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A TEST FACILITY FOR THE IN-VACUO ASSESSMENT OF DRY LUBRICANTS AND SMALL MECHANISMS AT CRYOGENIC TEMPERATURES

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ABSTRACT

The development of cryogenically cooled spacecraft mechanisms has created a requirement for efficient lubrication in vacuum at very low temperatures. Only dry lubrication is practicable under these conditions, but at present there exists a lack of basic tribological data upon which cryo-mechanism designers can base a reasoned choice of lubricant. This paper describes the development of a vacuum, cryogenic test facility which has been employed to test ball bearings lubricated with different types of dry lubricant. Initial measurements on the efficacy of these lubricants at a temperature of 20K are reported. The use of the facility to test gears and small mechanisms, and to measure thermal conductance across load-determined contact areas, at cryogenic temperatures is also discussed.

KEYWORDS: Cryogenics, Tribology, Solid lubrication, Space.

1. INTRODUCTION

Cryogenics is an important technology of the space age. It was first promoted on an industrial scale by the post-war space programmes. In particular, cryogenic fluids are important for spacecraft propulsion systems and most of the early research was in support of the development of seals and bearings for engine turbopumps. More recently, spacecraft mechanisms which operate at cryogenic temperatures, both in vacuum and in inert atmospheres, have become of interest. Indeed, over the next few years several space missions are planned which depend on instruments which require cryogenic cooling. These instruments include infrared detectors, superconducting devices, and a variety of 'telescopes'- infrared, X-ray, gamma-ray, and high energy. Infrared detectors are of particular importance in both astronomy and in earth applications instruments (such as instruments which monitor terrestrial weather, resources, and pollution). Future missions include the Infrared Space Observatory (ISO), the Far Infrared Space Telescope (FIRST), and the Upper Atmosphere Research Satellite (UARS).

Depending on their particular application, many low temperature devices will contain moving mechanical parts in which contacts may either be pure sliding (as in screw threads) or rolling with sliding (as in ball bearings). These components must be lubricated to ensure their proper operation. However, oils and greases will solidify and be rendered ineffective at cryogenic temperatures so that only dry (solid) lubrication is feasible. Whilst it is known that molybdenum disulphide (MoS_2), lead, and PTFE are effective lubricants under vacuum at or near room temperature, their frictional properties at very low temperature (20K and below) have been little studied. It is by no means clear how such lubricants behave at low temperatures and thus some means of characterising their tribological behaviour at 20K and below is required. This paper describes the development of a cryogenic test facility whose primary purpose is to facilitate the study of dry lubricants at low temperatures. Measurements of the torque and torque noise of bearings coated with PTFE, lead and MoS_2 when operating in vacuum at temperatures below 20K are reported.

2. THE CRYOGENIC TEST FACILITY

The cryogenic test facility is shown in figures 1 to 3. The refrigeration is produced by a Philips PGH 107S Stirling cycle cryogenerator. This is a two stage expansion engine which circulates helium gas at two quasi-independent stable temperatures - nominally 80K and 20K. Because of its attendant noise and vibration, the cryogenerator and its peripherals are installed outside the laboratory in which the cryostat is located. The cold helium gas is circulated between the cryogenerator and the cryostat by means of a vacuum insulated transfer line which contains four pipes - a 'supply' and a 'return' for each stage.

The cryostat consists of a high vacuum chamber of diameter 40cm, which contains two thermally isolated copper 'cryo-pots'. These cryo-pots are concentrically mounted, with the inner suspended from the outer by means of three pieces of PTFE. The outer cryo-pot is supported within the vacuum chamber by three stainless steel tubes which are welded to the wall of the vacuum chamber. Heat conduction along these tubes is reduced by including PTFE insulators in the supports.

The outer and inner cryo-pots are maintained at nominal temperatures of 80K and 20K respectively by the cryogenerator. Thermal contact to the individual cryo-pots is made by 'peening' the appropriate supply pipe into the base of the appropriate cryo-pot. The inner cryo-pot, in which test rigs are housed, has an internal diameter of 250mm and a depth of 225mm. Access to the cryo-pots is by way of removable lids on both the vacuum chamber and the cryo-pots.

