Effect of Bearing Structure on Oil-air Flow and Temperature of High Speed Ball Bearing by Combining Nonlinear Dynamic and CFD Model

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Effect of bearing structure on oil-air flow and temperature of high speed ball bearing by combining nonlinear dynamic and CFD model

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Abstract: To achieve effective cooling for high speed ball bearings, an investigation on the effect of bearing structure on oil-air flow and temperature inside bearing chamber is necessary. However, accurately defining boundary conditions of CFD model for high speed ball bearings has not been addressed completely. Adopting an improved dynamic model of bearings to calculate movements of balls and power loss to set the movement boundary and heat source of CFD model at high-low speeds and light-heavy loads. Then, rotational speed of cage and temperature of outer ring at various loads are tested to validate this proposed method. At high speeds, enlarging sealing degree of outlet not only reduces the temperature rise of bearings and improves the uniformity of temperature distribution, but also promotes the formation of oil-film on balls’ surfaces without increasing power loss. Yet it can reduce the temperature rise but can’t markedly improve the formation of oil-film at low and ultra-high speeds. Moreover, half birfield cage facing nozzle plays an important role in improving oil volume fraction inside the bearing cavity to reduce the temperature rise of bearings, and the next is birfield cage, they are again corrugated cage and half birfield cage back towards nozzle. These research results provide theoretical guidance for the improvement of bearing structure.

Keywords: Nonlinear dynamic model • CFD model • Oil-air flow and temperature • Sealing degree of outlet • Structure of cage

1 Introduction

Due to the widespread applications of high speed bearings on aircraft engines, high speed electric motors, and high speed trains, a plenty of attention has been paid to the temperature rise of high speed bearings, significantly influencing their rotating precisions, operation reliability and service life. Under certain conditions of loads, rotational speeds and lubrications, power loss and movement of balls and bearing structures crucially effect the oil-air flow and temperature distribution inside the bearing chamber. Therefore, accurately defining boundary conditions of CFD model based on engineering practice is important to study the effect of bearing structure on the oil-air flow and temperature distribution of high speed ball bearings.

The interaction between fluid and bearing components is complex when oil and air flow inside the bearings. Many scholars have been conducted the oil-air flow analysis by the experimental and numerical methods. Jeng and Gao [1] designed nozzle rings to study the lubrication effect with various nozzle locations based on an oil-air lubrication test rig. Akamatsu and Mori [2] investigated the effects of nozzle number, and ratio of diameter to length of nozzle on the temperature rise for weakening the internal air barrier. Aidarinis et al [3] analyzed the oil-air flow patterns inside bearing chamber adopting the laser Doppler velocimeter method, revealing the relation of oil-air flow characteristic with lubrication and heat transfer. Lee et al [4] used the particle image velocimeter method to study the influence of bearing structure and rotational speed on the air flow patterns. Moreover, Glahn et al [5] carried out the study on the flow patterns inside bearing cavity affecting crucially the heat transfer and thermal performance of bearings by establishing the CFD model. Koyama et al [6] emphasized the impacts of design parameters such as number and position of nozzle, flow velocity, and flow rate on the temperature distribution inside the bearing cavity. Wu et al [7] clarified the correlation between thermal characteristic and oil-air distribution inside the bearings using the VOF multiphase technique. Yan et al [8] conducted the flow performance analysis for high speed ball bearings under different air supply conditions, indicating the vortex distribution, air pressure difference, and thermal dissipation inside the bearings. Wu et al [9] elaborated the
modeling process for an accurate CFD model of high speed bearings to simulate the stratified air-oil flow inside the ball bearings, optimizing the configuration of oil jet cooling for the ball bearings. Yan et al [10] explained the effect of sealing condition on the internal flow patterns of high-speed ball bearings using the CFD simulation. However, existing CFD analysis of high speed bearings commonly adopted the empirical formulas to compute the heat generation and movements of bearings, which neglects the differential skidding, spin slipping, and gyroscopic sliding, material hysteresis of raceways, and shearing action of oil film, resulting in difficultly accurate obtaining movements of balls and power loss to set motion boundary and heat source for the CFD simulation. So, building the nonlinear dynamic model of high speed bearings with elastohydrodynamic lubrication (EHL) to obtain movements of balls and power loss of bearings is indispensable for the accurate CFD simulation.

For the dynamics of ball bearings, Jones [11] firstly studied the ball bearing system using the first mathematical theory according to the raceway control assumption. Harris [12] developed a quasi-static model, which cannot be applied to the time-varying operating conditions. Gentle [13] further improved the quasi-static model considering EHL traction, viscous force, cage drag, and deflection of balls. Subsequently, Jain [14] combined the semi-empirical EHL model to establish a complete dynamic model for analyzing the interaction between balls and cage. Bizare et al [15] formulated the force and moment equilibrium of ball bearings considering the EHL restore force to develop the nonlinear dynamic model with five degrees of freedom. Han and Chu [16] took into consideration the discontinuous contact between cage and balls, centrifugal and gyroscopic effects to establish the dynamic model for analyzing the skidding behavior of ball bearings. Gao [17] comprehensively considered the effects of hydrodynamic lubrication, thermal generation, Hertzian contact and kinematics of bearing components to study the skidding and over-skidding behaviors of high speed bearings. For the study on friction moments, Liu et al [18] proposed a time-varying friction moment calculation method considering elastic material hysteresis, slipping friction, viscous friction, and tangential friction of lubricant film between bearing components. In addition, the relationship between friction moments and shaft velocity, combined loads, lubrication conditions, surface and machining errors are studied [19-21]. These calculation criterions of friction moments are referenced for evaluating the power loss in this work. However, these dynamic models of ball bearings generally neglected the influence of asperity and hydrodynamic tractions on the force equilibrium of balls due to the internal friction forces between balls and raceways are quite small relative to the Hertzian contact loads at low speeds and heavy loads. At high speeds and light loads, the traction effects derived from the lubricant viscosity and macro-slippering of balls markedly enhance the internal friction forces and moments between bearing components, as a result, the asperity and hydrodynamic tractions must be considered for the force equilibrium of balls and their sliding behaviors. Moreover, asperity friction coefficient between balls and raceways at the mixed lubricant mode is closely associated with slide-to-roll ratio of balls, causing the internal friction forces of bearing components are varied real-timely during a rolling period of balls.

In this work, the CFD model of high speed ball bearings is firstly developed in Section 2, boundary conditions of which are determined based on the improved nonlinear dynamic model by considering the asperity and hydrodynamic traction forces, time-varying asperity friction coefficient and time-varying lubricant modes in Section 3. These nonlinear dynamic and CFD models are validated through experimental method in Section 5. Subsequently, sealing degree of outlet and structure of cage are varied to analyze their effects on the oil-air flow and temperature distribution of high speed ball bearings in Section 6.

### 2 Development of CFD model

When working at high speeds and light loads, various motion postures of balls and power loss crucially effect the oil-air flow and heat dissipation inside the bearing cavity. Exerting actual heat power and rotation and revolution of balls to the CFD model is important to investigate the two-phase flow behavior and temperature distribution in the bearing chamber. 7008C angular contact ball bearing is considered as the study object in this work, its partial structure parameters are listed in Table 1.

