Experimental research on dynamic thermal performance of silicon nitride all-ceramic ball bearings

S H Li¹,², Y H Wang*, C Wei¹, Z N Wang¹ and Z X Xia¹
¹ Mechanical Engineering College, Shenyang Jianzhu University, Shenyang, Liaoning 110168, China
² National-Local Joint Engineering Laboratory of NC Machining Equipment and Technology of High-Grade Stone, Shenyang, Liaoning 110168, China

*E-mail: yonghua0514@163.com

Abstract. In order to verify that the silicon nitride all-ceramic ball bearings have more excellent high-speed performance, thermal rise and vibration are the two main factors that restrict the bearing to move at high speed. In this manuscript, based on the establishment of the mechanical model of bearings, the structural model of all ceramic motorized spindle-bearings is established. The temperature field of the motorized spindle-bearing system is simulated through analysis of bearing heat generation. The thermal rise of silicon nitride all-ceramic ball bearings was tested by setting up all ceramic motorized spindle-bearing test platform under the operating speeds of 6,000, 9,000, 12,000, 15,000 and 18,000 rpm respectively, the results are compared with the simulation results. Vibration characteristics of the all-ceramic ball bearings were compared with those of similar steel bearings. The test results show that the thermal rise of all-ceramic ball bearings gradually tends to be stable at different rotational speeds after the running time exceeds 15 minutes, no-load thermal rise is less than 10°C, the maximum temperature of ceramic bearing is nearly 10°C lower than the maximum temperature of the simulated metal bearing. With the increase of rotating speed, the maximum amplitude of metal bearing without loading is 2.38 times of that of ceramic bearing. The conclusion shows that ceramic bearings have better dynamic characteristics and stable reliability than metal bearings, so they have better high-speed performance, providing reference value for the future design of the same type of all-ceramic bearings.

1. Introduction
Silicon nitride, as an important engineering structural ceramic material, has excellent properties as bearing material, such as high strength, high hardness, thermal shock resistance, good wear resistance, stable chemical properties, good insulation, good self-lubricating properties, etc. In addition, the ceramic material has low density, strong bearing capacity and low failure rate in high-speed environment, thus the working life is greatly increased [1, 2].

With the development of science and technology, new materials are widely used and gradually applied in the spindle-bearing system of CNC machine tools [3, 4]. Among them, the outer ring, inner ring and ceramic rolling elements of the all-ceramic bearing components are all made of ceramic materials, which are more used in spindle-bearing system. Ceramic ball bearings have excellent performance and good performance under high-temperature and high-speed environment, which can
guarantee the normal operation of high-speed equipment [5, 6]. The thermal rise and vibration characteristics of ceramic ball bearings will have a great impact on the accuracy and vibration characteristics of the equipment.

Bearing heat generation and vibration have great influence on spindle thermal rise, thermal displacement, rotation accuracy and vibration characteristics. Harris [7] calculated the heat generation of the contact unit inside the bearing and tested the bearing temperature according to the test, which verified that the local calculation of the heat generation of the bearing is relatively close to the actual working condition. Bossmans [8] established the thermal model of high-speed motorized spindle, deeply explored the factors such as thermal conductivity and thermal resistance, and analyzed the thermal rise characteristics of motorized spindle at different rotational speeds. Chen Xiao-an et al. [9] established a thermal-mechanical coupling dynamic model of angular contact ball bearings, and considered the influence of thermal displacement and pre-tightening mode in this process, and analyzed the friction loss and dynamic bearing stiffness of the spindle-bearing system in running state. Marhomy [10] put forward an analytical method to determine the stability of elastic spindle-bearing system by using nonlinear parameters. Wardle [11] established a model of dynamic characteristics of spindle-bearing system, which simplified the preload constant. Chen [12] established a model to determine the response of the spindle bearing system under high speed operation. Jones model was adopted for the bearing in the model, and only axial pretension was considered. Bert R. Jorgensen [13] established the dynamic model of the "spining-bearing" unit of motorized spindle based on the centralized parameter method in Timoshenko Beam theory, and studied its dynamic characteristics by numerical method. Altintas and Cao [14, 15] proposed how to establish the dynamic analysis model of spindle-bearing system by using finite element method under the influence of centrifugal force and bearing gyro torque. It is proved that the bearing dynamic model has good accuracy in predicting the stiffness, mode of vibration and natural frequency of contact force bearing.