2.1 The cooling power of the facility

The refrigeration produced by the unloaded cryogenerator is nominally 190W at 80K and 60W at 20K. During the commissioning of the facility, the refrigeration available at the 20K cryo-pot was measured using an electrical heater located on the 20K supply line (figure 3). The heater consists of two heating elements wound (in series) onto a stainless steel block which is clamped to the 20K supply pipe. (The heater is located outside the 20K cryo-pot because it is ultimately to be used in controlling the temperature without interfering with the test apparatus). The resistance below 40K was 39.8 ohms. A cryogenic linear temperature sensor (clts) was used to indicate the temperature. This was supplied with a three point calibration (295K, 77.3K, and 4.2K).

The cooling power below 19K is 25W. The cooling power of the 80K cryo-pot has not been measured.

3. TRIBOLOGICAL TESTS AT CRYOGENIC TEMPERATURE

The efficacy of three important solid film lubricants at cryogenic temperature has been examined by measuring the torque of dry lubricated ball bearings operating in vacuum at room temperature (293K) and at 20K. A ball bearing test rig, specially designed for use in the cryogenic facility, was employed for these tests.

3.1 The ball bearing test rig

The ball bearing test rig is shown in position in the vacuum chamber in figure 4. The apparatus is designed to test a pair of 20mm I.D. angular contact bearings arranged in a face-to-face configuration. It consists of three main parts: the bearing housing, the rotatable shaft, and the torque transducer. As spacecraft bearings are manufactured to high precision, it is usually important to choose the construction materials for both the bearing housing and the drive shaft with a view to minimising differential thermal contraction. In addition, it is important to avoid any volumetric changes associated with the austenite - martensite transformation which occurs in austenitic stainless steels when they are cooled to cryogenic temperatures (ref.1) . This latter problem, which is not well documented, has been avoided by choosing low carbon mild steel as the construction material for both the bearing housing and the bearing mounting located at the base of the shaft.

The housing in which the bearings are located is clamped to the base of the 20K cryo-pot. A 40N preload is applied by means of a stainless steel 'deadweight' which rests on the outer race of the top bearing.

The drive shaft consists of a stainless steel tube of diameter 0.66 cm and wall thickness 0.047cm. It is driven by a motor located on top of the vacuum chamber and enters the vacuum chamber through a ferrofluid, magnetic hollow shaft feedthrough. In order to facilitate the removal of the main flange of the vacuum chamber without disturbing the apparatus, the drive shaft contains a demountable coupling. This consists of three stainless steel pins which engage in sockets machined into a cylindrical piece of polyacetal. It is located just below the main flange. The shaft is connected at its bottom end to the bearing support by way of a second universal joint.

A carbon resistance thermometer is located at the base of the shaft as indicated in figure 4. The thermometer leads travel along the drive shaft and also contain demountable connections to assist in removing the main flange. The lower part of the drive shaft is cooled only by thermal conduction along the bearing housing and across the bearings themselves.

One novel feature of the apparatus is the method used to measure the bearing torque: a torque transducer is mounted integral with the drive shaft and rotates with the shaft. The usual practice of coupling the torque transducer to the bearing housing has not been adopted as this would require operating the transducer outside the recommended temperature range. The five leads required for its operation are connected to a slip ring via a vacuum feedthrough which is installed on-axis. Unfortunately, the slip ring only contains five rings so that, at the present time, the carbon resistor mounted at the base of the drive shaft can only be monitored when slow rotation speeds are employed.

3.2 Solid lubricants tested

Solid lubricants fall into one of three categories. These are:

a) Soft metallic films (e.g. lead, indium, gold, and silver).

b) Crystalline solids having laminar structure (e.g. molybdenum disulphide and tungsten diselenide).

c) Polymers (e.g. PTFE and polyimides).

In our tests we have selected a vacuum compatible lubricant from each of the above categories and examined its lubricating power in ball bearings operating at cryogenic temperatures. The chosen lubricants were lead, MoS_2, and PTFE. All three are commonly employed on spacecraft mechanisms operating at or near room tempertaure (say between -40 deg. C to +60 deg. C), but their comparative lubricating properties at very low temperature are not well documented. The manner in which lubricants were applied to the bearing surfaces is described below.

