New Design Concepts for Low Friction Plain Journal Bearings

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Abstract

An unlubricated journal bearing or bushing is simply a cylinder with a hole in the center. It normally supports a translating or rotating shaft. Examples include door hinges, caterpillar tracks and pistons. They are simple and inexpensive but can be prone to friction and wear. This can lead to them seizing or binding especially in dirty environments. This paper will present 2 new inexpensive designs for journal bearings which reduce the problem of wear and seizing inherent in existing journal bearings.

Keywords*:* Journal Bearings, Low Friction, Low Wear

1. Introduction

Bearings come in many different shapes and sizes. There are roller, ball and thrust bearings [1]. They can be grease lubricated, oil lubricated, linear or rotary [2]. The simplest type of bearing is a plain unlubricated journal bearing. It is simply a cylinder with a central hole that supports a rotating or sliding shaft. Normally made from a material such as brass or PTFE that has a low coefficient of friction [3]. These are simple and inexpensive but can suffer from variable friction and high wear in comparison with the lubricated or ball bearing types [3].

In order to design a low friction non-lubricated bearing it is necessary to understand the nature of friction. Friction between two sliding surfaces comes from three sources [4]. The first is adhesion. This is the molecular attraction between two surfaces in close contact. It is dependent on the material type and the kind of molecular bonds, such as van der Waals forces and capillary forces [5]. At a microscopic scale these molecular bonds can cause the surfaces locally to 'weld' together [6]. Breaking this weld forms tiny particles or debris.

The second contribution to friction comes from asperity interaction [7]. On a microscopic scale no surface is perfectly smooth. A series of peaks and valleys exists on all surfaces. When a peak from one surface tries to slide against a peak from the second surface a force is required to overcome this interaction. This force contributes to the friction coefficient.

The third contribution comes from particles or debris [8]. These particles may be present in the environment. For example, the caterpillar tracks of construction equipment have bearings operating in dirty environments. Even if the environment is clean, debris can be generated on the bearing surface from adhesion and asperity interaction, as shown in figure 1.

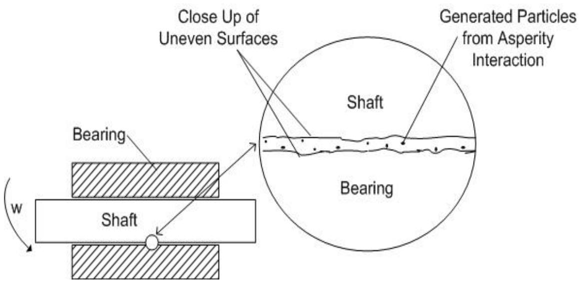

Figure 1: Asperity interaction and particle generation

This is why the coefficient of friction generally rises with short time periods [8]. It becomes a stable value when the rate of particle generation equals the rate at which particles are removed from the bearing surface. Wang [9] reviewed studies on wear mechanisms causing journal seizing failures.

Research carried out by Suh [4] challenged conventional views on the relative importance of the 3 friction sources. Traditionally it had been held that adhesion and asperity interaction were the primary sources of friction with plowing being a distant third [10]. Through a series of experiments, Suh demonstrated that this conventional thinking needed updating. He concluded that plowing is the dominant source of friction. The exact contribution from plowing obviously varies from situation to situation but to a first order of magnitude it can account for 50-60% of the friction coefficient. The remaining balance is distributed between adhesion and asperities. The implications of this mean that the friction coefficient is no longer just a function of the materials used. It also depends on time, the environment, surface roughness, relative surface hardness, load and design. Designs with undulating surfaces have been shown to have higher resistance to friction and wear [11]. This introduces additional complexity in the calculation or estimation of the friction coefficient but also presents opportunities. By providing a mechanism for the removal of particles, friction and wear can be reduced.

Traditional approaches to lower friction include increasing the harness of one surface [3], so that the plowing is only concentrated on one soft surface as shown in figure 2.

