Research on the influence of thermal expansion of steel shaft on dynamic characteristics of full ceramic bearing-rotor system

Jiancheng Guo¹,², Yuhou Wu¹,², Xiaochen Zhang¹,³, Yu Zhang¹,², He Wang¹,², Xu Bai¹,² and He Lu¹,²

Abstract

Bearing as the core components of high-grade CNC machine tools, with the constant change of internal clearance of bearing wear increasingly aggravated, spindle rotation accuracy reduced. In this paper, the idea of control variable method is used to explore the dynamic change of bearing clearance from different speeds and different radial loads. Starting with the dynamic model of bearing vibration, the theoretical model of rolling bearing with five degrees of freedom is established in this paper. The interaction force between steel shaft and ceramic bearing inner ring is calculated by Runge-Kutta method with elastic wall thickness ring theory, and the reduction of radial clearance of bearing is obtained. Therefore, a dynamic model of ceramic bearing considering the extrusion force of ceramic bearing inner ring is proposed. At the same time, the vibration test of steel shaft-all-ceramic bearing is designed and carried out. The test results show that under the same load, the higher the rotating speed, the shorter the time for the bearing-rotor system to reach temperature stability, and the root mean square of ceramic bearing-rotor system is obviously reduced. At the same speed, the greater the load, the more obvious the root mean square increase of the bearing-rotor system.

Keywords

Full ceramic bearings, bearing dynamics, radial clearance, dynamic characteristics

Date received: 13 March 2022; accepted: 6 June 2022

Handling Editor: Chenhui Liang

Introduction

With the continuous development of high-end technologies such as aero engines and gas turbine engines develop, the requirements for the performance and operation accuracy of rolling bearings are greatly improved. Full-ceramic bearings have the advantages of low density, high temperature resistance, corrosion resistance and so on. It can also maintain operation accuracy and high reliability under high vacuum, extreme temperature and large temperature difference.¹-³ Even in the absence of lubrication, full ceramic ball bearings can work reliably. Therefore, in extreme conditions, full-ceramic bearings are usually

¹School of Mechanical Engineering, Shenyang Jianzhu University, Shenyang, Liaoning, China
²National-Local Joint Engineering Laboratory of NC Machining Equipment and Technology of High-Grade Stone, Shenyang, Liaoning, China
³Key Laboratory of Fundamental Science for National Defense of Aeronautical Digital Manufacturing Process of Shenyang Aerospace University, Shenyang, Liaoning, China

Corresponding author:
Xiaochen Zhang, School of Mechanical Engineering, Shenyang Jianzhu University, Hunnan New District Hunnan East Road 9, Shenyang, Liaoning 110168, China.
Email: xczhang@sjzu.edu.cn
preferred. Full-ceramic bearings have been used in key components of aeroengines and gas turbines. However, full-ceramic bearings are usually installed in series on a steel base via a steel shaft. The thermal expansion of the steel shaft leads to a reduction in the radial clearance of the ceramic bearing, and aggravates the wear of the rolling elements and the inner and outer raceways of the bearing, which is not conducive to maintaining the running accuracy. Therefore, the vibration of ball bearing-rotor system is caused by the interaction between bearing components.

In the past several decades, the vibration of bearings has been studied extensively. The research showed that the bearing vibration was related to load, surface roughness and waviness, and was affected by working conditions such as speed, rotating torque and temperature. Xia calculated the variable speed stiffness of rolling bearings based on load-deflection relationship of Jones and Harris bearing dynamics model. Mao et al. studied the dynamic performance of bearings by improving the existing dynamic model through the variation rule of load distribution. Xu and Li introduced nonlinear force system into the dynamics of deep groove ball bearing to study bearing load characteristics and bearing fatigue life. Bovet and Zamponi studied the dynamic law of ball bearings under large torque loads and proposed a modeling method to predict the internal dynamic characteristics of bearing structures, paying special attention to the stress of shafts and bearing inner rings. Yang et al. analyzed vibration frequencies under different damage conditions and carried out spectrum analysis on bearing fault signals, and proposed a simple bearing dynamic fault feature method. Razpotnik et al. proposed a new analytical model of bearing stiffness to overcome the problem of unstable bearing system response in the transient region by using smooth processing. Pratiwi et al. studied the wear of ball bearings by using propeller dynamic identification technology, and analyzed the X, Y, and Z axes of ball bearings vibration by using fast Fourier transform principle. Wang et al. explored the impact of water-lubricated rubber bearings on friction and wear, theoretical analysis and experimental verification were used to illustrate the impact of different speeds, loads and changes in cooling water on the vibration of bearings under different rubber materials. Therefore, it is undeniable that working temperature has a significant impact on bearing performance and the interaction between bearing elements caused by changes in bearing internal clearance is worth studying. Shah and Patel established the motion coupling relation of shaft, bearing seat, ring and ball, and obtained the motion control equation. Liu et al. analyzed the thermal structure mechanism and proposed a closed-loop iterative modeling method to modify the heat source and thermal boundary conditions. Bizarre et al. considered the force and moment balance of five degrees of freedom angular contact ball bearings affected by elastohydrodynamic (EHD) lubrication, and deduced a complete nonlinear bearing dynamics model, evaluating the equivalent parameters of stiffness and damping for each contact under different load conditions. The study of Lioulios and Antoniadis, Ma et al., and Chen and Qu proved the effect of fit clearance on bearing vibration. There are few literatures on the effect of thermal expansion of steel shaft on the clearance change of all ceramic bearing, so it is very necessary to study the effect of thermal expansion of steel shaft on the vibration of bearing rotor system.