3.2.1 Lead. A feature of soft metal lubricants is that their effectiveness varies with film thickness. To obtain the lowest friction the lubricant should be applied in the form of very thin films, in the order 1 micron thickness. It is also preferable to employ such lubricants in conjunction with hard substrates - a condition readily met when dealing with bearing steels. The method of application employed in the present study was to deposit thin films of lead by the technique of ion plating (ref.2). This method gives rise to strong adhesion between film and substrate which in turn helps to ensure high film endurance. This method was employed to coat the inner and outer races of the test bearing; the balls were not coated. The ball bearings were fitted with a lead / bronze cage, the impregnated lead serving to replenish the lubricant as the cage wears.

3.2.2 Molybdenum disulphide. Molybdenum disulphide, an example of a laminar lubricant, was applied by the technique of magnetron sputtering (ref.3). Again this method leads to the formation of thin layers (typically 1 micron thick) which are well adhered to the substrate surface. This lubricant is of particular interest as, in vacuum, it gives rise to the lowest friction coefficient obtainable with solid lubricants (ref.3). The bearing races were coated with sputtered molybdenum disulphide whilst the balls were left uncoated. The bearings were fitted with steel cages, the pockets of which were also coated with sputtered MoS_2.

3.2.3 PTFE. Polymeric lubricants are mostly utilised by either rubbing transfer from a reservoir, such as for example a self-lubricating cage in a bearing, or, by manufacturing one of the sliding contacts from the chosen polymer. The former method was employed in the present tests in that bearings were fitted with a cage (ball retainer) made of a composite material comprising PTFE and glass fibre (strengthener). This material also contained a small amount of molybdenum disulphide. No lubricant coatings were applied to either the balls or the races, the cage being the sole source of lubrication. This method of lubrication relies on the transfer of PTFE from the cage to the balls and thence to the raceways. Thus a period of running-in is required to establish a transferred layer of lubricant. Accordingly, the bearings were run-in prior to testing.

3.3 The experimental procedure

All the measurements described in this paper were performed on 20mm I.D. angular contact bearings. The bearings were subjected to an axial preload of 40N and the shaft was rotated at 0.5 r.p.m.. Measurements of torque were made initially at room temperature in high vacuum (1.10^{-5} mbar). These measurements are characterised in terms of mean torque and peak-to-peak (hash width) torque. Following room temperature testing the temperature of the bearing housing was reduced to 20K and, by virtue of the colder surfaces within the test chamber, the pressure reduced to 5.10^{-8} mbar. Torque measurements were again taken. At each temperature condition the bearings were run over some 20 revolutions only; the intention being to determine what **immediate** effect, if any, operation at very low temperature had on the efficacy of the selected lubricants. Thus these measurements do not reflect the long term performance of such lubricated bearings, though such measurements are indeed planned.

One particular difficulty in measuring bearing torque in vacuum at low temperature is that a design compromise is required between the thermal and mechanical requirements of the apparatus. That is, it is necessary to minimise the 'heat leak' along the shaft to the bearing whilst at the same time making the shaft sufficiently robust and stable. The practical upshot of this is that it is necessary to choose between a 'hot' shaft and a high 'background torque' attributable to the eccentricity of the shaft. The background torque is periodic with the rotation, and is currently of the order of 50 g.cm peak to peak. However, a mean level can easily be identified and measured. These measurements are corroborated by the fact that room temperature measurements are consistent with measurements taken, for comparison, on a rig capable of higher accuracy.

Throughout the present measurements with the bearing housing at or below 20K, the temperature sensor located at the base of the shaft indicated a temperature of 140K. This shaft temperature is probably caused by the low thermal conductance of the bearing together with the heat leak to the bearing. (The heat leak is estimated to be 0.1W). A 'mechanical heat switch' is currently being designed to help cool the shaft.

3.4 Results and discussion

Table 1 lists the results obtained thus far. These results confirm that at room temperature in vacuum the lowest bearing torques are obtained using sputtered MoS_2. However, on decreasing the temperature to

20K an increase in torque is observed on all bearings. These increases are such as to lead to torques which are similar for all the bearings tested. However, the lowest noise is exhibited by the bearing treated with MoS_2.

To a first order approximation the torque developed by a dry lubricated angular contact ball bearing is proportional to the friction coefficient pertaining to the ball / race interface. Thus any thermally induced changes in bearing torque must derive from the frictional properties of the interface. It follows that any attempt at explaining such torque changes must, perforce, start with an explanation of how the friction coefficient might vary with temperature.

