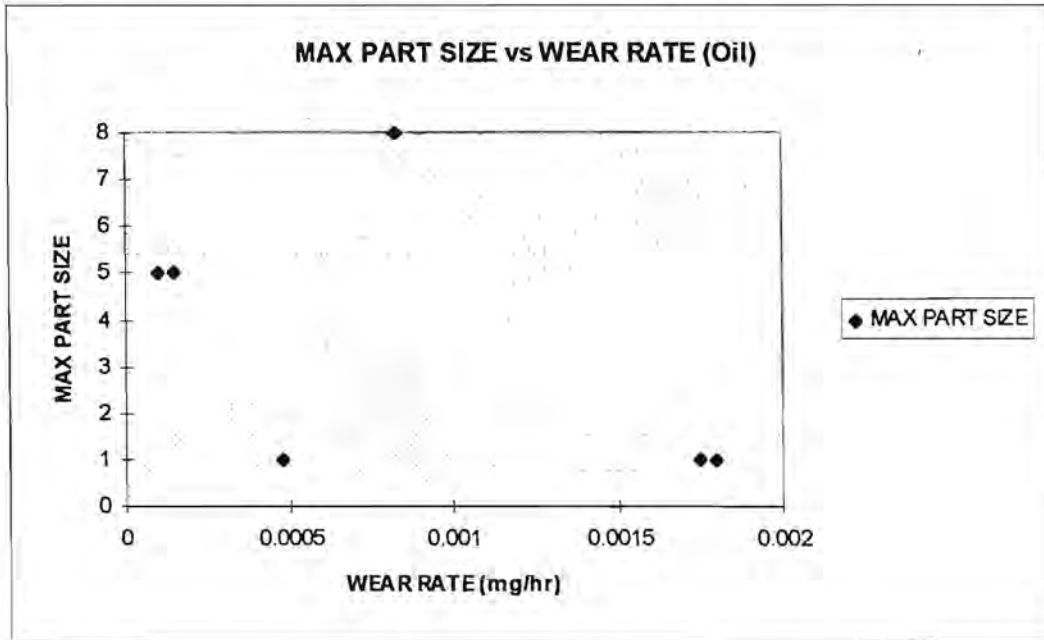


Fig 12 Maximum Particle Size vs Wear Rate (Oil)  
of the particle (for real size see Appendix 7 ).

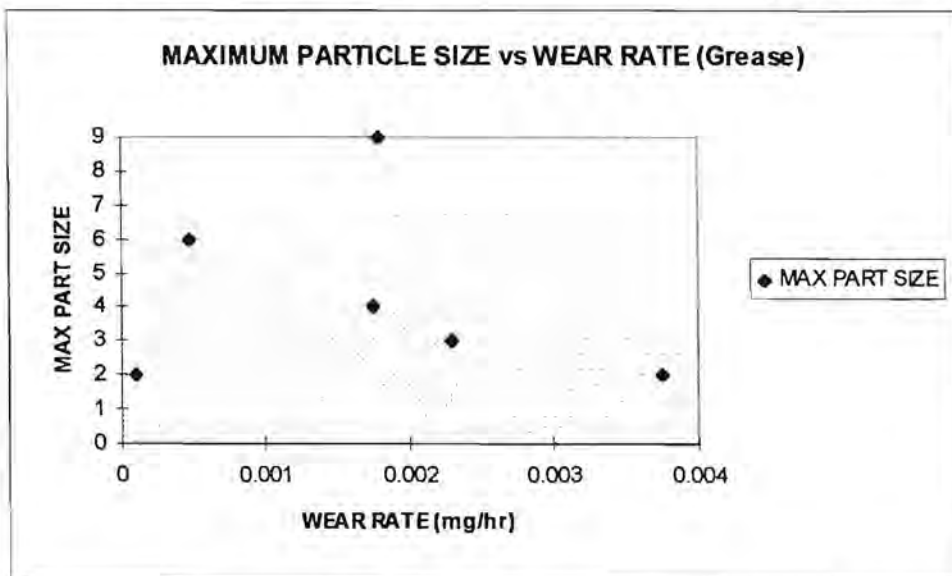


Fig 13 Maximum Size of Wear Particle vs Wear Rate (Grease)

### 6.5. Particle Morphology vs Wear

The particle morphology was the shape or form of the wear debris. The particle morphology can be used as an indication of the method of particle removal. The following different wear particles were distinguished:-1) Platelets

2) Chunks

3) Ribbons

4) Spheres

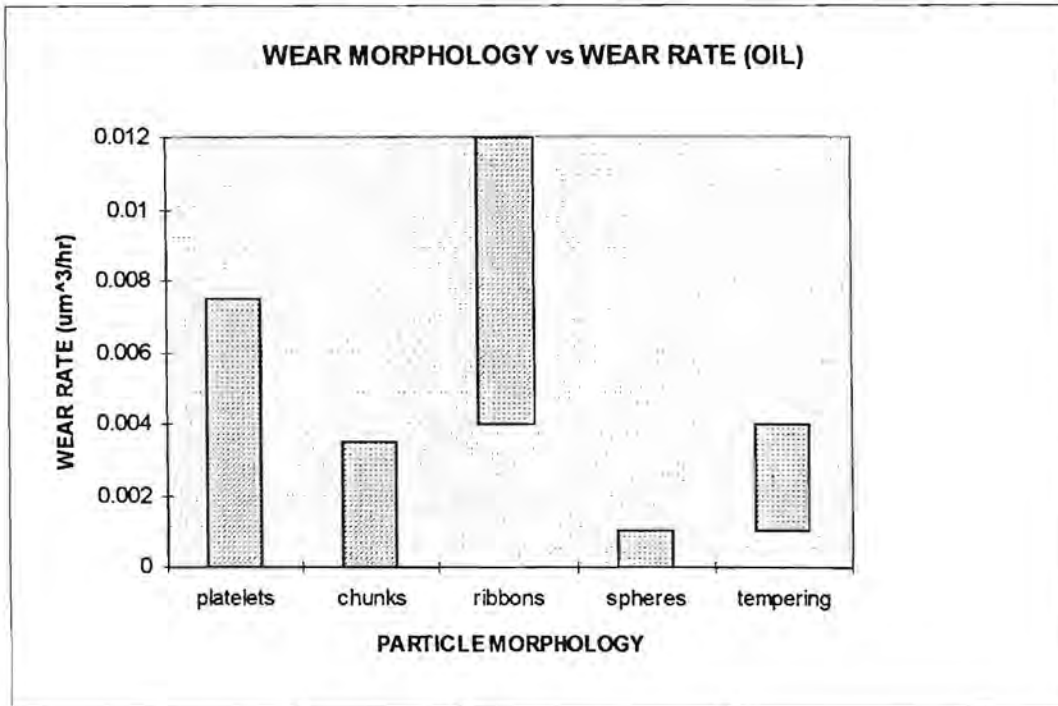


Fig. 14 Wear Particle Morphology vs Wear Rate

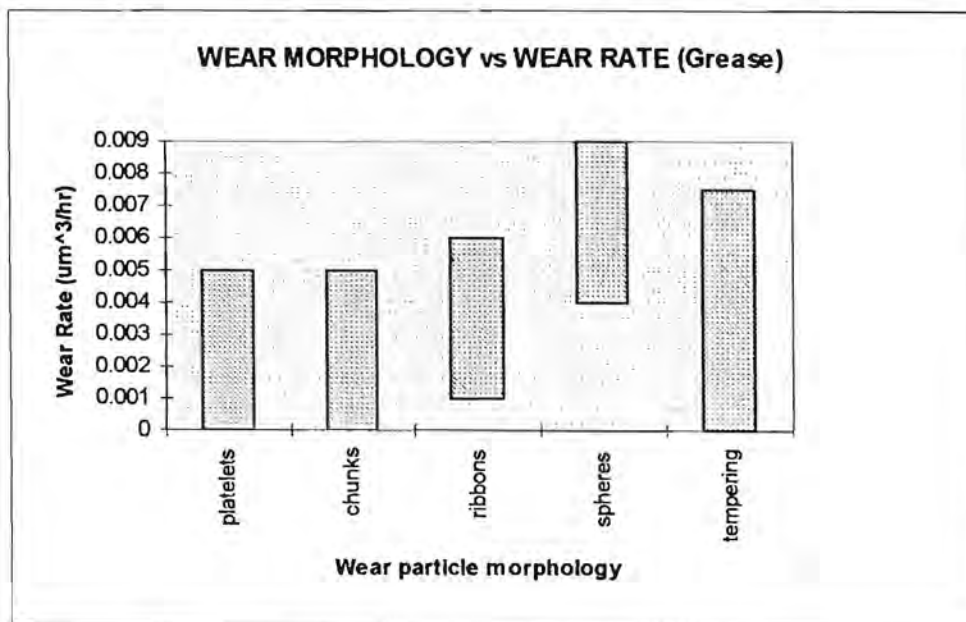


Fig. 15 Wear Rate vs Wear Particle Morphology (Grease)

### 6.6. Temperature vs Slip Amplitude

The temperature was measured at the contact zone, for both oil and grease, at a number of different slip amplitudes.

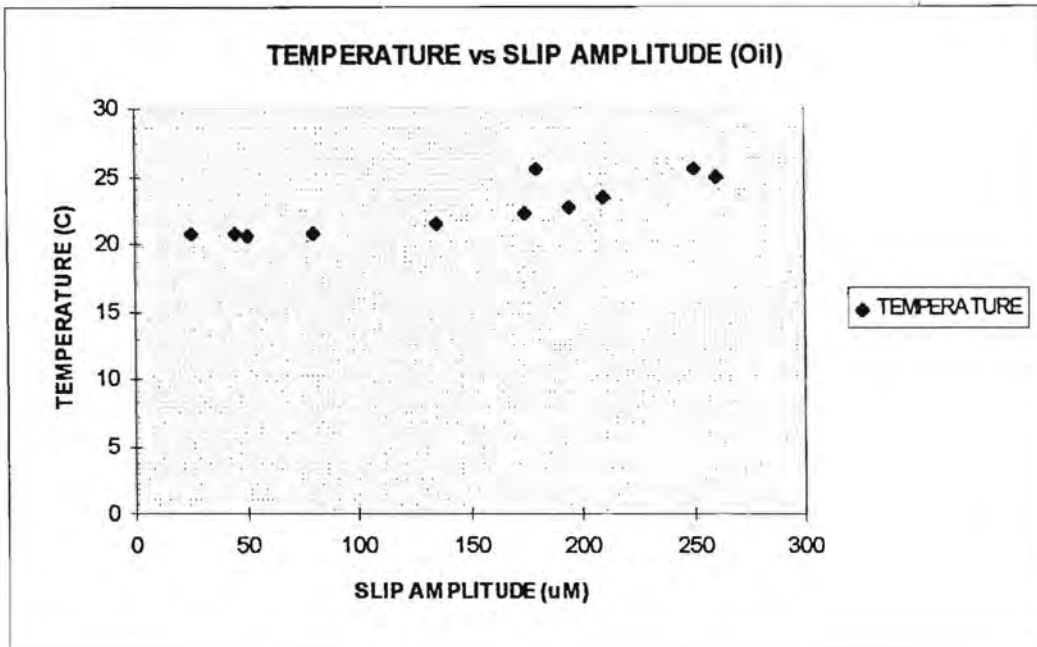


Fig. 16 Temperature vs Slip Amplitude (Oil)

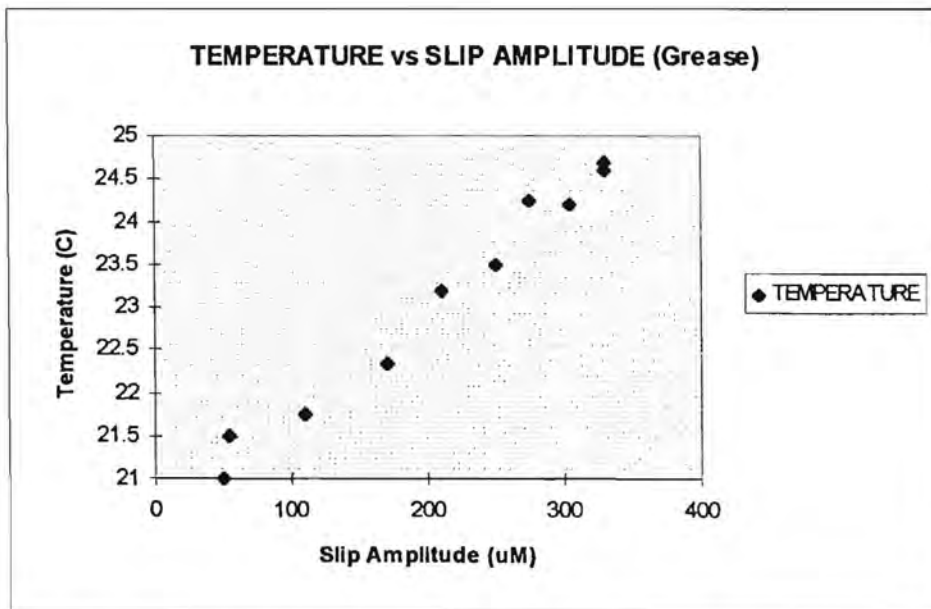


Fig. 17 Temperature vs Slip Amplitude (Grease)

### 6.7.Surface Profile

The surface profile was taken for each test. The general trends in the shape of the profile has been observed. It is very difficult to quantify a shape, thus the discussion of the surface profile will deal exclusively with visual observations.

## 7.DISCUSSION

The results achieved will be discussed in this section.

### 7.1.Slip Amplitude vs Wear

#### 7.1.1.Oil

The results suggest that there are two totally different wear mechanisms present, one operating at the low wear rates, and the other present at high wear rates (see Fig.18).

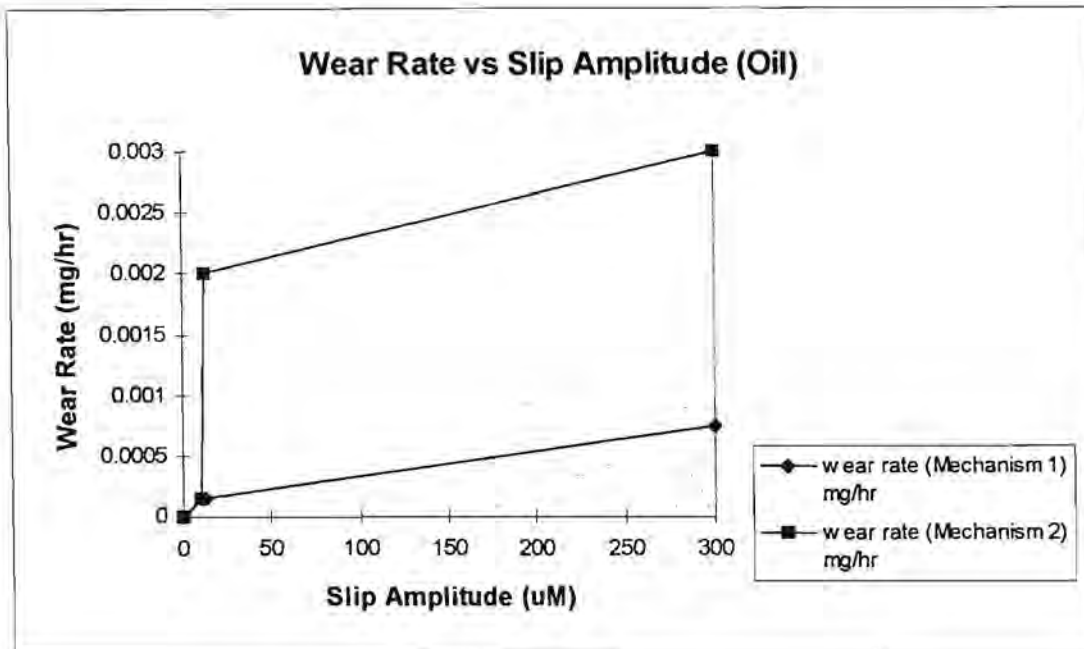


Fig. 18 Graph depicting the WearRate vs Slip Amplitude

At low slip amplitudes, smaller than 25uM, the wear rate was low. The wear particle morphology at these low slip amplitudes consists of platelets and chunky wear particles, which are both characteristic of fatigue wear mechanisms. It is surmised that the motion was accommodated by the elastic deformation of the contacting surfaces. Thus in this zone the vibration results in an alternating surface stress, which leads to subsurface crack growth. The crack

grows until it reaches a critical length or intersects with the surface, at which time the particle will be detached from the metal surface. This is consistent with the delamination theory of wear.