The contact load and contact angle of high-speed rolling bearing are the basis of analyzing and calculating thermal characteristics, stiffness characteristics and dynamic characteristics of the spindle-bearing system [16]. In this manuscript, considering the influence of the gyroscopic moment and centrifugal force to balance problems, based on the Hertz elastic contact theory of rolling bearing, all-ceramic ball bearing structure analysis model is established, through constructing the all-ceramic motorized spindle-bearing test platform, the rise of temperature of the silicon nitride all-ceramic ball bearing is tested, and compared with the vibration characteristics of similar steel bearing through the contrast test. It provides reference value for the future design of the same type all-ceramic bearing.

2. Establishment of bearing mechanical model

2.1. Geometrical relation of bearing internal deformation

As shown in figure 1, the relative positions of the j ball center and the center of curvature of the inner and outer ring of the raceway before and after the rolling bearing bears load at the Angle position \( \phi_j \). Assuming that the central position of the curvature of the outer raceway remains unchanged and the central position of the curvature of the inner raceway changes, \( \delta_a \) is the axial relative displacement, \( \delta_r \) is the radial displacement, and \( \theta \) is the angular deformation.

When no load is borne, the distance between the curvature centers of the inner and outer ring channels is:

\[
A = (f_i + f_e - 1)d_w
\]
Consider the effect of thermal displacement of the bearing, the axial distance and radial distance between the final positions of the curvature centers of the inner and outer raceways can be obtained from equations (2) and (3) at the ball \( j \) during constant pressure preload [17].

\[
A_j = A \sin \alpha_o + \delta_a + \theta R \cos \varphi_j + \delta_{aT} \tag{2}
\]

\[
A_j = A \cos \alpha_o + \delta_c \cos \varphi_j + \delta_{cT} \tag{3}
\]

At the position of ball \( j \) during positioning preloaded, considering the influence of bearing thermal displacement and centrifugal radial expansion, the axial distance and radial distance between the final positions of curvature centers of inner and outer raceways can be obtained by equations (4) and (5) [18].

\[
A_j = A \sin \alpha_o + \delta_a + \theta R \cos \varphi_j + \delta_{aT} \tag{4}
\]

\[
A_j = A \cos \alpha_o + \delta_c \cos \varphi_j + \delta_{cT} \tag{5}
\]

In the formula, \( \alpha_o \) is the contact angle before being loaded; \( \delta_{aT} \) and \( \delta_{cT} \) are the axial thermal displacement and radial thermal displacement respectively; \( \delta_{c} \) is the radial expansion of the bearing inner ring caused by centrifugal force under high-speed operation. The calculation method that affects it is shown in formula (6).

\[
\delta_{c} = \frac{\rho_i \omega^2}{16E_i} \left[ \left( 3 + \nu \right) d^2 + \left( 1 - \nu \right) (2R_i) \right]^2 \tag{6}
\]

In the formula \( \rho_i \) is the inner ring material density; \( d \) is the inner diameter of the inner ring; \( R_i \) is the diameter of inner ring channel; \( \nu \) is the inner circle material Poisson's ratio; \( E_i \) is the elastic modulus of inner ring material, and \( \omega \) is the revolving speed of inner ring.