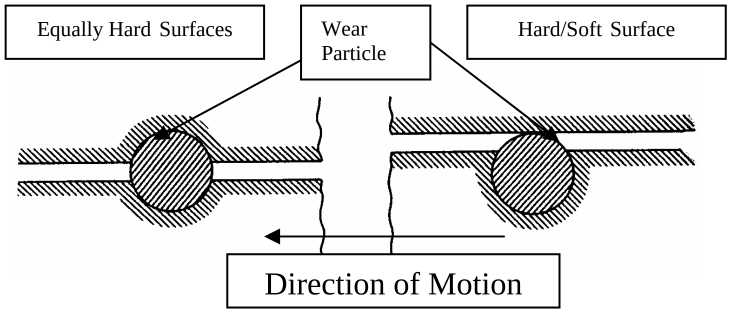

Figure 2: Surface plowing by debris

Another approach is to include particle traps which entrain moving particles [11]. This approach has not been very popular since they are expensive to manufacture and are prone to increased stress and fracture. Kohyama et al. [12] used a photolithography microtexturing process to etch microgroves into surfaces. This technique is not cost effective for plain commercial bearings.

Using a lubricant washes the surface and clears away much of the debris. This reduces the contribution from plowing. The lubricant also reduces surface to surface contact which reduces both the asperity and adhesion contributions. For these advantages, the penalty paid is increased cost and maintenance. There are certain environments where it is not possible to use a lubricant, such as clean rooms, high vacuum systems or at high temperatures.

In this paper, two plain journal bearing designs will be introduced. They both have the intention of producing a more steady friction coefficient and less wear by reducing the effects from plowing, asperity interaction and adhesion.

2. Concentric Ring Design

This proposed design consists of a series of concentric brass rings. Each ring has an inner diameter of 12.1mm and a thickness of 1mm. They are spaced 2.5mm apart and there are 9 rings in total. The shaft fits through the center of these rings. They are supported by a layer of plastic or any semi-flexible material. The plastic housing used in this design was selected to be polyurethane but that does not limit other plastics from being used instead. The inner and outer diameter of the plastic housing is 14.2 mm and 26 mm respectively. This assembly is enclosed in a stainless steel sheath, as shown in figure 3.

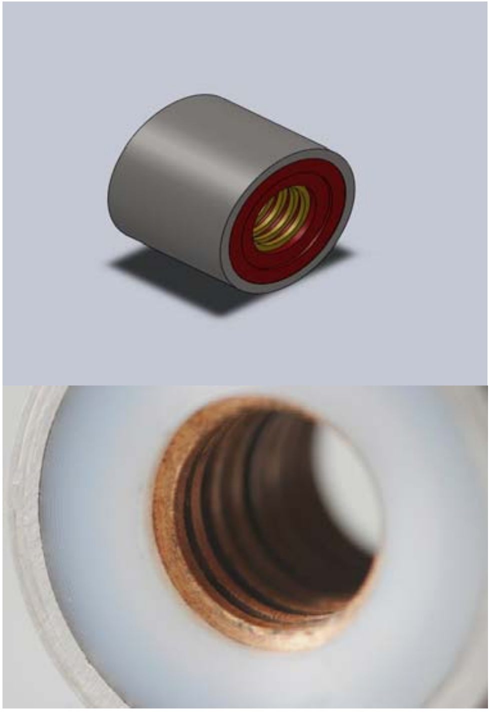

Figure 3: Concentric ring design concept

This design is intended to stabilize friction and wear between the shaft and bearing via the following mechanisms. Adhesion is reduced because there is less surface contact between the shaft and bearing. Take a 32mm long bearing that has been replaced by this design with 9 rings. Each ring is 1mm wide meaning the line of contact is reduced by a factor of 3.5. Assuming adhesion to be proportional to the contact area this will reduce the contribution from adhesion to friction.

Particles between the shaft and the bearing surface are not trapped there. They can easily fall out via the particle traps as shown in figure 4.

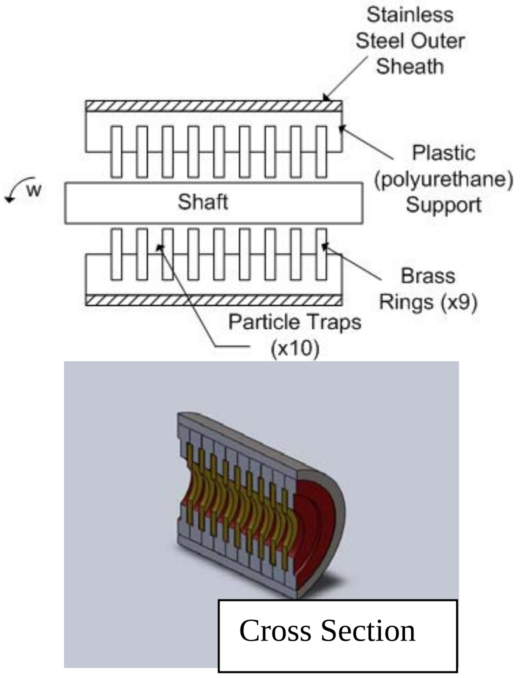

Figure 4: Particle traps in concentric ring configuration

The component from plowing is therefore considerably reduced.

