The effect of lubricant viscosity on the performance of full ceramic ball bearings

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Abstract

As a new high-end bearing product, full ceramic ball bearings are favoured in a variety. However, there have been few studies on the lubrication of full ceramic ball bearings. The purpose of this study is to reveal the relationship between the vibration and temperature rise of full ceramic angular contact ball bearings and the lubricant viscosity, and to improve the service life of the bearings. In this study, the effects of lubricant viscosity on the vibration and temperature rise of silicon nitride full ceramic angular contact ball bearings under different axial loads and rotation speeds were tested. Herein, a mathematical model of oil lubrication suitable for full ceramic ball bearings is established and the relationship between the lubricant viscosity, lubricant film thickness, outer ring vibration and temperature rise of the bearing is analyzed. It was found that the vibration and temperature rise first decrease and then increase with the increase of lubricant viscosity. In this range, there is an optimal viscosity value to minimize the vibration and temperature rise of the full ceramic angular contact ball bearing. The contact surface wear of the full ceramic angular contact ball bearing varies greatly under different lubricant viscosities. There is no obvious wear on the contact surface under optimal viscosity, and the service life of the bearing is greatly improved. These results can play an important role in revealing the lubricant mechanism of full ceramic ball bearings and improving their service life under optimal lubrication.

1. Introduction

With the progress of science and technology, the application environment and conditions of ball bearings such as high speed, high temperature, corrosion resistance, ability to operate in strong magnetic fields and oil-free lubrication, are becoming more and more demanding. Metal ball bearings with grease lubrication and oil lubrication have gradually failed to meet these requirements [1, 2]. As a new high-end product, full ceramic ball bearings are optimal in various extreme working conditions because of their material characteristics [3, 4]. As a basic guarantee for the operation of full ceramic ball bearings, optimal lubrication conditions are one of the important parameters affecting their performance. According to statistics, the failure ratio of full ceramic ball bearings caused by poor lubrication was about 72.2% [5, 6]. For full ceramic ball bearings, lubrication theory and methods are a vital part of the overall process of design, manufacturing and application. Lubricant viscosity, as a key parameter to improve lubrication efficiency and lubrication effect, is an important index to measure the performance of lubricating oil. In the lubrication process of full ceramic ball bearings, lubricant viscosity also plays an indispensable role in forming a good lubricant film between the ball and the ring [7–9]. Therefore, studying the effect mechanism of lubricant viscosity on the lubrication of full ceramic ball bearings is important.
for the selection of lubricating oil, the improvement of lubrication performance and the improvement of service life.

In recent years, extensive research has been conducted on the lubrication characteristics of ball bearings. Su et al analysed the lubricating grease technology of hybrid ceramic ball bearings, and carried out volatilization tests of different greases under the actual working conditions of cold compressors [10]. The feasibility of using grease lubricated hybrid ceramic bearings on cold compressors was confirmed. Yuan et al studied the temperature rise and vibration of metal bearings and hybrid ceramic bearings under different lubricant viscosities at high speeds [11]. Wen et al explained and elaborated various lubrication methods for silicon nitride ceramic bearings [12]. They predicted the future lubrication technology of silicon nitride ceramic bearings and proposed new development ideas for future lubrication technology. Considering the lubricant parameters, Huang et al analyzed the effect of lubricant viscosity and density on the dynamics of angular contact ball bearings under ideal and actual working conditions, and aimed to provide a data reference for the design of such bearings [13]. Based on the elastohydrodynamic lubrication geometry model of hydro-hybrid ceramic ball bearings, Zhang et al studied the effect of water gage on the pressure film thickness of water-lubricated hydro-hybrid ceramic ball bearings under different conditions [14]. Zhou et al compared the operating conditions of full ceramic ball bearings and non-lubricated steel ball bearings, and recorded their temperature rise, vibration and fatigue life [15]. Dmitrichenko et al established a dynamic model of ball bearings and studied the effects of different rheological models of lubricants and hydrodynamic pressure on the dynamics of bearings [16]. Wen et al examined the effect of outer raceway defects with different sizes and position angles on the bearing dynamics under high-speed working conditions [17]. Taking the high-speed angular contact ceramic ball bearing as the research object, He et al designed a high-speed grease test-bench, compared grease lubrication with oil mist lubrication, and studied the relationship between speed, load and temperature rise of high speed bearings [18]. Zhang et al obtained the calculation formulas of the elastohydrodynamic traction coefficients of three aviation lubricants through tests, and established the nonlinear dynamic differential equations of high-speed angular contact ball bearings [19]. Dong et al proposed a numerical dynamic analysis method for ball bearings [20]. Based on Hertz contact theory and elastohydrodynamic lubrication theory, the normal contact force between the ball and the raceway was calculated and the results were compared. Aiming at the high stiffness and uneven load under starvation lubrication of full ceramic bearings, Shi et al added uneven load conditions to the dynamic model to study the effects of ball diameter tolerance, ball arrangement and ball number between the interaction of the inner ring and the balls [21]. Combined with the low speed and heavy load conditions in startup and shutdown, Shi et al incorporated the dynamic performance of aerodynamic ball bearings into the hybrid thermal elastohydrodynamic lubrication theory, the accuracy of the numerical lubrication model under low speed and heavy load was verified by experiments [22].