The maximum slip amplitude at which the motion can be accommodated by the elastic deformation of the two contacting surfaces is a characteristic of the test rig. The stiffness of the test rig would be expected to play a major role in the size of the region in which elastic accommodation occurs. For a stiff system the elastic accommodation region would be expected to be small, increasing in size with a drop in the system's stiffness. In other words the size of the elastically accommodated region is inversely proportional to the stiffness of the system. In practice the elastically accommodated region will be dependant on the particular piece of equipment which houses the rolling element bearing.

At a slip amplitude of approximately 25 $\mu$ m there was a bifurcation in the wear rate curve. The one branch, wear mechanism I, follows a gradual linear increase. The second branch approximates a step, initially a very rapid increase in the wear rate which progresses into a slow linear increase. This bifurcation in the wear curve marks the onset of relative motion and the end of the elastically accommodated region.

The first wear mechanism was characterised by low wear rates. The wear rate increased linearly with respect to the slip amplitude, but was relatively low over the entire range of slip amplitudes for the first wear mechanism, i.e. the curve displayed a low gradient.

Platelets and chunky wear particles were found in the oil samples that corresponded with the first wear mechanism. Platelets are normally the result of either delamination wear or oxidation wear. From the analysis done it is not possible to tell whether the particles are oxidized or not, which is the only way to distinguish between oxidation wear and delamination wear. (The wear particles are normally oxidized for oxidation wear, and are clean metal for

delamination wear). It is however surmised by the author that the platelets were the result of oxidation wear, as oxidation wear has previously been found in well lubricated systems.

The chunky wear particles were the result of a fatigue wear mechanism. The fretting motion sets up sub-surface stresses which alternate depending on the direction of motion. This alternating stress causes cracks to grow in the sub-surface layer in a fashion very similar to fatigue crack growth. These cracks grow until they intersect with each other, and result in wear particles. Although the chunks are the result of a fatigue wear mechanism there is relative motion between the contacting surfaces, unlike in the elastically accommodated region.

The second branch approximates a step function. The wear rate increases until it reaches approximately 0.000243 mg/hr. At this point the wear rate increases in a linear fashion with respect to the slip amplitude. During this linear phase a second wear mechanism, different to the first, is present.

Both wear mechanism one and mechanism two, are present over the entire range of slip amplitudes that were investigated. It is important to determine why some tests exhibit wear mechanism one and other tests exhibit wear mechanism two, even though the slip amplitude, and all other parameters, may be very similar for both tests. It would be advantageous to be able to limit false brinelling to the first wear mechanism, as the wear rate is considerably less than for wear mechanism II.

The onset of the second wear mechanism was determined by the level of contamination of the oil. All the tests corresponding to the second wear mechanism had high levels of silicon contamination (see Appendix 7 ). In practise it is difficult to keep the oil free from contamination, and this proved difficult in this investigation. The lubricant bath was not completely enclosed, and it was thus possible for dust and other sources of silicon to enter the oil sample.

Ribbon shaped wear particles, which are characteristic of abrasive wear, were found in the oil samples corresponding with wear mechanism two, and will be discussed more thoroughly in section 7.6.. This would suggest that the silicon particles act as abrasives. This abrasion results in a high wear rate which would be destructive to the rolling element bearing.

There were two samples that lay on the line that represents the first wear mechanism, which have high levels of silicon contamination, and would thus be expected to lie on the curve representing the second wear mechanism. The low wear rates and absence of ribbon shaped wear particles suggests that the silicon did not penetrate into the contact zone. The factors that determine whether or not silicon penetrates into the contact zone are not known, and are outside the scope of this investigation. Using sealed bearings would obviously prevent silicon contamination, and thus the onset of abrasive wear.

### 7.1.2. Grease

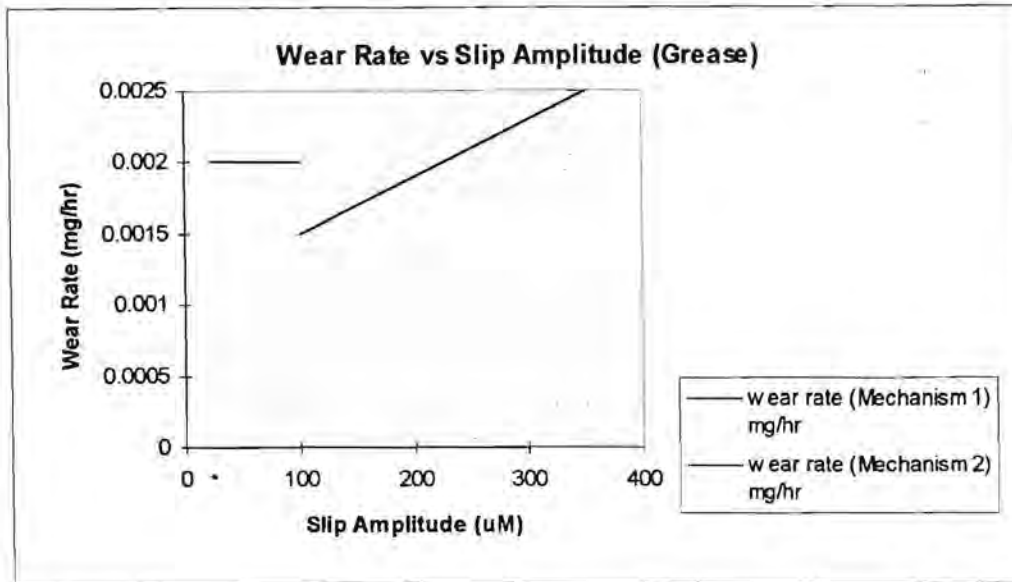


Fig. 19 Wear Rate vs Slip Amplitude (Grease)

The plot of the wear rate against the slip amplitude progresses through three definite stages for the tests run using grease.

For low slip amplitudes (smaller than 25um) the wear rate is low. At approximately 25um the wear rate increases as a step function. At approximately 0.000324mg/hr the rate of increase in the wear rate settles into a constant band until a slip amplitude of approximately 100um. From that point the wear rate drops to a value of approximately 0.000162 mg/hr, and increases in a slow linear fashion with respect to the slip amplitude.

For the first stage of the wear process, which occurs at low slip amplitudes (smaller than 25um) the relative motion is accommodated by the elastic deformation of the contacting surfaces. The elastic accommodation region for the grease tests corresponded closely to the elastically accommodated region for the oil tests. The wear particles at the low slip amplitudes are predominately chunks and platelets, which are characteristic of surface fatigue due to a changes in the surface stress field. The fact that the elastically accommodated regions for the oil and grease tests correspond so well is further proof that the motion is accommodated with out any sliding. If there was any relative sliding the lubricant film

would have an effect on the wear characteristics. The wear characteristics are however very similar, which leads to the deduction that there is no lubricant film, and thus no relative sliding.

The size of the region over which elastic accommodation occurs would be dependant on the test rig, and thus the equipment which houses the rolling element bearing. The region of elastic accommodation would be expected to increase as the stiffness of the bearing housing dropped, as discussed in section 7.1.1..

At slip amplitudes between 25um and 100um stick-slip contacting conditions would be expected. Fairly high wear rates are found at these slip amplitudes. Local welds would be expected which would be broken as soon as slip occurred. The wear particle morphology was not investigated at the slip amplitudes at which stick-slip contact conditions would be expected, thus it was not possible to confirm the existence of this region. It would be expected that the wear particles would exhibit signs of both sliding and fatigue wear mechanisms.

The third stage of the wear process is the slip stage, which begins at slip amplitudes of approximately 100um. At slip amplitudes above 100um the wear particles include spheres, the result of local overheating, and ribbons, which are the result of abrasion. For overheating and abrasion to occur relative motion is a prerequisite, which supports the hypothesis that slip occurs at this stage.

## 7.2.Wear Index vs Wear

The wear index is a count of the number of wear particles in a sample. It would thus be expected that the wear index would follow the amount of wear, if the size of the wear particles was constant for all the slip amplitudes that were tested.

### 7.2.1.Oil

It would be expected that the wear rate would be proportional to the product of the wear index and the average wear particle size. (The number of wear particles multiplied by the volume of the wear particles equals the total volume of the material removed).

The wear index of the test samples does not follow the wear rate, but stays at fairly constant level. The average particle size does not increase with respect to the wear rate either. This suggests that the wear index does not display sufficient resolution to display a change in the wear rate at such low wear rates.

According to the results there is a single point, at approximately 0.000162 mg/hr, which shows a wear index well above all the other points. The particle quantifier can not distinguish between the presence of a single large particle, or a number of smaller particles. The presence of a single large particle can thus give a false reading as to the amount of wear. The reading of the maximum particle size for that particular test reveals that a particle that was a lot larger than the rest of the particles was found. This suggests that the one large wear particle that was found in that particular test has resulted in a false reading being given as to the total amount of wear.

### 7.2.2. Grease

The results of the wear index versus the wear rate form two definite trends (see Figure 9 of the Results and Figure 20 below). The first trend is at the low wear rates associated with the elastic accommodation of the relative motion. The second trend is associated with the wear rates that represent the pure sliding. No lubricant samples were taken for the tests where the relative motion resulted in stick-slip contacting conditions.

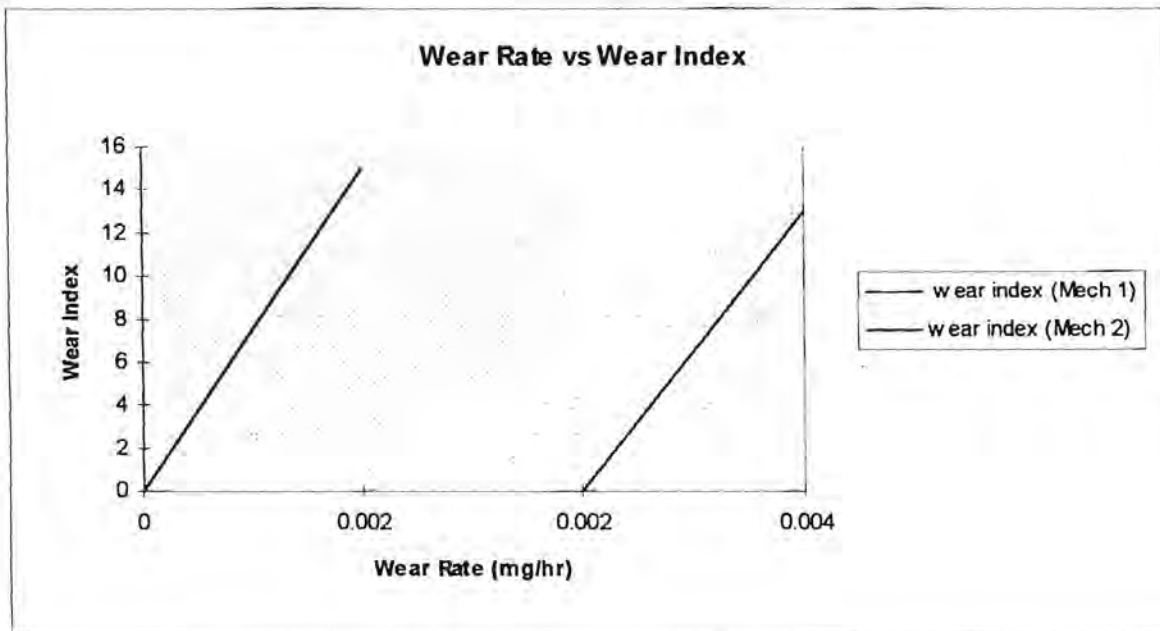


Fig. 20 Wear Rate vs Wear Index (Grease)

For the elastic accommodation of the relative motion the wear index increases fairly rapidly with respect to the wear rate. This is probably due to the presence of at least one very large wear particle in each of the lubricant samples. The presence of the large wear particles is recorded in the results relating to the maximum particle size. As reported earlier the particle quantifier can not distinguish between a single large wear particle and a number of small wear particles. The high wear index that is reported at these low wear rates is thus surmised to be due to the presence of a single or few large particles. A large

number of wear particles would not be expected at such low wear rates.

For the pure slip the wear index was found to increase linearly with respect to the wear rate. This suggests that the number of wear particles increases as the wear rate increases.

### 7.3.Average Particle Size vs Wear

An investigation of the average particle size should give an indication of which wear mechanism is present at any particular point, and give some information about the wear rate.

#### 7.3.1.Oil

The average particle size was found to be constant with respect to the total amount of wear for both of the wear mechanisms. The average size of the particles ranged in size from 5um to 20um (see Appendix 7). This is however misleading, as the size of the wear particles could well be a lot larger when they initially detach from the parent material. The particles do not leave the contact zone immediately, and are thus ground and crushed between the roller and the outer race, and are thus smaller when they are finally ejected.

The average size of the wear particle does not increase with respect to the wear rate, and the average particle size thus not give a good indication of the degree of false brinelling.

### 7.3.2. Grease

The average particle size displays two distinct trends, the first related to the elastically accommodated region, and the other associated with the pure sliding region (Figure 21). No lubricant samples were taken in the region which is associated with stick-slip. It is thus not possible to draw any

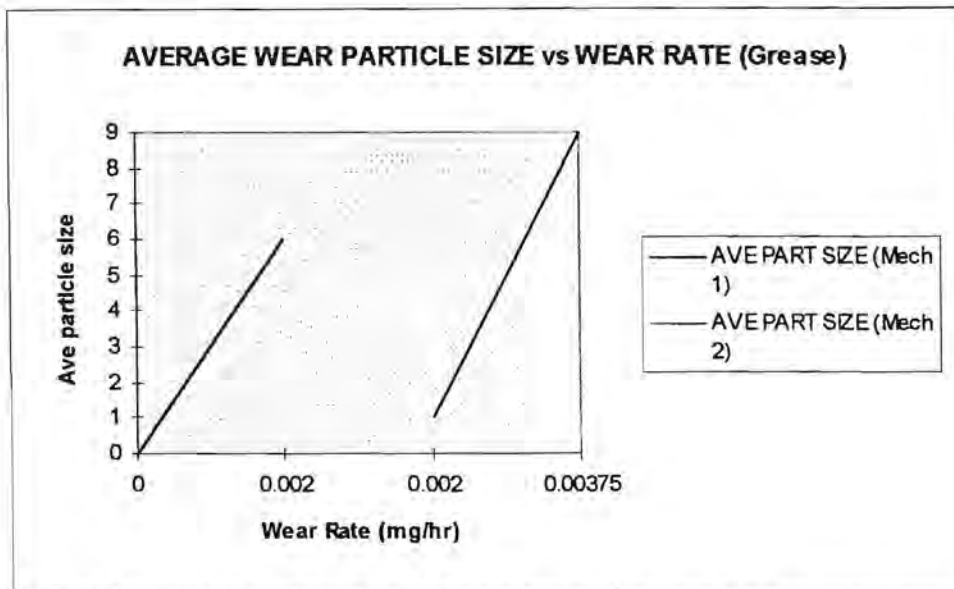


Fig.21 Average Particle Size vs Wear Rate

conclusions regarding this region.