In this study, a fully ceramic dynamic model considering the extrusion force of the inner ring of the ceramic bearing caused by the thermal expansion of the steel shaft is established in the rotor system composed of steel shaft and ceramic bearing during high-speed rotation. The expansion of steel shaft caused by the temperature rise of bearing under different rotational speeds and radial loads is analyzed, the extrusion force on the inner ring of ceramic bearing is calculated by Runge-Kutta method based on the theory of elastic wall thickness ring. The initial displacements are x0 = 10−6 m, y0 = 10−6 m, the initial velocity is 6000 rpm, and the applied loads are F = 1000, 1700, 2400, 3200,and 3900 N. The x-and y-displacements can be obtained by using the fourth-order Runge-Kutta algorithm to solve formula (17), and the experimental data are compared under the same conditions, thus verifying the correctness and effectiveness of the model.

**Experimental method**

**Experimental subject**

The bearing model used in the experiment is 6206 full ceramic silicon nitride deep groove ball bearing. Which is designed and manufactured by the research group, and the bearing accuracy is grade P4. Root mean square is a dimensional parameter, also known as effective value, which is defined as follows: 

\[ a_{rms} = \left( \frac{1}{T} \int_{0}^{T} a^2(t) dt \right)^{1/2}. \]

Root mean square represents the energy of vibration signal, which is essentially the average of time, because it has the characteristics of good stability and trend. In the vibration analysis of bearings, the root mean square of acceleration is used to characterize the vibration intensity. As shown in Figure 1, the inner and outer rings and rolling elements of the bearing are all made of silicon nitride material. The cage is peek. And the length of the steel shaft connecting the bearing is 13.35 cm. Since the silicon nitride ceramic material is a brittle material, the rigidity is high and it is impossible to use the traditional rigid bearing to directly press and assemble. Therefore, in this paper, liquid nitrogen is used to achieve ultra-low temperature...
cooling, and the steel shaft is cooled to shrink it, ensuring that the ceramic inner ring and the rotating shaft can be assembled without damage to improve the precision of mutual cooperation. The steel shaft-ceramic bearing rotor system is shown in Figure 2.

**Experimental equipment**

In order to verify the accuracy of the dynamic model in the temperature range of the bearing-rotor system, relevant tests were carried out on ABLT-9 bearing life strengthening testing machine in a factory. As shown in Figure 3. The testing machine is mainly composed of bearing-rotor, bearing seat, transmission system, loading system, lubrication system, electrical control system, computer monitoring system and other parts. The bearing-rotor is installed in the test head seat, the transmission system is transmitted by the motorized spindle, and the operation interface of the computer can control the rotation speed through the encoder. The loading system can provide the load required for the test. The lubrication system makes the test bearing fully lubricated under normal conditions for testing. The charge sensitivity of YD-1 6064 piezoelectric sensor is $6\times10^6$ $\text{PC} = \text{ms}^{-1}$, the frequency response is 1–1000 $(\text{Hz} \pm \text{dB})$, the working temperature is $-40^\circ \text{C} + 80^\circ \text{C}$, and the maximum measurable acceleration is $2000 \text{m/s}^2$. It has the advantages of wide working frequency, small size, lightweight, long service life, easy installation and not easy to damage, etc. It is one of the widely used sensors in the field of vibration test. The lead thermocouple PT100 temperature sensor has a response time of 1 s, stability, $R_0 \leq 0.04\%$, temperature range: $-20^\circ \text{C} - 500^\circ \text{C}$, anti-vibration grade: 10~2000 Hz, and adopts stainless steel shell, thus achieving the characteristics of good stability, high accuracy and strong seismic performance. The computer records the temperature and vibration information of bearing outer ring in real time through temperature sensor and vibration sensor, and monitors the operation of the machine.