At a dry lubricated sliding interface the friction coefficient, u, at constant load W, is governed by the shear strength of the lubricant film, s, and the true contact area, A. This follows from the equations (ref.4) below:

$$\text{Frictional force } F = u.W = s.A \qquad (1)$$

$$\text{Hence} \qquad u = \frac{s.A}{W} \qquad (2)$$

Thus the temperature dependence of the friction coefficient will depend on the temperature dependence of the product s.A.

Let us first examine the contact area "A". This will be determined by the substrate material which in our case is 52100 steel and which is common to all three tests described here. Given that the contact area is described by the Hertzian equations, we may say that, for a given load,

$$A = k.E^{-2/3} \qquad (3)$$

where E is the elastic modulus of the bearing steel and k is a geometric constant of proportionality. (Note that equation (3) is an approximation in which it is assumed that the square of Poisson's ratio for 52100 steel approximates to zero at all temperatures of interest).

The variation of A depends on how E changes with temperature. It is reported by Leadbetter (ref.5) that the elastic modulus of most metals and alloys increase by between 10% and 15% as temperature is reduced from 300K to 0K. Thus we would expect the contact area to decrease by, at most, 10%. Thus the predicted effect on friction coefficient of decreasing the temperature of the substrate from 300K to 20K is small and corresponds to a reduction of, at most, 10%.

We next address the question of how the shear strength of the tested lubricants might change with temperature.

Simon et al (ref.6) have measured the shear strength of lead as a function of temperature and found that on going from 300K to 20K the shear strength **increases** by a factor of 4.5. The shear strength of PTFE will also increase with decreasing temperature, a fact which accounts for the increased friction which accompanies the rubbing of this material against a counterface as the temperature is reduced (ref.7). However, we do not have quantitative data on the shear strength of PTFE as the temperature approaches 20K, although Minhas and Petruccelli (ref.8) report its value down to 200K at which temperature its shear strength is approximately twice its room temperature value. Still less is known of the temperature dependence of sputtered MoS_2. However, it is known that the shear strength of sputtered MoS_2 films is highly sensitive to water vapour even under conditions of high vacuum (ref.9). Thus the friction of sputtered MoS_2 is dependent on the surface population of adsorbed water molecules. The size of this population is determined by the net rate of desorption (or adsorption) of water molecules. Since the rate of desorption is an exponential function of temperature, we would expect it to be reduced to essentially zero at 20K such that any water molecule which is adsorbed on the MoS_2 is likely to remain there. On this basis we would predict an increase in friction coefficient, and hence an increase in bearing torque, at cryogenic temperatures.

In summary, the above analyses predict an increase in bearing torque proportional to sliding friction coefficient on reducing the temperature from 300K to cryogenic temperature, independently of whether the bearing is lubricated by PTFE, lead or MoS_2. Table 1 shows that this is precisely the case.

4. FUTURE WORK

The following activities will be undertaken at cryogenic temperatures on the ESTL cryogenic tribology facility.

4.1 The further assessment of dry lubricants

Friction coefficient, endurance and wear-rate will be measured using a pin-on-disc apparatus. This is a standard technique which involves sliding a hemispherically tipped 'pin' against a rotating disc coated with the lubricant under test. The wear-rate is calculated from the diameter of the pin wear-scar and the coefficient of friction is obtained from the torque transmitted to the (stationary) pins. Sputter-coated molybdenum disulphide, ion-plated lead and PTFE are again of particular interest.

4.2 Measurement of torque and wear of gears

Measurements will be made of torque and gear-tooth wear on plasma nitrided gears treated with solid lubricants, titanium nitride coated gears with (and without) solid lubricants and a polyimide pinion against stainless steel.

4.3 The testing of mechanisms

It will be possible to test small mechanisms in a simulated space environment. This will basically extend current mechanism testing facilities at ESTL to very low temperatures.

4.4 Thermal conductance across loaded contacts

Knowledge of the thermal conductance of loaded contacts is particularly important for equipment whose performance is temperature dependent. As an illustration, the system-to-instrument interface of many spaceborne infra-red telescopes is a bolted joint. Also, in many cryogenic spacecraft mechanisms the only means of heat transfer is by conduction through bearings. It is therefore important to have an understanding of the thermal conduction across the ball-to-raceway contacts. Methods of measuring and maximising the thermal conductance across loaded contacts in general will be studied.