In the elastically accommodated region the average size of the wear particles was found to increase linearly with respect to the wear rate. It would follow that the average size of the wear particles would have to increase with the wear rate if the wear index did not increase, as was the case. Thus the product of the wear index and the average particle size represents the total amount of wear.

As there is no relative motion in the elastically accommodated zone the detached wear particles will not be ground down in the contact zone after detaching from the parent material. It is thus possible to get an accurate measurement of the size of the particle when it detaches from the metal surface.

For the pure slip region, the average particle size increases linearly with respect to the wear rate. As with the argument forwarded above, the average size of the wear particles has to increase if the wear index remains constant, if the increase in the wear rate is to be recorded.

The increase in the average size of the wear particles is what the third body theory of wear, forwarded by Berthier et al, suggests. The increase in the wear rate in the slip region, is associated with an increase in the slip amplitude. As the slip amplitude increases the momentum of the wear particles increases, and the wear particles will spend less time in the contact zone before escaping. The wear particles will not be ground between the contacting surfaces as much as if they remained within the contact zone for an indefinite period.

#### 7.4. Maximum Particle Size

There is a good chance that the maximum size of the wear particles will not be a very good indicator of the amount of wear. This is because the wear particle is obviously at its largest when it is first detached from the surface. It will then be ground between the two surfaces, and will break up and decrease in size. The longer the wear particle stays in the contact zone the more chance there is of the particle being ground until it is a smaller particle. The maximum size of the wear particle will thus not be determined so much by the wear mechanism, but by the amount of time that the wear particle spends in the contact zone. According to the third body theory of fretting forwarded by Berthier et al (15) it would be expected that the time that the wear particle spends in the contact zone would be determined by the slip amplitude. The higher the slip amplitude, the higher the momentum and the less time the particle should spend in the contact zone.

### 7.4.1.Oil

The maximum size of the wear particles followed two different trends. These trends corresponded with the existence of the two different wear mechanisms that were found with the plots of the wear rate versus the slip amplitude. From Fig 22 it can be seen that a least squares fit line can be fitted through the points that correspond with the first wear mechanism,

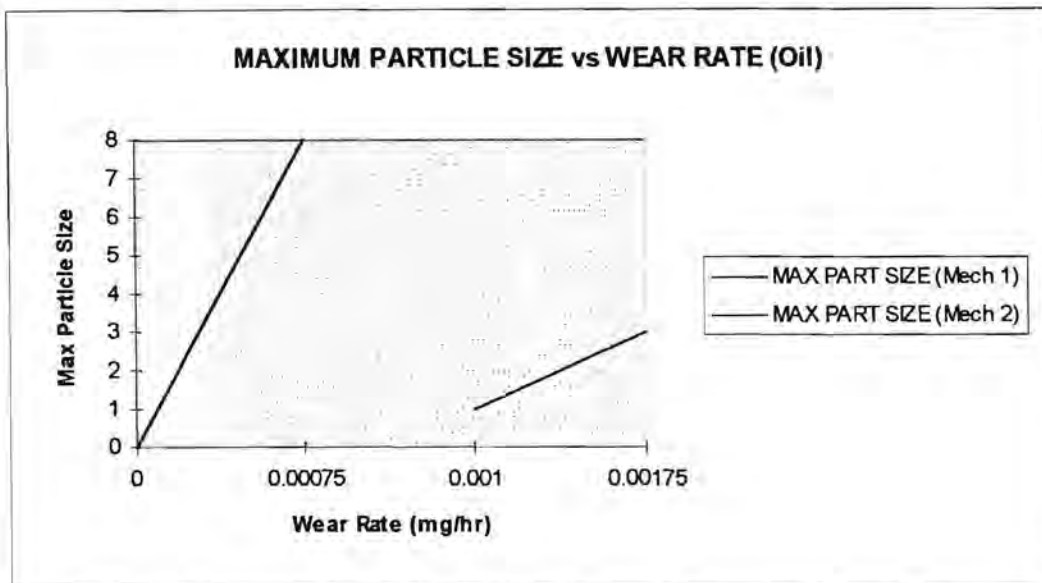


Fig.22 Maximum Particle Size vs Wear Rate (Oil)

and another line can be fitted through the points that correspond with the second wear mechanism. The size of the wear particles increases with an increase in the wear rate. As the slip amplitude increases for each separate wear mechanism the size of the wear particles that are ejected from the contact zone will increase. This is consistent with the models proposed by Colombie et al [6] which deal with the effect of the "third bodies" and "sinks". When the wear particles are ejected from the contact zone the wear "sinks" are emptied. According to the third body theory of wear the wear rate is determined by the sinks, and not the source material. Thus as particles are ejected, other particles are needed to fill the sinks, which results in a high wear rate. The presence of large wear particles at high slip amplitudes does not mean that the particles being detached from the surface are necessarily larger than the particles being detached at low slip amplitudes. This may be the case, but it can not be proved with-in the scope of this investigation.

The presence of large wear particles can thus be seen as a sign of a relatively high wear rate for the particular wear mechanism that is present at the time. To be able to tell what the actual wear rate is, it will be necessary to look at the wear particle morphology to determine which wear mechanism is present.

### 7.4.2. Grease

The plot of the maximum particle size versus the wear rate displays the presence of the elastically accommodated and the pure sliding regions. No lubricant samples were taken in the stick-slip zone, so no conclusions can be drawn about this region.

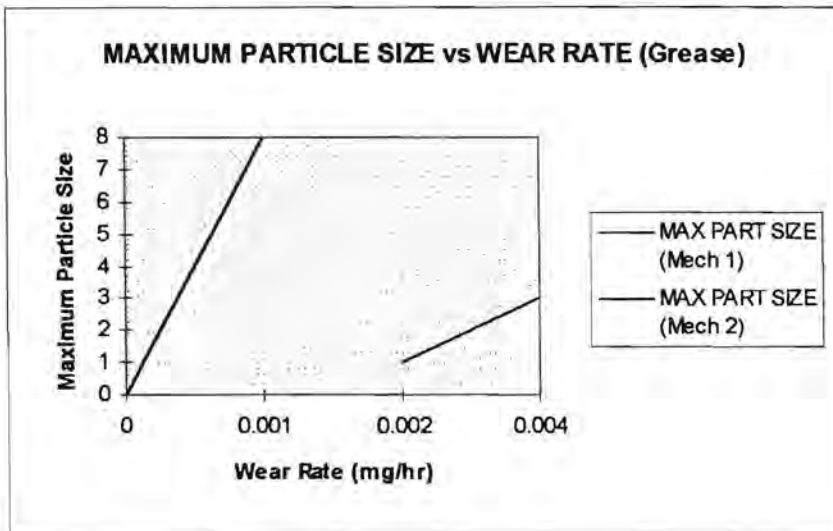


Fig.23 Maximum Particle Size vs Wear Rate (Grease)

The maximum particle size at the low wear rates, which corresponds to the region of elastic deformation, is large in relation to the true sliding region. In the elastic accommodation zone the entire contact area will experience fatigue conditions, which would result in a few large particles being produced. As there is no sliding the particles will not be ground in the contact zone, which will further ensure a large maximum particle size for this zone.

In the regions of pure sliding the maximum size of the wear particles is found to increase linearly with respect to the wear rate. This is in accordance with the third body theory of wear. The high wear rates correspond to high slip amplitudes, and at these high slip amplitudes the momentum of the wear particles

will be large, and it will thus be more likely that the wear particles will be able to escape from the contact zone. As the wear particles have more chance of escaping soon after their detachment for high slip amplitudes there will be less chance of the particles being ground into smaller particles.

### 7.5. Wear Particle Morphology

The morphology of the wear particles is determined by the wear mechanism, and what happens in the contact zone after the wear particle has been detached from the metal surface. A study of the wear particle morphology can thus reveal much about the wear process. Each of the basic wear mechanisms has a characteristic wear particle morphology. It should thus be possible to determine what basic wear mechanisms are present during false brinelling.

To support the third body theory of wear, no change in the wear particle morphology would be expected with respect to a change in the slip amplitude [15]. The change in the wear rate would be determined by the dynamics of the wear particles, and not the wear mechanism. There was however a change in the wear mechanism, which will be discussed below, which tends to dispute the theory that the wear rate is determined primarily by the wear particle dynamics. This does not however mean that the third body does not have an effect on the wear rate, only that it will be secondary.

#### 7.5.1. Oil

At the low slip amplitudes (smaller than 25 $\mu$ m) plate-like particles were found. These particles are very similar to the particles that result from delamination wear. As there is no relative motion it would be suspected that the plate-like particles are the result of a fatigue wear mechanism. This is because without any relative motion most other forms of wear will not be able to develop. It is thus surmised that the change in direction of the vibration will result in an alternating stress field around the contact zone. This alternating stress field results in crack growth, which will originate from inclusions and voids. The cracks will grow until they intersect with each other or with the surface. The platelets at these low slip amplitudes are thus produced by a very similar wear mechanism to the formation of the chunky wear particles.

Platelets were found in all the oil samples over the entire range of slip amplitudes that were tested, for wear mechanisms one and two. For sliding conditions

platelets are the result of either delamination wear, or oxidative wear. The wear particles that result from oxidative wear are plate-like in shape, and consist mainly of oxides, whereas delamination wear particles are clean metal. It was not possible from the analysis to tell whether the particle is an oxide or not, as the colour of the wear particle is not given. Oxidative wear is however present whenever there is metal to metal contact, even if there is lubrication, as reported by Sullivan (34), so it is likely that the platelets are due to oxidative wear.

At low wear rates, (smaller than 0.00182 mg/hr), which correspond to the first wear mechanism, chunky wear particles were present. The chunky wear particles, are the result of a fatigue mechanism. Delamination wear is the fatigue mechanism that is normally encountered in sliding, but the wear particles are normally thin. This is because the maximum shear stress is normally found just below the surface when one surface is sliding over another. The fatigue mechanism is accelerated if there are inclusions or voids in the subsurface. For the formation of chunky wear particles there must be either a number of inclusions or voids in the planes of failure, or else the stress distribution in the sub-surface layers is very different for fretting corrosion than to that found in sliding wear. It is very unlikely that voids and inclusions will be found in the same planes in a number of different samples, especially in a material with the purity of bearing steels. It would thus appear that the chunky wear particles are not the result of material defects. It can thus be assumed that the stress fields in the sub-surface layers for fretting are very different to those found in sliding wear, and for this reason it is possible for chunky wear particles to be formed.

At high wear rates, which correspond to the second wear mechanism, ribbon shaped wear particles were found in the oil sample.

Ribbon wear particles are normally the result of abrasion. The abrasive particles in this test were the silicon particles that were in the oil samples. The

silicon particles were harder than the metal surface, which is attested to by the ribbon shape. If the silicon particles were softer than the metal surface there would still be signs of wear, only ribbons would not be expected.

The situation that exists with false brinelling is a third body abrasive wear problem. The wear particle morphology of third body abrasion does not normally consist of ribbons, as the abrasive particles do not normally plough into the metal surface, but roll and occasionally slide, forming short wear scars. For the ribbon-shaped particles to be formed the abrasive particles will have to become embedded in one of the surfaces, and will then be able to plough into the other surface. This ploughing action will result in the ribbon shaped wear particles.

From a condition monitoring point of view much can be learnt from the particle morphology. Platelets can be expected for all slip amplitudes. For the first wear mechanism chunky wear particles will be present in the oil sample. For samples taken from tests that were predominantly of the second wear mechanism ribbon shaped particles would be expected.

Thus in summary, at low slip amplitudes (smaller than 25um) chunky wear particles and platelets were found in the oil samples, and it is suspected that these particles are the result of fatigue wear mechanisms. For wear mechanism one the wear particles are platelets and chunks, and it is suspected that the wear mechanisms present are oxidative wear which results in the platelets, and a fatigue wear mechanism which results in the chunks.

The second wear mechanism is characterised by ribbon shaped wear particles, the result of abrasion due to silicon contamination in the oil sample. Platelets, the result of oxidative wear, were also present.

### 7.5.2.Grease

At the wear rates that correspond with the elastic accommodation of the motion the wear particles consist of platelets and chunks. Both of these wear particle morphologies can be the result of a fatigue wear mechanism. The vibration would result in an alternating stress that would lead to sub-surface crack growth. These cracks would grow until they reach a critical length at which time the particles would detach from the metal surface.

No lubricant samples were taken in the stick-slip region, which occurs between 25um and 100um, so there is no wear particle morphology data for the stick-slip region.

The onset of relative motion, which occurs at a slip amplitude of approximately 100um, is marked by the onset of ribbon shaped wear particles. Ribbon shaped particles are characteristic of abrasive wear, which has as a prerequisite relative motion. Unlike the oil samples, abrasion is not associated with high silicon contamination levels. The abrasive particles must then be third bodies, or detached wear particles, that are in the contact zone. The other wear particles that are present in this range of slip amplitudes are chunks, the result of a fatigue wear mechanism, and platelets. The platelets are either the result of delamination wear or of oxidative wear.

For ribbon shaped particles to occur the abrasive particles will have to be harder than the metal surface. In previous investigations on abrasion it was found that work hardening of the material did not affect the rate of wear. This is because the material work hardens to its maximum hardness during the process of abrasion. Thus in this investigation it is unlikely that the abrasive particles are composed solely of the parent metal, as the abrasive particles would not be any harder than the metal surface. If the platelets are the result of oxidative wear it would then be possible for them, as an oxide, to be harder

than the metal surface, and abrasive wear could thus occur. This supports the supposition that the platelets are the result of oxidative wear.

At high wear rates, from approximately 0.000325 mg/hr to 0.000405 mg/hr, spherical wear particles and signs of tempering were found. Both are the result of local overheating. At high slip amplitudes it appears that grease is not able to dissipate the heat that is generated by the relative motion. An attempt was made to measure the contact temperature, but no marked increase was recorded. This suggests that the heating is localized to the micro-contacts of the parent material.

At very high wear rates (0.003645 mg) signs of sliding wear were found. This gives an indication of the threshold value of the slip amplitude between sliding and fretting. It is important to note that the wear rate for sliding wear is higher than that found in fretting. It is thus important to keep the slip amplitudes low, in order to prevent the system from becoming a sliding wear problem.

Thus in summary, the elastically accommodated region is characterised by platelets and chunks, the result of fatigue wear mechanisms. The onset of sliding is signalled by the appearance of ribbon shaped particles, a characteristic of abrasion. At higher slip amplitudes, and thus wear rates, spherical wear particles and tempering are present, which are a sign of local overheating.

### 7.6. Comparison of Oil and Grease

A comparison of the effect that the type of lubricant has on the rate of false brinelling is important when it comes to selecting a lubricant for a bearing that is likely to experience false brinelling.