By introducing new variables \( V_{1j} \) and \( V_{2j} \), the geometric compatibility equation of ball and raceway deformation is:

\[
(A_j - V_{1j})^2 + (A_j - V_{2j})^2 - [(f_i - 0.5)d_w + \delta_{aT}]^2 = 0 \tag{7}
\]

\[
V_{1j}^2 + V_{2j}^2 - [(f_i - 0.5)d_w + \delta_{cT}]^2 = 0 \tag{8}
\]
According to figure 1, the contact angle between the ball and the inner and outer raceway can be expressed as:

\[
\cos \alpha_{ij} = \frac{V_{2j}}{(f_e - 0.5)d_w + \delta_{ej}}
\]
\[
\sin \alpha_{ij} = \frac{V_{1j}}{(f_e - 0.5)d_w + \delta_{ej}}
\]

\[
\cos \alpha_{ij} = \frac{A_{2j} - V_{2j}}{(f_e - 0.5)d_w + \delta_{ej}}
\]
\[
\sin \alpha_{ij} = \frac{A_{1j} - V_{1j}}{(f_e - 0.5)d_w + \delta_{ej}}
\]

(9)

2.2. Ball force and balance equation

When the bearing is running at high speed, the ball is subjected to centrifugal force, gyro torque, and the contact load between the ball and the raceway. The force of the \(j\)th ball is shown in figure 2. \(F_{cj}\) is the centrifugal force, \(M_{gj}\) is the gyroscopic moment, \(Q_{ij}\) and \(Q_{ej}\) are the contact loads between the ball rolling body and the inner and outer raceways [17, 18, 19].

![Figure 2. Load diagram of the \(j\)th ball.](image)

The centrifugal force of the \(j\)th ball is calculated by the following equation:

\[
F_{cj} = \frac{1}{2} m_w \omega^2 \left( \frac{\rho_m}{\omega} \right)^2
\]

(10)

The gyroscopic moment of the \(j\)-th ball is calculated by the following equation:

\[
M_{gj} = J \left( \frac{\omega}{\omega} \right) \left( \frac{\omega}{\omega} \right) \omega^2 \sin \beta
\]

\[
J = \frac{1}{60} \rho_m \pi d_w
\]

(11)

(12)

In the equation, \(\rho_m\) is the material density.
The contact loads of the ball and the inner raceway are:

\[
Q_{ij} = K_{ij} \delta_{ij}^{3/2}
\]

(13)

\[
Q_{ej} = K_{ej} \delta_{ej}^{3/2}
\]

(14)

In the equation, \(K_{ij}\) and \(K_{ej}\) are the load deformation constants between the rolling element and the inner and outer raceways \([20]\).

According to the outer race control theory, it is believed that the frictional force only occurs on the outer race, taking \(\lambda_{ij}=0\) and \(\lambda_{ej}=2\).

The balance equation of the ball is:

\[
Q_{ij} \sin \alpha_{ij} - Q_{ej} \sin \alpha_{ej} - \frac{M_{ij}}{d_w} (\lambda_{ij} \cos \alpha_{ij} - \lambda_{ej} \cos \alpha_{ej}) = 0
\]

(15)

\[
Q_{ij} \cos \alpha_{ij} - Q_{ej} \cos \alpha_{ej} + \frac{M_{ij}}{d_w} (\lambda_{ij} \sin \alpha_{ij} - \lambda_{ej} \sin \alpha_{ej}) + F_{ij} = 0
\]

(16)

Assuming that the outer raceway of the bearing is fixed and the external load received by the bearing is the axial force \(F_a\) and the radial force \(F_r\), the balance equation of the inner raceway of the bearing is:

\[
F_a - \sum_{j=1}^{2} (Q_{ij} \sin \alpha_{ij} - \frac{M_{ij}}{d_w} \lambda_{ij} \cos \alpha_{ij}) = 0
\]

(17)

\[
F_r - \sum_{j=1}^{2} (Q_{ij} \cos \alpha_{ij} + \frac{M_{ij}}{d_w} \lambda_{ij} \sin \alpha_{ij}) \cos \varphi_j = 0
\]

(18)

During positioning preloading, the initial preloading force of the bearing has been changed due to the radial expansion amount \(\delta_c\) generated by centrifugal force, the axial thermal deformation amount \(\delta_{aT}\) at and the radial thermal deformation amount \(\delta_{rT}\) caused by the thermal rise of each component. The changed preloading force value needs to be calculated by equation (19).

\[
F_a = \sum_{j=1}^{2} \sin \alpha K_n \delta_n^{3/2}
\]

(19)