It can be seen from the above results that, the prior research on the lubrication of ball bearings has mainly focused on the lubrication of metal ball bearings and hybrid ceramic ball bearings, and that relatively few studies have focused on the lubrication of full ceramic ball bearings. Only a few experts have studied the lubrication of full ceramic ball bearings, and their work was only limited to the excellent characteristics of full ceramic ball bearings under standard lubrication conditions. The effects of lubricant characteristics (e.g. composition, viscosity, etc) on the service life of full ceramic ball bearings have not been quantified. Therefore, the present study used a BVT-1A bearing vibration measuring instrument to test the effects of lubricant viscosity under different working conditions on full ceramic angular contact ball bearings (hereinafter referred to as full ceramic ball bearings) that were made of silicon nitride ceramic materials. The outer ring vibration and the temperature rise of full ceramic ball bearings were measured under different lubricant viscosities. The vibration and temperature rise of full ceramic ball bearings and of metal ball bearings were compared experimentally. Combined with the test data, a mathematical model suitable for oil lubricated full ceramic ball bearings was established. Moreover, a S-4800 scanning electron microscope was used to observe the surface topography of the ceramic ball and cage after the test; the corresponding conclusions are discussed. The results of this paper are of great significance for selecting full ceramic ball bearing lubricant, improving its lubrication and service life, and providing data support for the lubrication of full ceramic ball bearings. In addition, the mathematical model of lubrication established in this paper also provides guidance for the selection of optimal viscosity of lubricants for full ceramic ball bearings in practical applications.

2. Mathematical model of the full ceramic ball bearing

2.1. Analysis of vibration characteristics of the full ceramic ball bearing

2.1.1. Contact analysis between ceramic balls and bearing rings

The friction and impact between ceramic balls and bearing rings cause the bearing vibration. In order to study the vibration of full ceramic ball bearings, it is particularly important to analyse the contact state between the
ceramic balls and bearing rings. In the calculation of the contact between angular contact bearing components, it is generally considered that there is a point contact deformation between the balls and the inner and outer rings. When the external force acting on the bearing is small, the mutual deformation between the ball and ring at the contact point will produce a large contact stress. In order to study the contact vibration between the ceramic balls and the rings, the contact stress of full ceramic ball bearing is analysed by the point contact method. The contact deformation conforms to the Hertz elastic contact theory.

According to the Hertz contact theory, the contact force $Q$ is:

$$Q = K \delta^{3/2}$$

where, $\delta$ is the contact deformation, and $K$ is the contact stiffness.

$$K = \frac{1}{3} E' (\Sigma \rho)^{\frac{1}{2}}\left(\frac{2}{b^*}\right)^{\frac{3}{2}}$$

$$\delta^* = \frac{2F}{\pi \left(\frac{E}{2\kappa^2E}\right)^{\frac{3}{2}}}$$

In these formulas, $E'$ is the elastic modulus parameter; $\kappa$ is the eccentricity parameter of the ellipse; and $E$ and $F$ are the first and second complete ellipse integrals, respectively. The contact force and length of the long and short radius of the ellipse in the contact area can be obtained by calculating the contact deformation.