**Experimental procedure**

The experiment was carried out in a dust-free laboratory with a temperature of 297.65 K $\pm$ 2 K and humidity of $\approx 60\%$. The geometric parameters of 6206 full ceramic silicon nitride deep groove ball bearing are shown in Table 1. Table 2 shows test performance related parameters. The encoder is controlled by the computer to set the speed. The speeds used in this article are 6000, 7000, 8000, 9000, and 10,000 rpm, the axial load is 200 N, acting as a preload, the radial loads are 1000, 1700, 2400, 3200, and 3900 N, and the lubrication conditions were oil-lubricated and non-lubricated. In order to investigate the influence of the temperature rise of the bearing-rotor system on the vibration of the bearing-rotor system, the bearing-rotor system runs for a period of time at 6000–10,000 rpm respectively. When the vibration is stable, take the root mean square and temperature values of the vibration for 10 min, and continue to run for a period of time. When the temperature of the bearing-rotor system reaches stability,
take the root mean square and temperature values of the vibration for 10 min. Change the radial load of the bearing and repeat the process of the above experiment.

Theoretical model

Calculation of radial clearance of ceramic bearing

The assembly, working temperature and rotating speed of bearing have great influence on the radial clearance of bearing. Therefore, it is necessary to analyze the change of radial clearance. The room temperature is $T_a$, the bearing temperature is $T_b$, and the steel shaft temperature is $T_s$. the radial interference $u_{ia}$ between the inner ring of the bearing and the rigid bearing due to thermal expansion can be obtained.

$$u_{ia} = d((T_s - T_a)a_s - (T_b - T_a)a_b)$$  (1)

Where, $a_s$ and $a_b$ is the linear expansion coefficient of the steel shaft and silicon nitride material, and the unit is $\text{mm/mm}^\circ\text{C}$, the inner diameter of ceramic bearing inner ring is $d$.

The outer diameter of ceramic bearing outer ring is $d_o$. The temperature of the outer ring is higher than the ambient temperature $T_a - T_o$. Then the expansion $u_{io}$ around the outer ring of the bearing is:

$$u_{io} = a_o d_o (T_o - T_a)$$  (2)

Table 1. Structure parameters of 6206 bearing.

| Parameters               | Values |
|--------------------------|--------|
| Outer diameter $D$ /mm   | 62     |
| Inner diameter $d$ /mm   | 30     |
| Bearing width $B$ /mm    | 16     |
| Number of rolling elements $z$ | 9     |
| Diameter of ball $Dw$ /mm | 9.525 |

Table 2. Test performance related parameters.

| Parameters                          | Values       |
|-------------------------------------|--------------|
| Density of silicon nitride (kg/m³)  | 3240         |
| Density of steel (kg/m³)            | 7820         |
| Elastic modulus of silicon nitride  | $3.17 \times 10^{11}$ |
| Elastic modulus of steel (Pa)       | $(2-2.117) \times 10^{11}$ |
| Linear expansion coefficient of      | $3.28 \times 10^{-6}$ |
| silicon nitride ($C^{-1}$)          |              |
| Coefficient of linear expansion of   | $(11.9-12.0) \times 10^{-6}$ |
| steel ($C^{-1}$)                    |              |

Influence of lubricating oil on bearing temperature rise

When the bearing runs at high speed, the heat of the bearing mainly comes from the friction between the friction pairs, and is also affected by the viscosity, density and surface tension of the lubricating oil, resulting in a certain friction torque, which directly affects the calorific value of the bearing. Therefore, the calorific value of the deep groove ball bearing is calculated by the Palmgren empirical formula method.

Calculation of the overall friction torque:

$$M = M_0 + M_1$$  (3)

In the formula, $M_0$ represents the relative torque generated by the bearing type, speed and lubricating oil characteristics, the unit is $N\cdot mm$; $M_1$ represents the friction torque generated by the rolling bearing load, the unit is $N\cdot mm$.

Calculation of $M_0$:

$M_0$ is the power consumption produced under the action of lubricating oil, and its expression is following:

$$M_0 = \begin{cases} 10^{-7} f_0(\eta n)^{2/3} D_w^3 & \eta \geq 2000 \\ 160 \times 10^{-7} f_0 D_w^3 & \eta < 2000 \end{cases}$$  (4)

Where $D_w$ is the average diameter of the bearing, and the unit is $\text{mm}$; $f_0$ is the correlation coefficient of different bearing types and different lubrication modes; $n$ is the rotating speed of bearing, and the unit is $r/\text{min}$; $\eta$ is the kinematic viscosity of lubricating oil at bearing working temperature, and the unit is $\text{mm}^2/\text{s}$. The selection rule of $f_0$ is that the light series bearings take the smaller value and the heavy series bearings take the larger value.

