Development of hydrodynamic micro-bearings

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Abstract. This paper describes the modelling and testing of mm-scale hydrodynamic bearings which are being developed to improve the efficiency of a cm-scale turbine energy harvester, whose efficiency was previously limited by poorly lubricated commercial jewel-bearings. The bearings were fabricated using DRIE and their performance was assessed using a custom built MEMS tribometer. Results demonstrate that acceptably low friction is achieved when low viscosity liquid lubricants are used in combination with an appropriate choice of friction modifier additive. Further reduction in friction is demonstrated when the step height of bearing is adjusted in accordance with hydrodynamic theory. In parallel with the experiments, hydrodynamic lubricant modelling has been carried out to predict and further optimize film thickness and friction performance. Modelling results are presented and validated against experimental friction data.

1. Introduction

Despite significant advances in MEMS design and fabrication technology over the past few decades, problems of high frication coefficient and wear rates have severely limited the performance, and so far prevented the commercial exploitation, of devices that include sliding components [1-3]. These issues arise due to surface forces (viscous drag, van der Waals and capillary forces) that dominate at small length scales due to high surface to volume ratios. In addition to this, silicon – the most common material for MEMS components – is brittle and has high surface energy [4].

Many researchers have attempted to devise effective lubrication methods for MEMS devices. For instance, bonding self-assembled monolayers (SAMs) to surfaces has been shown to be an effective means of preventing stiction and reducing friction in MEMS [5]. Such an approach is partially effective since SAMs reduce surface energy and make the contacting surfaces hydrophobic – hence preventing water vapour adsorption [6,7]. Another lubrication method that has been applied to the silicon surfaces of MEMS is dry coatings, such as diamond-like carbon (DLC), which has shown to give friction coefficients of less than 0.01 [8]. The above two lubrication methods, however, are not suitable to be applied to contacting surfaces that involve large amount of sliding, such as the bearings within a rotating machine. This is because SAMs and DLC coatings cannot replenish themselves and so have a finite wear life [9]. Vapour phase lubrication is another way to lubricate MEMS, which overcomes this problem. However, the expense of this method and the requirement for specialized hermetic sealing preclude its use in many applications [1].

Liquid lubrication is widely applied in macro-scale applications but is generally considered to be unsuitable for micro-scale devices as it has been assumed to result in excessive viscous drag [3] – a claim supported by observations of over damping and the poor performance of certain liquid lubricated MEMS devices [10]. However, it should be noted that the viscosity of the liquids used in those studies was relatively high; whereas recent research at imperial college [2,3] has in fact shown that MEMS-type contacts can achieve low friction provided sufficiently low viscosity lubricants are used.

The aim of this research is to further demonstrate the effectiveness of liquid lubricating MEMS devices by designing and incorporating micro-hydrodynamic thrust pad bearings into an existing cm-scale energy harvester. The bearings were fabricated using deep reactive ion etching (DRIE) and tested on a custom designed tribometer. To optimize these components, the bearing step height was controlled and friction results are compared to results from a lubrication model based on a finite difference solution to Reynolds' equation. The test results illustrate that the friction coefficient of the thrust pad bearing is lower than 0.04, and can be further reduced by including amine additives.

2. Experimental methods

2.1. Bearing fabrication

A range of hydrodynamic micro-thrust step bearings were designed and fabricated on a bonded silicon on insulator (BSOI) wafer using photolithography and DRIE. The resulting bearings consist of pairs of circular pads with radial steps to pressurize the lubricant as they rotate against each other (Figure 1 shows an SEM image of the bearing geometry). The key dimensions of the bearings are a 1 mm radius, a recessed channel angle of 45°, an inner recessed region with a radius of 0.5 mm, an outer radius recessed region with a radius of 0.95 mm and a step height ranging from 1 to 50 µm.

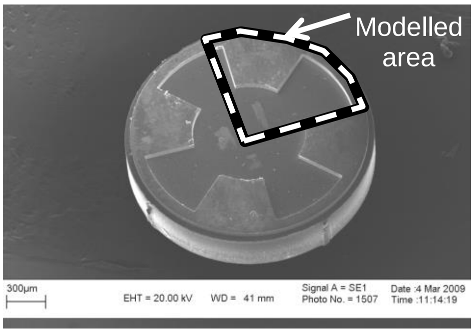

Figure 1. SEM image of a patterned MEMS thrust bearing [3].

