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A novel tribometer for the measurement of friction torque in microball bearings

Weixu Yang[a], Xiaoli Wang[a,*], Hanqing Li[b], Xintao Song[a]

[a] Department of Mechanical Engineering, Beijing Institute of Technology, Beijing 100081, China
[b] Microsystems Technology Laboratories, Massachusetts Institute of Technology, Cambridge, MA 02139, United States

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ABSTRACT

A novel tribometer for the measurement of friction torque in silicon-based microball bearings is demonstrated. First, a new configuration of microball bearing is proposed based on the contact mechanics analysis. The bearing is fabricated using halo etch and intermediate layer bond processing methods and the friction torque test platform is then developed. It is found that the microscale rolling friction torque between microball and raceway exhibits a sublinear relationship with load, which is different from the case of macroscale rolling, and the Si_3N_4 ball bearings generate larger friction torque than 440C ball bearings under the same load.

1. Introduction

The microball bearing is considered as the better mechanical support for the rotary MEMS owing to its lower friction coefficient than the planar contact bearing and the increased stability over the noncontact gas bearing [1–4]. However, the measurement of friction torque in microball bearings has been challenging up to now so that the quantitative relationship between friction torque and applied load of microball bearings due to size effects is unclear, which has become the bottleneck of the development of this kind of bearing technique.

In the past years, a range of tribometers were built to evaluate MEMS rolling friction and wear [5–11]. For example, Lin et al. [7] developed a linear tribometer composed of stainless-steel microballs, silicon stator and slider. It was found that the rolling friction coefficient between microballs and slider ranged from 0.006 to 0.01 at motor speeds of 90 and 120 rpm and load of 0.4*g*. Olaru et al. [8,9] introduced a rotary tribometer consisting of a driving disc, an inertial driven disc and three microballs which are rolling between the two discs. The spin-down friction tests was made under speeds of 30–210 rpm and loads of 8–33 mN. Later on, McCarthy et al. [10,11] presented a planar contact encapsulated microball bearing with 285-μm-diameter stainless steel microballs and a radial inflow microturbine platform for rolling friction measurement. In the spin-down friction testing, the friction torque of 0.0625–2.5 μN m was measured, but at the same time the excessive wear was found due to the direct contact between microball and bond interface in raceway and thereby the operating speed and stability were limited. Then Waits et al. [3] moved the bond interface away from the wear track, and the power law relationship between friction torque and applied load was derived from spin-down testing. The above researches are very meaningful, but some fundamental aspects related to the design of microball bearing still need to be further investigated. For example, the effects of bond interface offset and ball material on the contact characteristics between the microball and raceway have not been clear so that the design or even fabrication lack guidance and then is difficult to be improved.

In this paper, the contact mechanics analysis is made and a new configuration of microball bearing is then designed to improve the contact performance and simplify the fabrication. Then a friction torque test platform is developed, by which the rotor speed, angular position, normal load, gas supply pressure and flow rate can be monitored online. Finally, the quantitative relationship between friction torque and applied load of bearings with Si_3N_4 and the 440C stainless steel balls is derived from spin-down friction testing, respectively.

2. Contact mechanics and design

As shown in Fig. 1(a), the microball bearing consists of microballs and three silicon layers, i.e. capping layer, blade layer and raceway layer. Fig. 1(b) depicted radial cross section of the previous bearing raceway design [3]. To investigate the effects of bond interface offset *s* and microball material on contact performance, ball-radial raceway contact and ball-axial raceway contact are simulated. Schematic drawings of the contact model of ball-radial raceway and ball-axial raceway are depicted in Fig. 1(c) and (d), respectively. For ball-radial raceway contact, the applied load is radial centrifugal force $F_{centrifugal}$

* Corresponding author.
E-mail address: xiaoli_wang@bit.edu.cn (X. Wang).

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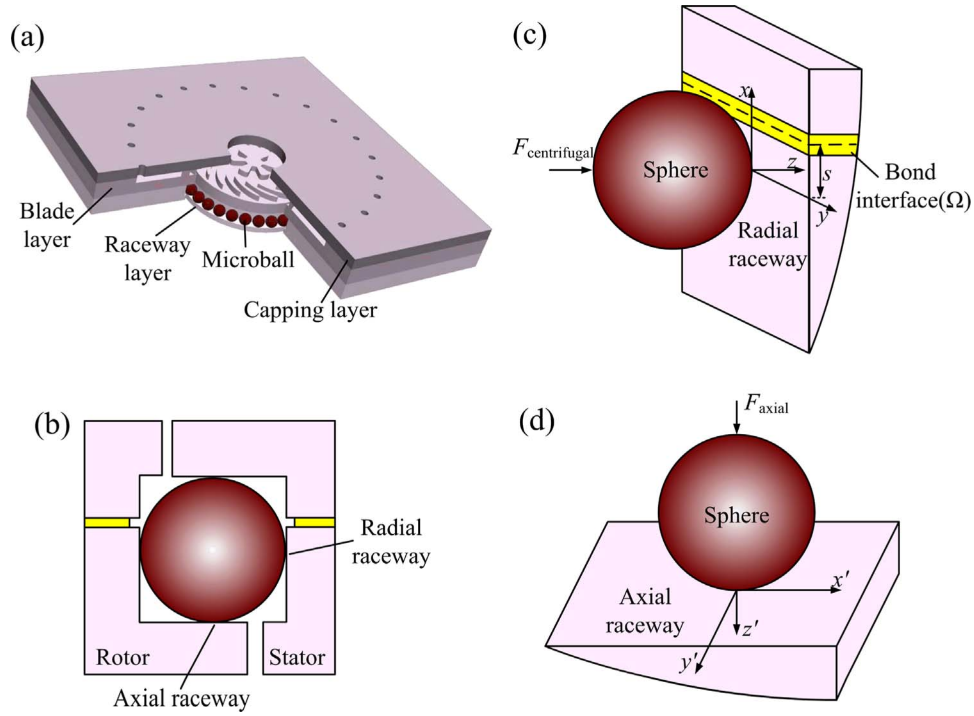