The analysis of contact deformation between ceramic balls and rings should consider the surface elastic deformation. According to the elastic theory, the elastic displacement of each point in the vertical direction can be expressed as:

$$v(x) = -\frac{2}{2\pi E} \int_{s_1}^{s_2} p(s) \ln (s - x)^2 ds + c$$

$$\times \frac{1}{E} = \frac{1}{2} \left(\frac{1 - \nu_e^2}{E_e} + \frac{1 - \nu_r^2}{E_r}\right)$$

where, $v(x)$ is the elastic displacement in the vertical direction. For the elasto-hydrodynamic lubrication, $p(s)$ is the fluid pressure distribution; $s$ is the additional coordinate on the $x$-axis, representing the distance between $p(s)$ and the origin of the coordinate; $s_1$ and $s_2$ are the start and end points of the load, respectively; $c$ is an undetermined constant that is incorporated into $h_0$; $E$ is the equivalent elastic modulus; $E_e$ is the elastic modulus of the outer raceway; $E_r$ is the elastic modulus of the balls; $\nu_e$ is the Poisson ratio of the outer raceway, and $\nu_r$ is the Poisson ratio of the ball bearings.

2.1.2. Vibration differential equation of the outer ring of full ceramic ball bearings

Generally, the bearing is assembled in the bearing housing and regarded as a rigid contact between the outer ring and the block is assumed. Since the thermal expansion coefficient of the full ceramic ball bearing material is small, the contact stiffness between the bearing and the seat will change significantly with changes of ambient temperature, which will affect the dynamic characteristics of full ceramic ball bearings. When analysing the dynamic characteristics of full ceramic ball bearings, the vibration of the outer ring needs to be considered.

The forces acting on the outer ring include the contact force and the friction caused by the interaction between balls and the outer ring, as well as the acting force of the bearing housing on it. When analysing the
interaction between the outer ring and the bearing housing, the bearing housing is considered to be fixed. The contact between the outer ring and the bearing housing is considered to be rigid. The force on the outer ring of a full ceramic ball bearing is shown in figure 1.

In figure 1, $\varphi_{oj}$ represents the position angle of the ceramic ball relative to the outer ring, and $\alpha_{oj}$ represents the centroid offset angle of the outer ring.

The vibration differential equation of the outer ring of full ceramic ball bearings can be rewritten as:

$$F_{ox} + \sum_{j=1}^{N} ((T_{ojo} - F_{ojo}) \cos \alpha_{oj} + Q_{ojo} \sin \alpha_{oj}) = m_{o} \ddot{x}_{o}$$

$$F_{oy} - \sum_{j=1}^{N} ((T_{ojo} - F_{ojo}) \cos \phi_{oj} - (T_{ojo} - F_{ojo}) \sin \alpha_{oj} \sin \phi_{oj} + Q_{ojo} \cos \alpha_{oj} \sin \phi_{oj}) = m_{o} \ddot{y}_{o}$$

$$F_{oz} - \sum_{j=1}^{N} ((T_{ojo} - F_{ojo}) \sin \phi_{oj} + (T_{ojo} - F_{ojo}) \sin \alpha_{oj} \cos \phi_{oj} - Q_{ojo} \cos \alpha_{oj} \cos \phi_{oj}) = m_{o} \ddot{z}_{o}$$

(4)

where, $F_{ox}$, $F_{oy}$, and $F_{oz}$ are the components of the external load along each coordinate axis of the outer ring; $m_{o}$ is the mass of the outer ring; $\ddot{x}_{o}$, $\ddot{y}_{o}$, and $\ddot{z}_{o}$ are the displacement accelerations of the outer centre of mass along their respective coordinates in the coordinate system $\{O, X, Y, Z\}$; $\omega_{o}, \omega_{yo}$, and $\omega_{zo}$ are the angular velocities of the outer ring along their respective coordinates in the coordinate system $\{O, X, Y, Z\}$; $I_{ox}$, $I_{oy}$, and $I_{oz}$ are the rotational inertias of the outer ring along their respective coordinates in the coordinate system $\{O, X, Y, Z\}$; $k_{o}$ is the curvature radius coefficient of the outer raceway; $r_{o}$ is the raceway radius of the outer ring, and $r_{o} = 0.5d_{m} - 0.5d_{n}k_{o} \cos \alpha_{oj}$.