**Table 1** 7008C ball bearing structure parameters

| Name                        | Symbol | Value |
|-----------------------------|--------|-------|
| Pitch diameter(mm)          | \(d_n\) | 54    |
| Contact angle               | \(\alpha^0\) | 15°   |
| Steel ball diameter(mm)     | \(D\)  | 6.35  |
| Steel ball number           | \(n_b\) | 18    |
| Outer raceway groove radius (mm) | \(r_o\) | 3.43  |
| Inner raceway groove radius (mm) | \(r_i\) | 3.43  |
| Guide face width of cage(mm) | \(W_c\) | 10.1  |
| Guide face diameter of cage(mm) | \(d_{gap}\) | 57.7  |
| Small diameter between rib guide face and cage(mm) | \(d_1\) | 49.9  |
| Large diameter between rib guide face and cage(mm) | \(d_2\) | 58.1  |
| Surface roughness of raceway(μm) | \(\sigma_r\) | 0.34  |
| Surface roughness of balls(μm) | \(\sigma_b\) | 0.1   |
According to some CFD modeling experiences in [7, 8, 22], the CFD model is established, as shown in Fig. 1. Nozzle fluid, outlet fluid and bearing chamber are considered as the fluid field. Nozzle flow and outlet flow are divided by hexahedral structured mesh, and bearing chamber are meshed with tetrahedral unstructured mesh. A gap of 1/20 of balls' radius between balls and raceways in the simulation configuration is defined to deal with the contact feature between balls and raceways. The sliding mesh plane method is used to deal with the data transfer in the flow field and at its edges. Moreover, the VOF technique [23] for multiphase fluid monitor is employed for tracking the oil-air two-phase flow, because this technique can trace the interface of oil-air based on the evaluation of volume fraction of one phase in a grid. In addition, because of the complexities of the relative movements between inner and outer rings, balls and cage, causing the complicated flow patterns at high rotational speeds, the RNG k-ε model is selected in this simulation due to its superiority of considering high strain rate, large curvature overflowing and high accuracy for the rotational flow [24]. Surfaces of inner ring, outer ring, cage and balls interacting with the fluid field are defined as the heat transfer walls (0.5 mm) to simulate the temperature distribution in the bearing cavity. The convection coefficients are calculated by following the energy conservation equation in the computation. The heat source is exerted at the contact areas between balls and inner and outer raceways. The heat power is divided into inner ring, outer ring and balls in a ratio of 1:1:2, according to the generation mechanism of power loss in Section 3. The gauge pressure and temperature of the ambient environment are defined as 0 Pa and 300 K. The oil-flow rate is 1L/min and the nozzle diameter is 1.5 mm. The viscosity of lubricant oil described in Table 2 are inputted into the CFD model.

Figure 1 Main mesh diagram of 7008C angular contact ball bearing.

At high speeds and light loads, the differential skidding, spin sliping, and gyroscopic sliding of balls markedly effect their self-rotation and revolution. It is indispensable to correctly exert two movements to balls to analyze the flow field variation during the CFD simulation. The global reference system (o-xyz) is fixed at the bearing center, as shown in Fig. 2. The local reference system (o-x'y'z') is positioned at the center of ball, and it rotates along the x-axis of the global reference system (o-xyz). Three angular velocity components ω_x', ω_y' and ω_z' of the ball around x', y' and z', are applied as its own movement boundary in the CFD model, and the orbital revolution speed ω_c of balls is imposed to the movement boundary of fluid field.

Figure 2 Reference system of balls and fluid domain in the CFD model.

Three different meshes have been validated to achieve the grid independence, as shown in Table 3. The results indicate that the differences of outlet oil flow rate and average oil volume fraction are less than 2.3%, suggesting that mesh density can be used in all the calculations.

Table 3 Calculated values under various mesh specifications

| Total number of cells | Minimum mesh volume (m³) | Outlet flow rate (kg/s) | Average oil volume fraction |
|-----------------------|--------------------------|-------------------------|-----------------------------|
| Mesh 1                | 653486                   | 2.023e-14               | 0.01425                     | 0.04092 |
| Mesh 2                | 841166                   | 1.988e-14               | 0.01429                     | 0.04185 |
| Mesh 3                | 1292855                  | 1.942e-14               | 0.01433                     | 0.04236 |

3 Nonlinear dynamic model of high speed ball bearings

To set accurately the motion boundary and heat source for the CFD simulation, establishing a nonlinear dynamic model is necessary by considering the asperity and hydrodynamic traction forces, time-varying asperity friction coefficient and time-varying lubricant modes. To analyze the force equilibrium of balls and inner and outer rings, the quasi-dynamic model of ball bearings is employed to calculate the load distribution, contact angle, contact area size, and so on, determining the friction force and moment of balls and raceways. To describe motions of balls and bearing rings, three coordinate systems are established, as shown in Fig. 3, which is consistent with that of CFD model described in Fig. 2. The local
reference system of balls ($o$-x'y'z'), rotated along the x-axis of the global reference system ($o$-xyz) fixing at the ball bearing center, is an orbital reference system, in which the ball has three angular velocity components $\omega_x$, $\omega_y$, and $\omega_z$ around $x'$, $y'$ and $z'$, respectively. The other moving coordinate system ($o$-x''y''z'') lies in the elliptic contact area between balls and raceways with major axis $x''$, minor axis $y''$ and $z''$ axis perpendicular to the contact patch. Thus, the displacements of inner ring in y and z directions (radial component, $\delta_i$), in x direction (axial component, $\delta_a$) and angular displacements around y and z directions can be obtained while outer ring is fixed.

3.1 Force equilibrium of balls and inner ring with EHL

Under the mixed EHL (a very thin lubricant film) condition, the dynamic behavior of balls, as presented in Harris [25], is subjected to the impacts of traction force (i.e. asperity and hydrodynamic tractions), contact pressure, centrifugal force and gyroscopic moment at high rotation speeds, which indicates that the contact angles at inner and outer raceways ($\alpha_i$ and $\alpha_o$) changes relative to the nominal contact angle ($\alpha_0$) and the center of ball no longer collinear with the groove curvature centers of inner and outer rings, as shown in Fig. 4. Accordingly, this is crucial for obtaining a new equilibrium position of balls to restore the force balance. The new ball equilibrium position ($X_{i1}$, $X_{i2}$) can be solved by the Pythagoras' theorem from Fig. 4:

$$ (BD \sin \alpha_0 + \delta_a + \gamma_i \cos \psi_j - X_{i1})^2 + (BD \cos \alpha_0 + \delta_x \cos \psi_j - X_{i2})^2 = [(f_i - 0.5)D + \delta_{ij}]^2 $$

where $B = f_i + f_o - 1$, $f = \pi/3D$, $f_o = \pi/2D$, $D$ is ball diameter, $\gamma_i$ is the radius of locus of raceway groove curvature centers, $\theta$ is the angular displacement of inner ring, $\psi_j$ is the angular position of ball.

![Figure 3](image1.png) Definition of three coordinate systems.