Fig. 1. Schematic of (a) microball bearings, (b) bearing raceway design [3], (c) ball-radial raceway contact and (d) ball-axial raceway contact.

which is related to the ball material density, while for ball-axial raceway contact, the applied load becomes axial force F_{axial} which depends on the applied thrust flow. Modeling methods of the two kinds of contact are similar, so only the contact model of microball and radial raceway (Fig. 1(c)) is described in detail.

Governing equations for the elastic contact problem are as follows [12],

$$F_{\text{centrifugal}} = \int_{\Delta} p(x, y)\mathrm{d}x\mathrm{d}y \tag{1}$$

$$h(x, y) = h_0(x, y) + u(x, y) - d_0 \tag{2}$$

$$h(x, y) \geq 0, \quad p(x, y) \geq 0, \quad h(x, y)p(x, y) = 0 \tag{3}$$

where $F_{\text{centrifugal}}$ is the radial centrifugal load, p is the radial contact pressure, h denotes the gap between two surfaces, h_0 is the original geometry gap before deformation, u describes the surface displacement, and d_0 is the relative 'rigid-body' approach. According to Hooke's law and equivalent inclusion method (EIM) [13], which treats the bond interface in radial raceway as homogenous inclusions with unknown equivalent eigenstrain, the following equation can be established [12],

$$C^n_{ijkl}C^{-1}_{klmn}\sigma^*_{mn} - \sigma^*_{ij} + C^n_{ijkl}\varepsilon^*_{kl} = \sigma^0_{ij} - C^n_{ijkl}C^{-1}_{klmn}\sigma^0_{mn} \tag{4}$$

where C_{klmn} and C^n_{ijkl} denotes elastic moduli of the silicon layer and bond interface, respectively, ε^*_{kl} is equivalent eigenstrain of the interface. σ^0_{ij} and σ^*_{ij} represents applied stress and eigenstress, respectively, and they can be calculated from the following equations [14],

$$\sigma^0_{ij}(x, y, z) = \mathbf{C}^{\sigma_{ij}}_p * \mathbf{p} \tag{5}$$

$$\sigma^*_{ij}(x, y, z) = \frac{-\mu}{4\pi(1-\nu)}\int_\Omega \Theta_{ij}[\varepsilon^*]\,\mathrm{d}x_1\mathrm{d}y_1\mathrm{d}z_1 \quad (x, y, z) \notin \Omega \tag{6}$$

$$\sigma^*_{ij}(x, y, z) = \frac{-\mu}{4\pi(1-\nu)}\int_\Omega \Theta_{ij}[\varepsilon^*]\,\mathrm{d}x_1\mathrm{d}y_1\mathrm{d}z_1 - 2\mu\varepsilon^*_{ij} - \frac{2\mu}{1-\nu}\nu\varepsilon^*_{ij}\delta_{ij} \quad (x, y, z) \in \Omega \tag{7}$$

where * symbolizes convolution, $\mathbf{C}^{\sigma_{ij}}_p$ is influence coefficient, vectors Θ_{ij} are the combinations of potentials, δ_{ij} represents the Kronecker δ, μ and ν are shear modulus and Poisson's ratio, respectively. Submitting Eqs. (5)–(7) into Eq. (4), equivalent strain and thus eigenstress can be obtained. The surface displacement u, composed of displacement due to contact load u° and displacement caused by homogeneous inclusions u^*, is given as [14],

$$u(x, y) = u^0(x, y) + u^*(x, y) = \mathbf{C}^{u_z}_p * \mathbf{p} - \frac{1}{2\pi}\int_\Omega \mathbf{U}^s_i[\varepsilon^*]\,\mathrm{d}x_1\mathrm{d}y_1\mathrm{d}z_1 \tag{8}$$

After obtaining the surface displacement u, h is updated in Eq. (2), and thus $\mathbf{p}$ is obtained based on Eqs. (1)–(3) [15]. Full flow chart for this computational procedure is depicted in Fig. 2. Numerical simulations are conducted by the semi-analytical method (SAM), conjugate gradient method (CGM) and fast Fourier transform technique (FFT)

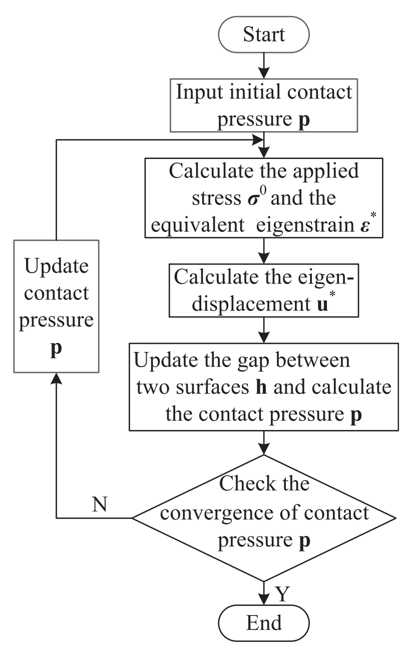

Fig. 2. Flow chart for the numerical calculation of ball-axial raceway contact.