2.2. Lubricant properties

2.2.1. Viscosity-pressure and viscosity-temperature effect

The pressure and temperature of the lubricant have a large effect on its viscosity. Based on the viscosity-pressure and viscosity-temperature equations proposed by Roelands, the relationship between the temperature, pressure and viscosity of the lubricant can be expressed as:

$$\eta = \eta_{0} \exp \left[ A_{1} \left( 1 + \frac{1}{1 + A_{2}p} \right)^{2} (A_{3} T - A_{4})^{s_{1}} \right]$$

(5)

where, $A_{1} = \ln (\eta_{0}) + 9.67$; $A_{2} = 5.1 \times 10^{-9} (\text{Pa}^{-1})$; $A_{3} = 1/(T_{o} - 138) (\text{K}^{-1})$; $A_{4} = 138/(T_{o} - 138) (\text{K}^{-1})$; $z_{o} = \alpha/(A_{1}A_{2})$; and $s_{0} = \beta/(A_{1}A_{2})$. Here, $\alpha$ is the viscosity coefficient in the Barus viscosity-pressure equation; $\beta$ is the thermal expansion coefficient in Reynolds’ Viscosity-temperature equation, with a value of 0.00065 m K$^{-1}$, and $\eta_{0}$ is the dynamic viscosity under atmospheric pressure.

2.2.2. Formulas of lubricant film thickness

For full ceramic ball bearings, the Dawson-Higginson film thickness formula is generally used to calculate the minimum lubricant film thickness [23]:

$$h_{min} = 2.65\alpha^{0.54}(\eta_{0}u_{c})^{0.7} \rho_{12}^{0.43} E^{0.03} W^{0.13}$$

(6)

where, $\alpha$ is the pressure viscosity coefficient of the lubricant; $\eta_{0}$ is the viscosity coefficient; $u_{c}$ is the average oil pouring speed; $\rho$ is the equivalent cylinder radius; $E'$ is the overall elasticity modulus, and $W$ is the maximum load per unit contact length.

For the internal state of the contact area of full ceramic ball bearings, under the pure rolling assumption, the entrainment velocity can be calculated by the kinematic behavior of the bearing. The calculation process can be referred to [24], and the specific formula is:

$$u_{c} = \frac{d_{m}}{2} [(1 - \gamma) (\omega_{i} - \omega_{m}) + \gamma \omega_{R}]$$

(7)

where, $\omega_{i}$, $\omega_{m}$ and $\omega_{R}$ are the angular velocities of the inner raceway, cage and balls, respectively.

$$\gamma = D \cdot \cos \alpha / d_{m}$$

$D$ is the diameter of the ball; $d_{m}$ is the pitch circle diameter, and $\alpha$ is the contact angle.

The dependence of lubricant viscosity on the shear rate of the lubricant film is the same as that of complex viscosity on the shear rate:

$$\eta^{*}(\omega) = \eta(\gamma)$$

(8)

where, $\eta$ is the dynamic viscosity of lubricant, and $\gamma$ is the shear rate. However, this law is only applicable to the linear viscosity-elastic behaviour. This is because the elastic-plastic behaviour of the shear and yield stress is mostly nonlinear. By adding amplitude variables to the formula, equation (5) can be changed to:

$$\eta^{*}(\gamma, \omega) = \eta(\gamma)$$

(9)

where, $\lambda_{m}$ is the strain amplitude when the angular shear rate $\omega$ is applied, and normally should be high enough.
2.2.3. Relationship between dynamic viscosity, lubricant film thickness and lubricating film pressure

Reynolds equation, as a basic equation for describing and calculating viscous fluid flow, is often used to analyse and determine the relationship between lubricant film thickness, lubricant film pressure and dynamic viscosity.

\[ \frac{\partial}{\partial \theta} \left( h \frac{\partial p}{\partial \theta} \right) + R^2 \frac{\partial}{\partial y} \left( h \frac{\partial p}{\partial y} \right) = 6 \eta \mu \frac{\partial h}{\partial \theta} \] (10)

where, \( p \) is the lubricant film pressure; \( h \) is the lubricant film thickness; \( R \) is the bearing radius; \( u \) is the translation speed of the axis neck surface along the circumference cut, \( u = R \omega \frac{\partial \theta}{\partial \theta} \); \( \omega \) is the rotational angle speed of the shaft neck; \( \eta \) is the dynamic viscosity of the lubricant oil; \( \theta \) and \( y \) are the circumferential and axial coordinates respectively of the bearings.