![Figure 4](image2.png) Ball center and raceway groove curvature centers before and after operation.

The forces acting on each ball, including the ball-raceway contact loads $Q_i$ and $Q_o$, traction force $F_t$ (asperity traction force $F_a$ and hydrodynamic traction force $F_h$), centrifugal force $F_c$, viscous drag force $F_v$ and gyroscopic moment $M_g$ are presented in Fig. 5. At high speeds and light loads, the internal traction force plays an indispensable impact on the dynamic behavior of balls, thus, the traction effect due to the lubricant viscosity and asperity contact is considered in this part. The equilibrium functions of the $j^{th}$ ball are listed as follow [26]:

$$ Q_o \cos \alpha_o - F_{c, i} \sin \alpha_i - Q_i \cos \alpha_i + F_{a, i} \sin \alpha_i - F_c = 0 $$

$$ Q_o \sin \alpha_o + F_{a, i} \cos \alpha_i - Q_i \sin \alpha_i - F_{a, i} \cos \alpha_i = 0 $$

$$ F_y y_i - F_{c, i} y_i + F_v = F_{cage} = 0 $$

$$ 0.5D(\gamma_o F_{x, i} + \gamma_i F_{x, i}) = M_{gy} $$

$$ 0.5D(\gamma_o F_{y, o} \sin \alpha_o + \gamma_i F_{y, i} \sin \alpha_i) = M_{gy} $$

$$ 0.5D(\gamma_o F_{z, o} \cos \alpha_o + \gamma_i F_{z, i} \cos \alpha_i) = M_{gy} $$
asperity traction force lubricant viscosity and film thickness under the EHL from the shearing action of lubricant is closely related to resulting in the asperity force formation between balls and raceways, EHL region due to the existence of surface roughness, component of equation is used to solve the film thickness between balls and raceways should be attained. To determine the asperity traction force and hydrodynamic traction force of balls, the lubricated contact pressure is replaced by Hertzian contact pressure as follows:

\[ h(x', y') = h_0 + \frac{x'^2}{2R_e'x'} + \frac{y'^2}{2R_e'y'} - \frac{2}{\pi \kappa} \int \int \frac{p(x'', y'')}{\sqrt{(x'' - x')^2 + (y'' - y')^2}} dx'' dy'' \]  

(12)

where \( h_0 \) is the approach between balls and raceways, \( R_e' \) and \( R_e'' \) are curvature ratios in \( x'' \) and \( y'' \) directions, \( \kappa \) is equivalent modulus of elasticity, \( \Gamma \) is continuous fluid domain, \( x_e \) and \( y_e \) are continuous fluid boundaries.

Under the condition of high speeds and light loads, the differential sliding and spin sliding of balls obviously impact the shearing action of lubricant. Wang et al [29] deduced the relative slipping velocities of balls on inner and outer raceways considering the combined effect of differential sliding and spin sliding of balls. The velocity differentials at any point \((x'', y'')\) in the inner raceway contact elliptical area are given by:

\[ \Delta v_{x''} = \omega_e \cdot \sin \alpha_i - \omega_e \cdot \cos \alpha_i + \omega_e \cdot \sin \alpha_i \]  

(13)

\[ \Delta v_{y''} = \omega_e \cdot \left( R_o'^{-0.5} - R_o'^{-0.5} \right) + \left( \frac{D}{2} \right)^{0.5} \]  

(14)

where \( \omega_e \) is the orbital revolution speed of ball, \( R_o \) is curvature radius of deformed surface in the outer raceway, \( d_m \) is bearing pitch diameter. Similarly, the velocity differentials at any point \((x'', y'')\) in the inner raceway contact elliptical area are listed as follow:

\[ \Delta v_{x''} = \frac{d_m (\omega_i - \omega_e)}{2} - \omega_e \cdot \cos \alpha_i + \omega_e \cdot \sin \alpha_i + (\omega_i - \omega_e) \cos \alpha_i \]  

(15)

\[ \Delta v_{y''} = \omega_e \cdot \left( R_i'^{-0.5} - R_i'^{-0.5} \right) + \left( \frac{D}{2} \right)^{0.5} \]  

(16)

where \( \omega_i \) is the revolution speed of inner ring, \( R_i \) is curvature radius of deformed surface in the inner raceway. Besides, because of the rotational speed differential, the spinning velocities on the outer and inner raceways are deduced as follow:

\[ \omega_{x''} = \omega_e \cdot \sin \alpha_i + \omega_e \cdot \cos \alpha_i \]  

(17)

\[ \omega_{y''} = \omega_e \cdot \sin \alpha_i + \omega_e \cdot \cos \alpha_i \]  

(18)

Further, the skidding velocity \( \Delta u(x'', y'') \) in equation (10) can be obtained as follows:
\[ \Delta w_{ij}(x', y') = \left( \Delta v_{ij}^x - \omega_{ij}^x x' \right) \frac{y'}{r_j} + \left( \Delta v_{ij}^y - \omega_{ij}^y y' \right) \frac{x'}{r_j} \]  \hspace{1cm} (19)

At high speeds and light loads, the asperity friction coefficient \( \mu_c \) mainly relies on the complex sliding and rolling of balls. Under different lubrication modes, the relation of asperity friction coefficient with the slide-to-roll ratio between two contact surfaces is determined [30, 31]. The empirical formula under the mixed lubrication is employed to correct the asperity friction coefficient, as follow:

\[ \mu_c = (-0.1 + 22.28s)e^{-181.4d_b} + 0.1 \]  \hspace{1cm} (20)

The slide-to-roll ratio \( s \) of balls on the inner and outer raceways can be solved, as follow:

\[ s_i = \frac{\Delta v_{i}^x}{V_o} \quad \text{and} \quad s_l = \frac{\Delta v_{l}^y}{V_l} \]  \hspace{1cm} (21)

where \( V_o \) and \( V_l \) are the average rolling velocities of balls at points \((x''_o, y''_o)\) and \((x''_l, y''_l)\), respectively, which can be calculated by:

\[ V_o = 0.5\left( \omega_{o} \sin \alpha_o - \omega_{o} \cos \alpha_o \right) \left[ \left( R_o^2 - x_o^2 \right)^{0.5} - \left( R_o^2 - a_o^2 \right)^{0.5} + \left( \frac{D}{2} - a_o \right)^{0.5} \right] + \frac{d_m \omega_c}{2} \]  \hspace{1cm} (22)

\[ V_l = 0.5\left( \omega_{o} \cos \alpha_o + \omega_{o} \sin \alpha_o + \omega_{o} \cos \alpha_o \right) \left[ \left( R_o^2 - x_o^2 \right)^{0.5} - \left( R_o^2 - a_o^2 \right)^{0.5} + \left( \frac{D}{2} - a_o \right)^{0.5} \right] + \frac{d_m \omega_c}{2} \]  \hspace{1cm} (23)