Calculation of $M_1$:

$M_1$ is the friction loss due to elastic hysteresis and local differential sliding and its expression is following:

$$M_1 = f_1 P_1 D_w$$  (5)

Where $f_1$ is the correlation coefficient of different bearing types and load sizes, $P_1$ is the calculated load of bearing friction torque (i.e. equivalent dynamic load of bearing), and the unit is $N$. 
Calorific value calculation of bearing:

\[ H = \frac{\pi}{30} Mn \]  

(6)

Where \( H \) is the total heating amount of rolling bearings, and the unit is \( \text{W} \).

Analysis of bearing force of ceramic shaft

As shown in Figure 4(a), Lim and Singh\(^3\) put forward a detailed load-displacement relationship, on the basis of which the influence of steel shaft expansion on the bearing structure is added. During the installation of the deep groove ball bearing, the inner ring of the ceramic bearing is connected with the steel shaft, and the outer ring of the bearing is fixed in the bearing seat, so the bearing force of the shaft is expressed as:

\[ F = \frac{1}{2} F_x F_y F_z / C_{138} T \]

(7)

The azimuth angle of the j-th bearing ball can be expressed as:

\[ \psi_j = \frac{2\pi}{N} (j - 1) + \psi_0 \]  

(10)

Where \( \psi_0 \) is the angle between the first rolling body and the x axis, \( N \) is the number of rolling elements, \( A_\theta \) and \( A_j \) are the distance between the center of curvature of the inner and outer ring raceways of the bearing when the bearing is not loaded and when the bearing is loaded, \( d_1 \) is the total contact deformation between the rolling body and the raceway, \( a_0 \) is the contact angle without load. According to the displacement direction of the inner ring, the radial displacement \( d_r \) and axial displacement \( d_z \) of the j-th rolling body:

\[ d_r = \delta_x \cos \psi_j + \delta_y \sin \psi_j - u_r \]

(11)

\[ d_z = \delta_z + (R_i + u_e)(\beta_x \cos \psi_j - \beta_y \cos \psi_j) \]

(12)

Where: \( R_i \) is the radial distance from the curvature center \( O_i \) of the ball bearing inner raceway to the bearing center \( O \), \( u_r \) is radial clearance of bearing, \( \delta_x, \delta_y, \delta_z \) denote the relative displacement of the inner ring along the x, y, and z directions, respectively, \( \beta_x \) and \( \beta_y \) denote angular displacements in the x and y directions, respectively.
According to Hertz contact theory,\textsuperscript{32} the contact force between j-th bearing ball and raceway can be expressed:

\[ Q_j = K_n \delta_j^p \]  

(13)

Where: \( Q_j \) is the normal contact load on the rolling body, \( K_n \) is the effective stiffness coefficient of inner raceway-ball-outer raceway, \( n \) is an exponent, Point-to-point contact: \( n = 1.5 \), the effects of centrifugal force and gyroscopic moment on ball bearings are ignored because these effects are considered only at extremely high rotational speeds. The components of the normal load \( Q_j \) in the j-th sphere along r and z in the negative direction are \( Q_j \cos \alpha_j \) and \( Q_j \sin \alpha_j \), respectively. As shown in Figure 4(c), the components of \( Q_j \cos \alpha_j \) in the negative directions along the x-axis and the y-axis are \( Q_j \cos \alpha_j \cos \psi_j \) and \( Q_j \cos \alpha_j \sin \psi_j \), respectively. The equilibrium equation can be obtained:

\[
\begin{align*}
\begin{bmatrix}
F_x \\
F_y \\
F_z \\
M_x \\
M_y
\end{bmatrix}
&= \sum_{j=1}^{N} \begin{bmatrix}
Q_j \cos \alpha_j \cos \psi_j \\
Q_j \cos \alpha_j \sin \psi_j \\
Q_j \sin \alpha_j \\
R_j \sin \alpha_j \cos \psi_j \\
-R_j \sin \alpha_j \sin \psi_j
\end{bmatrix} 
\end{align*}
\]  

(14)

Where \( \alpha_j \) is the contact angle under loading, as can be seen from Figure 4(c).

\[
\begin{align*}
\cos \alpha_j &= \frac{A_0 \cos \alpha_0 + \delta_{\alpha_j} + u_i - u_{i0}}{A_1} \\
\sin \alpha_j &= \frac{A_0 \sin \alpha_0 + \delta_{\alpha_j}}{A_1}
\end{align*}
\]  

(15)

