Assessment of thin film UHMWPE bearings for precision slideways

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Summary

In this extended abstract, results from tests on a novel, thin-film, Ultra-High-Molecular-Weight-Polyethylene (UHMWPE) bearing are presented. The goal of this investigation is to assess the performance and manufacture and to determine tribological conditions prevailing in the contact region. The bearings are comprised of a thin film of the polymer stretched over a spherical brass substrate. As of writing, two bearing geometries with substrate radii of 2.38 mm and 6.35 mm have been tested in air to determine both smoothness/repeatability of motion at traversal speeds of around 0.01 $mm \cdot s^{-1}$ to 1.00 $mm \cdot s^{-1}$ and durability under loads ranging from around 2.2 N to 28.0 N at speeds of up to 12.0 $mm \cdot s^{-1}$. Smoothness of motion of the 2.38 mm radius bearing was assessed by using it as the counterface for traversing a carriage along an optically polished Zerodur® datum. A stylus-based measuring system was then used to profile a smooth, flat glass specimen and the repeatability of specific surface features measured with successive traces being used as a measure of both bearing noise and repeatability. While it is recognized that there are numerous other sources contributing to the measured signal, we believe this does represent a conservative estimate of the uncertainty of this bearing system. With this in mind, it was apparent that features of amplitude less than 10 nm could be repeatably measured. Durability was assessed using a custom built pin-on-disc testing apparatus. Wear of thin films is a particularly difficult measurement and we have used the criteria of catastrophic failure of the film as a measure of the durability. While the 2.38 mm bearing was observed to fail in one experiment, the larger radius bearing was observed to survive after sliding distances of over 2 km at loads and speeds of 28.0 N and 4.2 $mm \cdot s^{-1}$, respectively[*].

Introduction

This bearing represents a variant on the thin-film polytetrafluoroethylene (PTFE) bearings that are used in some profilometers[1, 2] and the Tetraform™ [3] grinding machine for motion control with sub-nanometer performance. The objective of this new bearing design is to produce a low-cost, robust, vacuum-compatible and modular dry rubbing bearing with sub-nanometer performance. To operate at these levels, it is necessary to address stiffness, heat removal from the interface, wear, thermo-mechanical characteristics, accuracy of the datum counterface and frictional characteristics (in particular, stiction and load dependence)[4].

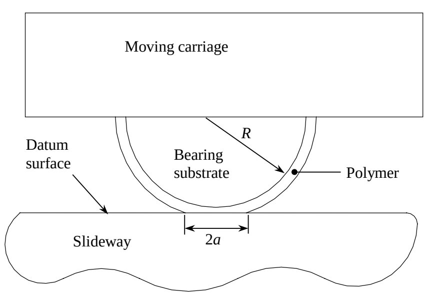

Fig. 1: Schematic diagram indicating key components of a precision slideway bearing utilizing a thin film polymer attached to a spherically shaped substrate.

While, initially, polymers might appear a poor choice of material for such demanding applications, there is plenty of evidence to suggest that they might provide a robust, inexpensive and easily fabricated solution. As far as can be determined, polymer bearings also appear to introduce little wear of the counterface that, in most designs, serves as a datum surface for controlled motion (linear or rotary). In contrast to fluid film and rolling element bearings that require either a pressurized or continuously coated fluid supply, polymeric bearings are passive mechanisms requiring no external supplies or connections. Competing bearing systems for linear slideway designs are the solid lubricant (steel on steel, lubricant filled metal composites on steel) and roller bearings.

Bearing Design

Given the criteria stated previously, two bearing designs have been built and tested as part of this study. The first design consists of a 4.76 mm (0.1875 inch) diameter brass sphere adhered to an aluminum substrate. A 0.15 mm (0.006 inch) thick UHMWPE film is then stretched over the sphere and held into place by a stamped metal cover, which is adhered to the substrate using a two-part epoxy while under a compressive load. A schematic illustration of the bearing is shown in Fig. 1. The dimensions of the bearing are approximately 10.0 mm × 10.0 mm × 12.7 mm. However, upon testing the bearing under various load conditions, it

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[*] Certain commercial equipment, instruments, or materials are identified in this article in order to specify the experimental procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose.

was found that the stress levels at the interface caused catastrophic failure at loads greater than 6.6 N at a surface speed of 4.7 mm·s^{-1}. To reduce the contact stress and increase the load capability of the bearing, a second bearing was designed with a radius of curvature nearly three times that of the original design. With suitable assumptions relating interface conditions, the contact radius of a rigid sphere in contact with a rigid flat surface through an elastic foundation can be estimated by[5]

$$a = \left(\frac{4PRh}{\pi E} \right)^{1/4} , \qquad (1)$$

where a is the radius of contact, P is the applied load normal to the surface, R is the radius of the sphere, h is the thickness of the film and E is the elastic modulus of the film. The contact area A, is given by

$$A = \pi a^2 . \qquad (2)$$

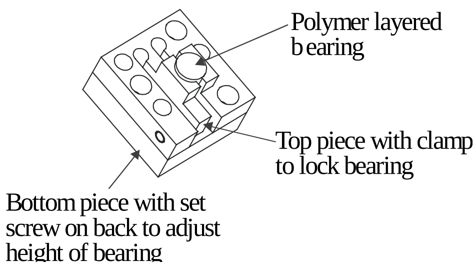

Fig. 2: 2nd generation bearing incorporating brass cylinder with a 6.35 mm radius of curvature.