Thus, the equation of traction force \( F_t \) is rewritten at different lubrication states, as follows:

\[ F_t = \mu_c \int_{-a}^{a} \int_{-b}^{b} \left[ \eta(p(x', y'), \sigma) \right] d^x d^y \times \Delta w_{ij}(x', y') \]  \hspace{1cm} \( \Lambda < 0.01 \)  \hspace{1cm} (24)

\[ F_t = \mu_c \int_{-a}^{a} \int_{-b}^{b} \left[ \eta(p(x', y'), \sigma) \right] d^x d^y \times \left[ \Delta w_{ij}(x', y') \right]^2 \]  \hspace{1cm} \( 0.01 \leq \Lambda < 3 \)  \hspace{1cm} (25)

\[ F_t = \int_{-a}^{a} \int_{-b}^{b} \left[ \eta(p(x', y'), \sigma) \right] \frac{\Delta w_{ij}(x', y')}{h(x', y')} dx' dy' \]  \hspace{1cm} \( \Lambda \geq 3 \)  \hspace{1cm} (26)

Where oil film parameter \( \Lambda = h_b / (\sigma_l + \sigma_o)^{0.5} \), which depends on the minimum oil film thickness \( h_b \), surface roughness of raceway and ball \( \sigma_l \) and \( \sigma_o \). \( \Lambda < 0.01 \) indicates asperity contact, \( 0.01 \leq \Lambda < 3 \) implies mixed EHL, and \( \Lambda \geq 3 \) denotes pure EHL.

3.3 Drag force calculation of balls

For the orbit motion of balls, the drag force derived from the lubricant-air mixture and the discontinuous contact between balls and cage should be considered in the force equilibrium of balls. The viscous drag force imposing on the ball can be calculated by [25]:

\[ F_i = \frac{\pi}{32} C_d \rho_e (D_i d_o \omega_c)^2 \]  \hspace{1cm} (27)

where \( \rho_e \) is the equivalent density of the lubricant-air mixture, \( C_d \) is the drag coefficient, which is determined based on[32].

Under the axial and radial loads, balls push the cage to rotate in the heavily loaded region due to the sufficient friction force between balls and raceways. Otherwise, the cage pushes balls forward in the light loaded region. Accordingly, the interaction between balls and the cage is significant for keeping rolling bearings working steadily. The derivation of contact force on balls provided by the cage refers to [16]. The relative displacements \( \delta_r \) and \( \delta_l \) between balls and the cage are described as follow:

\[ \delta_r = \frac{d_m}{2} \sqrt{2 - 2 \cos \left( \phi_{l,cage} - \frac{\pi}{n_b} (2j - 1) \right) \frac{d_m}{2} \sin \left( \frac{\pi}{n_b} \right) + c_r} \]  \hspace{1cm} (28)

\[ \delta_l = \frac{d_m}{2} \sqrt{2 - 2 \cos \left( \phi_{l,cage} - \frac{\pi}{n_b} (2j - 3) \right) \frac{d_m}{2} \sin \left( \frac{\pi}{n_b} \right) + c_r} \]  \hspace{1cm} (29)

where \( \phi_l \) is angular displacement for the \( j \)th ball, \( \phi_{l,cage} \) is angular displacement of the cage, \( c_r \) is the clearance between the ball and the cage, \( n_b \) is the number of balls. Thus, the contact force of the \( j \)th ball is given by:

\[ F_{cage} = k_{cage} (\xi_+ \delta_+ - \xi_- \delta_-) \]  \hspace{1cm} (30)

where \( k_{cage} \) is the contact stiffness between the ball and the cage (10^8 N/m). When \( \delta_+ \), \( \delta_- < 1 \), one has \( \xi_1 = 1 \). If not, one has \( \xi_1 = 0 \). If not, one has \( \xi_1 = 0 \).

3.4 Force equilibrium of inner ring

To solve the force equilibrium of balls at high speeds, some forces provided by balls are exerted to inner ring, which must be equilibrated with the combined loads. Thus, the balance of forces on inner ring is expressed as follow:
adapting to the fourth-order Runge-Kutta algorithm.

3.5 Calculation of friction moments for power loss

At high rotational speeds, the friction heat of ball bearings greatly affects their performance and life. Accordingly, it is crucial to evaluate accurately the friction moments of the high speed ball bearings for improving their service performance. The tangential friction moment $M_t$ of lubricant film caused by the shearing action of lubricant, differential slipping moments $M_d$ induced by the differential ones, rolling friction moments $M_r$ produced by the elastic material hysteresis, spinning friction moments $M_s$ created by the spin motion of balls, viscous friction moments $M_v$ generated by the oil-air mixture, and slipping moment $M_c$ engendered by the sliding friction between rings and cage are evaluated based on the nonlinear model of high speed ball bearings with EHL. They are formulated, respectively, as follow [18, 33, 34].

Tangential friction moment $M_t$ of lubricant film is given by:

$$M_t = \frac{Dn}{\omega} \int_{x}^{a} \int_{y}^{b} \left( \frac{\Delta t(x', y')}{h(x', y')} \right) dx' dy'$$  \hspace{1cm} (36)

Differential slipping moments $M_d$ induced by the differential ones are described by:

$$M_d = \frac{\eta}{\omega} \int_{a}^{a} \int_{b}^{b} \left( \frac{1}{(x/a)^2} \right) \eta(p(x', y')) \Delta t(x', y') d\alpha u(x', y') d\alpha$$  \hspace{1cm} (37)

Rolling friction moment $M_r$ produced by the elastic material hysteresis is represented by:

$$M_r = \frac{2\sigma Q_a a}{3\pi D} \left( \frac{1}{d_m - 2\Delta h} \right) + \frac{2\sigma Q_a a}{3\pi D} \left( d_m + 2\Delta h \right)$$  \hspace{1cm} (38)

where $\epsilon$ is the elastic hysteresis loss coefficient.

Spinning friction moments $M_s$ created by the spin motion of balls can be calculated by:

$$M_s = \frac{\eta}{\omega} \int_{a}^{a} \int_{b}^{b} \left( \frac{1}{(x/a)^2} \right) \eta(p(x', y')) \mathbf{u}(x', y') d\alpha$$  \hspace{1cm} (39)

Viscous friction moments $M_v$ generated by the oil-air mixture are obtained by:

$$M_v = \frac{\eta \rho C_m d_t}{1 - \frac{d_1}{d_2}}$$  \hspace{1cm} (40)

Slipping moment $M_c$ engendered by the sliding friction between rings and cage is assessed by:

$$M_c = \frac{\eta \rho \omega_c C_m d_1 d_2}{1 - \frac{d_1}{d_2}} \frac{1}{2}$$  \hspace{1cm} (41)

where $\eta$ is the dynamic viscosity of lubricant, $W_c$ is the total guide face width of cage, $C_m$ is a coefficient, $d_1$ is the guide face diameter of cage, $\omega_c$ is the angular velocity of cage, $d_2$ is the smaller diameter between rib guide face and cage, $d_3$ is the bigger diameter between rib guide face and cage.