At the same time, according to the formula derived above, the resultant force of all bearing balls on the inner ring of deep groove ball bearing can be expressed as:

\[
\begin{align*}
\begin{bmatrix}
F_x \\
F_y \\
F_z \\
M_x \\
M_y
\end{bmatrix}
&= \sum_{j=1}^{N} K_n \begin{bmatrix}
\left( (A_0 \sin \alpha_0 + \delta_{\alpha_j})^2 + (A_0 \cos \alpha_0 + \delta_{\alpha_j} + u_i - u_{i0})^2 - A_0 \right) \cos \alpha_j + \delta_{\alpha_j} + u_i - u_{i0} \\
\left( (A_0 \sin \alpha_0 + \delta_{\alpha_j})^2 + (A_0 \cos \alpha_0 + \delta_{\alpha_j})^2 \right) \cos \psi_j \\
R_j \left( A_0 \sin \alpha_0 + \delta_{\alpha_j} \right) \sin \alpha_j + \delta_{\alpha_j} + u_i - u_{i0} \\
-R_j \left( A_0 \sin \alpha_0 + \delta_{\alpha_j} \right) \cos \alpha_j \sin \psi_j \\
R_j \left( A_0 \sin \alpha_0 + \delta_{\alpha_j} \right) \cos \psi_j
\end{bmatrix}^{1.5}
\end{align*}
\]  

(16)

Experimental results

Influence of rotational speed on vibration of bearing-rotor system. As shown in Figure 5, the vibration of the bearing-rotor system in no-load condition without lubrication 10 min before the start is recorded. It can be seen from Figure 5 that the vibration of the bearing-rotor system increases with the increase of the rotating speed under the condition of no load and no lubrication. This is because of the existence of bearing radial clearance, bearing in high-speed rotation, the centrifugal force of the rolling body becomes larger, resulting in the impact of the bearing outer ring increases. As can be seen from Figure 6, under no-load condition without lubrication, the time for the temperature rise of bearing-rotor system to reach the temperature balance of bearing-rotor system with the increase of rotating speed is shortened, and the final temperature is maintained at about 33°C. The temperature for 9000 or 10,000 rpm seems to be still increasing volatility. Because a large amount of heat generated by bearing friction will be transferred to the bearing seat through heat conduction and heat convection, and then the bearing seat will dissipate heat to the outside world through heat convection. Heat is continuously
transferred between solids, between solids, and between solids and fluids. Finally, reach the equilibrium state, So the temperature balance is not absolutely stable, but relatively fluctuating. The heat-induced load and friction heat generation interact with each other during the operation of ball bearings. Under the action of radial load and rotating speed, the bearing produces friction heat generation, which makes the high bearing temperature affect the thermal-induced load of the bearing, and the thermal-induced load affects the friction heat generation produced by the bearing, which leads to the further increase of bearing temperature until the time when the bearing temperature tends to be stable is shortened with the increase of time. During this period, the inner and outer rings of ceramic bearings also become larger with the increase of temperature, but the increase is very small and almost negligible. In theory it is calculated that the maximum expansion of the steel shaft is 0.008 mm and the maximum extrusion stress is 76.7606 N. When the bearing rotor system reaches temperature stability, it can be seen from Figures 5 and 7 that when the rotational speed is 6000 and 1000 rpm, the vibration of bearing rotor system increases with the increase of time. After a period of time, through the comparison between Figures 5 and 7, it can be seen that the corresponding vibration after temperature rise in each time period is less than that before temperature rise. The root mean square of vibration of the 6000 rpm bearing rotor system drops from 3.5 to 3.2 m s$^{-2}$, that of the 7000 rpm bearing rotor system drops from 4.5 to 3.8 m s$^{-2}$. The root mean square vibration of 8000 rpm bearing-rotor system decreased from 5.5 to 4.8 m s$^{-2}$. The root mean square vibration of 9000 rpm bearing-rotor system decreased from 6 to 5.2 m s$^{-2}$. The root mean square vibration of 10,000 rpm bearing-rotor system decreased from 7.2 to 5.8 m s$^{-2}$. From this, it can be concluded that the root mean square of the vibration of the bearing-rotor system from 6000 to 10,000 rpm shows a downward trend under no-load and no-lubrication conditions, and the average drop is about 10%. As shown in Figure 8, the vibration of the bearing-rotor system under oil lubrication at no load 10 min before the start of recording. It can be seen from Figure 8 that the vibration of ceramic bearing-rotor system under oil lubrication also increases with the increase of rotating speed, but the vibration of bearing-rotor system under no-load oil lubrication is reduced to
one-third of that under no-load non-lubrication condition, and the collision and friction between rolling body and inner and outer rings of bearing are reduced due to the existence of oil film. As shown in Figure 9, the temperature of the inner ring of the bearing is about 30°C after 60 h. At this time, in theory the maximum expansion of the steel shaft is limited, the extrusion force on the inner ring of the bearing is also small, and the radial clearance of the bearing changes little, the vibration reduction of the ceramic bearing rotor system under the condition of no-load oil lubrication is relatively small.