2.2. Test rig

The bearings were evaluated using the custom built MEMS tribometer shown in Figure 2. This equipment is similar to the one used previously to study tribological behaviours of MEMS under high sliding conditions [11]. The setup uses a high-speed servo motor to rotate the bearings and measures the frictional torque from the rotation of the elastic suspension onto which the bearing is mounted by means of an optical lever technique.

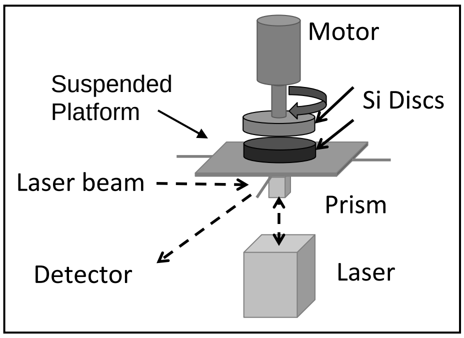

Figure 2. Schematic of tribometer used to characterize MEMS-scale bearings.

2.3. Pad bearing model

Modelling work was carried out in order to predict and further optimize micro-bearing film thickness and friction performance. As shown in figure 1, the bearing geometry can be divided into four identical parts, each containing a step, and the model is applied to each of the quarters. The quarter bearing geometry is then discretised into grid points and Reynolds' equation (in polar coordinates assuming only rotation and no squeeze film effects) is solved at each point using a finite difference approach:

$$p_{ij} = CN_{ij}\, p_{i,j+1} + CS_{ij}\, p_{i,j-1} + CE_{ij}\, p_{i+1,j} + CW_{ij}\, p_{i-1,j} + G_{ij} \qquad [1]$$

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where $p_{i,j}$ is the hydrodynamic pressure at point i,j and coefficients CN, CS, CE, CW and G are derived from Reynolds' equation. By incorporating the stiffness of the bearing supports, this approach can predict bearing performance in terms of pressure distribution, friction coefficient and film thickness under hydrodynamic conditions.

3. Results and Discussion

Figure 3 shows friction vs. speed (Stribeck) curves for one particular design of thrust pad bearing lubricated with a very low viscosity liquid (toluene, v = 0.55 mPas); the step height of the bearing in this test is 50 μm. It can be seen that when speed is greater than 500 rpm, the contact operates in the full film lubrication regime (surfaces are fully separated by liquid) and the friction coefficient is around 0.1. At low speed, however, there is insufficient entrainment of fluid to separate the surface and friction is high due to solid-solid contact. This problem can be effectively solved by adding a surface active friction modifier, octadecylamine (ODA). When 0.1% weight of ODA is blended with the lubricant, the friction coefficient at low speeds is reduced from over 1 to less than 0.2.

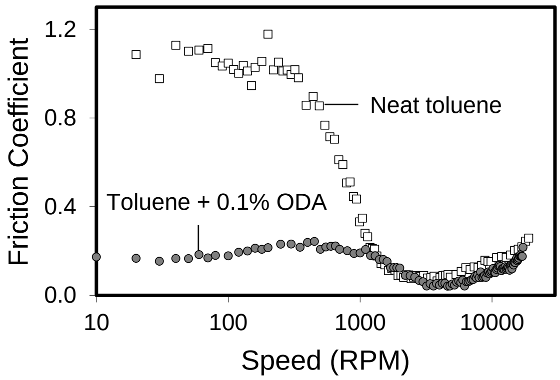

Fig. 3. Stribeck curve for MEMS thrust pad bearing showing effect of amine friction modifier.

Results from numerical simulations suggested that the performance of the mm-scale thrust pad could be further improved by adjusting the step height of the bearing. This is indeed the case, as shown experimentally in Figure 4, where it can be seen that the friction coefficient of the bearing reduces with decreasing step height. For instance, the friction coefficient of the bearing with a 1 μm-step at 10000 rpm is less than 0.02, whereas that for the 50 μm-step height bearing is approximately 0.3. This result is also in accordance with hydrodynamic theory. It should also be noted that the optimised step height of 1 μm provides low friction even when the higher viscosity liquid, hexadecane (v = 3.03 mPas), is used as the lubricant. This choice of lubricant is important in order to reduce evaporation.