At very low slip amplitudes, smaller than 25um, the wear rate is very similar for oil and for grease. The motion is accommodated by the elastic deformation of the two contacting surfaces, and the lubricant will have very little effect on the wear rate.

For the first wear mechanism in the oil tests, i.e. the oil sample is free from major contamination, the wear rate is well below the wear rates at comparable slip amplitudes for the grease tests.

At slip amplitudes between 100um and 200um the wear rate for the contaminated oil samples was higher than for the grease samples.

It would thus appear that as long as the oil is free from contamination, oil is a better lubricant for false brinelling conditions than grease is. Oil is however more susceptible to contamination than grease is, as any contaminants that fall into oil will go into suspension. Grease, due to its high viscosity, does not allow dust particles etc. to go into suspension easily. The potentially abrasive particles are thus kept distant from the contacting components if grease is used, which will limit the amount of wear.

### 7.7. Temperature vs Slip Amplitude

In the section dealing with the wear particle morphology it was found that grease could not dissipate the heat generated by the relative motion. It would thus be expected that an increase in the temperature of the contact zone could be recorded. A hole was bored into the specimen to within 5mm of the contact zone. A temperature probe was then placed in the hole in an attempt to read the temperature of the contact zone. If the change in the wear mechanism results in a change in the heat generated it would be expected that it could be detected.

#### 7.7.1. Oil

A linear increase was found between the temperature and the slip amplitude. There was no change in the gradient of the plot of the temperature versus the slip amplitude, which suggests that the change from the elastic accommodation to the first or second wear mechanism does not result in a measurable change in the contact temperature. The wear particle morphology did not indicate a large temperature rise either.

The gradual increase in the measured temperature would thus appear to be the result of the increase in the relative velocity between the two surfaces. The temperature increases by less than 5 degrees with an increase of slip amplitude of 240um. With such a small change in the measured temperature with respect to the slip amplitude it would be very difficult to detect the level of false brinelling by monitoring the contact temperature.

### 7.7.2.Grease

The increase in the temperature with respect to the slip amplitude was measured as a linear relationship. The temperatures measured were found to be very low, below 25°C (degrees Celsius). The presence of spherical wear particles and signs of tempering however shows that there were very high temperatures in the contact zone. From these two apparently contradictory results it can be assumed that the temperature effects are very localised. The high temperatures would be expected at the micro-contact points, and not over the entire Hertzian contact zone.

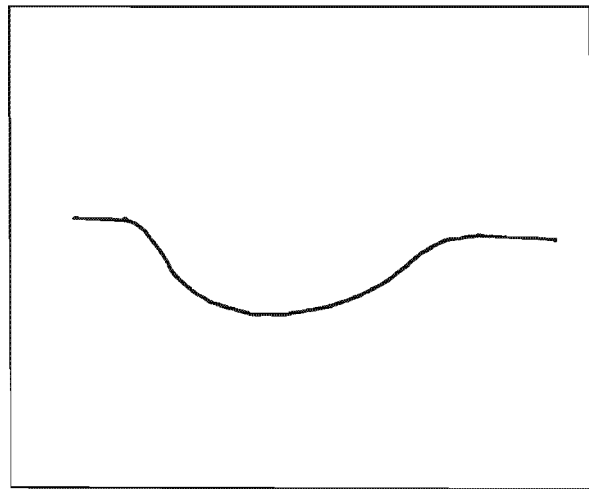
The temperature probe was within 5mm of the contact zone. This is closer than could be achieved in most bearings without causing damage. It would thus be difficult to get more accurate temperature measurements from a bearing in operation. Measuring the temperature would thus not provide a means of detecting the onset of false brinelling.

### 7.8. Surface Profile

The profile of the wear scar was measured as a means of calculating the amount of wear. Visual observations were made of the wear profiles, in an attempt to learn more about the mechanism of false brinelling.

It was found that the wear profiles were very similar for oil and for grease, and thus all observations apply to both lubricants.

The traces taken perpendicular to the direction of the vibration (figure 24) were found to be parabolic in shape. The depth of the scar was roughly proportional to the slip amplitude.



The width of the scar was also found to increase slightly with an increase in the slip amplitude. This is because the area of contact between the roller and the "outer race" will increase as more of the roller beds into the "outer race".

Figure 24 Wear Profile  
Perpendicular to Vibration

The profile of the scar in the direction of the vibration was also found to be the same for oil and grease. The scar was at its deepest in the middle of its length, for high and low slip amplitudes as shown in figure 25. The middle of the scar is the area that will experience the highest relative velocity. The cyclic motion associated with fretting means that the centre of the wear scar is constantly loaded, while the extremities of the scar are only loaded when the roller is at the ends of its path. Thus the combination of the relative velocity and the constant load results in the centre of the scar wearing more than the extremities.

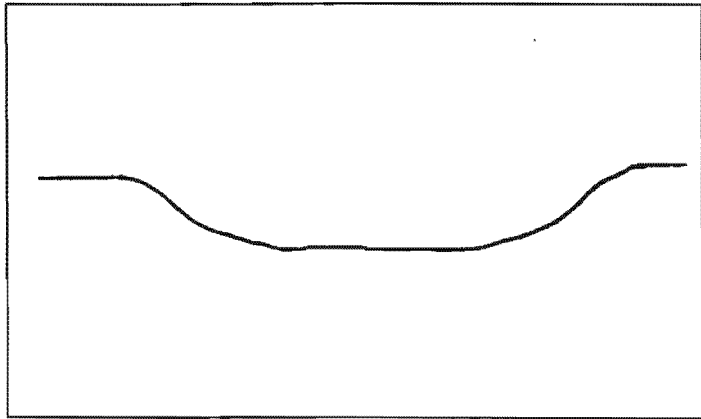


Figure 25 Wear Profile in Direction of Vibration

The profile of the scar was far more irregular for small slip amplitudes ( $<25\mu\text{m}$ ), than for high slip amplitudes. In figure 26 the irregular wear pattern is clearly illustrated. The irregular shape of the scar at low slip amplitudes must be the result of the stick-slip sliding, and chunk removal. This result is consistent with the results of a number of researchers [5, 48, 47, 49, 50, 51], who found a change in the wear mechanism at a particular slip amplitude. At the lower amplitudes, it is surmised by the researchers [17], that adhesion is the predominant wear mechanism, which would result in the irregular surface profile, as illustrated in figure 26.

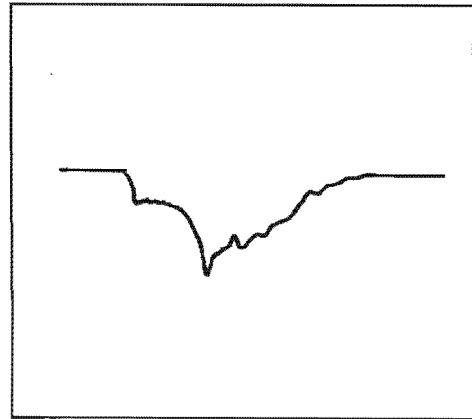


Figure 26 Wear Profile  
for Small Slip  
Amplitudes

## 8. CONCLUSIONS

Based on the findings of this investigation, the following conclusions can be drawn:

- (i) Slip Amplitude. The amount of wear was found to be dependant on the slip amplitude of the roller in relation to the race. For the oil tests it was possible at low slip amplitudes (smaller than 25um) for the relative motion to be accommodated elastically, which would result in a very low wear rate. At the onset of relative motion there is a bifurcation in the curve of the wear rate versus the slip amplitude. Wear mechanism one follows a gradual linear increase in the wear rate with respect to the slip amplitude. Wear mechanism two, which develops as a result of high silicon contamination levels, follows a step pattern. The wear rate increases very rapidly initially, and then settles down to a gradual linear increase. Ideally one would want to limit the wear process to the elastically accommodated region, as it is in this region that the lowest wear rates are found, but if this is not possible the slip amplitude must be limited to as small a value as possible.

For the grease tests, only one wear mechanism was found. The initial stage, slip amplitudes below 25um, is marked by elastic accommodation, and exhibits low wear rates. The next stage consists of a stick-slip region, with relatively high wear rates. The final stage is the slip stage. The wear rate in this stage is initially low, but increases in a linear fashion with respect to the slip amplitude. To limit the wear rate the slip amplitude should ideally be limited to the elastically accommodated region. If this is not possible it would be advantageous to actually by pass the stick-slip region, and operate in the early stages of the pure slip region. The size of the stick-slip region will be dependant on the equipment that is housing the rolling element bearing, and it will thus be necessary to determine the size of the stick-slip region for each piece of equipment. Determining the size of the stick-slip zone will be difficult for the complex structures that house bearings.

There are a number of ways of limiting the amount of vibration that is transmitted to a machine. One method is to stiffen the mounting of the machine that is producing the vibration. An other possible method is to place the effected equipment on vibration isolators, such as rubber feet. In very extreme cases the equipment could be placed on air bags. Both means of limiting the vibration change the natural frequency of the structure, moving the natural frequency away from the frequency of the vibration. The worst case would be when the natural frequency of the equipments foundation is equal to the frequency of the transmitted vibration.

- (ii) Reason for Change in Wear Rate. The oil tests exhibited two different wear mechanisms, wear mechanism one exhibiting a much lower wear rate than wear mechanism two. Both wear mechanisms were present over the entire range of slip amplitudes that were investigated. The parameter that determined which wear mechanism was present was the level of silicon contamination. High silicon levels resulted in high wear rates. To prevent the high wear rates that are associated with wear mechanism two it is essential that the oil be as clean as possible. This can be done by filtering the oil to ensure its cleanliness, or by exhibiting great care to seal the bearing housing from possible contamination.

The changes in the wear rate with respect to the slip amplitude for the grease tests were the result of changes in the contacting conditions. The elastic accommodation zone is the region which displays the least wear, and it would thus be advantageous to ensure that these contacting conditions prevail. The size of this region will be determined by the stiffness of the equipment that houses the bearing. The stiffer the housing, the smaller the elastically accommodated region. Thus when the equipment is not running it would be advantageous if the bearing housing could be made less stiff.

- (iii) Oil and Grease. The plot of the wear versus the slip amplitude reveals that the oil tests (Fig 6) resulted in less wear than the grease tests (Fig 7), if the first wear mechanism is predominant. This was found over the entire range of slip amplitudes that were investigated.

If the oil contained high levels of silicon contamination the wear rate increased considerably, until it is very similar to the wear rates found in the grease tests. Thus, if it is not possible to keep the oil clean, the use of oil and grease will yield similar life spans under false brinelling conditions.

- (iv) Third Body Wear Theory. The results achieved in this investigation support the third body theory of wear. According to this theory the wear process is determined to a large extent by the dynamics of the third body, or wear debris. The presence of wear particles in the contact zone of a lubricated system is detrimental to the life span of that particular system. In the grease tests for instance, the presence of the oxidized platelets in the contact zone resulted in abrasion, which would not have occurred if the wear particles had escaped from the contact zone as soon as they were detached from the parent material.

Further support for the third body theory of wear is offered by the size of the wear particles that were found in the lubricant samples. The magnitude of the maximum particle size was found to increase with respect to the slip amplitude for both the oil and grease tests, for the regions in which relative motion occurred. The average size of the wear particles was found to increase in the grease tests with respect to the slip amplitude. According to the third body theory

the increase in the slip amplitude would impart a larger momentum to the wear particles, and they would thus have a greater chance of escaping from the contact zone.

- (iv) Condition Monitoring. It was found that oil debris analysis revealed very little about the condition of the bearing. The wear index, the average particle size, and the maximum particle size followed certain trends, but these were dependant on which wear mechanism was present, not on the total amount of wear. From the present knowledge it would be necessary to analyze the above information in conjunction with the wear particle morphology, to firstly determine which wear mechanism is present. The results would still be very vague, and not much confidence could be placed in these results.

The wear debris in the grease samples revealed very similar results to those found for the oil samples. The parameters analyzed would have to be used in conjunction with the wear particle morphology to ascertain which contact conditions are predominant. More research will have to be done in this area before the condition of the machine can be determined with any confidence.

## 9. RECOMMENDATIONS

Based on the findings of this investigation the following recommendations can be made :

- (i) Further Research. It is strongly recommended that further investigations be done in the field of false brinelling. There are a number of areas that can be highlighted, these include :
- (a) Lubricants- The effect of the viscosity, the additives, in particular oxidation inhibitors, and type (i.e.grease or oil) of lubricant on the rate of false brinelling should be further investigated. The contact conditions that are prevalent during false brinelling lead to boundary lubrication conditions, thus lubricants which are suited to boundary lubrication conditions should be investigated. The effect of using greases with good heat dissipation qualities should be investigated.
  - (b) Reducing the Slip Amplitude- Methods of reducing the slip amplitude, and thus the amount of wear should be investigated. The stiffening of the equipments foundation and the development of a fixture that clamps the shaft to the bearing housing, reducing the slip amplitude to a fraction of its previous levels, should be investigated.
  - (c) Oil Contamination- The effect that different contaminants, such as ferrous particles, have on the wear rate should be investigated further. Not only the type of contaminant, but also the level of contamination should be investigated.
  - (d) Type of Rolling Element- The effect that the type of rolling element plays on the rate of false brinelling should be investigated. In this investigation cylindrical rollers were used. Ball and taper bearings should also be investigated to see if it is possible to reduce the amount of

wear by changing the type of bearing that is used.

- (e) Surface Treatment- In this investigation the surface finish was clean. The effect of a modification to the surface layer, such as chemical browning, should be investigated.
- (f) Frequency and Load- The effect that the load and the frequency has on the slip amplitude at which the contacting conditions change should be investigated further.

(ii) Ways to Limit False Brinelling. From the findings of this investigation it is possible to make a number of recommendations that will limit the amount of false brinelling. The actions to be taken to limit the amount of false brinelling are:

- (a) Lubricant- To minimize the amount of wear that attributable to false brinelling the use of oil in preference to the use of grease would be advised. The wear rate was lower for oil lubricated systems than for grease lubricated systems over the entire range of slip amplitudes tested. Every effort must be made to ensure the cleanliness of the oil, as oil containing contaminants results in wear rates very similar to those encountered if grease is the lubricant. In this regard the use of sealed bearings would be advised.

If grease is to be used it should have good heat dissipation properties to minimize the effects of high temperatures that occur at the micro-contacts.

- (b) Slip Amplitude- The wear rate was found to be dependant on the slip amplitude. To limit the amount of wear the slip amplitude must be kept as low as possible. Ideally the vibration should be limited to amplitudes that correspond to the elastic accommodation zone. This can be done by stiffening the foundations of the equipment producing the vibration. For more severe cases the stationary equipment can be isolated from the vibration by air bags and rubber vibration isolators.
  
- (c) Barring Motors- One way to prevent false brinelling is to slowly rotate the equipment, by means of a barring motor, thus preventing the conditions necessary for false brinelling. This method of prevention was not with-in the scope of this project, but is discussed in the literature that was reviewed.

### (iii) Condition Monitoring

From the results of this investigation it is possible to make a number of recommendations about what parameters to trend so that it is possible to detect false brinelling before it becomes critical.

- (a) Wear Index- By trending the wear index it would be possible to detect an increase in the amount of wear for grease lubricated bearings. The same would not hold true for oil lubricated bearings, as the wear index did not increase with an increase in the wear rate.
  