4.5 Future extension of the cryogenic facility

It is intended to reduce the base temperature of the facility to 4.2K by including a Joule-Thompson expansion valve. This valve will be independent of the cryogenerator although the cryogenerator will be used to precool the helium gas circulated through it. The valve will be used to cool a third cryo-pot which will be located inside the other two. The refrigeration available at 4.2K will nominally be 1W.

5 CONCLUDING REMARKS

A test facility, designed primarily for tribological studies at cryogenic temperatures, has been developed and commissioned. Preliminary measurements of the frictional torques developed by ball bearings treated with three commonly employed "space" lubricants (MoS_2, lead, and PTFE) indicate that frictional torque increases with decreasing temperature. This torque increase is attributed primarily to increases in the lubricants' shear strength as the temperature is decreased.

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TABLE 1

LUBRICANT	HOUSING TEMPERATURE	BEARING TORQUES (X10^-4Nm) MEAN	Peak - Peak
LEAD	295K	10	6
	17K	21	13
MoS_2	295K	5	8
	17K	26	8
PTFE	295K	21	38
	17K	24	57

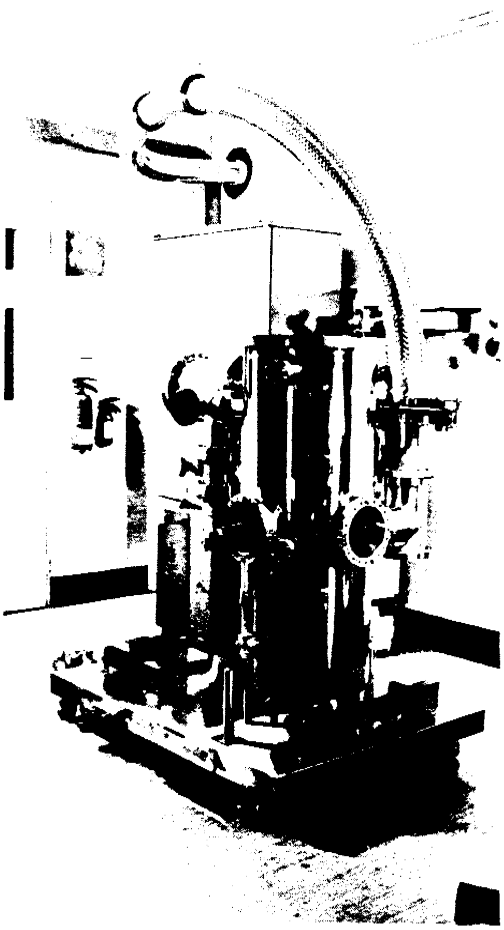

Figure 1. The ESTL cryogenic facility.

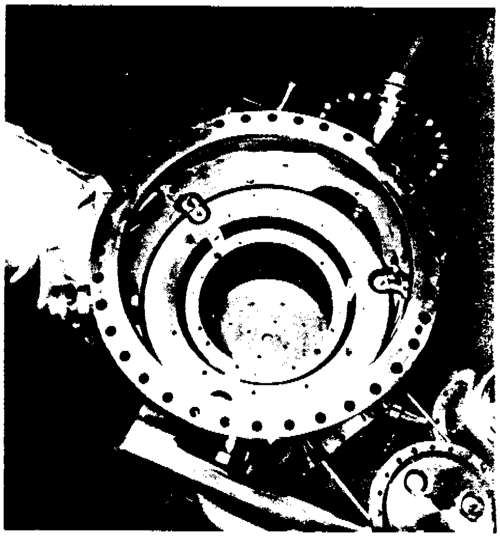

Figure 2. The copper cryo-pots within the vacuum chamber.