According to the friction moment formulas indicated above, the power loss $P_L$ of high speed ball bearings can be gained accurately, as follows:

$$P_L = (M_L + M_d + M_t + M_v + M_c) \omega_c$$  \hspace{1cm} (42)

In this work, the proposed model is just a further refinement of nonlinear dynamic models established by other scholars [15-17], the accuracy of which is further improved due to the following two advantages:

(1) This proposed nonlinear dynamic model considers the asperity traction force and hydrodynamic traction force for analyzing the force equilibrium of balls, which is because the traction effects derived from the lubricant viscosity and macro-slip of balls markedly enhance the internal friction forces and moments between bearing components at high speeds and light loads.

(2) The slide-to-roll ratio and lubricant mode of balls are changed continuously when balls pass through the primary and secondary loaded zones under axial and radial loads, so that asperity friction coefficient and lubricant conditions for each ball are adjusted real-time to evaluate accurately the traction forces and friction moments.

4 Calculation Flow

The computation flowchart is illustrated in Fig.6. First, initial values of quasi-dynamic model of ball bearings with Hertz contact are calculated by the quasi-statics method [35]. Then, this quasi-dynamic model is solved through the Newton-Raphson method and some dynamic results are outputted to the nonlinear dynamic model of ball bearings with EHL. Subsequently, film thickness $h(x', y')$, contact pressure $p(x', y')$, hydrodynamic traction force $F_{th}$, and viscous drag force $F_v$ computed by the EHL contact model are used to adjust the nonlinear parameters of dynamic model of ball bearings with EHL, in which hydrodynamic traction and viscous drag are responsible to force equilibrium and position equilibrium of balls. Thus, this proposed iterative algorithm for the high speed ball bearings with EHL is solved until attaining the force equilibrium of inner ring. The obtained $(\omega_1, \omega_r, \omega_c, \omega_v)$, $P_L$ are inputted into CFD model to define the motion of balls, fluid field and inner ring, and exert the heat source. So far, the boundary conditions of CFD model can be defined based on the movements of balls and power loss derived from the nonlinear dynamic model.
Validation of nonlinear dynamic and CFD models

To validate the reliability of nonlinear model of high speed ball bearings, 7008C angular contact ball bearing (as listed in Table 1) is considered as the study object in this work. The rotation speed of inner ring \( \omega_i \) is set as 10000 r/min, and the axial force \( F_a \) is varied from 50 N to 1000 N. Three angular velocity components \( \omega_{x'}, \omega_{y'} \) and \( \omega_{z'} \) for the ball of \( j=1 \) with both time and axial loads are obtained based on the proposed nonlinear model, as shown in Fig.7. In the convergent state, \( \omega_{x'}, \omega_{y'} \) and \( \omega_{z'} \) just get close to constant values (\( \omega_{x'} \approx -3589 \text{ rad/s}, \omega_{y'} \approx 0 \text{ rad/s}, \omega_{z'} \approx 1275 \text{ rad/s} \)) when \( F_a > 300 \text{ N} \), which is in good agreement with the analytical results of Han [16]. The discrepancy of variation of three rotational components between the proposed nonlinear model and Han’s results [16] is attributed to the different initial values for their iterative algorithms. Therefore, the nonlinear model of high speed ball bearings with EHL is believable for the following study on the effect of ball movement on the fluid field in the bearing cavity.

![Calculation flowchart of movements of balls and power loss for defining the boundary conditions of CFD model.](image)

Figure 6 Calculation flowchart of movements of balls and power loss for defining the boundary conditions of CFD model.

5 Validation of nonlinear dynamic and CFD models

To validate the reliability of nonlinear model of high speed ball bearings, 7008C angular contact ball bearing (as listed in Table 1) is considered as the study object in this work. The rotation speed of inner ring \( \omega_i \) is set as 10000 r/min, and the axial force \( F_a \) is varied from 50 N to 1000 N. Three angular velocity components \( \omega_{x'}, \omega_{y'} \) and \( \omega_{z'} \) for the ball of \( j=1 \) with both time and axial loads are obtained based on the proposed nonlinear model, as shown in Fig.7. In the convergent state, \( \omega_{x'}, \omega_{y'} \) and \( \omega_{z'} \) just get close to constant values (\( \omega_{x'} \approx -3589 \text{ rad/s}, \omega_{y'} \approx 0 \text{ rad/s}, \omega_{z'} \approx 1275 \text{ rad/s} \)) when \( F_a > 300 \text{ N} \), which is in good agreement with the analytical results of Han [16]. The
Figure 7 Variation of angular velocity components with time and axial loads (a) $\omega_x'$, (b) $\omega_y'$ and (c) $\omega_z'$, and (d) variation of $\omega_x'$, $\omega_y'$ and $\omega_z'$ in the convergent state with increasing axial load.

To further verify the dependability of this recommended model, the rotational speeds of cage at various axial forces are employed to conduct comparative analysis with test results of Pasdari [36] and Han’s analytical ones [16]. From Fig. 8, it can be found that when increasing gradually axial force, the ratio $\omega_{\text{cage}}/\omega_i$ derived from the proposed model has the same variation trend with the tested results. The peak value of $\omega_{\text{cage}}/\omega_i$ appearing at light axial forces is attributed to that the large slide-to-roll ratio of balls causes the intensified asperity friction force to push balls and cage, while at heavy loads, the pure rolling of balls avoids the asperity friction force resulting in the ratio $\omega_{\text{cage}}/\omega_i$ is dropped off lightly. Particularly, Han’s research results neglected the relation of asperity friction coefficient with the slide-to-roll ratio between two contact surfaces at high speeds and light loads so that the peak value of $\omega_{\text{cage}}/\omega_i$ couldn’t occur at light loads. According to these analyses, it is obvious that the developed nonlinear model in this work is reliable for calculating the dynamic behavior of ball bearings.

Figure 8 Variation rule comparison of the ratio $\omega_{\text{cage}}/\omega_i$ between present results and tested [36] and analytical [16] ones.

To validate the reliability of CFD model for the high speed ball bearings, three angular velocity components (shown in Fig. 7(d)), revolution and power loss (shown in Fig. 9(a)) derived from the nonlinear dynamic model are imported into the proposed CFD model. According to Figs. 7(d) and 9(a), axial forces {50, 100, 300, 1000} N are selected to compare the simulated results and tested ones. The temperature rise test is conducted by the experimental rig of high speed ball bearings, as shown in Fig. 9(b). Temperature sensors are assembled away from the nozzle. The ball closest to the nozzle is defined as the first ball at the azimuth of 0°. The technical data of the experimental apparatus is shown in Table 4.

