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# Investigation of skidding in angular contact ball bearings under high speed

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ABSTRACT

This paper proposes a dynamic model to investigate skidding in angular contact ball-bearings with considering the interaction between balls and raceways, cage and lubricant. The differential equations governing the motions of bearing elements are established and solved using a fourth-order Runge–Kutta algorithm; traction forces between balls and raceways are evaluated based on elastohydrodynamic lubrication theory. The results show that the applied axial load significantly influences the behavior of skidding due to the changes of internal load, orbital and rotation speeds of ball under different operating conditions; appropriate axial load can be determined to avoid severe skidding.

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## 1. Introduction

The angular contact ball bearing is widely used in numerous machinery systems for its high reliability and low power consumption, besides the ability to support radial and axial loads. However, skidding often occurs in rolling bearings under high speed and low load conditions, which can cause wear and incipient failure of the bearing ring and rolling element surfaces [1,2]. Cocks and Tallian [3] highlighted the effect of rolling element sliding in rolling bearing on the smearing damage. Gloeckner et al. [4] and Dotzel [5] pointed out that sliding occurring in raceway/ball contact makes significant contribution to the micro-spalling and thermal characteristics of high speed ball bearings in aircraft engine applications.

Various analytical and numerical models have been developed to analyze the skidding behavior of ball and roller bearings. For ball bearings, Jones [6,7] proposed the first mathematical theory to analyze the ball bearing system; however, his method requires the raceway control assumption to get solution. Later, based on Jones' model Harris [8,9] developed an improved model without using the raceway control assumption for axially loaded angular contact ball bearings. The results obtained by Harris more closely approximated the measured data reported by Poplawski and Mauriello [10]. The quasi dynamic model developed by Jones [6,7] and Harris [8,9] are commonly used nowadays, however it cannot be used to analyze combined radial and axial loads or time-varying operating conditions, both of which are important to analyze the motion in bearing system. Wang et al. [11] developed a quasi-static analysis model without the raceway control assumption to analyze the performance of angular contact ball bearings under the combined action of radial, axial loads and the shaft tilting moment along with the consideration of the effects of centrifugal force and gyroscopic moments, and compared with the results from the raceway control theory, it indicated that for high-speed operations, outer raceway control hypothesis is appropriate. Boness and Gentle [12,13] also developed a quasi-static model of a ball bearing. They considered elastohydrodynamic (EHD) traction forces, viscoelastic forces, cage drag, pocket friction, and elastic deflection of the bearing elements in detail, and achieved good agreement with existing experimental evidence. Absence of any ball-cage interaction and no consideration of dynamic effects cause the inadequacy of quasi-static models. Gupta [14] proposed a complete dynamic model to analyze the ball motion and skidding in ball bearings considering the interaction between balls and cage, where a semi-empirical EHD lubrication model is used to calculate friction force and moment; however, no detailed results about the ball motion and skidding characteristics are given. Jain [15] also investigated the skidding phenomenon in angular contact ball bearing, where a dynamic model to analyze the angular velocity of balls is established based on the internal load distribution calculated by a quasi-static model. Shao et al. [16] proposed an analytical model to study the skidding when a rolling element enters the loaded zone of the rolling element bearing under radial load and later extended it to consider the effect of the lubrication oil film [17]. Tu et al. [18] studied the skidding behavior during the acceleration process of a deep-groove ball-bearing

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**Nomenclature**

| | |
|---|---|
| $A$ | semi-major axis of ball-raceway contact ellipse, m |
| $b$ | semi-minor axis of ball-raceway contact ellipse, m |
| $d_m$ | bearing pitch diameter, m |
| $D$ | ball diameter, m |
| $r$ | ball radius, m |
| $r_o$ | outer raceway groove curvature radius, m |
| $r_i$ | inner raceway groove curvature radius, m |
| $K_o$ | load–deformation coefficient for ball and outer raceway contact, $N/m^{1.5}$ |
| $K_i$ | load–deformation coefficient for ball and inner raceway contact, $N/m^{1.5}$ |
| $F_a$ | externally applied axial load, N |
| $F_r$ | externally applied radial load, N |
| $F_c$ | ball centrifugal force, N |
| $F_d$ | viscous drag force, N |
| $m$ | ball mass, kg |
| $M$ | inner ring and shaft mass, kg |
| $n_i$ | bearing speed, r/min |
| $I$ | rotational inertia, kg $m^2$ |
| $\alpha_0$ | free contact angle, rad |
| $\alpha_o$ | ball–outer raceway contact angle, rad |
| $\alpha_i$ | ball–inner raceway contact angle, rad |
| $\beta$ | ball pitch angle, rad |
| $\omega_i$ | inner ring revolution speed, rad/s |
| $\omega_c$ | ball orbital revolution speed, rad/s |
| $\omega_{x'}$ | ball rotating angular velocity around $x'$ axis, rad/s |
| $\omega_{y'}$ | ball rotating angular velocity around $y'$ axis, rad/s |
| $\omega_{z'}$ | ball rotating angular velocity around $z'$ axis, rad/s |
| $\omega_s$ | ball spinning speed on the contact path, rad/s |
| $\Delta V$ | speed difference between two contacting surfaces, m/s |
| $\Delta_x$ | displacement of the inner ring center in $x$ direction, m |
| $\Delta_y$ | displacement of the inner ring center in $y$ direction, m |
| $\Delta_z$ | displacement of the inner ring center in $z$ direction, m |
| $\Delta'_{jy}$ | displacement of the ball center in $y'$ direction, m |
| $\Delta'_{jz}$ | displacement of the ball center in $z'$ direction, m |
| $\delta_{ij}$ | deformation between the inner-raceway groove curvature center and ball center, m |
| $\delta_{oj}$ | deformation between the outer-raceway groove curvature center and ball center, m |
| $\eta_0$ | lubricant viscosity at atmospheric pressure and temperature, Pa s |
| $\gamma$ | viscosity–pressure coefficient, $Pa^{-1}$ |
| $\zeta$ | viscosity–temperature coefficient, $^{o}C^{-1}$ |
| $K_c$ | the lubricant thermal conductivity, J/(kg K) |