- (b) Average Particle Size- Monitoring the average particle size will aid in detection of false brinelling in grease lubricated bearings. The amount of false brinelling would be increasing if the average particle size increased.

- (c) Maximum Particle Size- The maximum particle size increased with respect to the slip amplitude and wear rate. This increase in particle size was found for both the elastically accommodated region and the region of slip for both oil and grease. The presence of large particles would be a sign of a high wear rate, but the presence of small particles does not necessarily mean that the wear rate is low, as small particles can also be found at high wear rates depending on how long they spent in the contact zone.
  
- (d) Particle Morphology- The presence of ribbon shaped wear particles in oil samples was a sign of high wear. For grease samples the presence of spherical wear particles that have experienced high temperatures would be a sign of high wear rates.
  
- (e) Temperature- Monitoring the contact temperature would not reveal any information concerning the condition of the bearing as it was found that the increase in the temperature was at the micro-contacts, and could not be detected by a temperature probe.

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**APPENDIX 1**

The following calculations will deal with the error that arises from using a flat outer race in the test rig instead of a round outer race.

The width of the contact zone for two cylinders in contact is:

$$(1) \text{Hertz} \quad b = \frac{\sqrt{2 \frac{F}{\pi l} \left[ \frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2} \right]}}{\sqrt{\frac{1}{d_1} + \frac{1}{d_2}}}$$

For the test rig:

---


$$\begin{aligned} E_1 &= E_2 \\ \nu_1 &= \nu_2 \\ d_2 &= \text{infinite} \end{aligned}$$


---

Therefore Hertzian Contact width for the test rig is:

$$b = \sqrt{4 \frac{Fd}{\pi l} \left[ \frac{1-\nu_1^2}{E} \right]}$$

The constants for the bearing  
(N203):

---


$$\begin{aligned} E_1 &= E_2 \\ \nu_1 &= \nu_2 \end{aligned}$$


---

Therefore Hertzian Contact width for the bearing is:

$$b = \sqrt{\frac{4Fd_1d_2}{\pi lE} \left( \frac{1-\nu^2}{d_1+d_2} \right)}$$

Comparing the test rig with the bearing the following constants are equal:

$$\begin{aligned} d_r &= d_1 \\ l_r &= l_b \\ E_r &= E_b \\ F_r &= F_b \end{aligned}$$

The ratio of the width of the contact zone of the test rig to the width of the contact zone for the bearing will then be:

$$\frac{b_r}{b_b} = \frac{d_1 + d_2}{d_2}$$

For the N203 bearing this ratio will equal **1.199**

The equation for the contact stress is:

$$P_{\max} = 2 \frac{F}{\pi b l}$$

If the maximum contact stress for the bearing is divided the maximum contact stress for the rig the following expression will be derived:

$$\frac{p_b}{p_r} = \frac{b_r}{b_b}$$

The stress on the bearing will thus be **1.199** times the stress on the test rig for the same load.

APPENDIX 2

Calculations to determine the contact stress between the roller and the outer race for the test rig.

**Table I** Value of constants for the test rig.

---


$$\begin{aligned}
 l &= 7\text{mm} \\
 d &= 7\text{mm} \\
 F &= 5.8\text{gN} \\
 E &= 200 \times 10^9 \text{Nm}^{-2} \\
 \nu &= 0.3
 \end{aligned}$$


---

The width of the contact zone in the test rig is:

$$b = \sqrt{4 \frac{Fd}{\pi l} \left( \frac{1-\nu^2}{E} \right)}$$

Thus

$$b = 1.8155 \times 10^{-5} \text{m}$$

But

$$P_{\max} = 2 \frac{F}{\pi b l}$$

Therefore

$$P_{\max} = 2.8501 \times 10^8 \text{Nm}^{-2} = 285.01 \text{MPa}$$

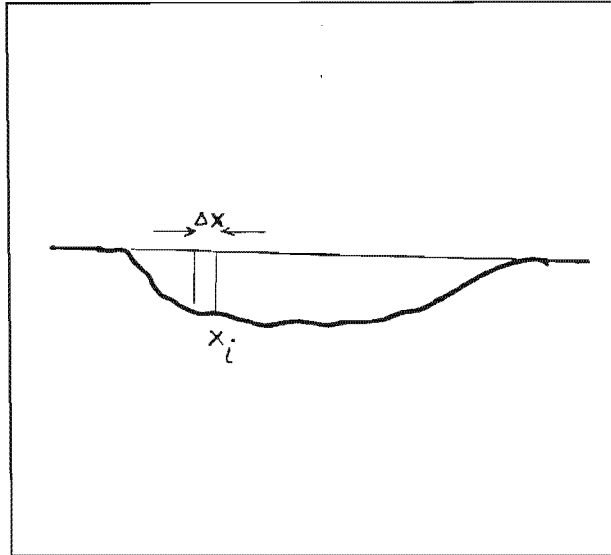
Therefore

According to the SKF Catalogue the basic static load rating is defined in accordance with ISO 76-1989 as the static load which corresponds to a calculated contact stress at the centre of the most heavily loaded rolling element. The maximum value for a

roller bearing is 4000 MPa, which is well above the value for the contact stress in the test rig.

APPENDIX 3

This appendix will deal with the procedure that was followed to determine the amount of wear from the profiles that were taken of the wear scar.



**Figure 1** Profile taken at right angles to the direction of vibration.

$$A_i = x_i \Delta x$$

$$A = \Delta x \sum_{i=1}^i x_i$$

Therefore

If the profile was assumed to be a rectangle with a depth equal

$$A = kx \frac{m}{2}$$

to half of the maximum depth of the recorded profile the following expression would hold

$$k = \frac{a}{x_m^2}$$

Solving for k:

k will serve as the constant that relates the area of the scar in cross-section to the volume.

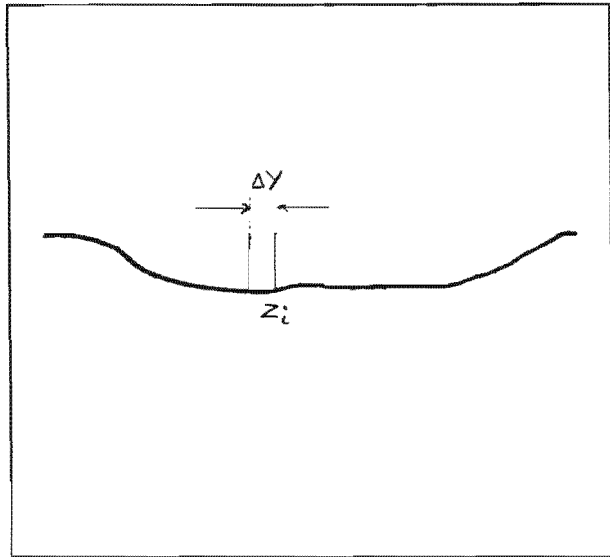


Figure 2 Profile of the wear scar in the direction of the vibration.

The volume of the scar will be equal to the sum of the areas of

the slices (Fig. 2) multiplied by the width of the slices. The area of the slice is calculated by relating the maximum depth of the scar to its area at right angles to the direction of the vibration.

Therefore

$$V_i = A_i \Delta y$$

$$V_i = \left(z_i \frac{k}{2}\right) \Delta y$$

Therefore

$$V = \frac{k\Delta y}{2} \sum_{i=1}^i z_i$$

APPENDIX 4Parameters that Affect the Rate of False Brinelling

1. Material
2. Load
3. Slip Amplitude
4. Frequency
4. Surface Hardness
6. Surface roughness
7. Lubricant
8. Additives in lubricant
9. Temperature
10. Oxygen concentration
11. Radial clearance in bearing
12. Radial load
13. Cage pocket clearance
14. Radial guidance of the cage
15. Roller diameter
16. Inclusions and voids in material
17. Direction of vibration
18. Type of bearing i.e. roller or ball)
19. Humidity

APPENDIX 5Specifications of apparatus(a) Talysurf

Manufacturers	Taylor-Hobson
Stylus: material	Diamond
Tip width	0.0025 mm
Force	100 mgf
Accuracy	better than 3% of full scale

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(b) Shaker

Manufacturer	Ling Dynamics Systems (Ltd)
Shaker type	406
Amplifier	IPO 100
Transformer tapping	direct coupled
Effective armature mass	0.2 kg
Displacement limit	approximately 8.8mm
Velocity limit	1.52 m/s
Maximum acceleration	490.5 m/ s <sup>2</sup>

---

(c) Proximity Probe

Manufacturer	Bently Nevada
Model Number	3301 05 02 12 10 02 00
Unthreaded length	5.1 mm
Overall case length	30.5 mm
Connector	male mini coax
Power supply	-24 Vdc
Load	10 k ohms
Conversion factor	12.22 mV/um

-----

(d) Oscilloscope

Manufacturer	Hewlett Packard
Model	

-----

(e) Oil

Manufacturer	Shell
Grade	SAE 30

-----

(f) Grease

Manufacturer	SKF
Grade	LGEP 2/1

Base	Lithium soap
Consistency	NLGI grade 2
Temperature range	-30 to +110 degrees C
Use	medium/large roller bearings under high load

(g) Test rig

Mass of load	5.8 kg
Load on roller	56.9 N
Contact Stress	285.01 MPa

-----

## APPENDIX 6

The results that are given for parameters such as the maximum and average particle size are not the actual magnitudes, but are codes that are related to the magnitudes of the parameters according to the following tables. All the graphs in the section dealing with the results use the code system as laid out in this section.

APPENDIX 7

The following are the results that were recieved from C.M.S. International.



# CMS INTERNATIONAL

## SUMMARY OF MACHINE HEALTH INDICATORS



### 1) STATUS

This gives an indication as to the overall health of a machine and is indicated by a number between 1 - 5, and is recorded as follows:

INDEX	COMMENT	POSSIBLE ACTION
1	Healthy Machine	Nil
2	Machine showing slight indication	Monitor all indications closely
3	Machine showing distinct indication	Monitor more frequently, use other monitoring techniques to gather more information. Plan corrective action
4	Machine showing distinct threatening indication	Take corrective action to reduce effects of the detected fault
5	Machine in serious distress	Plan earliest possible removal from service

### 2) FERROGRAPHIC PARAMETERS

Ferrographic analysis measures the following parameters. These are measured on a scale of 1 - 9 as follows:

CODE	AVE	MAX	DENS (>5um)	CNTM (>5um)
1	0 - 5	0 - 100	0 - 5	0 - 5
2	5 - 10	100 - 200	5 - 10	5 - 10
3	10 - 20	200 - 300	10 - 20	10 - 20
4	20 - 30	300 - 400	20 - 40	20 - 40
5	30 - 40	400 - 500	40 - 80	40 - 80
6	40 - 50	500 - 600	80 - 160	80 - 160
7	50 - 60	600 - 700	160 - 320	160 - 320
8	60 - 70	700 - 800	320 - 640	320 - 640
9	> 70	> 800	> 640	> 640

3) EXPLANATION OF PARAMETERS

AVE: Average size in microns of the metallic wear debris in a representative portion of the sample

MAX: Maximum size in microns of wear debris within a representative portion of the sample

DEN: Density index is the number of metallic particles larger than 5 microns in one square mm of a standardised sample

CONT: Contamination index is the number of non-metallic particles larger than 5 microns in one square mm of a standardised sample

\* AVE and MAX are measured in microns

\*\* DENS and CONT are measured by particle counting

4) PARTICLE CHARACTERISTICS:

Are defined according to shape and mechanism of removal as follows:

A - abrasive wear      P - platelets      C - chunky particles  
Fr - fretting wear      N - needle shaped wear particles  
R - ribbons produced by abrasive agents  
S - spherical particles      F - fused particles

Contact:ROY NOYES  
 Address:PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3233

TRIBO TECH INDUSTRIAL

Customer:UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (1 )  
 SAMPLE 1 (1 )  
 BEARING (BEAR)

STATUS

Lub Supp

1

S/N:  
 Res Per:ROY NOYES  
 Tel:  
 Fax:

Lub Type  
 SAE 30

Sample No. Date Y-M-D Meter Read Diagnosis			1 92-11-20 CINDY	LIMITS	110.0 94.00 92-11-20 Visc40 92-11-20
Status			1		
OIL CONDITION					
Visc40			94	110	15.00 92-11-20 Fe 92-11-20
OIL ADDITIVES					
Zn Ca Mg P Ba B Mo			1208 1238* 598 1 1 1 1	2000 250 2000 1500 50 50 5	1.00 5.00 92-11-20 Cu 92-11-20
WEAR METALS					
Fe Cr Al Cu Pb Sn Ni Sb			1 1 1 2 5 1 2 1	15 5 5 5 5 5 5 5	200.0 1.00 92-11-20 WI 92-11-20
CONTAMINANTS					
% Water Na Si			0 6 10	0 400 10	6.00 3.00 92-11-20 CONT 92-11-20
WEAR DEBRIS					
WI CONT AVE DENS MAX			1 3 2 1 2	200 6 5 6 5	

92-11-20 Diagnostic Message

A LOW DENSITY OF FINE FERROUS WEAR, AVERAGE SIZE 5 - 10 MICRONS, WITH WEAR OF UP TO 200 MICRONS.  
 A LOW DENSITY OF FINE MINERAL AND FIBRE CONTAMINATION PRESENT.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3233

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (1 )  
 SAMPLE 2 (2 )  
 BEARING (BEAR)

STATUS

Lub Supp

1

Lub Type  
 SAE 30

S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis				2 92-11-20 CINDY	LIMITS	110.0 91.00 92-11-20 Visc40 92-11-20
Status				1		
OIL CONDITION						
Visc40				91	110	15.00 92-11-20 Fe 92-11-20
OIL ADDITIVES						
Zn				1268	2000	1.00 92-11-20 Cu 92-11-20
Ca				1304*	250	
Mg				630	2000	5.00 92-11-20 WI 92-11-20
P				1045	1500	
Ba				1	50	1.00 92-11-20 CONT 92-11-20
B				1	50	
Mo				1	5	
WEAR METALS						
Fe				1	15	
Cr				1	5	
Al				1	5	
Cu				1	5	
Pb				1	5	
Sn				1	5	
Ni				1	5	
Sb				1	5	
CONTAMINANTS						
% Water				0	0	
Na				4	400	
Si				14*	10	
WEAR DEBRIS						
WI				14	200	
CONT				2	6	
AVE				3	5	
DENS				1	6	
MAX				6*	5	

92-11-20 Diagnostic Message

A LOW DENSITY OF MAINLY FINE FERROUS WEAR WITH A FEW LARGER WEAR PARTICLES OF UP TO 530 MICRONS.  
 LOW MINERAL AND FIBRE LEVEL.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 3 (GREASE) (3 )

STATUS

Lub Supp

2

Lub Type

S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis				921223-01001 92-12-18 0	LIMITS	110.0 92-12-18 Visc40 92-12-18
Status				2		
OIL CONDITION						15.00 92-12-18 Visc40 92-12-18
Visc40				0	110	
OIL ADDITIVES						9.00 92-12-18 Fe 92-12-18
Zn				3683*	2000	
Ca				34	250	
Mg				5	2000	
P				894	1500	
Ba				1	50	
B				8	50	
Mo				1	5	
WEAR METALS						5.00 92-12-18 Cr 92-12-18
Fe				9	15	
Cr				1	5	
Al				1	5	
Cu				3	5	
Pb				94*	5	
Sn				1	5	
Ni				1	5	
Sb				1	5	
CONTAMINANTS						200.0 92-12-18 WI 92-12-18
% Water				0	0	
Na				5	400	
Si				3	10	
WEAR DEBRIS						6.00 92-12-18 CONT 92-12-18
WI				11	200	
CONT				2	6	
AVE				3	5	
DENS				1	6	
MAX				4	5	