In the formula, \(\delta_n\) is the total deformation of the ball after loading, which can be obtained by equation (20); \(\alpha\) is the contact angle after loading, and \(K_n\) is the load deformation constant obtained by equation (18) \([21]\).

\[
\delta_n = \left[ (A \sin \alpha + \delta_a + \delta_{aT} + R \theta \cos \varphi)^2 + (A \cos \alpha + \delta_r + \delta_c + \delta_{rT})^2 \right]^{1/2} - A
\]

(20)

\[
\sin \alpha = \frac{A \sin \alpha_0 + \delta_a + \delta_{aT} + R \theta \cos \varphi}{\left[ (A \sin \alpha_0 + \delta_a + \delta_{aT} + R \theta \cos \varphi)^2 + (A \cos \alpha_0 + \delta_r \cos \varphi + \delta_c + \delta_{rT})^2 \right]^{1/2}}
\]

(21)
3. Motorized spindle-bearing system model

3.1. All ceramic motorized spindle-bearing structural model

The research in this manuscript takes the angular contact ball bearing 7009C as an example, and the performance analysis is carried out for different materials. The structural parameters of bearings with different materials are shown in Table 1.

| Bearing material | Inner diameter (mm) | Outer diameter (mm) | Initial contact angle (°) | Density (g/cm³) | Diameter of balls (mm) | Elasticity modulus (GPa) | Poisson ratio |
|------------------|---------------------|---------------------|---------------------------|-----------------|------------------------|--------------------------|--------------|
| Si₃N₄            | 45                  | 75                  | 15                        | 3.24            | 8.731                  | 320                      | 0.26         |
| Steel            | 45                  | 75                  | 15                        | 7.85            | 8.731                  | 206                      | 0.3          |

This manuscript uses the ceramic spindle as the prototype to study the dynamic characteristics of silicon nitride all-ceramic ball bearing, as shown in Figure 3. The front (7009C) and rear (7008C) ceramic bearings are angular contact ball bearings, the two bearings are paired and installed back to back. The bearing and shaft adopt interference fit. The rotor is installed on the shaft by hot pressing, and the air gap between the rotor and the stator is 0.3 mm. Spindle cooling uses a water chiller to adjust the temperature and flow rate generated at high speeds. The cooling water jacket is hot-pressed in the main shaft housing and has a rectangular cross section. In the running process of the high-speed ceramic motorized spindle, the dynamic characteristic of bearing is the main factor affecting the running precision of spindle [22]. The spindle bearing system adopted in this experiment is based on the high-speed motorized spindle model of 170-30/15 and all-ceramic ball bearing as support. The bearing material selected is silicon nitride and the cage is PEEK composite material. It has higher wear resistance, because the density is less than the steel bearing, the inertia force and centrifugal force produced is smaller, because the friction force is smaller the heat produced is lower, has a better development prospect.

![Figure 3](image.png)

**Figure 3.** The schematic diagram of spindle-bearing system structure:
1-Front bearing 7009C; 2-Cooling Jacket; 3-Rear bearing 7008C; 4-Rotor; 5-Stator; 6-Shaft.

3.2. Thermal field of the motorized spindle-bearing system

Based on the boundary conditions of the spindle parts, the simulation results through Comsol software show that the distribution of the thermal field of the motorized spindle-bearing system in Figure 4.
Figure 4. The temperature field of the motorized spindle-bearing system.

The steady-state temperature field cloud diagram of 170-HT-30 through simulation calculation is shown in figure 4, the simulation results show that the spindle thermal reaches steady state, the internal thermal distribution is very obvious, and the temperature at the rotor is the highest, up to 51 °C. The temperature difference between the front and rear bearings is also obvious. The temperature of the front bearing is slightly higher than that of the rear bearing, up to 47 °C. The main reason is that the front and rear bearing models are different. Under the same load, the internal contact state of the bearing is not the same. As a heat source, the heat output of the front bearing is greater than that of the rear bearing. Secondly, from the structure of the electric spindle, the front bearing is closer to the built-in motor, and the heat will also be transferred from the rotor to the front bearing, so that the thermal rise of the front bearing is slightly higher than that of the rear bearing.

