Research Article

Experimental Research on the Influence of Working Conditions on Vibration and Temperature Rise of Si_3N_4 Full-Ceramic Bearing Motors

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Working conditions such as lubrication, preload, and rotational speed have important influence on vibration and temperature rise of the spindle motor. In this study, controlled variable experiments are carried out on the silicon nitride (Si_3N_4) full-ceramic ball bearing and steel bearing of the same type, and the vibration signal characteristics and temperature rise of the spindle motor are tested and analysed, by changing the lubrication conditions, preloads, and rotational speeds of the spindle motor. Through the research, it is found that as the rotational speed increases, the vibration velocity of the Si_3N_4 full-ceramic bearing spindle motor under different preloads and lubrication conditions shows an overall increasing trend; kurtosis generally presents a downward trend and gradually flattens, indicating that although the vibration velocity increases at high speeds, the vibration signal shows a relatively stable state. As the rotational speed increases, the difference of vibration velocity under the condition of applying preload and no preload decreases, indicating that the influence of preload on the vibration of full-ceramic bearing spindle motor decreases with the increase in rotational speeds. At the same time, it is found that f_r and $5f_r$ have greater impact on the vibration of full-ceramic bearing spindle motor, where f_r is the frequency of the bearing in normal operation, and $5f_r$ is 5 times of the normal operating frequency. Lubrication conditions have little effect on the temperature rise of full-ceramic bearing spindle motor, and the temperature rise under nonlubricated conditions is even slightly lower than that under grease lubrication conditions. The research results show that the vibration velocity and temperature rise of Si_3N_4 full-ceramic bearing spindle motor are less than those of steel bearing with the same type, indicating that full-ceramic bearing has better performance than steel bearing under the same working conditions.

1. Introduction

As the main source of power, spindle motors are widely used in CNC machine tools, wind turbines, high-speed railway, and new energy automobiles and other fields [1–3], the performance and service life of spindle motors directly determine the service performance and life of mechanical equipment. The common sources of faults in spindle motors can broadly be classified as the followings [4, 5]: bearing faults, stator or armature faults, broken rotor bar and end-ring faults, and eccentricity-related faults. Among all types of

faults, bearing is one of the most prevalent factors, almost 40%–50% of all motor failures are bearing related, such as fatigue failures, spalling, crack propagation, friction and wear, electric corrosion, and so on. As an important index to evaluate the performance of spindle motor, rise in vibration and temperature affects the performance and life span of the spindle motor directly [6, 7]. There are many factors that cause the vibration and temperature rise of the spindle motor, and the reasons are also very complicated. For example, the vibration is mainly caused by rotor imbalance, uneven air gap, and abnormal bearings; temperature rise is

mainly caused by winding iron core heating, poor bearing lubrication, bearings wear, and other reasons; among which, the vibration and temperature rise caused by the bearing accounts for a large proportion [8, 9]. Excessive vibration and high temperature rise of the spindle motor not only will produce loud noises and aggravate bearing wear but also may cause damage to the insulation layer or even burn the motor, which will affect the performance and reduce the service life of the spindle motor. Therefore, it is particularly important to control the vibration and temperature rise of the spindle motor.

The main factors affecting the service performance of rolling bearings include the following: bearing materials, structure, manufacturing accuracy, preload, working temperature, and lubrication conditions. Hot isostatic pressed silicon nitride (HIPSIN) is considered to be the most suitable engineering ceramic material for bearings due to its low density, wear resistance, high rigidity, low expansion coefficient, self-lubricating, electrical insulation, and other excellent characteristics [10], and it has become a new direction for modern bearings and has become more and more widely used. There are many reasons for vibration and temperature rise in ceramic bearings during operation, such as frequent changes of rotational speeds, loads, external environment and other factors, friction and wear between the internal components of ceramic bearings, fatigue spalling and crack propagation of ceramic rings and rolling elements, cage fracture [11], existence of bearing clearance, viscosity and friction resistance of lubricating oil [12], and so on. Applying proper axial preloads [8] to the bearing can not only reduce the axial clearance of the bearing but also improve the stiffness of the shaft system to reduce vibration and noise [13]. Suitable lubrication [14, 15] can ensure effective lubrication of spindle motor bearings and reduce bearing wear and the generation of heat by friction, improving the performance and life of bearings. Therefore, it is of great significance to carry out researches on spindle motor vibration and temperature rise, starting from the bearing material, preload, and lubrication conditions.

Researchers have made many contributions in the fields of shaft vibration and motor temperature rise: Wang et al. [11] studied the thermal performance and temperature distribution of ball bearings in the traction motor, the influence of rotational speed and applied load variation on bearing temperature, and the temperature distribution law of each component in grease lubricated bearing are analysed. Jamadar and Vakharia [12] studied the effect of grease viscosity on mechanical vibrations associated with the damaged bearings, and the results show that when the grease viscosity is higher, the amplitude of the bearing presents a lower level, so the service performance of the bearing is improved. He et al. [16] studied the temperature fields of the air-cooled electric spindle by numerical methods, and they also experimentally studied the influence of wind speed on the temperature of the electric spindle and the relationship between rotational speed, vibration, and noise and found that the temperature, vibration, and noise increased with the rotational speed. The research results provide a basis for the optimal design of the air-cooled electric spindle. Kosaka et al. [17] analysed and

