Chinese Journal of Aeronautics, (2017), **xxx**(xx): xxx–xxx

CSAA

Chinese Society of Aeronautics and Astronautics
& Beihang University

Chinese Journal of Aeronautics

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Influences of preload on the friction and wear properties of high-speed instrument angular contact ball bearings

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Received 12 January 2017; revised 28 March 2017; accepted 25 May 2017

KEYWORDS

Angular contact ball bearings;
Bearing life;
Dynamic model;
Preload methods;
Wear rate

Abstract For starved-oil or solid lubrication of high-speed instrument angular contact ball bearings, friction heating and wear are the main reasons of bearing failures. This paper presents a dynamic wear simulation model to investigate the impacts of different preload methods and the changes of preload caused by wear on bearing wear life. The integral value QV of stress and sliding velocity in the contact ellipses between a ball and the inner and outer races determines friction heating and wear. The changes of QV with the friction coefficient and the wear volume under constant-force preload and fixed-position preload are analyzed. Results show that under the same initial preload, the QV decreases with an increase of the friction coefficient for both preload methods, and the latter is slightly larger. The wear of the ball and the race is equivalent to the ball diameter reduction. The QV of constant-force preload is almost not changed with a decrease of the ball diameter, but for fixed-position preload, the value decreases firstly and then increases substantially due to insufficient preload, and slipping occurs, the ball diameter is reduced by 0.025%, while the preload is reduced by 60.33%. An estimation of the bearing wear life under different preload methods requires a consideration of the changes in the wear rate of bearing parts.

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1. Introduction

In high-speed rotor systems for aviation, aerospace, and precision machine tools, pairs of preloaded angular contact ball bearings are commonly used. An appropriate axial preload, on one hand, can improve the rotation accuracy and support stiffness of the rotor, as well as reduce vibration and noise.[1,2] On the other hand, under high-speed and light-load conditions, by precisely controlling the preload, it is possible to effectively prevent orbit slipping and gyro sliding of balls,

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Peer review under responsibility of Editorial Committee of CJA.

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http://dx.doi.org/10.1016/j.cja.2017.07.006
1000-9361
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and reduce friction heating and wear.[3,4] Two main preload methods are often used in practice, constant-force preload and fixed-position preload, also called constant preload and rigid preload. The effects of preload on dynamic stiffness,[5,6] natural frequency,[5,7] vibration,[2,8] temperature rise,[9,10] and fatigue life[4,11] of spindle bearings with sufficient oil supply have been studied a lot. Some of the works have focused on preload optimization,[3,10–12] and revealed that the optimum preload can be determined by temperature rise, dynamic stiffness, and fatigue life for different speed ranges. Other researchers also investigated the impacts of different bearing preload methods on spindle dynamics. Li and Shin[13] presented the effects of bearing configuration on dynamic thermal and stiffness behaviors of high-speed spindles using a dynamic thermo-mechanical simulation model. Cao et al.[14] compared the effects of bearing preload methods on the dynamic performance of high-speed spindles by using a mathematical model as well as experiments, and pointed out that at high speeds and under cutting loads, the rigid preload method is more efficient in maintaining the dynamic stiffness of spindles than the constant preload one.

However, for starved-oil (lubricated by a plastic cage impregnated with a few milligrams of oil) or solid lubrication of high-speed instrument angular contact ball bearings, friction heating and wear are the main reasons of bearing failure. Liu et al.[15] established a wear life estimation model based on the quasi-static of Gyro-spin bearings for fixed-position preload. Gupta and Forster[16] built a numerical simulation model of wear for solid-lubricated ball bearings based on the overall dynamics of bearing elements. The time-averaged wear rates of balls, races, and the cage can be obtained by the computer model ADORE, which provides an analytical estimate of wear life for solid-lubricated ball bearings. However, researchers did not consider the impacts of different preload methods and the significant changes of preload caused by wear during the operation on the bearing wear life. This paper presents a coupled dynamic wear simulation model considering the differential sliding, spin sliding, and gyro sliding between balls and races for high-speed instrument rotor angular contact ball bearings. The changes of contact parameters and residual preload with the friction coefficient and wear volume under constant preload and fixed-position preload are analyzed. Thus, a foundation for more accurate prediction of bearing wear life can be laid.

2. Modeling and verification

According to the dynamic modeling method of rolling bearings in Refs. 16,17, a dynamic wear simulation model of gyro rotor angular contact ball bearings was built up. It is assumed that the mass centers of bearing components coincide with their geometric centers, balls and the cage have six degrees of freedom, the mass center of the outer race is fixed, the mass center of the inner race have three or two degrees of freedom with constant or rigid preload respectively, and both the inner and outer races can rotate around the axis. The effect of lubrication is considered by setting a reasonable friction coefficient. For starved-oil or solid lubrication of angular contact ball bearings, there is no hydrodynamic pressure effect in the ball/cage and cage/guide lands contacts. For all contacts, normal and tangential forces are calculated by the Hertz contact theory and the Coulomb friction law. The friction coefficient in the contact area is constant. A variable step-size of the fourth-order Runge-Kutta method is used to solve the differential equations of bearing motion.

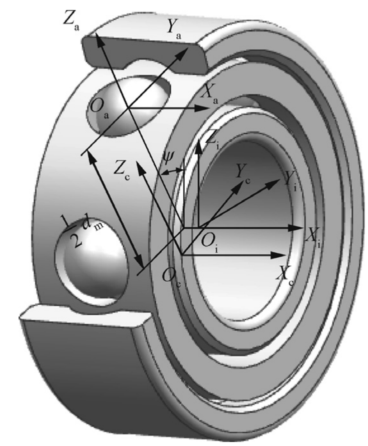

Fig. 1 Coordinate system.

2.1. High-speed rotor bearing dynamic model

2.1.1. Coordinate system and kinematic equations

The centroid motion of bearing parts is described in the inertial coordinate system, and the rotation about the centroid is described in the body fixed or azimuth coordinate system. The coordinate system is defined as shown in Fig. 1. The origin O_{i} of the inertial coordinate coincides with the center of locus of outer raceway groove curvature centers, X_{i} is along the bearing axial, and Z_{i} vertical up. The origin O_{a} of the azimuth coordinate frame is fixed to the ball center, X_{a} is in the axial direction of the bearing, and Z_{a} is in the radial direction, where d_{m} is the bearing pitch diameter and ψ is the ball azimuth angle. The cage coordinate frame origin O_{c} is fixed to the geometric center of the cage, X_{c} is along the cage axis, and Z_{c} points to the first pocket hole center.

It can be achieved through three successive rotations from the inertial coordinate system to the body fixed coordinate system.[17] The transformation matrix is as follows:

$$\boldsymbol{T}_{\mathrm{ib}} = \boldsymbol{T}(\eta,\xi,\lambda) = \begin{bmatrix} \cos\xi\cos\lambda & \cos\eta\sin\lambda + \sin\eta\sin\xi\cos\lambda & \sin\eta\sin\lambda - \cos\eta\sin\xi\cos\lambda \\ -\cos\xi\sin\lambda & \cos\eta\cos\lambda - \sin\eta\sin\xi\sin\lambda & \sin\eta\cos\lambda + \cos\eta\sin\xi\sin\lambda \\ \sin\xi & -\sin\eta\cos\xi & \cos\eta\cos\xi \end{bmatrix} \quad (1)$$

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