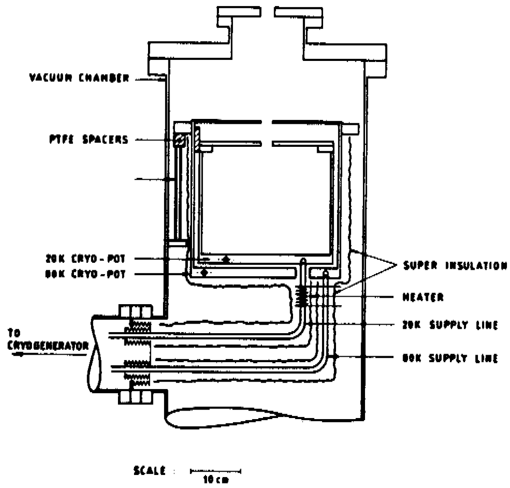

Figure 3. A schematic of the cryogenic facility.

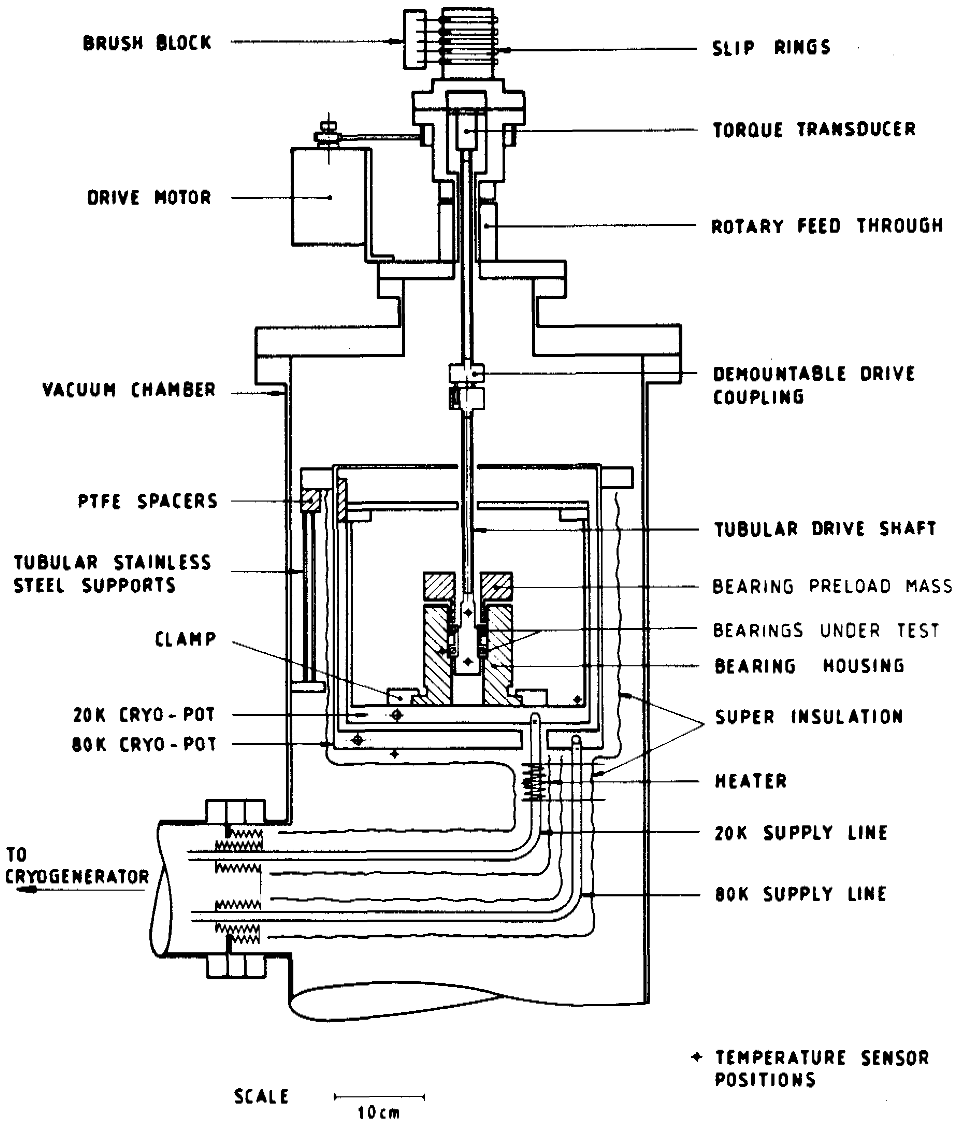

Figure 4. A schematic of the ball bearing test apparatus.