calculated the temperature fields of the disc permanent magnet motor using the equivalent thermal network method, and the temperature field distribution of each part of the motor was obtained. Liu and Zhang [18] considered the thermal-mechanical coupling factors and established a more accurate thermomechanical coupling dynamic model of the high-speed electric spindle; they quantitatively discussed the dynamic behavior of spindle system and the influence of thermal displacement on the dynamic behavior of spindle system. Zhang and Chen [19] established a thermal engine coupling model of spindle system, which were composed of three coupling submodels of bearing, thermal engine, and motorized spindle dynamics; they analysed the influence of thermal expansion of motorized spindle bearing on the dynamic performance of motorized spindle, and through the research, the effects of rotational speed, cooling water flow, and oil-gas pressure on the dynamic characteristics of spindle were noted. Li et al. [20] used the online adjustment test platform of electric spindle preload to study the influence of air supply pressure and rotational speed on the outer ring temperature rise of the motorized spindle front bearing. The research results show that the steady-state temperature rise of the high-speed motorized spindle increased with the increase in rotational speed and air supply pressure. Through the online control of the bearing preload, the active control of the thermal characteristics of the motorized spindle can be realized. Zhang et al. [21] took the optimized heat transfer coefficient as the boundary condition to calculate the internal temperature fields and thermal deformation of the high-speed spindle, established a high-speed motorized spindle thermal deformation prediction model, and verified the prediction accuracy. Other aspects, such as the relationship between temperature of the internal components of the bearing and friction heat generation power [22–24], influence of changes in bearing structural parameters, contact thermal resistance and bearing temperature on the nonlinear thermal characteristics [25], prediction of bearing temperature fields distribution through the thermal-mechanical coupling model [26], influence of thermal related fit clearance between the bearing outer ring and pedestal on the vibration characteristics of full-ceramic bearings [27], have also been extensively studied, and effective conclusions have been drawn.

It can be seen from the above that researchers have conducted in depth researches on shaft vibration and temperature rise and have obtained many valuable results. However, most of the present studies focus on traditional steel bearings and relatively few involve full-ceramic bearings. Compared with traditional steel bearings, ceramic bearings are more suitable for occasions with high speed, heavy load, and electric corrosion, such as spindle motors, because of their low density, wear resistance, high stiffness, low expansion coefficient, and electrical insulation, making it a new direction of rolling bearing research. Although when the difference between the characteristics of new ceramic materials and traditional metal materials are considered, various macro- and microfactors in ceramic bearing assembly and multi-interface system have extremely complex effects on bearing service performance, which brings many new problems, such as the multifield coupling relationship

between the rotational speed, load, and temperature fields and the elastic-plastic deformation, vibration noise, and thermal effect of ceramic bearings during operation. Therefore, in this study, the experimental method is used to explore the influence of lubrication conditions, preload, and rotational speed on the vibration and temperature rise of Si_3N_4 full-ceramic bearing and steel bearing spindle motors, to verify the application effect of full-ceramic bearing in spindle motor. This study verifies that the full-ceramic bearings have more excellent performance compared with traditional steel bearings in the aspects of reducing vibration and temperature rise. It has certain scientific significance and engineering application value to structure optimization, design, manufacture, and service performance of full-ceramic bearing for different working conditions.

2. Experimental Method

2.1. Development of Experimental Bearings. The most common processing method of Si_3N_4 and other engineering ceramic materials is grinding. Compared with other processing methods such as chemical processing and optical processing, the grinding has the advantages of simple processing technology, high processing efficiency, high dimensional accuracy, low surface roughness, and few surface and internal cracks; relatively high reliability engineering ceramic parts can be obtained through this processing method. The grinding process of full-ceramic bearings are divided into the grinding process of ceramic balls, inner ring, and outer ring. The processing route is shown in Figure 1. The structure and samples of the developed Si_3N_4 full-ceramic ball bearings are shown in Figure 2.

2.2. Assembly of Spindle Motor with Full-Ceramic Bearings. Generally, the inner ring of the bearing and the rotating shaft are interference fit. In traditional assembly process of steel bearing spindle motor, if the interference between steel bearing and rotating shaft is small, it can be assembled by direct pressing, but such installation will not only damage the matching accuracy of bearing and rotating shaft but also affect the operation accuracy of bearings, thus affecting the service performance of bearings. If the interference is large, it can be assembled by hot fitting. However, for full-ceramic bearings, due to the nonmagnetic properties and low thermal expansion coefficient of ceramic materials, the hot fitting method cannot be used. Therefore, in this study, liquid nitrogen is used to achieve ultralow temperature cooling, and the rotating shaft is cooled to make it shrunken, ensuring the clearance between the ceramic inner ring and the rotating shaft and achieving damage-free assembly to improve the assembly and running accuracy of the full-ceramic bearing spindle motor. After the bearing assembly is completed, the rotor system is dynamically balanced, and then, the spindle motor assembly is completed. In the static test of the assembled spindle motor, the radial runout of the spindle motor is in the range of $1\text{ }\mu\text{m}$ to $2\text{ }\mu\text{m}$. The assembled spindle motor is shown in Figure 3.

2.3. Experimental Devices and Procedure. The experiments are carried out in the dust-free laboratory with ambient temperature of $25\text{ }^\circ\text{C} \pm 2\text{ }^\circ\text{C}$ and humidity of $\leq 75\%$. The model of the spindle motor tested in the experiments is Z18-44P0XA30, and the performance parameters are shown in Table 1. The bearings used in the experiments are 6206 Si_3N_4 full-ceramic bearings (the materials of inner ring, outer ring, rolling element, and cage are Si_3N_4 , Si_3N_4 , and PEEK, respectively) and traditional 6206 steel bearings (the materials of inner ring, outer ring, rolling element, and cage are all bearing steel). Structure parameters and material parameters of the tested bearings are shown in Tables 2 and 3. The experimental devices and schematic diagram are shown in Figure 4.

The experimental devices and schematic diagram are shown in Figure 4. The spindle motor used in the experiments is produced by CTB Co., LTD, Beijing, China, and a frequency converter is used to control the start, stop, and rotational speed of the spindle motor. During the experiments, OFV5000 high-performance laser vibrometer and signal acquisition equipment produced by German Polytec Company are used to perform noncontact measurement of the vibration signal of the spindle motor. The vibration test point is the shaft end of the spindle motor. No direct contact is required during the experiments, and the vibration of the rotating shaft can be measured completely without distortion. At the same time, different test channels, such as vibration displacement, velocity, and acceleration can be selected according to actual needs. The frequency range is from DC to 24 MHz, and the vibration velocity is as high as $\pm 25\text{ m/s}$ and displacement resolution up to $0.1\text{ }\mu\text{m}$. The temperature rise is collected by the American FLIR infrared thermal imager, which can not only observe the real-time thermal image of the spindle motor during operation but also perform process control and prealarm through infrared video and related temperature measurement data. The Si_3N_4 full-ceramic bearings tested in the experiments are designed and manufactured by the research team, with P4 grade, and the steel bearings are SKF bearings of the same grade purchased.