**Influence of radial load on vibration of bearing-rotor system.** To explore the influence of radial load on the vibration of the bearing-rotor system, relevant tests were carried out under the condition of oil lubrication. The rotational speeds were set at 6000, 7000, 8000, 9000, and 10,000 rpm, and the radial loads were applied at 1000, 1700, 2400, 3200, and 3900 N, respectively. Observe the root mean square vibration and temperature rise of the bearing-rotor system. As shown in Figures 11 to 15, the vibration of the bearing-rotor system at the same speed gradually increases as the radial load increases. When the radial load is 1000 and 1700 N, the speed ranges from 6000 to 10,000 rpm, and the root mean square of the bearing vibration is basically stable at about 6–8 m s⁻², indicating that the two radial loads of 1000 and 1700 N belong to the light load range of the bearing, so the sensitivity of ceramic bearing to the changes of the two radial loads is not high. Bearing stiffness changes little, but when the speed reaches 10,000 rpm, bearing vibration increases significantly, because it is approaching the critical speed of ceramic bearing-rotor system. When the radial load is 2400, 3200, and 3900 N, the vibration of the bearing obviously increases with the speed from 6000 to 10,000 rpm. Because 2400 to 3200 N belong to the medium load range of the bearing, the stiffness of the ceramic bearing changes significantly, so the vibration

---

*Figure 8.* Vibration of bearing-rotor system under oil lubrication condition.

*Figure 9.* Temperature rise of bearing rotor system at different speeds under oil lubrication.

*Figure 10.* Vibration of bearing rotor system under oil lubrication when the temperature reaches equilibrium.
of the bearing changes greatly. 3900 N belongs to the heavy load range of ceramic bearings. The vibration of the bearing at 10,000 rpm is maintained at $18 \text{ m s}^{-2}$ which is about 1.5 times the vibration of the bearing at 6000 rpm. Mainly because silicon nitride has a small coefficient of thermal expansion, most of the heat generation of ball bearings comes from the spin of rolling body and inner ring raceway. When the rotating speed of the bearing is 6000 rpm, the spin ratio of the ball is small, and the heat generated by rolling friction between the rolling body and the inside and outside of the bearing is small. The flowing lubricating oil will take away most of the heat and cool the bearing at the same time, so the temperature rise of the bearing is relatively low in the first 20h in Figure 16. As shown in Figures 16 to 20, at the same speed, the time for the temperature rise of the bearing rotor system to reach stability gradually decreases with the increase of radial load. Because the larger the radial load, the larger the contact area between the rolling elements and the inner and outer rings of the bearing, and the more heat generated by the friction of the bearing. The maximum temperature is maintained at around 55°C. At this time, in theory the maximum expansion of the steel shaft is calculated to be 0.013 mm, and the maximum tensile stress generated is 142.5554 N. By comparing Figures 11 and 21, Figures 12 and 22, Figures 13 and 23, Figures 14 and 24, and Figures 15 and 25, it can be

---

**Figure 11.** Vibration values of bearing-rotor system with different radial loads under 6000 rpm oil lubrication.

**Figure 13.** Vibration values of bearing-rotor system with different radial loads under 8000 rpm oil lubrication.

**Figure 12.** Vibration values of bearing-rotor system with different radial loads under 7000 rpm oil lubrication.

**Figure 14.** Vibration values of bearing-rotor system with different radial loads under 9000 rpm oil lubrication.
seen that the mean square value of the vibration of the bearing rotor system is significantly smaller than that of the bearing rotor system before the temperature rise. The reason for this phenomenon is that the heat expansion of the steel shaft increases the pressure on the inner ring of the bearing, making the radial clearance of the bearing smaller, and the impact of the rolling body on the inner ring of the bearing smaller.