Table 4  Technical data of experimental apparatus.

| Apparatus and sensor | data          |
|----------------------|--------------|
| Temperature sensor   | Pt1000, -70-500°C |
| Vibration sensor     | JHT-II-B, ±15g |
| Axial force          | Hydraulic loading 0-30 kN |
| Radial force         | Hydraulic loading 0-30 kN |
| Oil flow transducer  | FT-110, 0-3.0L/min |
| Motorized spindle    | 0-15000 r/min |

Fig. 10(a) illustrates the temperature distribution of outer raceway at axial force of 100 N. Through extracting the temperature values at the contact area between balls and outer raceway, temperature curves at different azimuth angles can be obtained, and the outer surface temperature of outer ring at varied axial forces are measured, as shown in Fig. 10(b). It can be seen that outer surface temperatures are about 5 degrees cooler than those of outer raceway by comparing the tested results with simulated ones, because of the heat dissipation between outer surface and outer raceway. This suggests the proposed CFD simulation is reliable and power loss calculation derived from the nonlinear dynamic model is reasonable.
Effect of bearing structure on oil-air flow and temperature of high speed ball bearing by combining nonlinear dynamic and CFD model

6. Results and discussion

In engineering applications, the open oil-air lubrication method is applied to high speed bearings, as shown in Fig. 11(a), causing a large amount of lubricating oil to quickly flow out of the bearing cavity. For this, it is necessary to improve outlet’s structure to attain a certain sealing degree of outlet for reducing the outflow of oil-air, as described in Fig. 11(b). In the CFD model, sealing degrees of outlet [0%, 50%, 75%] are selected to study their effects on oil-air flow and temperature distribution inside the bearing chamber, as presented in Fig. 11 (c).

6.1 Effect of sealing degree of outlet at light and heavy loads

From Fig. 7(d), it can be seen that three angular velocity components $\omega_x'$, $\omega_y'$ and $\omega_z'$ of the ball have no obvious changes when axial force is large than 300 N. At light loads($F_a < 300$ N), three angular velocity components $\omega_x$, $\omega_y$ and $\omega_z$ of the ball are changed significantly with increasing axial force. Moreover, it can be seen from Fig. 9(a) that the revolution of ball has no obvious change when axial force is large than 200 N, while it fluctuates distinctly when $F_a < 200$ N. Power loss is firstly decreased and then be increased gradually with increasing the axial force. Due to the different motion states of the ball and power loss at light loads and heavy loads, the movements of the ball and power loss at axial forces 100 and 1000 N is adopted to define the boundary conditions of CFD model for studying the effect of sealing degree of outlet on oil-air flow and temperature inside the bearing cavity. Nozzle is positioned at azimuth angle of 0º (the ball of j=1). At this moment, rotation speed of inner ring $\omega_i$ is 10000 r/min, and radial force is 0 N.

For sealing degree of 0%, oil volume fraction distribution inside the bearing cavity at 100 N and 1000 N is described in Fig. 12. It is clear that much oil-air is distributed along outer raceway near nozzle, resulting in that temperature inside the bearing cavity near nozzle is smaller than that in other part of bearing cavity, as shown in Fig.13. This is because angular speed $\omega_z'$ of the ball entrains oil-air to outer raceway to dissipate the friction heat, as presented in Fig. 14. Yet, it can be found that at heavy load ($F_a= 1000$ N), oil volume fraction in the downstream of nozzle is larger than that at light load ($F_a= 100$ N), which is attributed to that at heavy load, angular speed $\omega_z$ of the ball rotates oil-air to the downstream of nozzle to enhance the oil volume fraction with respect to that at light load, as a result, the uniformity of temperature distribution inside bearing cavity at heavy load is better than that at light load, as shown in Fig.13. Even so, the distribution zone with high oil volume fraction is very small so that it is difficult to dissipate uniformly the friction heat inside bearing cavity. Besides, at light loads, the entrainment effect of the ball induced by angular velocity $\omega_z$ causes oil-air to surround the ball to facilitate the formation of oil-film, as shown in Fig. 14(b). At heavy load, angular speed $\omega_z'$ of the ball deviate the oil-air flow from the ball surface resulting in the difficulty in forming the oil-film, as shown in Fig. 14(a). These suggests that it is necessary to facilitate the formation of oil-film and heat dissipation by enlarging sealing degree of
outlet.

Figure 12  Oil volume fraction when sealing degree is 0%: (a) 1000 N, and (b) 100 N.

Figure 13  Temperature distribution inside the bearing camber when sealing degree is 0%: (a) Fa=1000 N, (b) Fa=100 N.

Figure 14  Streamline distribution around the ball when sealing degree is 0%: (a) Fa=1000 N, (b) Fa=100 N.

Fig. 15 presents the distribution of oil volume fraction at three kinds of sealing degrees. It can be seen that at heavy ($F_a=1000$ N) and light ($F_a=100$ N) loads, oil volume fraction inside the part of bearing cavity far away from nozzle are about 0.0, 0.001 and 0.005, respectively, when sealing degrees are 0%, 50% and 75%, despite it has no obvious change inside the part of bearing cavity near nozzle. What's more, oil volume fraction is markedly enhanced when sealing degree is 75% relative to that when sealing degree is 50%. This means oil-air can dissipate more friction heat to improve the uniformity of temperature distribution on outer and inner raceways, as shown in Figs. 16 and 17.

Figure 15 Distribution of oil volume fraction at various sealing degrees when $\omega_i=10000$ r/min: (a) 1000 N, and (b) 100 N.

From Figs. 16 and 17, it can be found that temperature of inner
and outer raceways far away from nozzle is significantly decreased by enlarging the sealing degree of outlet and yet, this method plays little effect on temperature decrease of inner and outer raceways near nozzle. However, it is clear that the uniformity of temperature distribution on outer and inner raceways is enhanced remarkably when sealing degree is 75%. Moreover, temperature distributed on inner raceway is higher than that on outer raceway when sealing degree is 75%, but temperature distributed on inner raceway is lower than that on outer raceway when sealing degree are 0% and 50%. This discrepancy is depended on the entrainment effect of balls, as shown in Figs. 18(b) and 19(b). When sealing degree is 75%, more entrainment effects of balls induced by angular velocities $\omega_x$ and $\omega_y$ are generated compared with that at sealing degree of 50% and 0%, leading to more lubricating oil distributed along outer raceway to take away friction heat. Besides, temperature of inner and outer raceways at $F_a=100$ N is lower than that at $F_a=1000$ N, which is attributed to power loss obtained from the nonlinear dynamic model has no nothing to do with sealing degree of outlet. Therefore, reasonable sealing degree of outlet can obviously reduce the temperature rise of bearings and improve the uniformity of temperature distribution on outer and inner raceways.

**Figure 16** Temperature distribution of outer and inner raceways at various sealing degrees when $\omega_i=10000$ r/min at $F_a=1000$ N.

**Figure 17** Temperature distribution of outer and inner raceways at various sealing degrees when $\omega_i=10000$ r/min at $F_a=100$ N.

**Figure 18** Streamline distribution inside bearing cavity and entrainment effect of balls when sealing degree is 75% at $F_a=100$ N: (a) streamline distribution, (b) entrainment effect.
Figure 19 Streamline distribution inside bearing cavity and entrainment effect of balls when sealing degree is 75% at $F_a=1000$ N: (a) streamline distribution, (b) entrainment effect.

From Figs. 18 and 19, it can be seen that ball 18 is surrounded by dense streamlines when sealing degree is 75%, which is more beneficial to the formation of oil-film on the ball surface with respect to that at sealing degrees of 0% and 50%. Further, more entrainment effects of balls occur compared with that at sealing degree of 50% and 0%, which means balls can be lubricated not only by oil-gas injection but also by entrainment. Additionally, intensive streamlines are distributed along inner raceway relative to that at sealing degrees of 0% and 50%, indicating that temperature of inner raceway is further reduced (illustrated in Figs. 16(b) and 17(b)), although much lubricating oil is entrained to outer raceway. Therefore, enlarging sealing degree of outlet can facilitate the formation of oil-film on balls’ surfaces.