In the experiments, the rotational speed of spindle motor is changed under the conditions of the controlling preload and lubrication conditions unchanged; the vibration and temperature rise experiments of Si_3N_4 full-ceramic ball bearings and steel bearings are carried out. The rotational speeds selected in the experiments are 2000 r/min–16000 r/min, with an interval of 2000 r/min, a total of 8 groups of rotational speed conditions. The preload condition refers to adding a corrugated spring washer to the shaft end of the spindle motor with a preload of 300 N; the lubricating condition refers adding an appropriate amount of grease to the bearings, and the model of grease is KLUBER NBU15. The experimental procedure is shown in Figure 5.

3. Vibration Frequency Characteristics and Heat Generation of Bearings

3.1. Frequency Characteristics of Vibration. In the bearing system, rotational frequency and its multiplication frequency are the main frequency components of the bearing

FIGURE 1: Processing route of full-ceramic bearing.

(a)

(b)

FIGURE 2: Structure and samples of full-ceramic bearings. (a) Structure of 6206 full-ceramic bearing. (b) Samples of Si_3N_4 full-ceramic bearings.

FIGURE 3: Assembly of the spindle motor.

TABLE 1: Performance parameters of the spindle motor.

Parameters	Values
Rated power (kW)	4.0
Rated voltage (V)	330
Rated current (A)	9.3
Rated frequency (Hz)	103
Maximum speed (rpm)	16000
Rated torque (N·m)	12.7
Series (P)	4
Moment of inertia (kg·m ²)	$5.4e-3$

TABLE 2: Structure parameters of 6206 bearing.

Parameters	Values
Inner diameter d (mm)	30
Outer diameter D (mm)	62
Pitch diameter d_m (mm)	46
Bearing width B (mm)	16
Diameter of ball D_w (mm)	9.525
Raceway radius of inner ring r_i (mm)	4.91
Raceway radius of outer ring r_e (mm)	5
Number of rolling elements z	9
Radial clearance u_r (μ m)	14

vibration. In addition, the vibration caused by the rolling frequency of the rolling elements in the inner and outer raceways also has a significant impact on the performance of bearings. The calculation methods of the characteristic frequency of each component of the bearing are as follows [28, 29].

As shown in Figure 2, definition

$$\begin{aligned}
 r_i &= f_i D_w, \\
 r_e &= f_e D_w, \\
 \gamma &= \frac{D_w \cos \alpha}{d_m},
 \end{aligned} \tag{1}$$

where f_i and f_e are the coefficients of raceway curvature radius of the inner ring and outer ring, respectively.

The relationship between the pitch diameter and the diameter of the inner and outer rings of the bearing is as follows:

$$\begin{aligned}
 d_m &= \frac{1}{2}(d_i + D_e), \\
 d_i &= d_m - D_w \cos \alpha, \\
 D_e &= d_m + D_w \cos \alpha.
 \end{aligned} \tag{2}$$

Assuming that the outer ring is fixed and the inner ring rotates with the rotating shaft, the linear speed of the inner ring raceway can be expressed as follows:

$$v_i = \frac{1}{2} \omega_r d_i. \tag{3}$$

Assuming that the rolling elements run in pure rolling state in the raceway of inner and outer rings, the rotational speed of the cage can be expressed as

$$v_c = \frac{1}{2} v_i = \frac{1}{4} \omega_r d_i = \frac{1}{4} \omega_r (d_m - D_w \cos \alpha). \tag{4}$$

Expressed in the form of frequency as

$$f_c = \frac{1}{2} f_r \left(1 - \frac{D_w}{d_m} \cos \alpha \right) = \frac{1}{2} f_r (1 - \gamma). \tag{5}$$

Therefore, the rotational frequency of the inner ring and outer ring relative to the cage can be expressed as

$$\begin{aligned}
 f_{ic} &= f_r - f_c = \frac{1}{2} f_r (1 + \gamma) \\
 f_{ec} &= f_c = \frac{1}{2} f_r (1 - \gamma).
 \end{aligned} \tag{6}$$

TABLE 3: Material parameters of the bearing.

Parameters	Si_3N_4	GCr15
Density (g cm^{-3})	3.2	7.85
Coefficient of thermal expansion (K^{-1})	3.5×10^{-6}	3×10^{-5}
Modulus of elasticity (GPa)	310	207
Poisson's ratio	0.26	0.30
Thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)	35	45

FIGURE 4: Experimental devices and schematic diagram: (a) Frequency converter. (b) Spindle motor. (c) Laser vibrometer. (d) Vibration data acquisition instrument. (e) Infrared thermal imager. (f) Temperature data acquisition. (g) Experimental schematic diagram.

FIGURE 5: Experimental procedure.

Assuming that the number of rolling elements is z , the rotational frequency of the rolling elements passing through the inner and outer ring raceways can be expressed as

$$f_i = \frac{Z}{2} f_r (1 + \gamma) \quad (7)$$

$$f_e = \frac{Z}{2} f_r (1 - \gamma).$$

The rotational frequency of the rolling elements can be expressed as:

$$f_b = f_{ic} \frac{d_i}{D_w} = \frac{d_m f_r}{2D_w} (1 - \gamma^2). \quad (8)$$

When the vibration signal is transformed by the fast Fourier transform, if the frequency spectrum contains the high vibration value at the above characteristic frequency, it indicates that the characteristic frequency has contributed to the vibration of the bearings.

3.2. Heat Generation inside the Bearings. Heat inside the bearing is mainly generated by friction between the rolling elements, inner ring, and outer ring of the bearing. Palmgren [30] has summarized the calculation formulas of the bearing friction torque under load and no-load conditions through a large number of experiments.