4. Experimental test and analysis of dynamic performance of all ceramic motorized spindle-bearing system

4.1. Temperature Vibration test

The test conditions are shown in table 2, in which the bearing speed starts from 6000 rpm and increases by 1000 rpm every time until 18000 rpm.

| Lubrication mode of bearings | Oil intake (ml/h) | Pressure (Mpa) | Cooling water temperature (°C) | Cooling water flow (L/min) | Room temperature (°C) | Bearing preload (N) |
|-----------------------------|------------------|----------------|--------------------------------|---------------------------|-----------------------|---------------------|
| Oil-air                     | 5                | 0.25           | 18                             | 10                        | 25                    | 400                 |

The equipment used during the test is shown in figure 5.
Figure 5. Test equipment.

The measurement of the vibration acceleration and thermal of the ceramic motorized spindle is shown in figure 6. The basic parameters of ceramic motorized spindle are shown in table 3.

Figure 6. The measurement of the vibration acceleration and thermal of the ceramic motorized spindle.

Table 3. Basic parameters of spindle.

| Basic parameters of spindle | 170-30/15 ceramic spindle | 170HT30 metal spindle |
|-----------------------------|---------------------------|-----------------------|
| Rated speed (rpm)           | 30000                     | 30000                 |
| Rated voltage(V)            | 350                       | 350                   |
| Rated frequency(Hz)         | 1000                      | 1000                  |
| Pole logarithm              | 4                         | 4                     |
| Shaft material              | Si$_3$N$_4$ ceramic        | 40Cr steel            |
| Bearing inner and outer ring material | Si$_3$N$_4$ ceramic        | GCr15 steel           |
| Bearing ball material       | Si$_3$N$_4$ ceramic        | GCr15 steel           |

4.2. Experimental results and analysis
The experiment on the bearing outer ring thermal as shown in figure 6. At the operating speed of 6000~12000 rpm, the curve of the thermal of the outer ring of the ceramic bearing changes with time, as shown in figure 7. Before the running time of 15min, it can be seen that the thermal rise at different speeds has an obvious upward trend. Friction heat generation is the main reason for the heat
generation of high-speed electric spindle bearings. In addition to the differential friction heat generation between the ball and the track, there is also the ball gyro Rotating friction heat generation and sliding friction heat generation between the ball and the cage [23]. When the motorized spindle started to operate, the ceramic bearings were still in the adaptation stage, and the coolant circulation was not smooth enough. Oil and gas lubrication does not play a certain role in bearings. After the running time exceeds 15 min, it can be seen that the thermal rise of the ceramic bearing gradually stabilizes at different speeds, and load thermal rise is less than 10°C. At this time, as the rotation time is longer, the high temperature resistance and self-lubricity of the ceramic bearing are more reflected. The performance of the spindle at high speed tends to be stable, and the life can be extended.

Compared with the simulation results in Figure 4, it can be seen that the maximum temperature of ceramic bearing is 36.5°C, which is nearly 10°C lower than the maximum temperature of the simulated metal bearing. It can be seen that ceramic bearings have better high-speed performance.

![Figure 7](image_url)

