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Time to Seizure Comparisons between Different Plain Journal Bearings

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Abstract
This paper is a continuation of the research which was presented at the first TSME conference. In that paper [1], two new concepts for plain journal bearings were presented. Their design takes advantage of the fact that the coefficient of friction is not just a function of the specific materials sliding together but also a function of time, load and environment. The coefficient of friction will increase with time until a steady state value is reached, where the rate of particle generation matches the rate at which particles are removed. The new designs took advantage of this information by incorporating particle traps and compliant surfaces to reduce friction. Since then an experiment has been designed and built to test both new design concepts against a regular plain journal bearing. The experimental data did not produce absolute data on the friction coefficient but it did allow a relative comparison between the different bearings. A time to seizure was produced in every case. The new concepts produced time to seizures ranging from 3 times greater than the regular bearing to over 5 times greater.
Keywords: Journal Bearings, Low Friction, Low Wear, Seizure Times

1. Introduction

Journal bearings are widely used in engineering applications. Typically they are used in low cost situations where the performance requirements are not as stringent as in the case with ball bearings [2]. Journal bearings are simple and inexpensive but can suffer from variable friction and high wear in comparison with the lubricated or ball bearing types [3]. An example of a plain journal bearing is shown in Figure. 1.

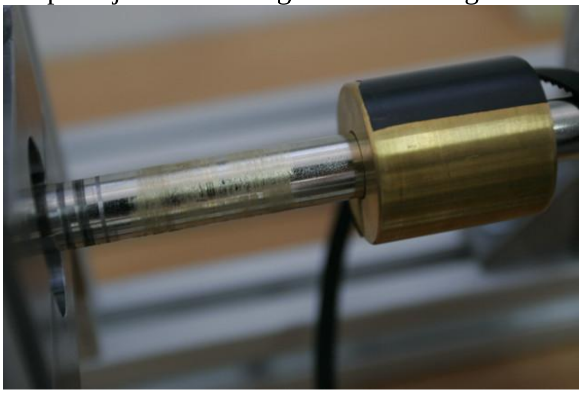

Figure. 1 Plain journal bearing on a shaft

In a paper published at the first TSME conference [1] two novel inexpensive bearing concepts were proposed. They both took advantage of fundamental studies on the nature of friction by Suh et al. [4]. Other researchers have examined the specific components of friction, such as adhesion [5], asperity interaction [6] and plowing [7]. Both of the designs proposed before were intended to reduce the effects of friction from each of these three components. This is achieved through having less surface contact, compliant surfaces and particle traps. In this paper an experiment was designed and built in order to compare the performance of these bearings with that of a regular plain journal bearing. When a bearing generates enough particles on its surface it can seize. Eventually, given enough time and use all journal bearings seize. By developing a test which runs the bearings until they seize a relative comparison can be made.

Both of the new designs shall be briefly described once more before describing the experiment. More complete details of their design can be found in [1].

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 in a housing of polyurethane. This assembly is enclosed in a

2mm thick stainless steel sheath, as shown in Figure. 2.

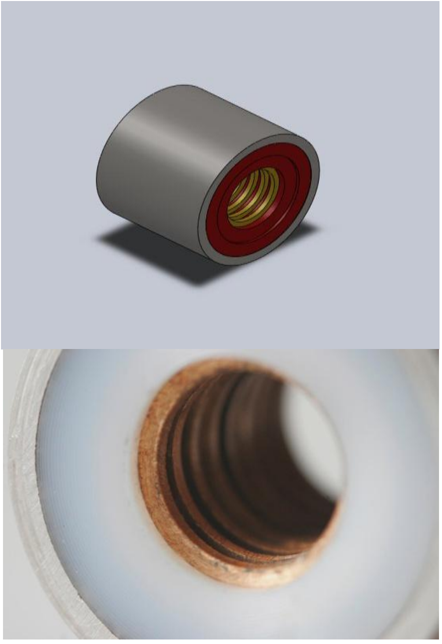

Figure. 2 Concentric ring design concept

Adhesion should be reduced because there is less surface contact between the shaft and bearing.

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. 3.