When the pressure distribution of lubricant film is solved, it is usually expressed in a dimensionless form for convenient calculation. Taking the dimensionless variables as \( \tilde{\eta} = \eta \omega \), \( \tilde{h} = h / \eta \), \( \tilde{p} = p / p_0 \), the equation can be written as:

\[ \frac{\partial}{\partial \theta} \left( \tilde{h} \frac{\partial \tilde{p}}{\partial \theta} \right) + \left( \frac{R}{L} \right)^2 \frac{\partial}{\partial y} \left( \tilde{h} \frac{\partial \tilde{p}}{\partial y} \right) = \frac{\partial \tilde{h}}{\partial \theta} \] (11)

where, \( L \) is the bearing width, and \( p_0 = \eta \omega R^2 / \tilde{c}^2 \).

2.3. Film thickness ratio and lubrication state

The operating conditions of bearings determine the lubrication status. For instance, the full ceramic ball bearing, rotation speed and load size will directly affect the lubricant film thickness. Therefore, appropriate oil film thickness should be designed according to the specific situation. When calculating the lubricant film thickness between contact pairs, the design parameter is usually the film thickness ratio \( \lambda \):
Equation (12) can also be written as:

$$h_{\text{min}} \geq 1.225 [\lambda] \cdot \sqrt{R_{a1}^2 + R_{a2}^2}$$  \hspace{1cm} (13)$$

where, $h_{\text{min}}$ is the minimum film thickness between the contact areas, and $\sigma_1$ and $\sigma_2$ are the root mean square deviations of two contact surfaces. The approximate relationship between the root mean square deviation $\sigma$ and the arithmetic mean deviation $R_a$ is $\sigma = (1.2 - 1.25) R_a$. $\lambda$ is the film thickness ratio, and $[\lambda]$ is the allowable film thickness ratio. The full ceramic ball bearing in this test is P4 ultra-precision ball bearing, and the surface roughness between the balls and rings is 0.03 microns.

Usually, when $\lambda > 3$, it can be assumed that the bearing assembly is in a state of completely elastic fluid lubrication, and the micro-bulges on the surface do not contact each other; under these conditions, bearing life can be increased to several times the rated life. When $1 < \lambda < 3$, the situation can be regarded as a partial elastic...
fluid lubrication or mixed lubrication state. Practically, it has been proved that most of high-side contact parts such as gears, cams and rolling bearings operate in a state of partial elastic fluid lubrication. Considering the effect of lubricant, additives and temperature, if the conditions close to elastohydrodynamic (EHD) lubrication are obtained, $\lambda$ should be in the range from 1.5 to 2.

2.4. Calculation process
The calculation process chart of the mathematical model is shown in figure 2.

3. Performance test of full ceramic ball bearings with different lubrication viscosities

3.1. Test bearings and oil
The test bearings were 7007 C silicon nitride full ceramic ball bearings. The inner and outer rings of the bearing and the balls were made of silicon nitride ceramic, and the cage was made of Bakelite. Its structural parameters are shown in table 1.

The test lubricant was Jinmei brand lubricating oil. According to the viscosity classification requirements of industrial liquid lubricants [25], the selected lubricant viscosity grade, kinematic viscosity value and density value are shown in table 2.

3.2. Test schemes
The test was carried out on the BVT-1A bearing vibration measuring instrument shown in figure 3 (a). The effect of lubricant viscosity on the vibration and temperature rise of full ceramic ball bearings was tested under fixed bearing rotation speed or under fixed axial load by the control variable method. The specific test schemes were:

1. The fixed rotation speed was 1500 rpm, and the axial loads were selected as 100, 300, 600 and 1000 N, respectively, to measure the bearing under different viscosities.
2. The fixed axial load was 600 N, and the rotation speeds were selected as 500, 1500, 3000 and 5000 rpm, respectively to measure the bearing under different viscosities. In addition, another test was conducted under the working conditions of bearings, with an axial load of 600 N and a rotation speed of 1500 rpm. The lubricant with a viscosity of 32.0 mm$^2$ s$^{-1}$ was selected for oil-bath lubrication [26]. In this latter test, the vibration and temperature rise of silicon nitride full ceramic ball bearing and metal ball bearing were compared and analysed under the same working conditions.