Friction torque caused by lubricant viscosity under no-load condition:

When $\nu_n < 2000$,

$$M_0 = 160 \times 10^{-7} f_0 d_m^3. \quad (9)$$

When $\nu_n \geq 2000$,

$$M_0 = 10^{-7} f_0 (\nu_n)^{2/3} d_m^3, \quad (10)$$

where f_0 is the coefficient related to bearing type and lubrication, ν is the dynamic viscosity of the lubricant, and n is the rotational speed of the bearing.

For load related friction torque M_1 ,

$$M_1 = f_1 P_1 d_m, \quad (11)$$

where f_1 refers to the coefficient related to the bearing type and load, and P_1 refers to the equivalent dynamic load of the bearing.

Therefore, the bearing friction torque can be expressed as: $M = M_0 + M_1$.

During the working process of the bearings, the rolling elements are subjected to centrifugal force, which makes the friction torque of the inner ring and outer ring different. Therefore, according to the difference of inner ring and outer ring raceway radius d_i and d_e , the total friction torques are divided into inner and outer ring friction torque components, that is, inner ring friction torque is $M_i = 0.5M \cdot D_w/d_i$, and outer ring friction torque is $M_e = 0.5M \cdot D_w/d_e$.

Under the condition of high-speed rotation, in addition to the rotation of inner ring and the revolution of the rolling elements, the rotation sliding between rolling elements and inner raceway of the bearing is also one of the main movements. The friction torque produced by rotation sliding has a great influence on the total friction torques, which should be considered in the calculation of heat generation. Therefore, the heat generated by rotation friction is added to above formula, so as to make the empirical formula more accurate.

$$M_s = \frac{3\mu Q a E_{(\eta)}}{8}, \quad (12)$$

where M_s is the rotation friction torque of the rolling elements, a is the major semiaxis of the contact ellipse, μ is the friction coefficient between the rolling elements, inner ring raceway, and outer ring raceway, Q is the normal contact load between the rolling elements, inner ring raceway, and outer ring raceway, and $E_{(\eta)}$ is the second-type elliptic integral function of the raceway contact area.

The heat generation in the contact area between the inner ring, outer rings, and rolling elements is related to the friction torque in the contact area and the rotational speed of the inner ring, outer rings, and rolling elements. According to the different raceway diameters of the inner ring and outer ring, the rotation friction torque M_s is also divided into friction torque M_{si} and M_{se} related to the raceway size.

Heat generated in the inner ring is given as

$$H_i = 10^{-3} W_c \cdot M_i + 1.047 \times 10^{-4} M_{si} \cdot n_m \cdot z. \quad (13)$$

Heat generated in the outer ring is given as

$$H_e = 10^{-3} W_c \cdot M_e + 1.047 \times 10^{-4} M_{se} \cdot n_m \cdot z, \quad (14)$$

where z is the number of rolling elements.

4. Results and Discussion

4.1. Influence of Working Conditions on Vibration

4.1.1. Variation of Vibration Velocity and Kurtosis. A large number of researches [31–36] have shown that, as a numerical statistic reflecting the distribution characteristics of random variables, kurtosis is particularly sensitive to shock signals, and it can well reflect the large pulse signal generated by bearing vibration, especially suitable for early fault diagnosis of bearings. The vibration velocity and kurtosis of Si_3N_4 full-ceramic bearings and steel bearings spindle motors under different working conditions are shown in Figures 6–8, respectively. Of which, the kurtosis index K is a numerical statistic reflecting the distribution characteristics of random variables, and it is a fourth-order cumulant. The calculation formula is given as follows:

$$K = \frac{\int_{-\infty}^{+\infty} [x(t) - \bar{x}]^4 p(x) dx}{\sigma^4}, \quad (15)$$

where $x(t)$ is the instantaneous amplitude, $\bar{x}$ is the mean amplitude, $p(x)$ is probability density, and σ is standard deviation.

As shown in Figure 6(a), under the condition of grease lubrication without preload, the vibration velocity of the spindle motor shows a continuous upward trend with the increase in rotational speed. Under the condition of preload applied with grease lubrication, the vibration velocity shows a trend of first rising, then falling, and then rising with the increase in rotational speed. When the rotational speed is not higher than 8000 r/min, the vibration velocity continues to rise. Within the range of 8000 r/min~10000 r/min, the vibration velocity shows a downward trend. After 10000 r/min, the vibration velocity continues to increase. When the rotational speed is lower than 8000 r/min, the preload has greater influence on the vibration velocity. With the increase in the rotational speed, the rising speed of the vibration velocity with preload is higher than that without preload, indicating that the impact of preload on the vibration velocity decreases gradually. As shown in Figure 6(b), with the increase in rotational speed, the kurtosis of the spindle motor with and without preload present a trend of first decline and then stable; the decline rates are 94.2% and 87.6%, respectively. The maximum kurtosis occurs when the rotational speed is 2000 r/min, indicating that the spindle motor has a high-amplitude vibration signal at low speed. With the increase in rotational speed, the change of kurtosis tends to be gentle after 6000 r/min, indicating that the vibration signal is relatively stable. The kurtosis of the spindle motor with preload applied at lower rotational speed is obviously greater than that with no preload, but as the rotational speed increases, the gap decreases.

Vibration velocity and kurtosis changes of Si_3N_4 full-ceramic bearing spindle motor with the increase in rotational speed under nonlubricated condition is shown in Figure 7. As shown in Figures 7(a) and 7(b), the change trend of vibration velocity and kurtosis are similar to that under grease lubrication condition because the ceramic

material has certain self-lubrication performance. However, for the vibration velocity, when the rotational speed is 8000 r/min, the vibration velocity under the condition of applying preload is even higher than that under the condition of no preload, which may be caused by the resonance because the rotational speed is close to the first-order critical speed of the spindle motor in this condition. For kurtosis, when the rotational speed is lower than 10000 r/min, the kurtosis value without preload under nonlubricated condition is significantly higher than that with applying preload; this may be due to that under the condition of no preload and nonlubrication, the axial clearance of full-ceramic bearing is bigger and looseness phenomenon occurs, resulting in the generation of vibration signals with high amplitude.