Fig. 20 describes the distribution of drag force for balls at various sealing degrees: (a) 1000 N, and (b) 100 N.

6.2 Effect of sealing degree of outlet at high and low speeds

Fig. 21 shows the rotation and revolution speeds of balls and power loss when changing rotation speed of inner ring at $F_a=1000$ N. Obviously, $\omega_{x}$ and $\omega_{y}$ are increased significantly with increasing rotation speed of inner ring, and angular velocity $\omega_{y}$ is almost 0 due to the restraint of axial force. Besides, power loss and revolution speed of balls are increased gradually with increasing rotation speed of inner ring. To study the effect of sealing degree of outlet on oil-air flow and temperature distribution inside the bearing chamber at low, high and ultra-high speeds, revolution and rotation speeds of balls and power loss at {5000, 10000, 15000} r/min are selected to define the boundary conditions of CFD model.

To sum up, enlarging sealing degree of outlet not only reduces the temperature rise of bearings and improves the uniformity of temperature distribution, but also promotes the formation of oil-film on balls’ surfaces without increasing power loss.
Effect of bearing structure on oil-air flow and temperature of high speed ball bearing by combining nonlinear dynamic and CFD model

Figure 21 Rotation and revolution speeds of balls and power loss when changing rotation speed of inner ring at $F_a=1000$ N.

For the open lubrication method in engineering applications, Fig. 22 presents the streamline distribution inside the bearing cavity at low, high and ultra-high speeds. It can be found that oil-air flows out the bearing cavity and no balls are surrounded balls at low speed (5000 r/min), suggesting the difficulties in forming the oil-film on balls’ surfaces and dissipating friction heat. When $\omega_i=10000$ r/min, high pressure near nozzle (showing in Fig. 22(d)) facilitates oil-air to spray the ball resulting in the formation of oil-film on balls’ surfaces, and inner raceway takes away plenty of oil-air because of the high rotation speed of inner ring (as shown in Fig. 22(b)), particularly, entrainment effects of a few balls near nozzle on oil-air occur. At rotation speed of 15000 r/min, high pressure near nozzle deviate oil-air flow so that a little oil-air is entrained by balls and took away by inner ring (as shown in Fig. 22(c)), that is, a large amount of oil-air is leaked from the bearing cavity. These phenomena indicate that open lubrication method is not conducive to the formation of oil-film and heat dissipation. For this, it is necessary to improve the outlet seal to facilitate the formation of oil-film and temperature reduction.

Figure 22 Streamline distribution and pressure distribution inside the bearing cavity at sealing degree of 0%; (a) $\omega_i=5000$ r/min (b) $\omega_i=10000$ r/min, (c) $\omega_i=15000$ r/min, (d) pressure distribution at $F_a=1000$ N.

Figs. 23 and 15(a) present the distribution of oil volume fraction at low, high, and ultra-high speeds when changing sealing degrees. It is obvious that oil volume fraction inside the part of bearing cavity far away from nozzle is increased when enlarging sealing degree of outlet, which means temperature distributed on inner and outer raceways far away from nozzle can be reduced, as shown in Figs. 24 and 16. Moreover, when rotation speed
is 15000 r/min, oil volume fraction inside bearing cavity is lower than that at rotation speeds of 5000 and 10000 r/min, in spite of adopting sealing degree of 75% at 15000 r/min. It can be inferred that the high speed flow of oil-air induced by inner raceway causes centrifugal force to make oil-air flow out of the bearing cavity, thus, temperature of inner and outer raceways at sealing degree of 75% is not obviously reduced with respect to that at sealing degree of 50%, as shown in Fig. 24(c) and (d). Therefore, large sealing degree of outlet should be adopted at low and high speeds, yet it should be appropriately reduced at ultra-high speeds.

**Figure 23** Distribution of oil volume fraction at various sealing degrees when $F_a=1000$ N: (a) 5000 r/min, (b) 15000 r/min.

**Figure 24** Temperature distribution of inner and outer raceways at various sealing degrees when $F_a=1000$ N: (a) for outer ring at 5000 r/min, (b) for inner ring at 5000 r/min, (c) for outer ring at 15000 r/min, and (d) for inner ring at 15000 r/min.

For sealing degree of 75%, streamline distribution inside the bearing cavity at various rotation speeds of inner ring is described in Fig.25. Clearly, at 5000 r/min, streamlines distributed inside the bearing cavity become denser, yet no streamlines surround balls, which is because balls with low angular velocities $\omega_x$ and $\omega_z$ difficulty entrain oil-air to form the oil-film on balls’ surfaces. At 10000 r/min, not only streamlines distributed along inner raceway become more intensive, but also more entrainment effects of balls induced by angular velocities $\omega_x$ and $\omega_z$ occur, as shown in Fig. 25(b), which suggests that at high speeds, large sealing degree of outlet not only reduces the temperature rise of bearings, but also promotes the formation of oil-film on balls’ surfaces. At
15000 r/min, only two balls near nozzle entrain oil-air to lubricate balls’ surfaces although angular velocities $\omega_x$ and $\omega_z$ of balls are very high, and no more dense streamlines appear inside the bearing cavity, as shown in Fig. 25(c). To summarize, enlarging sealing degree of outlet not only reduces the temperature rise of bearings, but also promotes the lubrication of balls at high speeds, yet it can reduce the temperature rise but can’t markedly improve the formation of oil-film at low and ultra-high speeds.

6.3 Effect of cage structure

Presently in engineering applications, there are three distinct cage structures are assembled into bearings: birfield cage, corrugated cage, half birfield cage back towards nozzle or facing nozzle, as shown in Fig. 26. Their associated geometries and assemblies have measurable effects on oil-air flow and temperature distribution inside the bearing chamber. For this, four cases (Case 1: birfield cage, Case 2: corrugated cage, Case 3: half birfield cage back towards nozzle, Case 4: half birfield cage facing nozzle) are selected to study their influences on oil-air flow and temperature rise, when rotation speed of inner ring $\omega_i$ is set as 10000r/min, axial force $F_a$ is 1000 N, radial force $F_r$ is 0 N, and sealing degree is 75%.