**Simulation results**

As shown in Figure 26, the rotating speed is 6000 rpm and the radial load is 1000 N, and the trajectory of the steel shaft-ceramic bearing rotor center is not easy to be deformed in the light load range due to the large stiffness of the ceramic bearing, so the trajectory of the shaft center is approximately circular, thus essentially explaining the stability of the ceramic bearing in the light load range. As shown in Figure 27, the rotating speed is 6000 rpm and the radial load is 2400 N. The track diagram of steel shaft ceramic bearing rotor axis shows that the ceramic bearing has obvious deformation within the medium load range, and the motion track of the axis is approximately an ellipse. Therefore, the deformation of ceramic bearing is greatly affected by the load within the medium load range. The reason why the ceramic bearing rotor increases with the increase of load in the medium load range is also expounded theoretically. As shown in Figure 28, the rotating speed is 6000 rpm, the radial load is 3900 N,
the steel shaft ceramic bearing rotor axis trajectory, the ceramic bearing deformation is more obvious in the heavy load range, and the movement trajectory of the axis changes more obviously compared with the medium load, which is similar to a flat ellipse, so the deformation of the ceramic bearing is more affected by the load in the heavy load range, The vibration of bearing rotor system is more intense. Comparing Figure 26 with Figure 27 and Figure 28, the drop of the axis rotation trajectory under light load is more obvious than that under medium load and heavy load, because the elastic modulus of silicon nitride is large, the deformation of bearing is small under light load, and the clearance of bearing is almost unchanged with the initial clearance after load, the axial trajectory of bearing under gravity of steel shaft is more obvious than that under medium load and heavy load.

Figure 19. Temperature rise of bearing-rotor system with different radial loads under 9000 rpm oil lubrication.

Figure 20. Temperature rise of bearing-rotor system with different radial loads under 10,000 rpm oil lubrication.

Figure 21. Vibration diagram after temperature rise of different radial load bearing rotor system under 6000 rpm oil lubrication.

Figure 22. Vibration diagram after temperature rise of different radial load bearing rotor system under 7000 rpm oil lubrication.
Conclusion

In this study, control variables were used to conduct relevant tests on the rotor system composed of Si3N4 full-ceramic ball bearings and steel shafts. Test analysis and theoretical model analysis were conducted by changing different speeds and radial loads, and the following conclusions were obtained:

1. In the rotor system of steel shaft full ceramic deep groove ball bearing, the heat generated by the
A high-speed rotating bearing leads to the expansion of the steel shaft and has a great impact on the vibration of the bearing rotor system. Considering the dynamic model of the extrusion force of the thermal expansion of the steel shaft on the inner ring of ceramic bearing, the calculated results are close to the experimental data.

2. Bearing—The vibration and temperature rise of the rotor system increase with the increase of speed and radial load. Among them, the order of the greatest influence on the deformation of the ceramic bearing is: heavy load, medium load, and light load.

3. Appropriately reducing the tolerance of interference fit between steel shaft and all-ceramic silicon nitride ball bearings can improve the service life of ceramic bearings under high speed and heavy load and reduce the wear of ceramic bearings.

Author contributions
The manuscript was written through contributions of all authors. All authors have given approval to the final version of manuscript.

Declaration of conflicting interests
The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding
The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This project is supported by National Natural Science Foundation of China (Grant no. 52005352, 51975388), and Key Laboratory of Vibration and Control of Aero-Propulsion System, Ministry of Education, Northeastern University (VCAME202007), and Open Fund of Key Laboratory of Fundamental Science for National Defense of Aeronautical Digital Manufacturing Process of Shenyang Aerospace University (SHYSYS202107).

ORCID iDs
Jiancheng Guo https://orcid.org/0000-0002-8347-5946
Xu Bai https://orcid.org/0000-0002-0746-2043