Increasing the substrate radius from 2.38 mm to 6.35 mm theoretically corresponds to a 60 % increase in contact area and a nearly 40 % reduction in contact stress given the assumptions stated above. As of writing, we are undertaking a theoretical investigation using continuum mechanics models and finite element studies to obtain a more complete model of this complex problem.

Based on the results and analysis described previously, a second bearing was designed. To keep the overall bearing size at a manageable level, a 12.7 mm brass sphere was machined to a 6.35 mm diameter cylindrical surface at the equator leaving the polar cap(s) as a spherical bearing surface with a 6.35 mm radius of curvature. The polymer is stretched over the pin and pushed through a reamed hole by a setscrew where the polymer/pin combination is secured by a squeeze clamp as shown in Fig. 2.

Table 1: Pin-on-disk results for the 2.4 mm radius UHMWPE bearing on float glass.

Direction	Load (N)	Speed (mm·s^{-1})	Dynamic Friction Coefficient	Comments
Forward	2.2	4.2	0.19	Bearings show no appreciable wear characteristics. Contact area shows little to no effect of sliding after a short run-in period.
		8.3	0.19	
	4.4	4.2	0.18	
		8.3	0.18	
	6.6	4.2	0.16	
		8.3	0.16	
	11.5	4.2	0.19	Bearing failed after approx. 700 meter traverse
		8.3	na	
Reverse	2.2	4.2	0.19	Bearings show no appreciable wear characteristics. Contact area shows little to no effect of sliding after a short run-in period.
		8.3	0.19	
	4.4	4.2	0.18	
		8.3	0.18	
	6.6	4.2	0.16	
		8.3	0.16	
	11.5	4.2	0.19	Bearing failed after approx. 700 meter traverse
		8.3	na	

The entire assembly can then be mounted to arbitrary structures. The total bearing and clamping system has dimensions of 25.4 mm × 38.1 mm × 12.7 mm. Although larger compared to the original bearing, this design allows for easy assembly and replacement of warn or damaged polymer material and the setscrew can act as a fine leveling mechanism if desired.

Results

To qualify these bearings for use in nanometer level systems, a number of important parameters would have to be tested. For the use in precision motion systems the criteria used to evaluate the performance of the bearings were friction/stiction, wear and repeatability. Two instruments were developed to evaluate these parameters. The first instrument was based on a pin-on-disk system commonly utilized for tribological investigations. The pin-on-disk system was designed to evaluate the friction/stiction and wear of the bearing sliding on a glass counterface. The second instrument is a profilometer based test apparatus that evaluates the repeatability of the bearings under loads during rectilinear motion.

Pin-on-disk

Both bearing designs have been tested for frictional force and durability over extended periods of time. Table 1 shows the results from the 2.4 mm radius bearing for various speeds and loads.

From the results given in Table 1 it is clear that the friction coefficient reduces as the applied load increases. However, there is a dramatic change in friction as the load increases over 6.6 N. It was observed that for loads over 6.6 N the bearing experienced catastrophic failure. In these cases the polymer tore from the substrate allowing for metal on glass interaction. From these observations a second bearing was designed allowing for an increase in radius of curvature and the ability to replace worn and/or damaged features. The new bearing was then tested in a similar manner and the results shown in Table 2.

As expected, the coefficient of friction decreased with an increase in load. At loads above 16.5 N the lateral force sensor became saturated and requires some modifications to properly evaluate the friction at higher loads. Fig. 3 contains examples of the data observed for the 6.35 mm diameter bearing. Included are plots of the friction coefficient as a function of time for short-term forward and reverse directions and long-term durability evaluations.

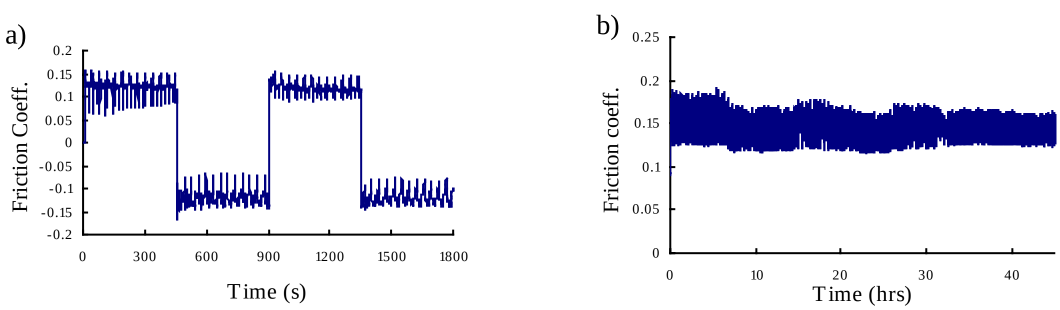

Fig. 3: a) Friction coefficient as a function of time for the 2nd generation bearing under an 11.45 N load at 4.2 mm·s^{-1}. Data includes three changes in direction. b) Friction coefficient as a function of time for a 6.65 N normal load over a 45 hour period.