Variation of vibration velocity and kurtosis of spindle motor with steel bearings at different rotational speeds are shown in Figure 8. As shown in Figure 8(a), under the condition of grease lubrication without preload, the vibration velocity increases first and then decreases with the increase in the rotational speed. When the rotational speed is lower than 8000 r/min, the vibration velocity increases rapidly with the increase in the rotational speed. After 8000 r/min, the vibration velocity shows a slow downward trend. Under the condition of applying preload and grease lubrication, the vibration velocity presents a trend of first rising, then falling, and then rising as the rotational speed increases. When the rotational speed is not higher than 8000 r/min, the vibration velocity continues to rise. In the range of 8000 r/min~10000 r/min, the vibration velocity shows a slight downward trend. After 10000 r/min, the vibration velocity continues to rise again. After 8000 r/min, the difference of the vibration velocity under the conditions of applying preload and no preload is gradually reduced, indicating that as the rotational speed increases, the influence of the preload on the vibration velocity decreases gradually. As shown in Figure 8(b), with the increase in the rotational speed, the kurtosis shows a continuous downward trend, under both conditions of applying preload and no preload, and the rate of decrease is 83.8% and 73.5%, respectively. Compared with Si_3N_4 full-ceramic bearings, the downward trend is gentler. The maximum kurtosis occurs at 2000 r/min, indicating that the spindle motor has a high-amplitude vibration signal at low speed. When the rotational speed is 2000 r/min and 12000 r/min, the spindle motor with preload and without preload have almost the same kurtosis value, and when the rotational speed is between 2000 r/min and 12000 r/min, the kurtosis with preload is significantly greater than that without preload, but as the rotational speed increases, the gap gradually decreases. When the rotational speed is higher than 12000 r/min, the kurtosis with preload becomes lower than that without preload.

4.1.2. Effect of Frequency Multiplication. Performing fast Fourier transform on the measured vibration signals, the frequency domain components of the vibration signals are observed and analysed, as shown in Figures 9–11.

FIGURE 6: Variation of vibration velocity and kurtosis for Si_3N_4 full-ceramic bearings under grease lubrication condition. (a) Variation velocity. (b) Kurtosis.

FIGURE 7: Variation of vibration velocity and kurtosis for Si_3N_4 full-ceramic bearings under nonlubricated condition. (a) Variation velocity. (b) Kurtosis.

Effect of frequency multiplication on the vibration velocity of Si_3N_4 full-ceramic bearings spindle motor under the condition of grease lubrication and applying preload is shown in Figure 9(a). As can be seen from Figure 9(a), the vibration velocity is mainly affected by 1 frequency multiplication (f_r) and 5 frequency multiplication ($5f_r$), and as the rotational speed increases, the overall vibration velocity is on the rise. When the ceramic bearings are running at low speed (less than 5000 r/min), $5f_r$ contributes the most to the vibration velocity, but for medium and high speed (higher than 6000 r/min), f_r become the largest contributor to the

vibration velocity, followed by $5f_r$. The $2f_r$ and $3f_r$ have little effect on the vibration velocity and the change is relatively stable. Figure 9(b) shows the effect of frequency multiplication on the vibration velocity of the Si_3N_4 full-ceramic bearing spindle motor under the condition of no preload. When the rotational speed is not higher than 4000 r/min, the effect of each frequency multiplication on vibration velocity basically the same, but when the rotational speed is higher than 4000 r/min, f_r has the greater effect on the vibration velocity, the second is $5f_r$; with the increase in the rotational speed, the effect on the vibration velocity shows a trend of

FIGURE 8: Variation of vibration velocity and kurtosis for steel bearings under grease lubrication condition. (a) Variation velocity. (b) Kurtosis.

FIGURE 9: Effect of frequency multiplication on the vibration velocity for Si_3N_4 full-ceramic bearings under grease lubrication condition. (a) Condition of applying preload. (b) Condition of no preload.

first increasing, then decreasing and then increasing. When the rotational speed is 10000 r/min, the vibration velocity of f_r and $3f_r$ are almost the same, and when the rotational speed is 12000 r/min, the vibration velocity of $2f_r$ and $5f_r$ are almost the same. When the rotational speed is higher than 10000 r/min, the axial clearance of the bearing increases due to the lack of preload, resulting in looseness, so the effect of $2f_r$ on vibration velocity is greater than that of $3f_r$.

Figure 10 shows the effect of frequency multiplication on the vibration velocity of Si_3N_4 full-ceramic bearing spindle

motor under nonlubricated condition. As can be seen from Figure 10(a) that under the condition of applying preload, the variation trend of frequency multiplication effect on vibration velocity is basically the same as that under the same condition with grease lubrication. This is because the ceramic material has certain self-lubrication performance, and the lubrication state has little effect on the vibration velocity of Si_3N_4 full-ceramic bearing under the condition of applying preload. Under the condition of no preload, as shown in Figure 10(b), the $3f_r$ under nonlubricated condition

FIGURE 10: Effect of frequency multiplication on the vibration velocity for Si_3N_4 full-ceramic bearings under nonlubricated condition. (a) Condition of applying preload. (b) Condition of no preload.

FIGURE 11: Effect of frequency multiplication on the vibration velocity for steel bearings under grease lubrication condition. (a) Condition of applying preload. (b) Condition of no preload.

fluctuates more than that under grease lubrication condition. This may be because there is no grease filling and lead to the bearing axial clearance increases.