**Figure 7.** The temperature of the outer ring of the bearing changes with time at different speed.

The ceramic bearing and the rotating shaft of the ceramic motorized spindle adopt interference fit, and the vibration of the spindle can reflect the dynamic characteristics of the inner ring of the ceramic bearing. The acceleration amplitude is used to measure the intensity of the vibration, and the data collected in the motorized spindle operation by the Non-contact laser vibrometer are exported and sorted out. Under the working conditions of 6000, 9000, 12000, 15000, 18000 rpm, ceramic motorized spindle and metal motorized spindle respectively take 2000 sets of data for spectrum analysis, and obtain the vibration frequency domain of ceramic motorized spindle at 6000~18000 rpm as shown in Figure 8. As can be seen from Figure 8, under the working condition of 6000 rpm, the fast Fourier transform (FFT) transform shows that the first-order frequency is 99.06 Hz, and the amplitude is 0.0158 m/s². The second-order frequency is 199.63 Hz, and the amplitude is 0.0258 m/s². The third-order frequency is 299.20 Hz, and the amplitude is 0.0234 m/s². Under the working condition of 9000 rpm, the first-order frequency is 149.72 Hz, and the amplitude is 0.0623 m/s². The second-order frequency is 299.94 Hz, and the amplitude is 0.0037 m/s². The third-order frequency is 450.68 Hz, and the amplitude is 0.0266 m/s². Under the working condition of 12000 rpm, the first-order frequency
is 200.63Hz, and the amplitude is 0.082m/s². The second-order frequency is 399.77Hz, and the amplitude is 0.0229 m/s². The third-order frequency is 601.16 Hz, and the amplitude is 0.025 m/s². Under the working condition of 15000 rpm, the first-order frequency is 249.29Hz, and the amplitude is 0.1121m/s², the second-order frequency is 500.34Hz, and the amplitude is 0.0291 m/s². The third-order frequency is 750.38 Hz, and the amplitude is 0.0284 m/s². When the bearing vibration signal increases with the number of revolutions, the double frequency amplitude is larger in the low frequency region than in the high frequency region. The vibration amplitude of each frequency level tends to change steadily while the number of revolutions of the motorized spindle increases, no-load vibration is less than 0.16m/s². Because of the excellent characteristics of silicon nitride material, such as high temperature resistance, high rigidity, good precision retention, good self-lubricating performance, etc., the thermal rise of the motorized spindle makes the thermal deformation of the internal ceramic bearing small, thus affecting its own natural frequency small.

![Figure 8. Vibration frequency domain of silicon nitride all-ceramic ball bearings.](image)

The vibration frequency domain of metal motorized spindle at 6000~18000 rpm is shown in figure 9. As can be seen from figure 5, under the working condition of 6000 rpm, the fast Fourier transform (FFT) transform shows that the first-order frequency is 99.73Hz, and the amplitude is 0.0122m/s². The second-order frequency is 199.72 Hz, and the amplitude is 0.005 m/s². The third-order frequency is 300.81Hz, and the amplitude is 0.0063 m/s². Under the working condition of 9000 rpm, the first-order frequency is 150.39Hz, the amplitude is 0.0712m/s². The second-order frequency is 300.26 Hz, the amplitude is 0.0661 m/s². The third-order frequency is 449.68Hz, and the amplitude is 0.0097 m/s². Under the working condition of 12000 rpm, the first-order frequency is 199.63Hz, the amplitude is 0.1161m/s². The second-order frequency is 399.77 Hz, the amplitude is 0.0458 m/s². The third-order frequency is 601.16Hz, and the amplitude is 0.0168 m/s². Under the working condition of 15000 rpm, the first-order frequency is 250.26Hz, the amplitude is 0.1808m/s². The second-order frequency is 500.34 Hz, the amplitude is 0.0732 m/s². The third-order frequency is 749.38Hz, and the amplitude is 0.0321 m/s². Under the working condition of 18000 rpm, the first-order frequency is 282.14Hz, the
amplitude is 0.3743 m/s$^2$. The second-order frequency is 601.16 Hz, the amplitude is 0.0072 m/s$^2$. The third-order frequency is 939.99 Hz, and the amplitude is 0.0483 m/s$^2$. The vibration amplitude of each frequency of the motorized spindle increases with the increase of the number of revolutions. On the one hand, the thermal rise causes the thermal deformation of the internal parts of the motorized spindle, that is, the shape of each part changes, which may affect its own natural frequency. On the other hand, it should be the result of the increase of electromagnetic force after the shape change, no-load vibration is less than 0.40 m/s$^2$.

![Figure 9. Vibration frequency domain of metal bearings.](image)

It can be seen from the experimental data that as the speed increase, the amplitude fluctuation of ceramic bearings is much smaller than that of metal bearings. The amplitude fluctuation corresponding to each order of frequency at each speed is also smaller, and the maximum amplitude of metal bearing without loading is 2.38 times of that of ceramic bearing. It can be seen that ceramic bearings have better dynamic characteristics and stable reliability than metal bearings.