92-12-18

Diagnostic Message

STATUS 2 DUE TO THE PRESENCE OF ABNORMAL WEAR ie. PLATELETS AND CHUNKS.  
 A LOW DENSITY OF MAINLY FINE WEAR, AVERAGE SIZE 20 MICRONS, WITH A FEW  
 WEAR PARTICLES OF UP TO 400 MICRONS PRESENT.  
 A LOW MINERAL AND FIBRE LEVEL.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 4 (GREASE) (4 )

STATUS

Lub Supp

2

Lub Type

S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis				921223-01002 92-12-18 0	LIMITS	110.0 92-12-18 Visc40 92-12-18
Status				2		
OIL CONDITION						
Visc40				0	110	15.00 92-12-18 Fe 92-12-18
OIL ADDITIVES						
Zn Ca Mg P Ba B Mo				3839* 34 4 927 1 3 1	2000 250 2000 1500 50 50 5	8.00 92-12-18 Cr 92-12-18
WEAR METALS						
Fe Cr Al Cu Pb Sn Ni Sb				8 1 1 3 99* 1 1 1	15 5 5 5 5 5 5 5	5.00 92-12-18 Cu 92-12-18
CONTAMINANTS						
% Water Na Si				0 6 1	0 400 10	200.0 92-12-18 WI 92-12-18
WEAR DEBRIS						
WI CONT AVE DENS MAX				9 3 3 1 9*	200 6 5 6 5	9.00 92-12-18 CONT 92-12-18

92-12-18 Diagnostic Message

A LOW DENSITY OF MAINLY FERROUS WEAR, PLATELETS, RIBBONS AND CHUNKS, AVERAGE SIZE 20 MICRONS WITH WEAR OF UP TO 1200 MICRONS. TRACE COPPER AND OTHER NON-MAGNETIC METALLIC WEAR. TEMPERING INDICATES HEAT STRESS. LOW MINERAL AND FIBRE LEVEL.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 5 (GREASE) (5 )

STATUS

Lub Supp

2

Lub Type

S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis				921223-01003 92-12-18 0	LIMITS	110.0 92-12-18 Visc40 92-12-18
Status				2		
OIL CONDITION						15.00 92-12-18 Visc40 92-12-18
Visc40				0	110	
OIL ADDITIVES						8.00 92-12-18 Fe 92-12-18
Zn				3811*	2000	
Ca				35	250	
Mg				4	2000	
P				924	1500	
Ba				1	50	
B				2	50	
Mo				1	5	
WEAR METALS						5.00 92-12-18 Cr 92-12-18
Fe				8	15	
Cr				1	5	
Al				1	5	
Cu				3	5	
Pb				99*	5	
Sn				1	5	
Ni				1	5	
Sb				1	5	
CONTAMINANTS						200.0 92-12-18 WI 92-12-18
% Water				0	0	
Na				7	400	
Si				1	10	
WEAR DEBRIS						6.00 92-12-18 CONT 92-12-18
WI				15	200	
CONT				1	6	
AVE				6*	5	
DENS				1	6	
MAX				3	5	

92-12-18 Diagnostic Message

A LOW DENSITY OF MAINLY FERROUS ABRASIVE WEAR, PLATELETS, MAINLY RIBBONS, CHUNKS AND SPHERICAL PARTICLES. AVERAGE SIZE 50 MICRONS, WITH WEAR OF UP TO 300 MICRONS. TEMPERING AND SLIDING WEAR. LOW MINERAL AND FIBRE LEVEL.

gea

Contact:ROY NOYES  
 Address:PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer:UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 6 (GREASE) (6 )

STATUS

Lub Supp

2

Lub Type

S/N:  
 Res Per:ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis			921223-01004 92-12-18 0	LIMITS	110.0 92-12-18 Visc40 92-12-18
Status			2		
OIL CONDITION					15.00 92-12-18 Visc40 92-12-18
Visc40			0	110	
OIL ADDITIVES					3.00 92-12-18 Fe 92-12-18
Zn			3600*	2000	
Ca			33	250	
Mg			4	2000	
P			864	1500	
Ba			1	50	
B			1	50	
Mo			1	5	
WEAR METALS					5.00 92-12-18 Cr 92-12-18
Fe			3	15	
Cr			1	5	
Al			1	5	
Cu			3	5	
Pb			91*	5	
Sn			1	5	
Ni			1	5	
Sb			1	5	
CONTAMINANTS					200.0 92-12-18 WI 92-12-18
% Water			0	0	
Na			6	400	
Si			1	10	
WEAR DEBRIS					6.00 92-12-18 CONT 92-12-18
WI			12	200	
CONT			1	6	
AVE			4	5	
DENS			1	6	
MAX			2	5	

92-12-18 Diagnostic Message

A LOW DENSITY OF FINE MAINLY FERROUS ABRASIVE WEAR, PLATELETS, RIBBONS, CHUNKS AND SPHERES AVERAGE SIZE 30 MICRONS WITH WEAR OF UP TO 200 MICRONS. TEMPERING PRESENT.

xū

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 7 (OIL) (7 )

STATUS

Lub Supp

2

Lub Type

S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

Sample No. Date Y-M-D Meter Read Diagnosis			921223-01005 92-12-18 0	LIMITS	110.0 ----- 79.00 92-12-18 Visc40 92-12-18
Status			2		
OIL CONDITION					
Visc40			79	110	15.00 ----- 92-12-18 Fe 92-12-18
OIL ADDITIVES					
Zn Ca Mg P Ba B Mo			1317 1281* 663 1160 1 3 1	2000 250 2000 1500 50 50 5	5.00 ----- 92-12-18 Cr 92-12-18
WEAR METALS					
Fe Cr Al Cu Pb Sn Ni Sb			1 1 1 1 5 1 1 1	15 5 5 5 5 5 5 5	5.00 ----- 92-12-18 Cu 92-12-18
CONTAMINANTS					
% Water Na Si			0 5 15*	0 400 10	200.0 ----- 92-12-18 WI 92-12-18
WEAR DEBRIS					
WI CONT AVE DENS MAX			1 2 3 1 5	200 6 5 6 5	6.00 ----- 92-12-18 CONT 92-12-18

92-12-18 Diagnostic Message

A LOW DENSITY OF FINE ABNORMAL WEAR, PLATELETS, CHUNKS AND SPHERICAL PARTICLES, AVERAGE SIZE 20 MICRONS, WITH WEAR OF UP TO 500 MICRONS. TEMPERING AND SLIDING WEAR PRESENT. A LOW DENSITY OF MINERAL AND FIBRE.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 8 (OIL) (8 )  
 S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

STATUS

Lub Supp

1

Lub Type

Sample No. Date Y-M-D Meter Read Diagnosis			921223-01006 92-12-18 0	LIMITS	110.0 89.00 92-12-18 Visc40 92-12-18
Status			1		
OIL CONDITION					
Visc40			89	110	15.00 92-12-18 Fe 92-12-18
OIL ADDITIVES					
Zn			1317	2000	1.00 92-12-18 Cr 92-12-18
Ca			1301*	250	
Mg			672	2000	5.00 92-12-18 Cu 92-12-18
P			1168	1500	
Ba			1	50	
B			3	50	
Mo			1	5	
WEAR METALS					
Fe			1	15	1.00 92-12-18 WI 92-12-18
Cr			1	5	
Al			1	5	
Cu			1	5	
Pb			9*	5	
Sn			1	5	
Ni			1	5	
Sb			1	5	
CONTAMINANTS					
% Water			0	0	1.00 92-12-18 CONT 92-12-18
Na			4	400	
SI			18*	10	
WEAR DEBRIS					
WI			1	200	
CONT			2	6	
AVE			3	5	
DENS			1	6	
MAX			1	5	

92-12-18 Diagnostic Message

A VERY LOW DENSITY OF FERROUS WEAR AVERAGE SIZE 20 MICRONS, WITH WEAR OF UP TO 100 MICRONS. MAINLY PLATELETS AND A FEW CHUNKS PRESENT.

Contact: ROY NOYES  
 Address: PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

TRIBO TECH INDUSTRIAL

Customer: UNIVERSITY OF CAPE TOWN (UCT )  
 TEST RIG (TEST)  
 TEST RIG SAMPLES (RIG )  
 SAMPLE 10 (OIL) (10 )  
 S/N:  
 Res Per: ROY NOYES  
 Tel:  
 Fax:

STATUS

Lub Supp

2

Lub Type

Sample No. Date Y-M-D Meter Read Diagnosis				921223-01008 92-12-18 0	LIMITS	110.0 89.00 92-12-18 Visc40 92-12-18
Status				2		
OIL CONDITION						
Visc40				89	110	15.00 92-12-18 Fe 92-12-18
OIL ADDITIVES						
Zn Ca Mg P Ba B Mo				1297 1281* 663 1148 1 3 1	2000 250 2000 1500 50 50 5	1.00 92-12-18 Fe 92-12-18
WEAR METALS						
Fe Cr Al Cu Pb Sn Ni Sb				1 1 1 1 4 1 1 1	15 5 5 5 5 5 5 5	5.00 92-12-18 Cr 92-12-18
CONTAMINANTS						
% Water Na Si				0 4 18*	0 400 10	1.00 92-12-18 Cu 92-12-18
WEAR DEBRIS						
WI CONT AVE DENS MAX				32 3 3 1 8*	200 6 5 6 5	200.0 32.00 92-12-18 WI 92-12-18
						6.00 92-12-18 CONT 92-12-18

92-12-18 Diagnostic Message

A LOW DENSITY OF FERROUS AND NON-MAGNETIC METALLIC ABNORMAL WEAR, PLATELETS, CHUNKS AND RIBBONS, AVERAGE SIZE 20 MICRONS WITH A FEW LARGER WEAR PARTICLES OF UP TO 800 MICRONS. TEMPERING SLIDING WEAR AND A COUPLE OF CUPRIC WEAR PARTICLES PRESENT.  
 PITTING PRESENT ON PLATELETS AND NOT MANY RIBBONS PRESENT.

UNIVERSITY OF CAPE TOWN  
 Attn:ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3240

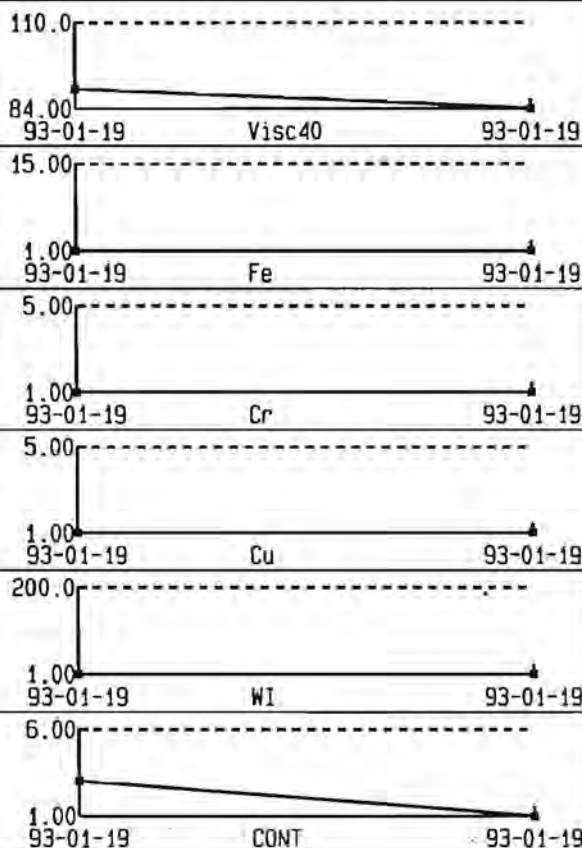
CMS INTERNATIONAL (PTY) Ltd

P O Box 418  
 ISANDO

1600  
 Tel:(011) 622 3625  
 Fax:(011) 615 7473

TRIBO TECH INDUSTRIAL		Customer:UNIVERSITY OF CAPE TOWN	(UCT )	Level 3 Make:
STATUS	Lub Supp	TEST RIG	(TEST)	Model:
	Lub Type	TEST RIG SAMPLES	(RIG )	Level 4 Make:
1		SAMPLE 9 (OIL)	(9 )	Model:
		Res Per:ROY NOYES		S/N:
		Tel:		
		Fax:		

Sample No.		921223-01007	930121-01006	
Bottle No.		8000717	8000973	
Date Y-M-D		92-12-18	93-01-19	
Meter Read		0	0	
Oil Hours		0	0	
Oil/Fil Chng	/	/	/	
Part Type		PR	P	LIMITS
Diagnosis				
Status		2	1	
OIL CONDITION				
Visc40		90	84	110
OIL ADDITIVES				
Zn		1311	1268	2000
Ca		1297*	1315*	250
Mg		668	714	2000
P		1163	1209	1500
Ba		1	1	50
B		3	1	50
Mo		1	1	5
WEAR METALS				
Fe		1	1	15
Cr		1	1	5
Al		1	1	5
Cu		1	1	5
Pb		8*	2	5
Sn		1	1	5
Ni		1	1	5
Sb		1	1	5
CONTAMINANTS				
% Water		0	0	0
Na		5	1	400
Si		19*	6	10
WEAR DEBRIS				
WI		1	1	200
CONT		3	1	6
AVE		2	3	5
DENS		2	1	6
MAX		1	1	5



92-12-18 Diagnostic Message

A LOW DENSITY OF FINE WEAR, PLATELETS AND RIBBONS, WITH WEAR OF UP TO 100 MICRONS. LOW MINERAL LEVEL. MORE FIBRE IN THIS SAMPLE THEN THE OTHERS.