Figure 11(a) shows the effect of frequency multiplication on the vibration velocity for steel bearing spindle motor under the condition of grease lubrication and applying preload. As can be seen from Figure 11(a) that similar to Si_3N_4 full-ceramic bearings, the major influences on the vibration velocity are f_r and $5f_r$, and the contributions varies

with the increase in the rotational speed. For example, in the range of 2000 r/min–5000 r/min and 9000 r/min–12000 r/min, the contribution of $5f_r$ on the vibration velocity is greater. However, in the range of 6000 r/min–8000 r/min and 13000 r/min–16000 r/min, the contribution of f_r on vibration velocity is greater. Figure 11(b) shows the effect of frequency multiplication on the vibration velocity for steel bearing spindle motor under the condition of grease lubrication and no preload. It can be found that $3f_r$ has the

greatest effect on the vibration velocity, followed by $5f_r$, and $2f_r$ has the least effect. This is due to that under the condition of no preload, with the increase in rotational speed, the clearance between the rolling elements and raceways in the bearing becomes larger caused by the centrifugal force, resulting in greater impact of $3f_r$ and $5f_r$ on the vibration.

4.1.3. Comparison of Si₃N₄ Full-Ceramic Bearings and Steel Bearings. Figure 12 shows the vibration velocity comparison of Si₃N₄ full-ceramic bearings and steel bearings under grease lubrication condition, with the increase in rotational speed. Among them, Figure 12(a) is the condition of applying preload, and Figure 12(b) is the condition of no preload. As can be seen from Figure 12, the vibration velocity of Si₃N₄ full-ceramic bearing is always lower than that of steel bearing, regardless of whether the preload is applied or not. Under the condition of applying preload, compared with steel bearing, the maximum reduction in vibration velocity of Si₃N₄ full-ceramic bearing is 45.6%, which occurs at the speed of 4000 r/min. Under the condition of no preload, the maximum reduction in vibration velocity is 55.3%, which occurs at the speed of 6000 r/min. Thus, it can be seen that the vibration performance of Si₃N₄ full-ceramic bearings in the spindle motor is superior under the same working conditions.

4.2. Influence of Working Conditions on Temperature Rise. Infrared thermal imager is a kind of equipment that uses infrared detector and optical imaging objective lens to receive the infrared radiation energy distribution pattern of the measured target and then embody it on the photosensitive components of the infrared detector. It does not need to directly touch the object to monitor the thermal image in the infrared wavelength spectrum. The thermal image corresponds to the thermal field distribution of the surface layer of the object. The difference of different tones indicates the different temperature of the measured object. The common infrared thermal imager works in the midinfrared (3~5 μm) or far-infrared (8~14 μm) band, respectively. The schematic diagram of the infrared thermal imager analysis/monitoring system used in the experimental research is shown in Figure 13.

4.2.1. Temperature Rise under Different Working Conditions. The running-in procedure used in the test is as follows: the start-up rotational speed is 1000 r/min, and the step of speed-up is 1000 r/min; the rotational speed is increased every 10 min before 6000 r/min, and every 20 min after 6000 r/min; after reaching the limit speed, the spindle motor maintains the speed and runs for a total time of 6 hours. During the period, the infrared thermal imager is used to collect the temperature data of the spindle motor.

Figures 14 and 15 show the variation of temperature rise at different parts of the steel bearing and Si₃N₄ full-ceramic bearing spindle motor. As can be seen from Figures 14 and 15, whether steel bearings or Si₃N₄ full-ceramic bearings, the spindle motor presents the highest temperature rise at the

position of front bearing, followed by the rear bearing, and with the lowest temperature rise at the shaft end. This is because for the front and rear bearings, the rotational friction of the bearing and electromagnetic heating between the stator and rotor can lead to the temperature rise, leading to the rise in temperature of the two positions higher than the shaft end. At the same time, due to the shaft end is exposed to the air, the convection heat dissipation effect is better, so the temperature rise is relatively lower. Although there is a fan to cool down the spindle motor during operation, the fan blows from the rear bearing to the front bearing, so the heat dissipation effect of the rear bearing is better than the front bearing, resulting in the lower temperature rise of the rear bearing than the front bearing.

Comparing Figures 14(a) and 14(b), it can be found that under the same working conditions, the temperature rise of the spindle motor with preload is significantly higher than that without preload. For the front bearing, rear bearing, and shaft end, the temperature rise of the spindle motor under the condition of applying preload is 21.3%, 18.3%, and 18.5%, respectively, higher than that with no preload applied. This is because the preload reduces the gap between the rolling elements, inner ring, and outer ring, which increases the friction between the rolling elements, the inner ring, and outer ring during the bearing operation, resulting in the increase in heat generation. Therefore, in the process of designing and assembling the spindle motor, an appropriate preload should be selected.

The temperature rise comparison of Si₃N₄ full-ceramic bearing spindle motor under grease lubrication and non-lubrication conditions are shown in Figures 15(a) and 15(b). As can be seen from Figure 15, for both the conditions of grease lubrication and nonlubrication, the temperature rise of the spindle motor shows a trend that first rises rapidly and then gradually stabilizes. The temperature rise under non-lubricated condition is not much different from that under grease lubrication condition and even slightly lower than the temperature rise under grease lubrication condition. This is because Si₃N₄ ceramic material has certain self-lubrication performance, and self-lubrication film between the ceramic balls, inner ring, and outer ring can be formed in the process of operation, reducing the generation of friction heat. For Si₃N₄ full-ceramic bearings with grease lubrication, grease not only plays a certain role in lubrication but also plays a certain role in hindering the operation of bearings, increasing the heat generation of bearings. At the same time, the addition of grease is not conducive to heat dissipation, so the temperature rise is slightly higher than that without lubrication.

4.2.2. Comparison of Temperature Rise between Si₃N₄ Full-Ceramic Bearings and Steel Bearings. Figure 16 shows the comparison of the temperature rise for Si₃N₄ full-ceramic bearings and steel bearings spindle motor. It can be seen from Figure 16 that under the same working conditions, the temperature rise of Si₃N₄ full-ceramic bearing is always lower than that of steel bearing. For front bearing, back bearing, and shaft end, the temperature rise

FIGURE 12: Vibration velocity comparison of Si_3N_4 full-ceramic bearings and steel bearings under grease lubrication condition. (a) Condition of applying preload. (b) Condition of no preload.

FIGURE 13: Schematic diagram of infrared thermal imager analysis/monitoring system.

FIGURE 14: Temperature rise of steel bearings under grease lubrication condition. (a) Condition of applying preload. (b) Condition of no preload.