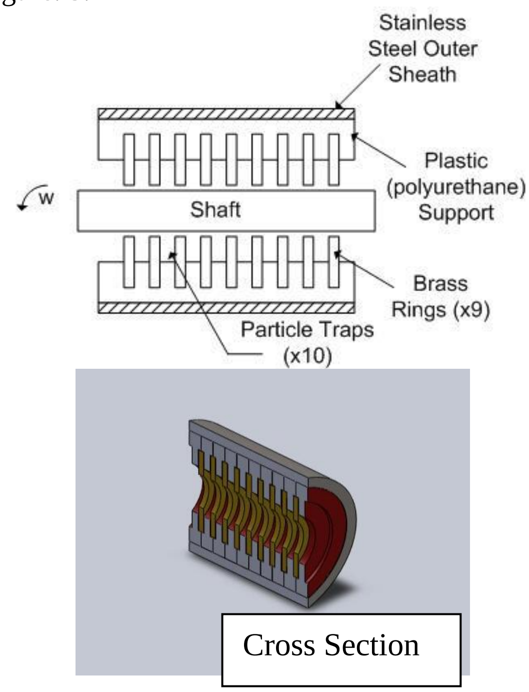

Figure. 3 Particle traps in concentric ring configuration

The component from plowing is therefore considerably reduced.

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.

Figure. 4 shows a picture of the completed design. The bearing length is 26mm and is designed for a shaft diameter of 12mm.

Figure. 4 Concentric ring bearing

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 was introduced. This is simply a stainless steel helical spring coil which the shaft is mounted through. The outside of the spring is supported with a steel sheath. The plastic support structure may be used in the same way as the previous design, if necessary. The spring used in Figure. 5 was manufactured from 3mm diameter wire and has a 4mm pitch. Two other spring bearings were manufactured with 2mm diameter stainless steel wire and a 5mm and 3mm pitch respectively.

Figure. 5 Inner spring bearing

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. 6 displays this mechanism.

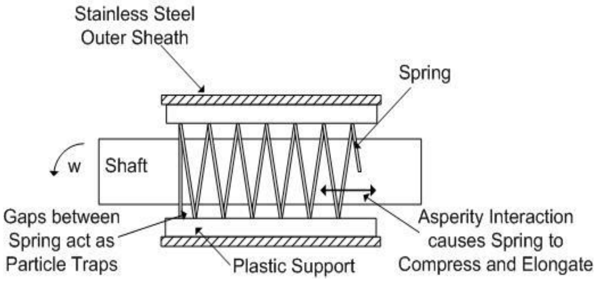

Figure. 6 Asperity interactions with inner spring design

The manufactured prototype for this design is shown in Figure. 5. The length is 32mm and it accommodates a shaft diameter of 12mm. A provisional patent was filed in Thailand, (patent pending #1001000704) for these designs.

4. Test Set Up

The next step was to design and build a test bed to test these bearings. Their performance should be compared with the plain bronze bearing shown in Figure. 1. A preliminary test bed was constructed. It applied a load to the bearing under test and a shaft was rotated inside the bearing. The power going into the motor was recorded along with the angular shaft speed. The time before the bearing eventually seized was recorded.

The test bed that was built is shown in Figure. 7. It consisted of the following basic components:

- 12mm stainless steel shaft for testing
- Motor for driving the shaft (A salvaged NA4565D motor from NISCA motors)
- Bearing under test
- Belt/pulley system for power transmission
- Power Supply (Acbel 330 Watt power supply obtained from a salvaged computer)
- Aluminum frame
- 2 supports for the shaft
- Standard weight for applying the load
- Voltmeter/Ammeter
- Digital Tachometer

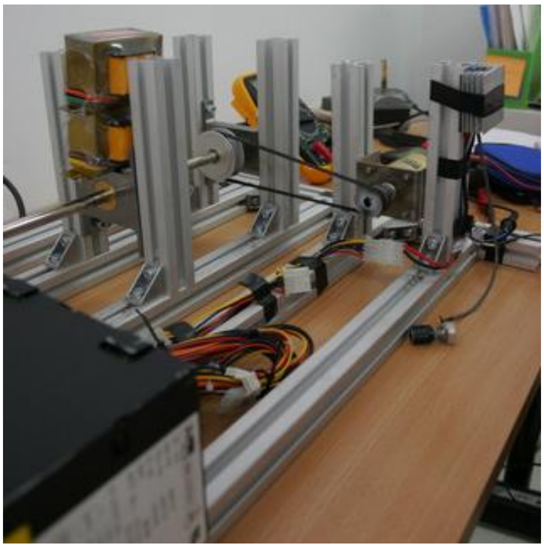

Figure. 7 Test Bed

5. Test Procedure

1. The bearing under test was setup and the shaft was aligned through it
2. A 2kg load was placed on the experimental bearing housing
3. The power supply was turned on and the shaft speed was adjusted until the angular speed was 750RPM
4. The voltage and current going to the motor was recorded every minute
5. The test was stopped when the motor stalled or overheated