A preliminary test was carried out to assess durability of the 6.35 mm bearing. At a load of 6.65 N, the bearing was run at a surface speed of 4.2 mm·s^{-1} for a period of 70 hours with no appreciable change in characteristics. In a further effort to determine the limit of performance, the bearing was subject to a maximum test-rig load of 27.8 N. Subsequent testing at a speed of 4.2 mm·s^{-1} for a period of 145 hours (total traverse distance of 2.2 km) indicated no visually observable damage or change in frictional characteristic.

Throughout all experiments, it was possible to infer a small stick/slip type behavior. Generally, the friction coefficient as measured appeared to increase by a factor of 0.05 upon reversal of traverse direction. A slightly higher initial friction was measured after the bearing had been in stationery contact with the counterface for a protracted time period (tens of minutes to days). As of writing, it is not possible to separate this variation between true bearing phenomena and instrument artifact.

Profilometer Test

The profilometer test evaluated the repeatability of the bearings sliding over a glass (Zerodur®) guide. A capacitance based contact probe measured the deviation of the rectilinear stage normal to the direction of motion as described previously. Data was collected using LABView™ software and written to spreadsheet where the data analysis was done. Initial analysis consisted of removing the tilt of the sample by removing a straight line fit to the original data. The residual values are the parasitic motion of the stage and represent the level of repeatability of the stage. Examples of the residual motion with the slideway being pushed and then pulled are shown in Fig. 4. Repeatability of the stage with the tilt removed was on the order of ±11.0 nm over a 10 mm travel range.

Table 2: Pin-on-disk results for the 6.35 mm radius UHMWPE bearing on float glass.

Direction	Load (N)	Speed ($mm \cdot s^{-1}$)	Dynamic Friction Coefficient	Comments
Forward	2.2	4.2	0.19	Bearings show no appreciable wear characteristics. Contact area shows little to no effect of sliding after a short run-in period.
	4.4	4.2	0.17	
	6.6	4.2	0.15	
	11.5	4.2	0.14	
	16.5	4.2	0.13	
	27.8	4.2	saturated	Outside sensors range
Reverse	2.2	4.2	0.19	Bearings show no appreciable wear characteristics. Contact area shows little to no effect of sliding after a short run-in period.
	4.4	4.2	0.17	
	6.6	4.2	0.15	
	11.5	4.2	0.14	
	16.5	4.2	0.13	
	27.8	4.2	saturated	Outside sensors range

From Fig. 4, it is apparent that there is a difference between profiles when the actuator is pushing or pulling the carriage. It is noted that when the micrometer is pushing the carriage, deflections due to bending moments generated by the offset between the line of action of the actuator force and the center of friction of the moving platform create an increase in the bending moment itself. In contrast, when pulling the carriage, perturbations create moments that tend to self-correct the error and reduce the magnitude. With this in mind, it is expected that the efficiency of the decoupling mechanism (cross roller bearing) between the actuator and moving carriage will worsen. In Fig. 4a, a periodic variation can be observed that closely corresponds to the pitch of the micrometer. However, even within this periodic variation, the profile is seen to repeat at nanometer levels. When the actuator is pulling the carriage, repeatable profiles are obtained with a total ‘band’ of deviations being within a few nanometers for the three profiles shown.

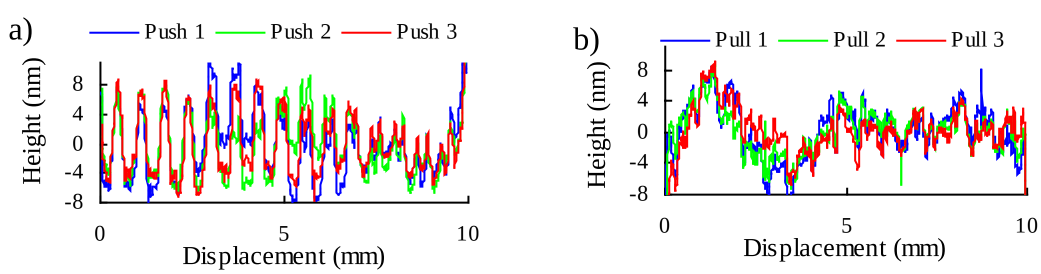

Fig. 4: a) Repeatability measurement of the profilometer height instrument as the rectilinear slide is being pushed. b) Repeatability measurement of the profilometer height instrument as the rectilinear slide is being pulled.

Acknowledgements

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