5. Conclusion

In this manuscript, through the establishment of all-ceramic motorized spindle bearing test platform, the thermal rise of silicon nitride all-ceramic ball bearing was tested, and compared with the same type of metal bearing temperature simulation. The vibration characteristics of the bearings were compared with those of similar type steel bearings. The conclusions are as follows:

- Based on the rotation time of the motorized spindle is longer, the thermal rise of the ceramic bearing gradually stabilizes at different speeds, the maximum temperature of ceramic bearing is nearly 10 °C lower than the maximum temperature of the simulated metal bearing, indicating that the high temperature resistance and self-lubricity of the ceramic bearing play a role, so that the performance of the spindle at high speed tends to be stable and the life is extended;
- With the increase of spindle speed, the vibration amplitude of each order frequency of ceramic spindle changes little and tends to be stable. The vibration amplitude of each frequency of the metal motorized spindle increases with the increase of revolution, which indicates that the
ceramic bearing has obvious frequency-time response characteristics, good dynamic characteristics and stable reliability;

- As the rotation speed of the electric spindle increases, the vibration amplitude of the ceramic motorized spindle at each order frequency changes less and tends to stabilize. The vibration amplitude of each frequency of the metal motorized spindle increases with the rotation speed. The maximum amplitude of the metal bearing is 2.38 times that of the ceramic bearing under no load, indicating that the ceramic bearing has better dynamic characteristics and reliability than the metal bearing.

Acknowledgments
This work was supported by NSFC Grant Numbers 51975388, Liaoning Natural Science Foundation 2019-MS-266, Shenyang Science and Technology Project RC170216. The Project is also sponsored by “Liaoning BaiQianWan Talents Program”.

References
[1] Moazami G M, Jenabali S A and Nazarboland A 2009 Mater Des. 6 2283-95
[2] Bai X T, Wu Y H, Zhang K 2017 Journal of Sound and Vibration 410 35-48
[3] Umehara N, Kirtane T, Gerlick R, Jain V K and Komanduri R 2006 Int. J. Mach. Tool. Manu. 46 151-95
[4] Golabczak A and Koziarski T 2005 Assessment method of cutting ability of CBN grinding wheels Int. J. Mach. Tool. Manu. 45 1256-60
[5] Ohta H and Kobayashi K 1996 Journal of sound and vibration 192 481-93
[6] Engel T, Lechler A and Verl A 2016 CIRP.ANN. 65 353-56
[7] Harris T A 2001 Rolling Bearraig Analysis (New York: John Wiley & Sons Ina)
[8] Bossmanns B and Tu J F 1999 Int. J. Mach. Tool. Manu. 39 1345-66
[9] Chen X A, Liu J F and He Y 2013 Chinese Journal of Mechanical Engineering 11 135-42
[10] El-Marhomy A A 1999 Heat and Mass Transfer 35 334-44
[11] Wardle F P, Lacey S J and Poon S Y 1983 Precision Engineering 5 175-83
[12] Chen C H, Wang K W and Shin Y C 1994 Int. J. Vib. Acoust. 4 506-22
[13] Bert R J and Yung C S 1998 J. Manuf. Sci. Eng. 120 387-94
[14] Cao Y and Altintas Y 2007 Int. J. Mach. Tool. Manu. 47 1342-50
[15] Altintas Y and Cao Y 2005 Manufacturing Technology 54 379-82
[16] Houupert L 2010 CAGEDYN: a contribution to roller bearing dynamics calculations part II: description of the numerical tool and its output Tribology Transaction 1 10-21
[17] Li H, He H and Liu L 2012 Bearings 9 37-39
[18] Guo X and Wang Y 2013 Bearings 4 30-33
[19] Than V T and Huang J H 2016 Tribology International 96 361-72
[20] Wang W, Hu L, Zhang S et al. 2014 Mechanism and Machine Theory 82 154-72
[21] Ma C, Yang J, Zhao L et al. 2015 Applied Thermal Engineering 86 251-68
[22] Zhang K, Wang Z et al. 2020 Adv. Mech. Eng. 12 1-12
[23] Chen G, Wang L and Gu L 2007 Journal of Aerospace Power 1 163-68