From Fig. 27, it can be found that many streamlines in Case 2 and 3 flow out of the bearing chamber so that few streamlines are distributed inside the part of bearing chamber away from nozzle, which means oil volume fraction inside the part of bearing chamber away from nozzle is low in Case 2 and 3(as shown in Fig. 28). This is because that the corrugated geometry (describing in Fig. 26(b)) of corrugated cage is beneficial to the outflow of
oil-air from bearing cavity. For Case 3, when half birfield cage backs towards nozzle, its effect on oil-air flow is almost the same as that of corrugated cage. As a result, high temperature rise of inner and outer raceways appears inside the part of bearing chamber away from nozzle for Case 2 and 3, as presented in Fig. 29. For birfield cage in Case 1, it prevents the outflow of oil-air from bearing cavity because of its ring-like structure. This facilitates inner raceway to carry much oil-air to the part of bearing chamber away from nozzle, which can be explained by streamline density inside the part of bearing chamber away from nozzle. Accordingly, higher oil volume fraction in Case 1 compared with that in Case 2 and 3 can dissipate more friction heat, resulting in that temperature of inner and outer raceways in Case 1 is lower than that in Case 2 and 3, as described in Fig. 29. When half birfield cage faces nozzle in Case 4, it not only impedes the outflow of oil-air but also facilitates the storage of oil-air into pocket clearance so that more oil-air is carried to the part of bearing chamber away from nozzle by half birfield cage and inner raceway, causing densest streamlines appear here relative to that in Case 1, 2 and 3. Thus, oil volume fraction in Case 4 is further increased inside the part of bearing chamber away from nozzle to take away more friction heat with respect to that in Case 1, 2 and 3, inducing the lowest temperature rise of inner and outer raceways in Case 4 relative to that in Case 1, 2 and 3, as observed in Figs. 28 and 29. Therefore, half birfield cage facing nozzle is beneficial to improve oil volume fraction inside the bearing cavity to reduce the temperature rise of bearings.

Figure 27 Streamline distribution inside bearing cavity: (a) Case 1, (b) Case 2, (c) Case 3, and (d) Case 4.
Effect of bearing structure on oil-air flow and temperature of high speed ball bearing by combining nonlinear dynamic and CFD model

7. Conclusions

(1) At light loads, angular velocity $\omega_x$ causes oil-air to surround the ball to facilitate the formation of oil-film. At heavy loads, angular speed $\omega_z$ of the ball deviate the oil-air flow from the ball’s surface resulting in the difficulty in forming the oil-film.

(2) When sealing degree is 75%, more entrainment effects of balls induced by angular velocities $\omega_x$ and $\omega_z$ are generated compared with that at sealing degree of 50% and 0%, leading to more lubricating oil distributed along outer raceway to take away friction heat, as a result, temperature distributed on inner raceway is higher than that on outer raceway.

(3) At high speeds, enlarging sealing degree of outlet not only reduces the temperature rise of bearings and improves the uniformity of temperature distribution, but also promotes the formation of oil-film on balls’ surfaces without increasing power loss. Yet it can reduce the temperature rise but can’t markedly improve the formation of oil-film at low and ultra-high speeds.

(4) Half birfield cage facing nozzle plays an important role in improving oil volume fraction inside the bearing cavity to reduce the temperature rise of bearings, and the next is birfield cage, they are again corrugated cage and half birfield cage back towards nozzle.

8 Declaration

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Availability of data and materials
The datasets supporting the conclusions of this article are included within the article.

Authors’ contributions
The author’s contributions are as follows: Lin Hua was in charge of the whole trial; Guiqiang Zhao wrote the manuscript; Song Deng and Dongsheng Qian assisted with sampling and laboratory analyses.

Competing interests
The authors declare no competing financial interests.

Consent for publication
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Figures

Figure 1
Main mesh diagram of 7008C angular contact ball bearing.

Figure 2
Reference system of balls and fluid domain in the CFD model.
Figure 3

Definition of three coordinate systems.
Figure 4

Ball center and raceway groove curvature centers before and after operation.
Figure 5

Forces applied on the ball.
Figure 6

Calculation flowchart of movements of balls and power loss for defining the boundary conditions of CFD model.
Figure 7

Variation of angular velocity components with time and axial loads (a) $\omega x'$, (b) $\omega y'$ and (c) $\omega z'$, and (d) variation of $\omega x'$, $\omega y'$ and $\omega z'$ in the convergent state with increasing axial load.
Figure 8

Variation rule comparison of the ratio $\omega_{cage}/\omega_i$ between present results and tested [36] and analytical [16] ones.
Figure 9

Illustration of (a) revolution and power loss at different axial forces, and (b) experimental rig of high speed ball bearings.
Figure 10

Illustration of (a) temperature distribution on outer raceway when $Fa = 100N$, (b) comparison of temperature between simulated and tested results.
Figure 11

Illustration of sealing degree for bearing chamber: (a) open lubrication method in engineering applications, (b) improved outlet's structure, and (c) various sealing degrees in CFD model.
Figure 12

Oil volume fraction when sealing degree is 0%: (a) 1000 N, and (b) 100 N.
Figure 13

Temperature distribution inside the bearing camber when sealing degree is 0%: (a) Fa=1000 N, (b) Fa=100 N.

Figure 14

Streamline distribution around the ball when sealing degree is 0%: (a) Fa=1000 N, (b) Fa=100 N.
Figure 15

Distribution of oil volume fraction at various sealing degrees when $\omega_{i}=10000$ r/min: (a) 1000 N, and (b) 100 N.
Figure 16

Temperature distribution of outer and inner raceways at various sealing degrees when $\omega_i=10000$ r/min at $F_a=1000$ N.
Figure 17

Temperature distribution of outer and inner raceways at various sealing degrees when $\omega_i=10000$ r/min at $F_a=100$ N.
Figure 18

Streamline distribution inside bearing cavity and entrainment effect of balls when sealing degree is 75% at Fa=100 N: (a) streamline distribution, (b) entrainment effect.
Figure 19

Streamline distribution inside bearing cavity and entrainment effect of balls when sealing degree is 75% at Fa=1000 N: (a) streamline distribution, (b) entrainment effect.
Figure 20

Distribution of drag force for balls at various sealing degrees: (a) 1000 N, and (b) 100 N.
Figure 21

Rotation and revolution speeds of balls and power loss when changing rotation speed of inner ring at $F_a=1000$ N.
Figure 22

Streamline distribution and pressure distribution inside the bearing cavity at sealing degree of 0%: (a) $\omega_i = 5000$ r/min (b) $\omega_i = 10000$ r/min, (c) $\omega_i = 15000$ r/min, (d) pressure distribution at $Fa=1000$ N.
Figure 23

Distribution of oil volume fraction at various sealing degrees when $F_a=1000$ N: (a) 5000 r/min, (b) 15000 r/min.
Figure 24

Temperature distribution of inner and outer raceways at various sealing degrees when Fa=1000 N: (a) for outer ring at 5000 r/min, (b) for inner ring at 5000 r/min, (c) for outer ring at 15000 r/min, and (d) for inner ring at 15000 r/min.
Figure 25

Streamline distribution inside the bearing cavity at sealing degree of 75%: (a) $\omega = 5000 \text{ r/min}$ (b) $\omega = 10000 \text{ r/min}$, (c) $\omega = 15000 \text{ r/min}$. 
Figure 26

Cage structure and assembly: (a) Case 1, (b) Case 2, (c) Case 3, and (d) Case 4.
Figure 27

Streamline distribution inside bearing cavity: (a) Case 1, (b) Case 2, (c) Case 3, and (d) Case 4.
Figure 28

Distribution of oil volume fraction inside bearing chamber for various geometries and assemblies of cage.
Figure 29

Temperature distribution of outer and inner raceways for various geometries and assemblies of cage.