The effects from asperity interaction should also be reduced. The supporting rings are not rigidly fixed in position; rather they are contained in pockets of polyurethane. Since the housing can compress and expand the rings have some flexibility in their position. The amount of motion depends of the stiffness of the plastic housing. Modeling the plastic as a spring, a locally compliant condition exists as shown in figure 5.

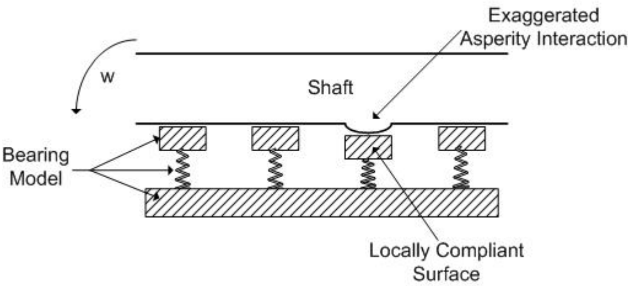

Figure 5: Asperity interaction with a locally compliant surface

If there are any large asperities on the shaft the ring can move as the shaft rotates against it. It can compress against the compliant surface. This reduces the friction from asperity interaction. Of course should the asperity break off, the particles generated can easily fall in between the ring gaps.

Another advantage of this design is that it is globally compliant. Take a plain bearing with a high load applied to either end. Figure 6 shows what happens to the load distribution. It becomes concentrated in just 2 small regions. This leads to high stress concentrations and wear in these regions.

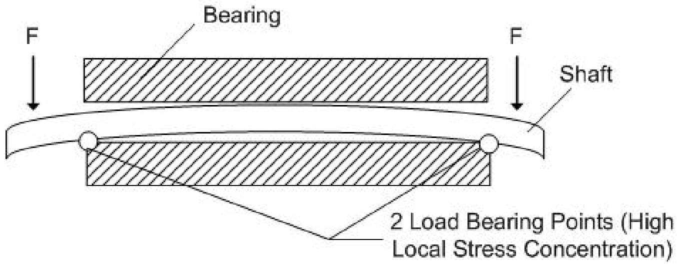

Figure 6: Load distribution in a plain journal bearing

The concentric ring design has a globally compliant surface. This causes a more uniform stress distribution throughout the bearing. This evens out stress concentrations and lowers wear. Figure 7 shows a model which demonstrates this basic principle.

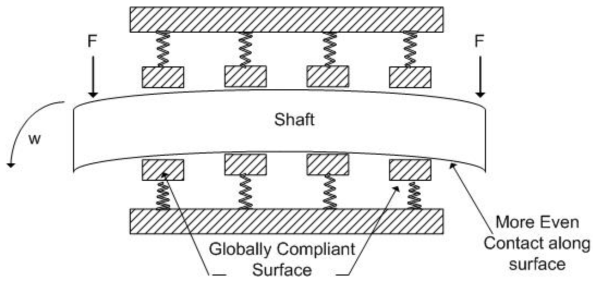

Figure 7: Globally compliant surface

Figure 8 shows a picture of the completed design. The bearing length is 32mm and is designed for a shaft diameter of 12mm.

Figure 8: Concentric ring bearing

The cost of a brass bearing having the same dimensions as the concentric ring design is ฿200. The cost of a single prototype concentric ring bearing is approximately ฿350. Using the cost of the pure raw material as a base and estimating the manufacturing cost to be 40% gives us a rough estimate of the production level cost. Accounting for a 30% profit margin, the final cost works out to be similar to the plain brass bearing, ฿200.

3. Inner Spring Design

In an effort to reduce cost and yet maintain many of the advantages of the concentric ring design an alternative design concept shall be introduced. This is simply a stainless steel helical spring coil which the shaft is mounted through. The spring used in this design was manufactured from 3mm diameter wire and has a 4mm pitch. The outside of the spring is supported with a steel sheath. The plastic housing may be used in the same way as the previous design if necessary. However for this particular design it may not be needed.