93-01-19 Diagnostic Message

No abnormal indication

UNIVERSITY OF CAPE TOWN  
 Attn:ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3240

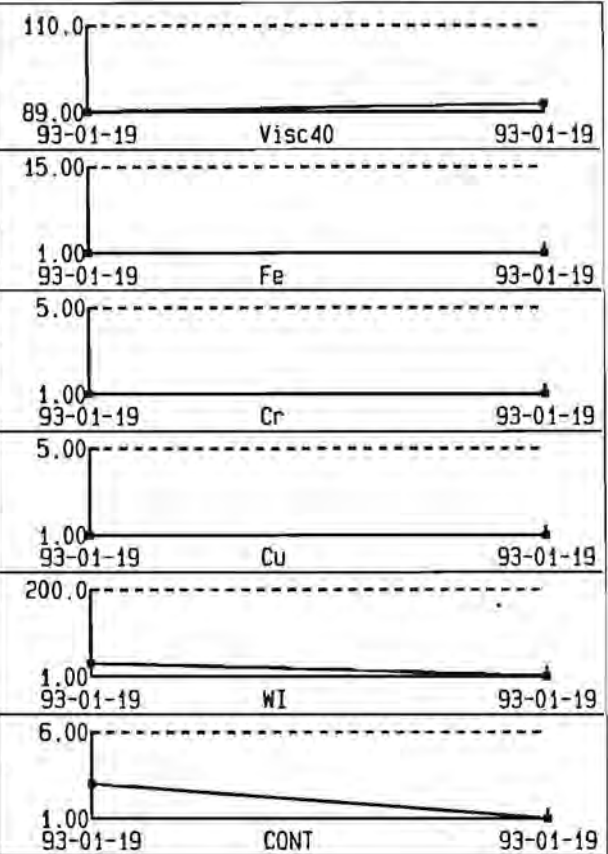
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 ISANDO

1600  
 Tel:(011) 622 3625  
 Fax:(011) 615 7473

TRIBO TECH INDUSTRIAL		Customer:UNIVERSITY OF CAPE TOWN	(UCT )	Level 3 Make:
STATUS	Lub Supp	TEST RIG	(TEST)	Model:
	Lub Type	TEST RIG SAMPLES	(RIG )	Level 4 Make:
1		SAMPLE 10 (OIL)	(10 )	Model:
		Res Per:ROY NOYES		S/N:
		Tel:		
		Fax:		

Sample No.		921223-01008	930121-01003	
Bottle No.		8000718	8000970	
Date Y-M-D		92-12-18	93-01-19	
Meter Read		0	0	
Oil Hours		0	0	
Oil/Fil Chng	/	/	/	
Part Type		PCR	PR	LIMITS
Diagnosis				
Status		2	1	
OIL CONDITION				
Visc40		89	91	110
OIL ADDITIVES				
Zn		1297	1209	2000
Ca		1281*	1250*	250
Mg		663	680	2000
P		1148	1143	1500
Ba		1	1	50
B		3	1	50
Mo		1	1	5
WEAR METALS				
Fe		1	1	15
Cr		1	1	5
Al		1	1	5
Cu		1	1	5
Pb		4	7*	5
Sn		1	2	5
Ni		1	1	5
Sb		1	1	5
CONTAMINANTS				
% Water		0	0	0
Na		4	1	400
Si		18*	7	10
WEAR DEBRIS				
WI		32	1	200
CONT		3	1	6
AVE		3	3	5
DENS		1	1	6
MAX		8*	1	5



92-12-18 ————— Diagnostic Message —————  
 A LOW DENSITY OF FERROUS AND NON-MAGNETIC METALLIC ABNORMAL WEAR, PLATELETS, CHUNKS AND RIBBONS, AVERAGE SIZE 20 MICRONS WITH A FEW LARGER WEAR PARTICLES OF UP TO 800 MICRONS. TEMPERING SLIDING WEAR AND A COUPLE OF CUPRIC WEAR PARTICLES PRESENT.  
 PITTING PRESENT ON PLATELETS AND NOT MANY RIBBONS PRESENT.

93-01-19 ————— Diagnostic Message —————  
 VERY LITTLE WEAR. MAINLY NON-MAGNETIC WEAR.

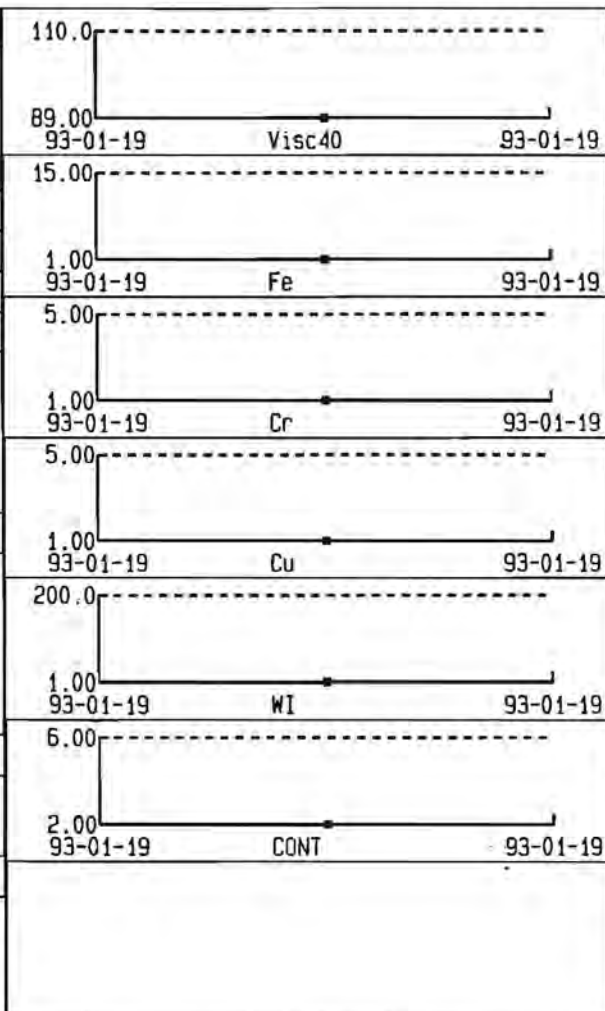
UNIVERSITY OF CAPE TOWN  
 Attn:ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3240

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 ISANDO  
 1600  
 Tel:(011) 622 3625  
 Fax:(011) 615 7473

TRIBO TECH INDUSTRIAL		Customer:UNIVERSITY OF CAPE TOWN TEST RIG TEST RIG SAMPLES SAMPLE 11 (OIL) Res Per:ROY NOYES Tel: Fax:	(UCT ) (TEST ) (RIG ) (11 )	Level 3 Make: Model:
STATUS <b>1</b>	Lub Supp SHELL  Lub Type OMALA 460			Level 4 Make: Model: S/N:

Sample No.				930121-01007	
Bottle No.				8000974	
Date Y-M-D				93-01-19	
Meter Read				0	
Oil Hours				0	
Oil/Fil Chng	/	/	/	P /	
Part Type					
Diagnosis					LIMITS
Status				1	
OIL CONDITION					
Visc40				89	110
OIL ADDITIVES					
Zn				1302	2000
Ca				1345*	250
Mg				729	2000
P				1241	1500
Ba				1	50
B				1	50
Mo				1	5
WEAR METALS					
Fe				1	15
Cr				1	5
Al				1	5
Cu				1	5
Pb				1	5
Sn				1	5
Ni				1	5
Sb				1	5
CONTAMINANTS					
% Water				0	0
Na				1	400
Si				9	10
WEAR DEBRIS					
WI				1	200
CONT				2	6
AVE				2	5
DENS				1	6
MAX				1	5



93-01-19 Diagnostic Message

A LOW DENSITY OF FINE, MAINLY NON-MAGNETIC WEAR PRESENT WITH WEAR OF UP TO 100 MICRONS.

UNIVERSITY OF CAPE TOWN  
 Attn: ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

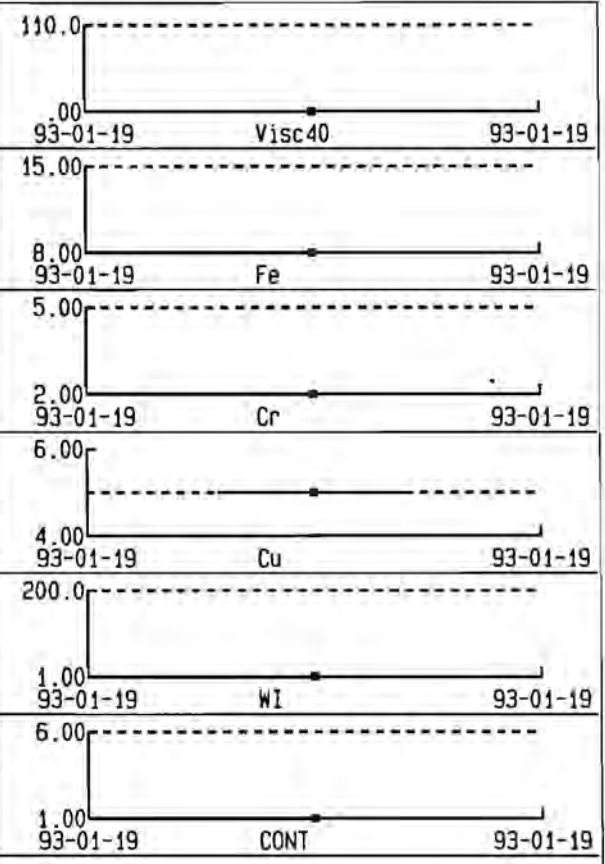
CMS INTERNATIONAL (PTY) Ltd

P O Box 418  
 ISANDO

1600  
 Tel: (011) 622 3625  
 Fax: (011) 615 7473

TRIBO TECH INDUSTRIAL		Customer: UNIVERSITY OF CAPE TOWN TEST RIG TEST RIG SAMPLES SAMPLE 12 (GREASE) Res Per: ROY NOYES Tel: Fax:	(UCT ) (TEST ) (RIG ) (12 )	Level 3 Make: Model:
STATUS	Lub Supp		Level 4 Make: Model: S/N:	
1	Lub Type			

Sample No.				930121-01005	
Bottle No.				8000972	
Date Y-M-D				93-01-19	
Meter Read				0	
Oil Hours				0	
Oil/Fil Chng	/	/	/	P /	
Part Type					
Diagnosis					LIMITS
Status				1	
OIL CONDITION					
Visc40				0	110
OIL ADDITIVES					
Zn				3401*	2000
Ca				60	250
Mg				6	2000
P				880	1500
Ba				1	50
B				1	50
Mo				1	5
WEAR METALS					
Fe				8	15
Cr				2	5
Al				2	5
Cu				5	5
Pb				97*	5
Sn				1	5
Ni				1	5
Sb				1	5
CONTAMINANTS					
% Water				0	0
Na				7	400
Si				1	10
WEAR DEBRIS					
WI				1	200
CONT				1	6
AVE				1	5
DENS				1	6
MAX				1	5



93-01-19 Diagnostic Message  
 No abnormal indication

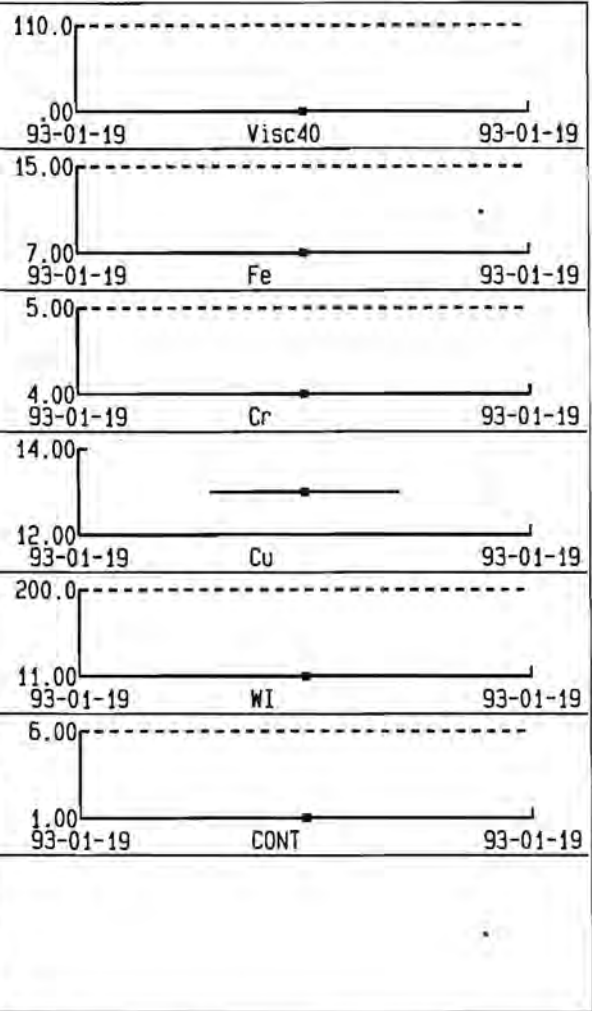
UNIVERSITY OF CAPE TOWN  
 Attn:ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel:021 - 650 3235  
 Fax:021 - 650 3240

CMS INTERNATIONAL (PTY) Ltd

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 ISANDO  
 1600  
 Tel:(011) 622 3625  
 Fax:(011) 615 7473

TRIBO TECH INDUSTRIAL		Customer:UNIVERSITY OF CAPE TOWN	(UCT )	Level 3 Make:
STATUS 1	Lub Supp SHELL	TEST RIG	(TEST)	Model:
	Lub Type OMALA 460	TEST RIG SAMPLES	(RIG )	Level 4 Make:
		SAMPLE 13 (GREASE)	(13 )	Model:
		Res Per:ROY NOYES		S/N:
		Tel:		
		Fax:		

Sample No.		8000968	
Bottle No.		0	
Date Y-M-D		93-01-19	
Meter Read		0	
Oil Hours		0	
Oil/Fil Chng	/	PCS /	
Part Type	/	C	LIMITS
Diagnosis	/		
Status		1	
OIL CONDITION			
Visc40		0	110
OIL ADDITIVES			
Zn		2611*	2000
Ca		93	250
Mg		6	2000
P		662	1500
Ba		1	50
B		1	50
Mo		1	5
WEAR METALS			
Fe		7	15
Cr		4	5
Al		6*	5
Cu		13*	5
Pb		155*	5
Sn		13*	5
Ni		23*	5
Sb		1	5
CONTAMINANTS			
% Water		0	0
Na		29	400
Si		8	10
WEAR DEBRIS			
WI		11	200
CONT		1	6
AVE		9*	5
DENS		1	6
MAX		4	5



93-01-19 Diagnostic Message  
 A VERY LOW DENSITY OF FERROUS AND NON-MAGNETIC METALLIC ABNORMAL WEAR,  
 AVERAGE SIZE IS HIGH (>70 um), WITH TRACES TEMPERING AND SLIDING WEAR PRESENT  
 WEAR OF UP TO 400 MICRONS PRESENT.