3.3. Processing of vibration and temperature rise signals
The signal processing device is shown in figure 3 (b). A THA9 paperless recorder was used to record the temperature rise. The ambient temperature of the test bench and the temperature of the outer ring were read from the temperature display, and the difference between them was calculated. Thus the temperature variation of the bearing after stable operation under the lubrication of different lubricant viscosities was obtained. The bearing vibration given by the digital display device of the OFV5000-High performance laser Doppler vibrometer was used to determine the root mean square value of the vibration acceleration of full ceramic ball bearing under different lubricants. The acceleration levels of bearing vibration were calculated to describe the vibration of the full ceramic ball bearing [27]. The calculation formula is:

$$L = 20 \log \frac{a}{a_0}$$

where, $L$ refers to the acceleration level of bearing vibration, decibels (dB); $a$ is the root mean square value of vibration acceleration or the peak value of bearing vibration acceleration at a certain frequency, meters per second (m s$^{-1}$); and $a_0$ is the reference acceleration, with a value of $9.81 \times 10^{-3}$ m s$^{-2}$. 

| ISO viscosity grade | kinematic viscosity (mm$^2$ s$^{-1}$) | kinematic viscosity at the beginning of the test (mm$^2$ s$^{-1}$) | density (kg m$^{-3}$) |
|--------------------|-------------------------------------|-------------------------------------------------|---------------------|
| 2                  | 2.2                                 | 10.2                                            | 879.3               |
| 10                 | 10.0                                | 19.6                                            | 875.1               |
| 22                 | 20.0                                | 32.4                                            | 882.3               |
| 32                 | 32.0                                | 48.0                                            | 880.0               |
| 46                 | 50.0                                | 74.1                                            | 881.7               |
| 100                | 100.0                               | 198.3                                           | 885.4               |
| 220                | 220.0                               | 350.4                                           | 886.7               |
| 320                | 320.0                               | 517.2                                           | 891.4               |
After 4 h of preliminary rotation, the data were recorded every 2 min, and the averages of ten groups of data were taken as the experimental data points.

4. Test results and analysis

4.1. Effect of lubricant viscosity on the vibration of full ceramic ball bearings

The vibration acceleration level of full ceramic ball bearings under different lubricant viscosities is shown in Figure 4.

It can be seen that under current conditions, the vibration acceleration level of full ceramic ball bearings shows a trend of first decreasing and then rising. The vibration of full ceramic ball bearings will be more severe when the lubricant viscosity is high or low. When the lubricant viscosity is between 2.2 and 32.0 mm$^2$/s$^{-1}$, the vibration acceleration level of full ceramic ball bearing in operation decreases rapidly as the lubricant viscosity increases. This is because the lubricant viscosity is low, leading to a sparse lubricant film between the ceramic balls and the ring, and insufficient lubricant film thickness. The lubricant film cannot completely isolate the two contact surfaces; hence the full ceramic ball bearing is in a mixed lubrication state [28]. There are micro-bulges on the balls and the raceway, and they directly rub and collide with each other [29]. When the balls rotate in the cage under the action of friction, the vibration of the bearing increases rapidly and reaches a peak value. The lower the lubricant viscosity is, the more micro-bulges on the two surfaces directly contact between each other, resulting in very severe vibrations. When the lubricant viscosity is 32.0 mm$^2$/s$^{-1}$, the vibration of the full ceramic
ball bearing is optimal. In this situation, the balls and the raceway are completely separated by the lubricant film, and the full ceramic ball bearing is in a fluid lubrication state. However, with any additional increase of the lubricant viscosity, the thickness of the lubricant film between the contact areas increases. When this happens, the ceramic balls need to overcome the large friction created by the lubricant film, and this excessive internal friction of the lubricant film added to the external friction will lead to a large vibration effect. In addition, angular contact bearings should have enough axial loads to eliminate bearing clearance in operation, so that there is interference inside the bearings; and thus the stable movement of each component should be guaranteed. When the load is not large enough, unstable movement will appear between the balls and the cage, which will significantly increase the bearing vibration. This was also confirmed in the experiment. In the tests under different axial loads, with the increase of the axial load, the vibration acceleration level of the bearing decreased with the increase of the axial load [30].

4.2. The effect of lubricant viscosity on the temperature rise of full ceramic ball bearings

The temperature rise of full ceramic ball bearings with different lubricant viscosities is shown in figure 5. It can be seen that with the increase of lubricant viscosity, the temperature rise is U-shaped, which is similar to the curve of the vibration acceleration level with lubricant viscosity. For low viscosity, the lubrication is mixed and the lubricant film is thin, resulting in direct contact and friction between the ceramic balls and micro-bulges. The temperature rise will increase. As the lubricant viscosity increases; in this case, the two contact surfaces are gradually separated by lubricant, and the micro-bulge contact between the two surfaces decrease. Meanwhile, a complete lubricant film is gradually formed between the two contact areas, and the temperature rise decreases substantially.