using a dynamic model, and found that skidding decreases with increasing load; however, this model can only be applicable for deep groove ball bearings, for angular contact ball bearings the skidding behavior would be more complicated because the spin axis of ball is not aligned with the bearing axis and the gyroscopic effects cannot be neglected.

For roller bearing, Dowson and Higginson [19] analyzed the effects of film thickness and frictional force on cage slip. Smith [20] reported the effect of oil viscosity, diametral clearance and outer race temperature on cage slip. Poplawski [21] analyzed the effects of oil temperature on cage slip. Hamrock and Jacobson [22] analyzed the effect of curvature on the cage and roller slip. Zhang [23] and Li and Chen [24] studied high speed roller bearing skidding by considering various factors such as the radial load, shaft speed, radial clearance and oil viscosity. Jacome et al. [25] presented a numerical model of roller bearing for mechanical event simulations based on the finite element method to study sliding between the rollers and races.

In the related experimental studies, Hirano [26] carried out several experimental investigations on the ball motion in thrust angular contact ball bearing, and found that, gross ball slip and cage instability occur when the value of the parameter $zF_c/F_a$ (where $z$ is the number of ball, $F_c$ is the centrifugal force of ball and $F_a$ is the axial load) exceeds 0.1. Based on the empirical criterion proposed by Hirano, Liao and Lin [27] investigated the skidding behavior of high-speed ball bearings under radial and axial loads with quasi-static analysis technique to determine internal load distribution. Kliman [28] also proposed a similar expression that $zF_c/F_a$ equals to $\cos(\alpha_s)$ which suggests that ball skidding could be reduced by minimizing the difference between the dynamic contact angles. The two criterion proposed by Hirano and Kliman are in disagreement about its magnitude. Further examination by Poplawaski [10] and Boness [12] suggested that this parameter alone is not sufficient to completely define onset of skidding in ball bearings. Selvaraj and Marappan [29] developed an experiment equipment to investigate the effect of operating parameters e.g. the shaft speed, radial load, viscosity of the lubricating oil, number of rollers and temperature on the cage slip of a cylindrical roller bearing. Li et al. [30] developed a skid damage test rig to simulate the dynamic contact between bearing ring and roller, trying to reveal the reason why the skid damage of roller bearing sometimes occurs and/or sometimes does not under high speed and low load condition at the same slip rate.

To study the ball motion and skidding characteristics under combined axial and radial loads in angular contact ball bearings, a dynamic model is developed in this paper. Considering the geometric relationship and interaction between raceways, balls, cage and lubricant, differential equations governing the motions of bearing elements are established. In this model, the motion of each ball is considered with six degrees of freedom, the inner race motion with three degrees of freedom, and the cage has one rotational degree of freedom. The frictional forces at contact interfaces are calculated based on EHD lubrication theory. The skidding behaviors of angular contact ball bearing under different operating conditions are investigated.

## 2. Theoretical model

In order to describe the motion of the rolling element, three coordinate systems are established as shown in Fig. 1. The first coordinate system ($O$-$xyz$) is fixed at the bearing center with $z$ axis coinciding with the bearing axis; the second coordinate system ($O$-$\boldsymbol{x'y'z'}$) is a moving coordinate system axis $\boldsymbol{z'}$ axis parallel with bearing axis and the coordinate origin attached to the center of rolling element, rotating around $z$ axis with speed $\omega_c$. In this coordinate, the ball has three angular velocity components $\omega_{x'}$, $\omega_{y'}$, $\omega_{z'}$ around $x'$, $y'$ and $z'$ axis, respectively. The third one is also a moving coordinate system $x''y''z''$ with $x''$-axis and $y''$-axis lying in the plane of the contact patch between ball and raceways and $z''$-axis perpendicular to the contact patch.

### 2.1. Contact deformation and contact force

Fig. 2 shows the relative locations of the ball center and the raceway groove curvature centers of angular contact ball bearing before and after combined axial and radial loads are applied at high speed. It is assumed that the outer raceway is fixed in

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