6. Results

The principle behind the testing procedure was to be capable of comparing the relative performance of several bearings. The power going associated with the spinning shaft is given by:

$$P_{shaft} = \vec{\tau}.\omega \tag{1}$$

Where $\vec{\tau}$ is the torque on the shaft in Newton Meters and ω is the shaft angular speed in radians per second. The torque is given by:

$$\vec{\tau} = \vec{r}\, x\, \vec{F} \tag{2}$$

Where $\vec{r}$ is the displacement vector and $\vec{F}$ is the force on the shaft. Since the angle between the radius and the applied force is 90˚ and only the magnitudes are of interest, (2) becomes $\tau = r.F$. The force, F, comes from the weight placed on the shaft N, and the friction coefficient, μ. They are related through (3).

$$F = \mu.N \tag{3}$$

Combining Eqs.(1) – (3) gives:

$$P_{shaft} = \mu . N . r . \omega \qquad (4)$$

Or the friction coefficient is given by:

$$\mu = \frac{P_{shaft}}{N . r . \omega} \qquad (5)$$

Ideally the force required to turn the shaft would be measured with a digital torque meter. However given budget constraints this was not a feasible option. Instead the power going into the motor was estimated by measuring the voltage, V and current, I.

$$P_{motor} = \vec{V} . \vec{I} \qquad (6)$$

The power going into the motor is not going to be equal to the power going to the shaft. For a start there are losses in the motor and losses associated with the transmission system. If an assumption is made that the total losses, K_{loss} are a constant percentage of the incoming power then:

$$K_{loss} P_{motor} = P_{shaft} \qquad (7)$$

K_{loss} could be up to 50% or more of the input power. For this experiment we do not know what K_{loss} is. Substituting (7) into (5) gives the final value for the magnitude of the friction coefficient:

$$\mu = \frac{K_{loss} V . I}{N . r . \omega} \qquad (8)$$

All of the variables in (8) are measured during the test, except for the loss coefficient. If this loss coefficient remains constant during separate tests, a reasonable assumption since the motor efficiency is constant, the relative friction coefficients between the different bearings may be obtained. Valuable qualitative data may still be extracted from the experiments, even though the precise value for the friction coefficient eludes us.

Figure. 8 displays the data for one set of experiments. The conditions for each experiment are the same. The plot shows the friction coefficient divided by the loss coefficient on the y-axis. On the x-axis is the number of rotations. The data with the diamond symbol is the original