FIGURE 15: Temperature rise of Si_3N_4 full-ceramic bearings under different lubrication conditions. (a) Condition of grease lubrication. (b) Condition of no lubrication.

of Si_3N_4 full-ceramic bearing spindle motor is 12.5%, 17.5%, and 16.8% lower than that of steel bearing spindle motor, respectively. Under grease lubrication condition, the temperature rise of Si_3N_4 full-ceramic bearing spindle motor with preload is almost equal to that of steel bearing spindle motor without preload, whereas the temperature rise of Si_3N_4 full-ceramic bearing spindle motor under

nonlubricated condition is even slightly lower than that of under lubrication condition. Thus, it can be seen that the temperature rise performance of Si_3N_4 full-ceramic bearings in the spindle motor is better under the same working conditions, and for full-ceramic bearings, the amount of grease added should be lower than that of steel bearings.

FIGURE 16: Temperature rise comparison of Si_3N_4 full-ceramic bearings and steel bearings under grease lubrication conditions.

5. Conclusion

In this study, controlled variable experiments are carried out on Si_3N_4 full-ceramic ball bearings and steel bearings of the same type; the vibration signal characteristics and temperature rise of the spindle motor are tested and analysed by changing the lubrication conditions, preloads, and rotational speeds of the spindle motor. The following conclusions are obtained:

- (1) With the increase in rotational speed, the vibration velocity of Si_3N_4 full-ceramic bearing spindle motor generally increases and the kurtosis decreases under different preload and lubrication conditions, indicating that the vibration signal is relatively stable, even though the vibration velocity increases at high speed.
- (2) With the increase in rotational speed, the difference of vibration velocity under condition of applying preload and no preload decreases, indicating that the effect of the preload on the vibration of full-ceramic bearing spindle motor decreases with the increase in rotational speed; At the same time, f_r and $5f_r$ have greater impact on the vibration of the full-ceramic bearing spindle motor, followed by $2f_r$ and $3f_r$.
- (3) Lubrication conditions have little effect on the temperature rise of full-ceramic bearing spindle motor, and the temperature rise under nonlubricated conditions is even slightly lower than that under grease lubrication conditions.
- (4) Under same working conditions, compared with steel bearings, the vibration velocity of full-ceramic bearing spindle motor decreases by 45.6%–55.3% and the temperature rise decreases by 12.5%–17.5%.

The vibration velocity and temperature rise of the full-ceramic bearing spindle motor are less than those of steel bearings with the same type. It shows that Si_3N_4 full-ceramic bearing has better performance than steel bearing under same working conditions.

Data Availability

The data used to support the findings of this study are included within the paper.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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References

- [1] D. Mu and X. Wang, "Application of high-speed spindle in intelligent high-speed drilling and tapping center machine tool," *IOP Conference Series: Materials Science and Engineering*, vol. 493, no. 1, pp. 1–6, 2019.
- [2] Y. Wu and L. Zhang, *Intelligent Motorized Spindle Technology*, Springer Nature, Berlin, Germany, 2020.
- [3] H. Cao, X. Zhang, and X. Chen, "The concept and progress of intelligent spindles: a review," *International Journal of Machine Tools and Manufacture*, vol. 112, pp. 21–52, 2017.
- [4] S. Nandi, H. A. Toliyat, and X. Li, "Condition monitoring and fault diagnosis of electrical motors-A review," *IEEE Transactions on Energy Conversion*, vol. 20, no. 4, pp. 719–729, 2005.
- [5] C. Bianchini, F. Immovilli, M. Cocconcelli, R. Rubini, and A. Bellini, "Fault detection of linear bearings in brushless AC linear motors by vibration analysis," *IEEE Transactions on Industrial Electronics*, vol. 58, no. 5, pp. 1684–1694, 2010.
- [6] L. Fan, J. Chen, K. Zhang, L. Zhang, and H. Shi, "Operating state evaluation model of motorized spindle based on composite index," in *Proceedings of the 2017 29th Chinese Control And Decision Conference (CCDC)*, pp. 5639–5645, IEEE, Chongqing, China, 2017.
- [7] Z. Liu, W. Chen, D. Li, and W. Zhang, "Theoretical analysis and experimental study on thermal stability of high-speed motorized spindle," *Industrial Lubrication & Tribology*, vol. 69, no. 6, pp. 1049–1065, 2017.
- [8] K. Zhang, Z. Wang, X. Bai, H. Shi, and Q. Wang, "Effect of preload on the dynamic characteristics of ceramic bearings based on a dynamic thermal coupling model," *Advances in Mechanical Engineering*, vol. 12, no. 1, pp. 1–18, 2020.
- [9] S. H. Li, Y. H. Wang, C. Wei, Z. N. Wang, and Z. X. Xia, "Experimental research on dynamic thermal performance of silicon nitride all-ceramic ball bearings," *IOP Conference Series: Materials Science and Engineering*, vol. 1009, no. 1, Article ID 012031, 2021.
- [10] H. Yan, Y. Wu, S. Li, L. Zhang, and K. Zhang, "The effect of factors on the radiation noise of high-speed full ceramic