References
1. Dong Y, Zhou Z and Liu M. Bearing preload optimization for machine tool spindle by the influencing multiple parameters on the bearing performance. Adv Mech Eng 2017; 9: 1687814016689040.
2. Wu Y, Yan H, Li S, et al. Calculation on the radiation noise of ceramic ball bearings based on dynamic model considering nonlinear contact stiffness and damping. J Sound Vib 2020; 479: 115374.
3. Shi H, Li Y, Bai X, et al. Investigation of the orbit spinning behaviors of the outer ring in a full ceramic ball bearing-steel pedestal system in wide temperature ranges. Mech Syst Signal Process 2021; 149: 107317.
4. Burgmeier L and Poursaba M. Ceramic hybrid bearings in air-cycle machines. J Eng Gas Turbine Power 1996; 118: 184–190.
5. Shi HT, Bai XT, Zhang K, et al. Influence of uneven loading condition on the sound radiation of starved lubricated full ceramic ball bearings. J Sound Vib 2019; 461: 114910.
6. Zhang L, Wei X, Wang J, et al. Experimental study on the lubrication and cooling effect of graphene in base oil for Si3N4/Si3N4 Sliding Pairs. Micromachines 2020; 11: 160.
7. Zaretsky EV, Chiu YP and Tallian TE. Ceramic bearings for use in gas turbine engines. J Mater Eng 1989; 11: 237–253.
8. Li M, Wang Y, Chen W, et al. Temperature rise characteristics for angular-contact ball bearings with oil-air lubrication based on fluid-solid conjugate heat transfer. Adv Mech Eng 2021; 13: 1687814021990927.
9. Sopanen J and Mikkola A. Dynamic model of a deep-groove ball bearing including localized and distributed defects. Part 2: Implementation and results. Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics 2003; 217: 213–223.
10. Lioulios AN and Antoniadis IA. Effect of rotational speed fluctuations on the dynamic behaviour of rolling element bearings with radial clearances. Int J Mech Sci 2006; 48: 809–829.
11. Cao H, Niu L, Xi S, et al. Mechanical model development of rolling bearing-rotor systems: a review. Mech Syst Signal Process 2018; 102: 37–58.
12. Kanematsu W. A review of rolling contact fatigue behavior of silicon nitride focusing on testing practices and crack propagation analysis. Wear 2018; 400–401: 10–20.
13. Yunlong W, Wenzhong W, Shengguang Z, et al. Effects of raceway surface roughness in an angular contact ball bearing. Mech Mach Theory 2018; 121: 198–212.
14. Zhang X, Han Q, Peng Z, et al. A new nonlinear dynamic model of the rotor-bearing system considering preload...
and varying contact angle of the bearing. Commun Nonlinear Sci Numer Simul 2015; 22: 821–841.
15. Yang W, Wang X, Li H, et al. A novel tribometer for the measurement of friction torque in microball bearings. Tribol Int 2017; 114: 402–408.
16. Petersen D, Howard C, Sawalhi N, et al. Analysis of bearing stiffness variations, contact forces and vibrations in radially loaded double row rolling element bearings with raceway defects. Mech Syst Signal Process 2015; 50–51: 139–160.
17. Wang Y, Cao J, Tong Q, et al. Study on the thermal performance and temperature distribution of ball bearings in the traction motor of a high-speed EMU. Appl Sci 2020; 10: 4373.
18. Sheng X, Li B, Wu Z, et al. Calculation of ball bearing speed-varying stiffness. Mech Mach Theory 2014; 81: 166–180.
19. Mao Y, Wang L and Zhang C. Influence of ring deformation on the dynamic characteristics of a roller bearing in clearance fit with housing. Int J Mech Sci 2018; 138–139: 122–130.
20. Xu L and Li Y. An approach for calculating the dynamic load of deep groove ball bearing joints in planar multi-body systems. Nonlinear Dyn 2012; 70: 2145–2161.
21. Bovet C and Zamponi L. An approach for predicting the internal behaviour of ball bearings under high moment load. Mech Mach Theory 2016; 101: 1–22.
22. Yang ZZ, Wu JG, Qin B, et al. ADAMS dynamics simulating and analysis of vibration signal for deep-groove ball bearings. Appl Mech Mater 2013; 312: 254–257.
23. Razpotnik M, Čepon G and Boltežar M. A smooth contact-state transition in a dynamic model of rolling-element bearings. J Sound Vib 2018; 430: 196–213.
24. Pratiwi MA, Ikhsan M, Octavianto RD, et al. Dynamic characterization of ball bearing in turbine propeller using Bump Test Method. S/NERGI 2021; 25: 135–140.
25. Wang H, Liu Z, Zou L, et al. Influence of both friction and wear on the vibration of marine water lubricated rubber bearing. Wear 2017; 376–377: 920–930.
26. Shah DS and Patel VN. A dynamic model for vibration studies of dry and lubricated deep groove ball bearings considering local defects on races. Measurement 2019; 137: 535–555.
27. Liu J, Ma C, Wang S, et al. Thermal-structure interaction characteristics of a high-speed spindle-bearing system. Int J Mach Tools Manuf 2019; 137: 42–57.
28. Bizarre L, Nonato F and Cavalca KL. Formulation of five degrees of freedom ball bearing model accounting for the nonlinear stiffness and damping of elastohydrodynamic point contacts. Mech Mach Theory 2018; 124: 179–196.
29. Ma C, Liu J and Wang S. Thermal contact conductance modeling of barring outer ring/bearing housing interface. Int J Heat Mass Transf 2020; 150: 119301.
30. Chen G and Qu M. Modeling and analysis of fit clearance between rolling bearing outer ring and housing. J Sound Vib 2019; 438: 419–440.
31. Lim TC and Singh R. A review of gear housing dynamics and acoustics literature. Washington, DC: National Aeronautics and Space Administration, 1989.
32. Harris TA and Mindel MH. Rolling element bearing dynamics. Wear 1973; 23: 311–337.