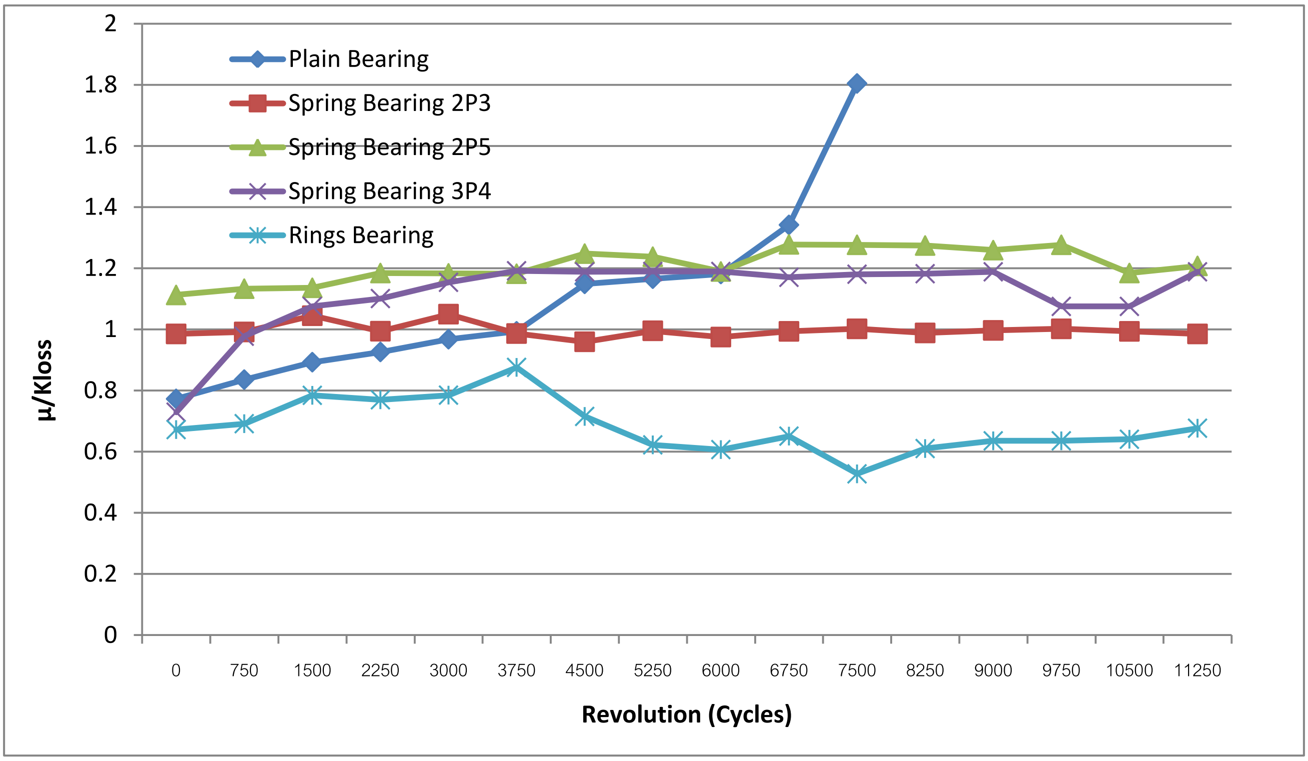

Figure. 8 Friction coefficient divided by loss coefficient for several bearings versus the number of revolutions

plain journal bearing, see Figure. 1. It begins at 0.8 and seizes up completely after approximately 7500 revolutions. The maximum ${}^{\mu}/_{K_{loss}}$ that the motor can deliver is 1.8. The light blue line is the data for the 'ring' bearing. It starts out at a slightly lower value of 0.7 and stays fairly constant over the course of the experiment, 11250 revolutions. The next 3 data sets come from 3 separate 'spring' designs. They each contain a slightly different spring. The brown line has a 2mm stainless steel spring with a 3mm pitch. The green line has a 2mm diameter stainless steel spring with a 5mm pitch. The purple line has a 3mm diameter stainless steel spring with a 4mm pitch. All appear to be fairly flat over the course of the experiment with the 2mm diameter 3mm pitch having the average lowest value.

A durability test was done on the 3 different designs. This was a similar experiment except that the shaft was spun inside the bearing until it seized. The length of time it took to completely seize up was recorded. The results are shown in Figure. 9. At a value of ${}^{\mu}/_{K_{loss}} = 1.8$ the motor seized. For the plain journal bearing the time it took to seize was approximately 10 minutes. The time it took for the spring bearing with the 2mm diameter 3mm pitch spring inside to seize was approximately 37 minutes. The experiment for the ring design was run for 55 minutes and then stopped as the motor was getting hot. At no point during the 55 minutes did the motor come close to stalling.