- angular contact ball bearings,” *Shock and Vibration*, vol. 2018, Article ID 1645878, 9 pages, 2018.
- [11] Y. Wang, J. Cao, Q. Tong et al., “Study on the thermal performance and temperature distribution of Ball bearings in the traction motor of a high-speed EMU,” *Applied Sciences*, vol. 10, no. 12, p. 4373, 2020.
 - [12] I. M. Jamadar and D. Vakharia, “Correlation of base oil viscosity in grease with vibration severity of damaged rolling bearings,” *Industrial Lubrication & Tribology*, vol. 70, no. 2, pp. 264–272, 2018.
 - [13] K. Zhang, Z. Wang, H. Shi, X. Bai, and Z. Wang, “Research on vibration characteristics of a ceramic spindle based on the reverse magnetic effect,” *Shock and Vibration*, vol. 2019, Article ID 6934087, 15 pages, 2019.
 - [14] L. Wang and H. Xu, “Experimental study on the dynamic performance of a new high-speed spindle supported by water-lubricated hybrid bearings,” *Shock and Vibration*, vol. 2016, Article ID 8297834, 8 pages, 2016.
 - [15] K. Zhang, X. Wu, X. Bai, Z. Wang, D. Zou, and J. Sun, “Effect of the lubrication parameters on the ceramic ball bearing vibration in starved conditions,” *Applied Sciences*, vol. 10, no. 4, p. 1237, 2020.
 - [16] Q. He, Y. Shen, F. Ren, L. Li, and A. Volinsky, “Numerical simulation and experimental study of the air-cooled motorized spindle,” *Proceedings of the Institution of Mechanical Engineers - Part C: Journal of Mechanical Engineering Science*, vol. 231, no. 12, pp. 2357–2369, 2017.
 - [17] T. Kosaka, Y. Kano, N. Matsui, and C. Pollock, “A novel multi-pole permanent magnet synchronous machine with SMC bypass core for magnet flux and SMC field-pole core with toroidal coil for independent field strengthening/weakening,” in *Proceedings of the 2005 European Conference on Power Electronics and Applications*, September 2005.
 - [18] J. Liu and P. Zhang, “Thermo-mechanical behavior analysis of motorized spindle based on a coupled model,” *Advances in Mechanical Engineering*, vol. 10, no. 1, pp. 1–12, 2018.
 - [19] P. Zhang and X. Chen, “Thermal-mechanical coupling model-based dynamical properties analysis of a motorized spindle system,” *Proceedings of the Institution of Mechanical Engineers - Part B: Journal of Engineering Manufacture*, vol. 230, no. 4, pp. 732–743, 2016.
 - [20] S. Li, T. Yang, Y. Liu, Z. Zhang, and G. He, “Temperature rise experiment and prediction modeling for key position of preload adjustable motorized spindle,” *Journal of Physics: Conference Series*, vol. 1965, no. 1, Article ID 012129, 2021.
 - [21] L. Zhang, W. Gong, K. Zhang et al., “Thermal deformation prediction of high-speed motorized spindle based on biogeography optimization algorithm,” *International Journal of Advanced Manufacturing Technology*, vol. 97, no. 5, pp. 3141–3151, 2018.
 - [22] W. Hannon, “Rolling-element bearing heat transfer-part I: analytic model,” *Journal of Tribology*, vol. 137, no. 3, Article ID 031102, 2015.
 - [23] W. Hannon, “Rolling-element bearing heat transfer-part II: housing, shaft, and bearing raceway partial differential equation solutions,” *Journal of Tribology*, vol. 137, no. 3, Article ID 031103, 2015.
 - [24] W. Hannon, T. Barr, and S. Froelich, “Rolling-element bearing heat transfer-part iii: experimental validation,” *Journal of Tribology*, vol. 137, no. 3, Article ID 031104, 2015.
 - [25] V.-T. Than and J. H. Huang, “Nonlinear thermal effects on high-speed spindle bearings subjected to preload,” *Tribology International*, vol. 96, pp. 361–372, 2016.
 - [26] C. Rabreau, J. Kekula, M. Ritou et al., “Influence of bearing kinematics hypotheses on ball bearing heat generation,” *Procedia CIRP*, vol. 77, pp. 622–625, 2018.
 - [27] H. Shi, X. Bai, K. Zhang, Y. Wu, and Z. Wang, “Effect of thermal-related fit clearance between outer ring and pedestal on the vibration of full ceramic ball bearing,” *Shock and Vibration*, vol. 2019, Article ID 8357807, 15 pages, 2019.
 - [28] H. T. Shi, X. T. Bai, K. Zhang, Y. H. Wu, and G. D. Yue, “Influence of uneven loading condition on the sound radiation of starved lubricated full ceramic ball bearings,” *Journal of Sound and Vibration*, vol. 461, Article ID 114910, 2019.
 - [29] Y. Wu, H. Yan, S. Li, K. Zhang, and L. Zhang, “Calculation on the radiation noise of ceramic ball bearings based on dynamic model considering nonlinear contact stiffness and damping,” *Journal of Sound and Vibration*, vol. 479, Article ID 115374, 2020.
 - [30] A. Palmgren, *Ball and Roller Bearing Engineering*, SKF Industries Inc., Philadelphia, 1959.
 - [31] B. Eftekharijrad, M. R. Carrasco, B. Charnley, and D. Mba, “The application of spectral kurtosis on acoustic emission and vibrations from a defective bearing,” *Mechanical Systems and Signal Processing*, vol. 25, no. 1, pp. 266–284, 2011.
 - [32] S. S. Udmale, S. S. Patil, V. M. Phalle, and S. K. Singh, “A bearing vibration data analysis based on spectral kurtosis and ConvNet,” *Soft Computing*, vol. 23, no. 19, pp. 9341–9359, 2019.
 - [33] J. Tian, C. Morillo, M. Azarian, and M. Pecht, “Motor bearing fault detection using spectral kurtosis-based feature extraction coupled with K-nearest neighbor distance analysis,” *IEEE Transactions on Industrial Electronics*, vol. 63, no. 3, pp. 1793–1803, 2015.
 - [34] H. Zhang, X. Chen, Z. Du, and R. Yan, “Kurtosis based weighted sparse model with convex optimization technique for bearing fault diagnosis,” *Mechanical Systems and Signal Processing*, vol. 80, pp. 349–376, 2016.
 - [35] W. Guo, P. W. Tse, and A. Djordjevic, “Faulty bearing signal recovery from large noise using a hybrid method based on spectral kurtosis and ensemble empirical mode decomposition,” *Measurement*, vol. 45, no. 5, pp. 1308–1322, 2012.
 - [36] F. Immovilli, M. Cocconcelli, A. Bellini, and R. Rubini, “Detection of generalized-roughness bearing fault by spectral-kurtosis energy of vibration or current signals,” *IEEE Transactions on Industrial Electronics*, vol. 56, no. 11, pp. 4710–4717, 2009.