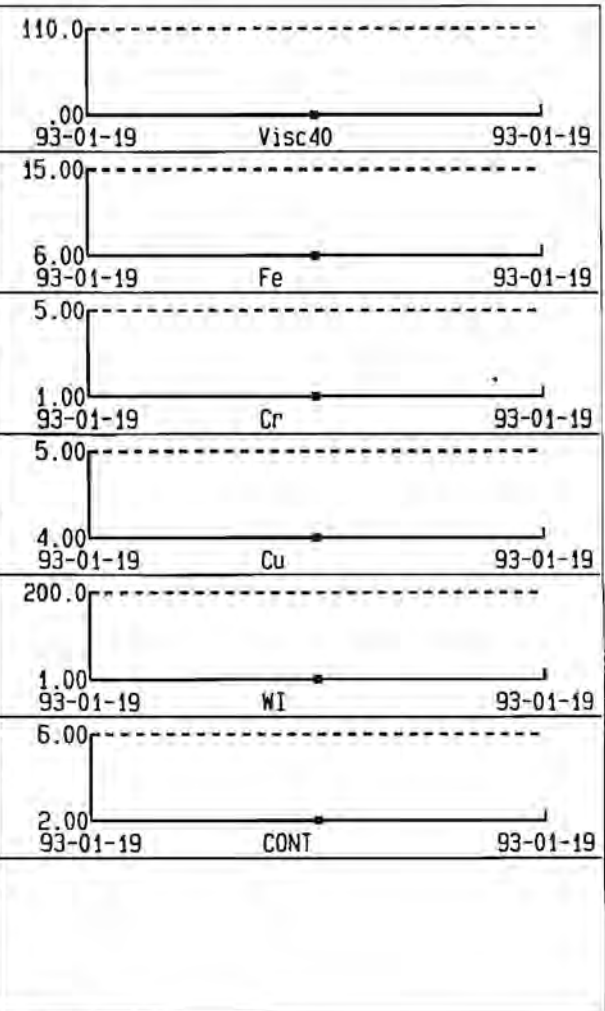
UNIVERSITY OF CAPE TOWN  
 Attn: ROY NOYES  
 PRIVATE BAG  
 RONDEBOSCH  
 CAPE  
 7700  
 Tel: 021 - 650 3235  
 Fax: 021 - 650 3240

CMS INTERNATIONAL (PTY) Ltd

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 ISANDO  
 1600  
 Tel: (011) 622 3625  
 Fax: (011) 615 7473

TRIBO TECH INDUSTRIAL		Customer: UNIVERSITY OF CAPE TOWN TEST RIG TEST RIG SAMPLES SAMPLE 14 (GREASE) Res Per: ROY NOYES Tel: Fax:	(UCT ) (TEST ) (RIG ) (14 )	Level 3 Make: Model:
STATUS	Lub Supp		Level 4 Make: Model: S/N:	
1	Lub Type			

Sample No.				930121-01008	
Bottle No.				8000975	
Date Y-M-D				93-01-19	
Meter Read				0	
Oil Hours				0	
Oil/Fil Chng	/	/	/	P /	
Part Type					
Diagnosis					LIMITS
Status				1	
OIL CONDITION					
Visc40				0	110
OIL ADDITIVES					
Zn				3504*	2000
Ca				61	250
Mg				7	2000
P				900	1500
Ba				1	50
B				1	50
Mo				1	5
WEAR METALS					
Fe				6	15
Cr				1	5
Al				3	5
Cu				4	5
Pb				93*	5
Sn				1	5
Ni				1	5
Sb				1	5
CONTAMINANTS					
% Water				0	0
Na				7	400
Si				3	10
WEAR DEBRIS					
WI				1	200
CONT				2	6
AVE				1	5
DENS				1	6
MAX				1	5



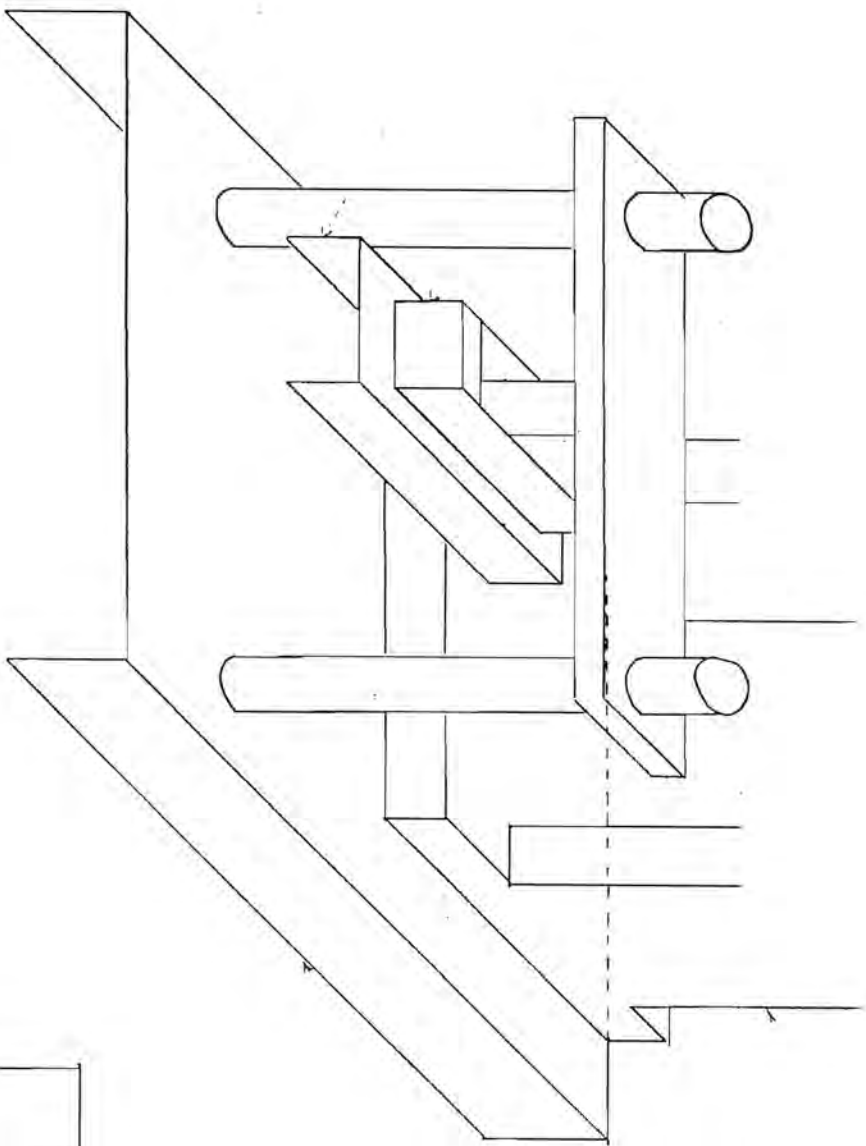
93-01-19 Diagnostic Message  
 No abnormal indication

APPENDIX 8

The drawings used in the construction of the test rig are provided in this section.

VIBRATING ARM

OIL BATH

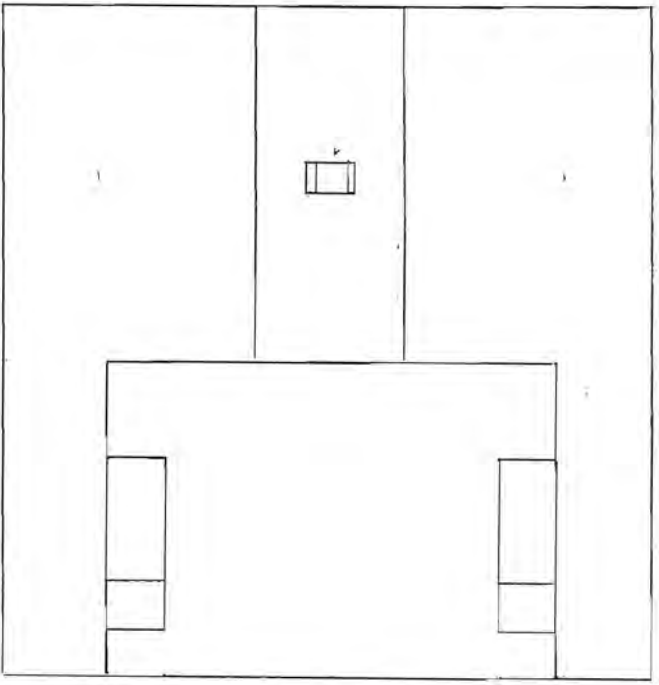


SHAKER FRAME

FRAME

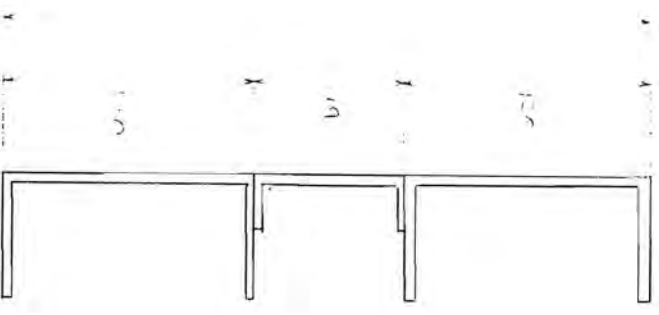
TEST RIG  
(ASSEMBLED)

LOAD  
CARRIER



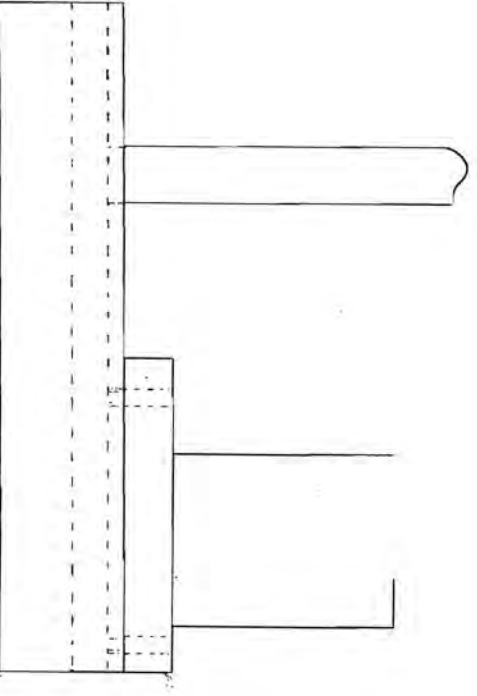
250

1  
4  
8



100  
100  
100

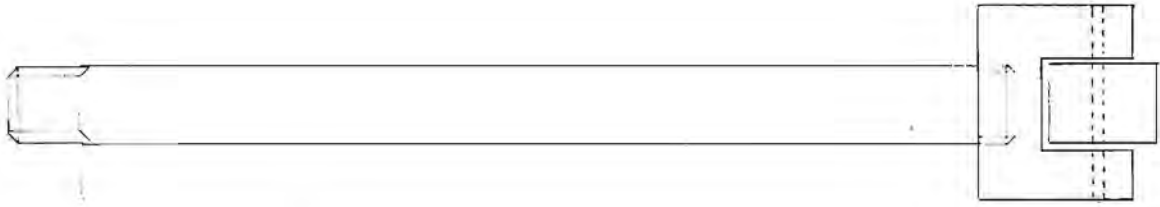
SHAKER STAND  
EXISTING



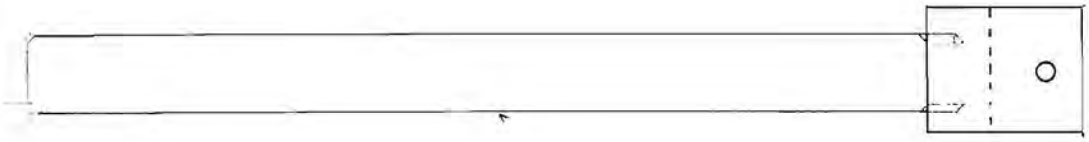
100  
100

100

FRAME OF TEST  
RIG



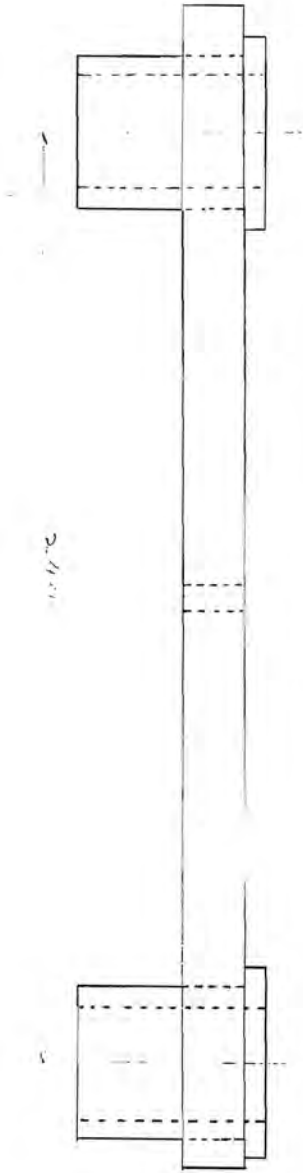
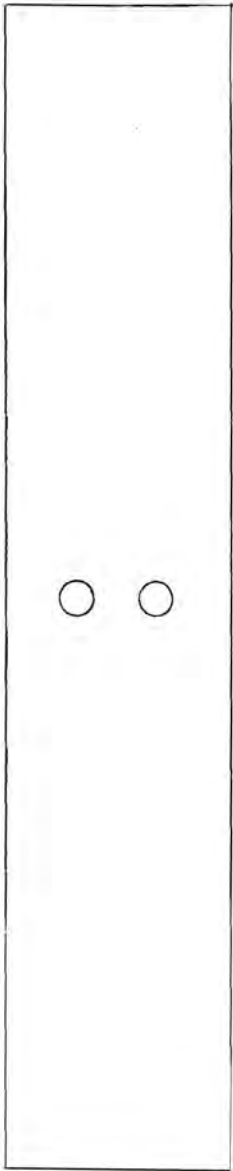
118



116

10mm Hex

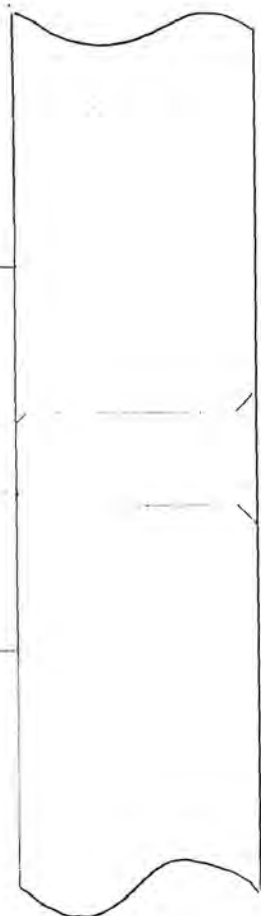
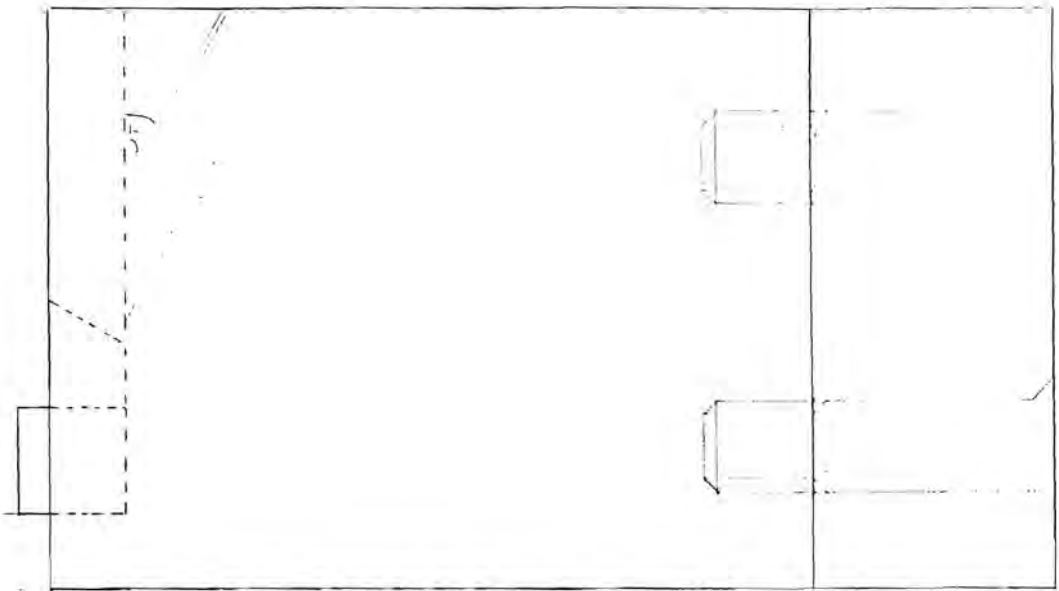
LOAD CARRIER  
(FITS UNDER VIBRATING ARM)



2.4.11  
2.4.11

2.4.11  
2.4.11

UPPER PLATE OF UPPER  
SPECIMEN FIXTURE



NOTE: TOP PLATE SHOWN,

BUT NO DETAIL IS SHOWN

UPPER SPECIMEN  
(ROLLER) FIXTURE

THREADED HOLE FOR  
OIL BATH

102 X 51 CHANNEL

6mm PLATE WELDED  
ONTO CHANNEL AND  
DRILLED TO MATCH SHAFER

6  
125  
51

150

VIBRATING  
ARM