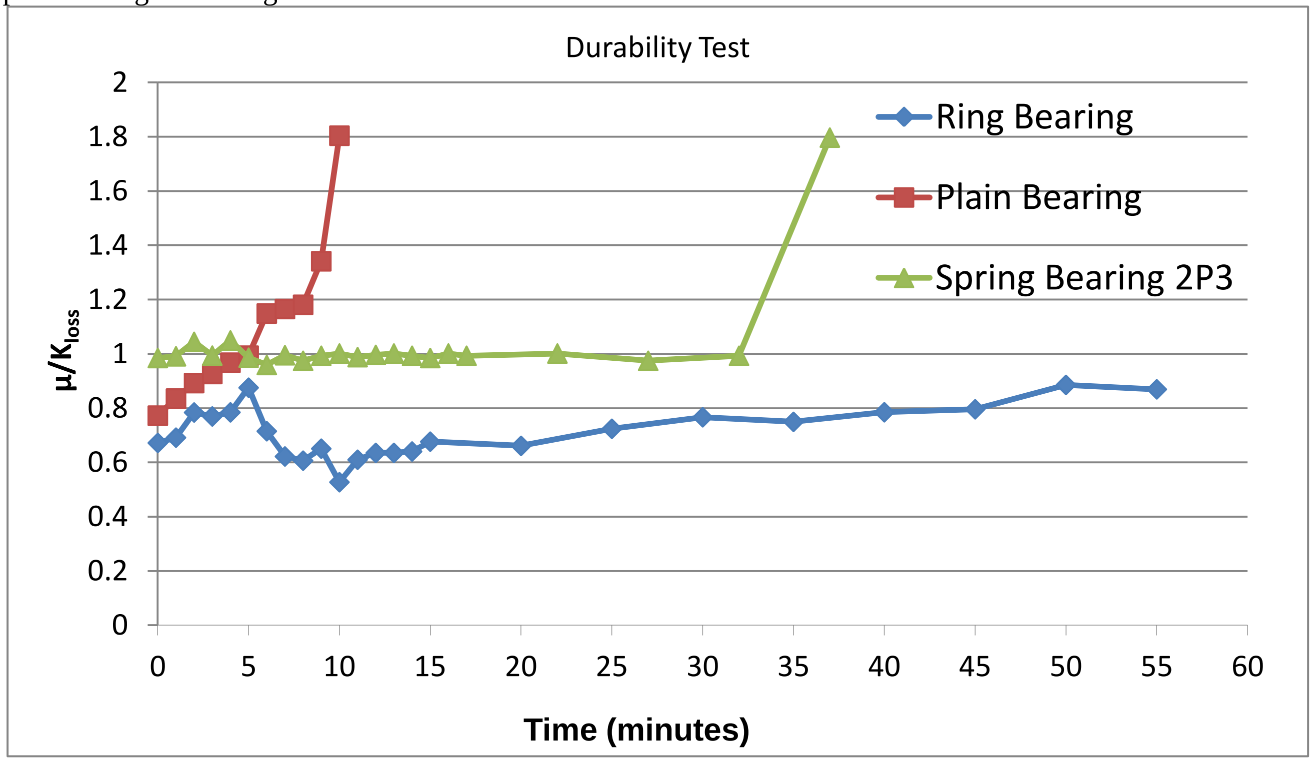

Figure. 9 Durability Test for three Bearing Styles

7. Discussion

In this paper two designs have been presented and initial testing carried out. The experiment allowed for the relative performance between the three different bearings to be observed. This has provided useful information.

The first conclusion is that for a regular bearing the friction coefficient is not constant. As can be seen from Figure. 8 the coefficient of friction depends on time as well as material properties of both interacting surfaces. This is consistent with the conclusions from Prof. Suh [8]. From the same figure it is possible to conclude that the new bearing designs have a friction coefficient that is more constant in time when compared with the regular bearing. The ring bearing has a lower coefficient of friction but the spring bearing initially appears to have a higher friction coefficient than the regular bearing. There is no satisfactory explanation at the moment as to why this is so.

After each test the stainless steel shaft was examined. It containing surface marks and scratches and was very rough. This was the case no matter which bearing was used. The shaft was replaced for each test. Each new test always started with two clean new surfaces.

Durability testing was run with the results displayed in Figure. 9. It shows that the best performing spring bearing lasted 3.7 times longer than the regular bearing. The ring bearing had outlasted the regular bearing by a factor of 5.5 before the test had to be stopped. It is unknown how much longer it would have be capable of running for.

There were issues with the experimental procedure. The most obvious was an inability to measure the shaft power directly. It had to be measured indirectly by monitoring the power to the motor. Also the test was not automated. Each piece of data had to be manually written down. This took some time and the conditions of the test may have changed as data was being collected. This probably accounts for some of the fluctuations in the data. The fluctuations were not commented on as it is impossible to tell if they are real or an artifact of the data recording time delay. The reason for these issues was due to financial and time constraints.

The ability of these bearings to handle radial loads needs to be addressed. The compressive strength of polyurethane is 10 times less than that of a metal. However at 138MPa it is still over 50,000 times stronger that the stress it was subjected to during this test. Since the purpose of the bearings are not only to support a load but to allow the load to move it suggests that the limit of not allowing motion would be reached far before the bearing fails in compressive strength. Other stresses such as fatigue, thermal loading and creep need to be addressed and quantified.

The next step is to design and build an experiment that takes into account these deficiencies and allows the collection of more accurate data. The experiment will need to measure the power on the shaft directly and take the data automatically. The exact friction coefficient and the effect of different spring size, ring size, plastic support and housing material needs to be characterized. The present results are very encouraging and can be used a guide for further research. Presently this work is being undertaken at Chiang Mai University.

8. Acknowledgments

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9. References

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