The temperature rise of the bearing is at a minimum when the lubricant viscosity is 32.0 mm² s⁻¹. Then, as the lubricant viscosity increases, the film thickness between the contact areas increases. Although the temperature rise of the bearing increases because of the large friction of the ceramic balls, it is still less than that caused by the direct friction of micro-bulges under low viscosity. Comparing the test results under these two test conditions, it can be found that the temperature rise of the full ceramic ball bearing under the axial load is significantly lower than that at variable rotation speed. This is consistent with the vibration effect test of lubricant viscosity. This is because with increased speed, the temperature rise and entrainment velocity of the bearing increase significantly [31]. The lubricant viscosity decreases with the increase of temperature, and the lubricant film between the two contact surfaces becomes thin. In addition, the formation of a stable lubricant film under the entrainment velocity is more difficult. Therefore, the higher the speed, the more apparent the dynamic characteristics of the bearing.

4.3. Performance comparison test of full ceramic ball bearing and metal ball bearing

Figure 6 shows the vibration and temperature rise of full ceramic ball bearings and metal ball bearings under 1500 rpm of speed, 600 N of axial load and 32.0 mm² s⁻¹ of lubricant viscosity. The metal ball bearing is a type of 7007 C and the material is GCr15 alloy bearing steel. In the comparison chart of the vibration acceleration, it can be seen that the vibration acceleration fluctuation range of full ceramic ball bearings is significantly smaller than that of metal ball bearings. This indicates that the vibration frequency and variation of full ceramic ball bearings
are lower than those of metal ball bearing. In the test, the root mean square vibration acceleration of the full ceramic ball bearing was 15.095 m s\(^{-2}\), while that of the metal ball bearing was 22.031 m s\(^{-2}\), a reduction of 31.5%. It can be seen from figure 6(b) that in the first 120 min, the temperature variation rate of the metal ball bearing is significantly higher than that of the full ceramic ball bearing. As the test progresses, the temperature variation rates of the two bearings begin to decrease. When the test lasts for about 180 min, the temperatures variations of the two bearings tend stabilize. Once the temperatures have stabilized, the temperature of the metal ball bearing is about 11.7 °C higher than that of the full ceramic ball bearing.

4.4. Simulation results and analysis

Figures 7 and 8 are the comparison diagrams between the simulated data and the experimental data. The red curves are the simulated results, and the blue curves are the experimental results under the same conditions. It can be seen that the curves are basically the same, but there are differences in the corresponding data under different operating conditions. The simulation results are basically consistent with the test results when the lubricant viscosity is 20.0 mm\(^2\) s\(^{-1}\) and 32.0 mm\(^2\) s\(^{-1}\), respectively. When the rotation speed is 1500 rpm; the axial load is 300 N, and the lubricant viscosity is 100.0 mm\(^2\) s\(^{-1}\), the maximum relative error between the simulation results and the test results is 0.543%. When the load is 600 N, the speed is 5000 rpm and the lubricant viscosity is 2.2 mm\(^2\) s\(^{-1}\), the maximum relative error between the simulation results and the test results is 0.621%. Similarly, when analysing the effect of viscosity on the temperature rise of full ceramic ball bearings, the maximum relative errors under these two conditions are 3.35% and 4.18%, respectively. These errors are within the allowable range. Therefore, the comparison between the above quantitative simulation calculation and the test data shows the feasibility of the lubrication model and the reliability of the test data.
Through the established mathematical model, the relationships between lubricant film thickness, lubricant film pressure and dynamic viscosity of the lubricant under different viscosities is obtained, and the corresponding lubrication can be determined. In the working environment of full ceramic ball bearings, optimal lubrication is the key to ensuring normal operation. According to the established mathematical model, the ideal film thickness ratio $\lambda (1.5 < \lambda < 2)$ is substituted into equation (8) to obtain the lubricant film thickness $h$. The
Reynolds equation is solved by the finite difference method and the corresponding range of lubricant viscosity is obtained. The operating conditions are selected as 600 N of axial load and 1500 rpm of bearing speed. Taking $\lambda$ as 1.5, the corresponding lubricant viscosity can be calculated to be $28.4 \text{ mm}^2 \text{ s}^{-1}$. When $\lambda$ is 2, the lubricant viscosity is $33.5 \text{ mm}^2 \text{ s}^{-1}$. Therefore, the optimal viscosity range of lubricant is $28.4–33.5 \text{ mm}^2 \text{ s}^{-1}$ under the set conditions. Comparing the test data, it can be seen that full ceramic ball bearings also show optimal dynamics when using a lubricant with a viscosity grade of 32. In the experimental test, the 32 grade lubricant resulted in the optimal bearing dynamics, which also corresponds to the simulation results. Once again, the rationality of the mathematical model and the reliability of the experimental data are proved.