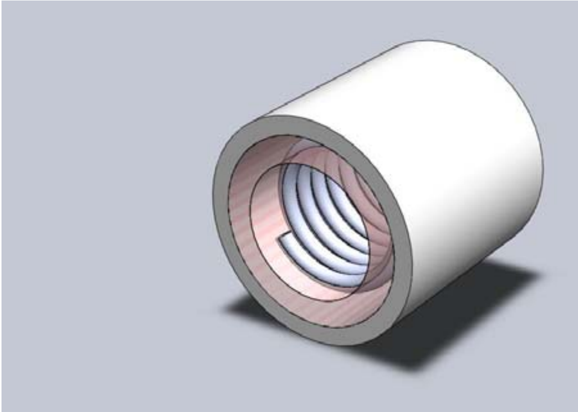

Figure 9: Inner spring bearing

Similarly to the concentric ring concept, adhesion is reduced because of the reduced contact area. Plowing is reduced because debris or small particles can fall between the spring spaces. Finally asperity interaction is reduced because the spring is a compliant surface. If a plastic housing is used the mechanism for reducing the asperity is exactly the same as shown in figure 5. However if there is no plastic housing a reasonable argument could be made that the spring is capable of compressing or elongating to accommodate the uneven surface. If there is a large asperity on the shaft, in theory it should only make contact with the spring once per revolution. In a regular bearing this asperity would be in constant contact with the surface. Figure 10 displays this mechanism.

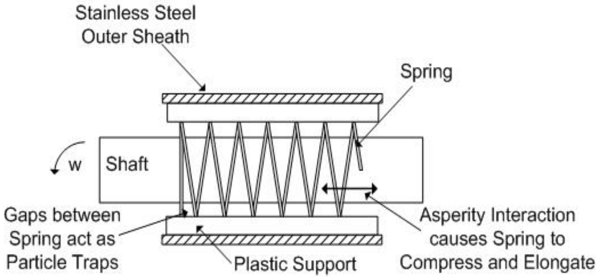

Figure 10: Asperity interaction with inner spring design

This design is also globally compliant. If a load is applied to both ends of the shaft the spring will tend to compress or buckle in the center. This will help to distribute the load. If a plastic housing is used the effects are shown in figure 7. Even without the plastic housing the spring will still buckle by itself. Eliminating the plastic housing reduces the cost but probably will reduce the compliance of the surface.

The single prototype cost was ฿400. The largest cost was the springs, which were manufactured to the required specifications for ฿ 250 each. In quantities of 1000 pieces the springs can be produced for ฿50 each. This would give a similar production level overall bearing cost of ฿200.

The manufactured prototype for this design is shown in figure 10. The length is 32mm and it accommodates a shaft diameter of 12mm. The inner spring diameter is 12.1mm; the spring wire diameter used was 3mm stainless steel with a 4mm pitch. A provisional patent was filed in Thailand, (patent pending #1001000704) for these designs.

Figure 11: Spring bearing design

Test Set Up

The next step is to design a build a test bed to test these bearings. Their performance should be compared with the plain bronze bearing. The test bed will apply a fixed load to the shaft inside the bearing and measure the torque and angular velocity over time. The coefficient of friction can then be calculated and displayed as a function time. The test bed design is underway.

4. Conclusions

In this paper 2 designs have been presented which should have the properties of lower friction and wear than traditional plain bearings. During use they should take much longer to seize. By examining friction and wear at a fundamental level insights were gained as to how bearings could be improved. Both of the designs should reduce the plowing component of friction by providing particle traps. The component from asperity interaction will be reduced due to the compliant nature of the surface. Adhesion should be reduced because there is less surface contact between the shaft and bearing. In a plain bearing the coefficient of friction tends to rise with time, it is expected that this will not happen in these designs.

Potential problems or issues with these bearings revolve around the compliant surface. Will the bearing be rigid enough to position the shaft accurately? What about the bearings repeatability and resolution? What will be its range for speed and acceleration? Will a preload be required? The only way to answer these questions is through thorough testing. Depending on the application the compliant surface can be made more rigid if needed at the expense of losing some of its flexibility. There is flexibility in the designs to allow for an optimal solution for a given application. If these bearings function as intended there may be applications where they can replace lightly lubricated bearings.

Estimates have shown that these designs can be economically competitive with plain bearings. Of course, these designs are merely concepts at this stage. Their exact properties cannot be quantified until properly tested. It is proposed to carry out this testing in the next phase of research.

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