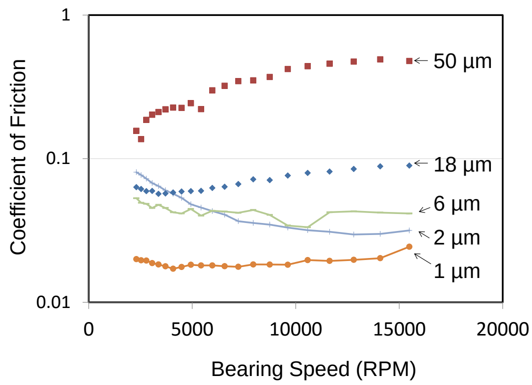

Fig. 4. Stribeck curve for MEMS thrust pad bearing showing effect of varying step height.

Bearing performance was also modelled using a finite difference solution to Reynolds' equation. The model was developed to predict the performance of the thrust pad bearing (*e.g.* pressure distribution, friction coefficient and film thickness) when operating in the full film hydrodynamic regime. The resulting hydrodynamic pressure distribution for one quarter of the 1 μm step height bearing is illustrated in Figure 5. The validation of the model is shown in Figure 6 where the friction coefficient vs. speed curves of a 1μm-step height bearing obtained from both simulation and experiment are compared. The film friction in the model is obtained by integrating the shear stress within the liquid, therefore, as expected, the simulation result does not match the experimental results when the speed is lower than 3000 rpm (*i.e.* when surface contact occurs and bearing operates in the boundary lubrication regime). However, when the speed is higher than 5000 rpm, and the bearing operates in the full film regime, the simulation result agrees well with the experimental result.

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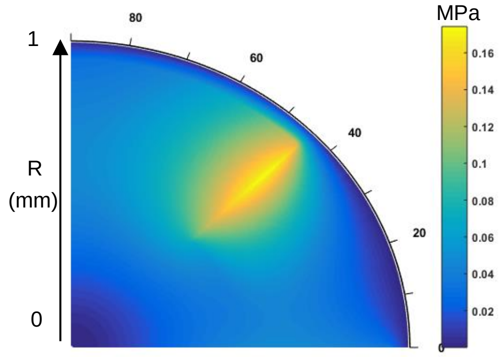

Figure 5. Hydrodynamic pressure in ¼ of pad bearing, predicted by finite difference model.

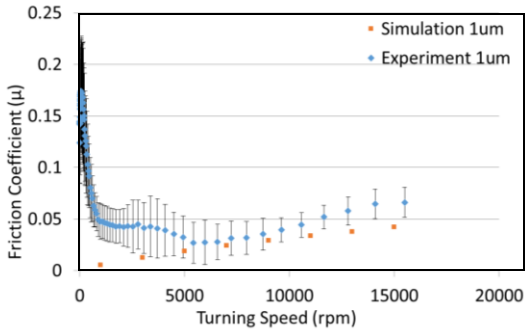

Figure 6. Comparison between of outputs from finite difference model and bearing test.

4. Conclusions

A series of mm-scale thrust bearings were successfully designed and fabricated with DRIE. With low viscosity lubricant, both simulation and experiment showed that the friction coefficient of an optimized bearing (1 μm step height) can be lower than 0.02 when operating in the high speed, full film regime. In addition, the performance of the bearing at low speeds can be significantly improved by including an amine friction additive friction modifier. The hydrodynamic thrust pad bearings will be integrated into the cm-scale turbine energy harvester shown in Figure 7, where it is expected they will offer improved performance compared to the existing jewel bearings.

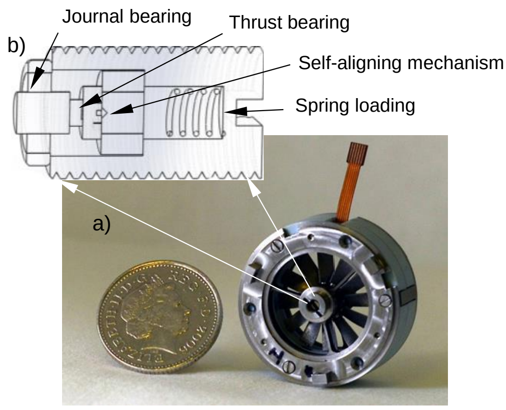

Figure 7. a) Photograph of turbine energy harvester [12], b) drawing of bearing and housing.

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