5. The effect of lubricant viscosity on surface morphology

Figures 9 and 10 show the surface morphology of ceramic balls and cages from disassembled test bearings that had been tested with lubricants of different viscosities; the images were obtained with a S-4800 cold field emission scanning electron microscope. It can be seen that when the viscosity is $2.2 \text{ mm}^2 \text{ s}^{-1}$, the wear of the ceramic balls is more serious. There are many dense surfaces and deep scratches and pits. Because of the direct contact between the ceramic balls and the Bakelite cage in the mixed lubrication case, there is obvious layering and material adhesion on the surface. The hardness of the two materials and the material characteristics are also different. When the ceramic balls rotate in the cage, the cage material will produce significant abrasive wear. The cage material falls off and the chips separate become randomly attached to the cage and to the surface of the ceramic balls. This will greatly affect the performance of the full ceramic bearings.

6. Discussion

Based on the Reynolds equation, the lubricant film thickness formula, the viscosity temperature and viscosity pressure formulae, combined with the effect of lubricant film thickness and pressure on ceramic balls and the dynamic model of the of ceramic bearing outer ring, a mathematical model of the effect of lubricant viscosity on the vibration and temperature rise of full ceramic ball bearings was established. The lubrication of full ceramic ball bearings corresponding to the lubricant viscosity was determined. Comparing the test results with the simulation results, it was found that the data curves of the two are basically the same. After analysis and calculation, it was found that the maximum relative error between the simulation results and test results was 4.18%. During the experiment, eight different grades of lubricants were used. Through the experiment, it can be seen that the temperature rise and vibration level of the bearing were the same, and there was an optimal viscosity value that maximizes bearing performance. These two dynamic characteristics were well-optimized when a 32 grade lubricant was employed. The optimal viscosity range obtained by the mathematical analysis model was $28.4–33.5 \text{ mm}^2 \text{ s}^{-1}$, which is consistent with the experimental data. For some lubricant viscosities, the surface damage of silicon nitride full ceramic ball bearings was obvious after operation. When the viscosity was too low or too high, there were many dense scratches and deep pits on the surface of the ceramic balls, and material adhesion and layering appeared on the surface of the cage. When the lubricant viscosity was close to the optimal range, the above phenomena were minimized and the surface damage to the ceramic balls and the cage was greatly reduced.

7. Conclusion

In this study, a mathematical analysis model of oil lubrication for full ceramic ball bearings was established. The vibration and temperature rise of bearing outer ring were calculated by this mathematical model. It was found that the lubricant viscosity has a great effect on the frictional wear of full ceramic ball bearings. The bearing surface damage was the minimum when the optimum viscosity value was used. The comparison between the simulation and experimental data shows that the established data model has excellent accuracy, and can be used to guide the practical application of full ceramic ball bearings. It also provides reference for the determination of lubrication conditions of full ceramic ball bearings.

As the lubricant viscosity of the silicon nitride full ceramic angular contact ball bearing increases, the vibration and temperature rise of the outer ring decreases at first and then increases. The rotation speed has a more obvious effect on the performance of full ceramic ball bearings than does the bearing load.

Under the same operating and lubrication, the vibration and temperature rise of full ceramic ball bearings and metal ball bearing are different. The vibration and temperature rise of the outer ring of full ceramic ball bearings are significantly lower than those of metal ball bearings. Through calculation, it can be found that the
root mean square of vibration acceleration of the outer ring of full ceramic ball bearings decreases by 30% on average, and the temperature rise decreases by 21% on average compared to metal ball bearings.

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Data availability statement

The data generated and/or analysed during the current study are not publicly available for legal/ethical reasons but are available from the corresponding author on reasonable request.

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