Study on Stiffness of Angular Contact Ball Bearing and Its Evolution Rule under Excessive Preload

Xianghe Yun¹, Ning Li¹, Baogang Wen² and Qingkai Han¹*

¹ School of Mechanical Engineering, Dalian University of Technology, Dalian, PR China
² School of Mechanical Engineering and Automation, Dalian Polytechnic University, Dalian, PR China

E-mail: qk.han@hotmail.com

Abstract. There are many factors influencing the bearing stiffness variation, among which, preload has more significant effect. In addition, bearing stiffness is developing accompanying with its performance deterioration, which leads to vibration variation of bearing-rotor system. In this paper, system support stiffness model and system dynamics model of micro-turbine bearing-rotor system test rig are built to obtain the bearing stiffness identification method by model simplification based on modal theory. The identification method is validated adopting modal test. The stiffness evolution rule and rotor vibration of different stages are achieved via employing the identification method to extract bearing stiffness over the process of bearing run-to-failure. The results indicate that bearing stiffness is the main source of system support stiffness. Bearing stiffness will change during the whole life, which affects the vibration responses in time and frequency domain. Rotor stability is influenced by cage movement.

1. Introduction
Rolling bearing is main load-carrying component of rotary machinery, whose stiffness variation gives rise to the oscillation of bearing vibration[1], at the same time, it has the impression on safety and stable revolution of rotor. During the service course of bearing, its stiffness will alter accompanying with the function decay of bearing, which has an impact on the bearing-rotor system vibration. Some researches [2-7] demonstrated that the factors bringing about the variation of bearing stiffness are axial preload, inner radial clearance, speed and so on. Thereinto, axial preload not only has notable effect on bearing stiffness, but also on bearing life. The ball will skid and roll intermittently if the preload is too small, which will induces the instabiliy of cage-ball. Then, the vibration amplitude of bearing is one or two orders of magnitude higher than that as bearing is in the normal operation[8]. However, the higher preload, the more heat production[9], it will cause the expansion of inner friction moment and shorten the bearing service life[10]. Therefore, appropriate preload not only extends the service life[11], but also strengthens the stiffness[2, 11, 12], depresses the vibration[2, 4], in addition, guarantees the stability of rotor. Consequently, the research on the influence of preload on bearing stiffness, the bearing stiffness evolution law of the whole life and rotor vibration characteristics should be carried out in detail, thus, it will explain bearing vibration mechanism fundamentally and provide guidance for bearing condition-based maintenance and stiffness design of rotor system.
In recent years, people pay their attention to study the effect of preload acting on bearing stiffness and vibration, bearing life, rotor stability and so on. Alfares et al. [4] advocated that bearing stiffness is increasing and vibration is decreasing with the strengthening of preload. However, bearing vibration keeps steady as preload exceeds to a certain value. Cao et al. [6] presented that bearing stiffness increases gradually along the axial and horizontal radial direction, while the stiffness along the vertical radial direction decreases firstly and then increases with the increment of preload. Gunduz et al. [13] proposed that the natural frequency of rotor system grows up along with the increment of preload. In addition, many researches [14, 15] introduced that the rigid preload mechanism is more efficient than constant preload mechanism in maintaining the dynamic stiffness of spindles.

People usually introduced the L_P fatigue life theory, which is firstly proposed by Lundberg and Palmgren[16], to gain a clear idea of the relationship between preload and bearing service life. Hagiw et al. [17, 18] determined the optimal preload by analyzing the relationship between bearing service life of shaft ball bearing and preload. Kaczor et al. [19] observed the influence of preload acting on the angular contact bearing service life. Zhang et al. [20] advocated a quasi-dynamic model to invest the load distribution of ball bearing standing arbitrary preload based on the contact state of ball-raceway, in the same time, it was employed to study the impact of external loads, rotating speed and preload on ball-raceway contact state and calculate the fatigue life of ball bearing in a running state. Some researchers took advantage of advanced algorithm to diagnose bearing initial fault[21] and prognose the remaining useful life[22].

In the aspect of rotor stability, inner radial clearance and axial preload have a notable effect on the rotor stability. Tiwari and Gupta et al. [3] firstly proposed the variation of inner radial clearance induces a third region of instability, moreover, it resulted in the appearance of regions of periodic, subharmonic and chaotic behavior in a large extent. Harsha et al. [23, 24] advocated that the rotational speed has the same effect with inner radial clearance. Bai et al. proved that the rotor stability is strongly dependent on the inner radial clearance, axial preload and radial force [25], for instance, inner radial clearance and rotational speed brought about three routes to instability, which are the period-doubling bifurcation, the secondary Hopf bifurcation and the boundary crisis [26], axial preload has a profound effect on bifurcation margins [27], the non-linear characteristics, hertz contact forces and internal clearance of ball bearings led to the occurrence of subharmonic resonance at the same time [28]. Zhang et al. [29] maintained that bearing contact angular changes notably in company with the axial preload, the contact angular significantly contributes to the ball passage frequency and its coupling frequency with rotational frequency. With considering the influence of contact angular, the main resonance region of bearing-rotor system switches to the lower speed ranges. Furthermore, Ashtekar et al. [30] analyzed that the rotor orbits affected by different preload. Allaire et al. [31] studied the influence of bearing parameter acting on synchronous vibration of turbine-gearbox-generator system. Xiao et al. [32] obtained the variety of bearing parameters and rotor stability affected by different preload. Long et al. [33] analyzed the critical speed of rotor influenced by static and dynamic stiffness, respectively.

From the above, people can draw a conclusion that the existing studies are mainly focused on discussing the influence of axial preload on bearing stiffness, life and rotor stability by analytic method. Nevertheless, people do not have an insight into bearing stiffness, its stiffness evolution law of the whole life and development of rotor axial orbit through these conclusions. For that reason, bearing stiffness identification method is proposed by means of simplification of support stiffness model and bearing-rotor system dynamics model of micro-turbine. Next, the bearing stiffness evolvement rule is achieved by introducing vibration data of bearing service period to the forementioned method. The rest of this paper are arranged as follows: In section 2, the theoretical background of bearing stiffness identification method, test equipment and location of sensors are illustrated. Section 3 presents bearing stiffness, its evolution law of the whole life and rotor vibration characteristics. The conclusions are made in Section 4.

2. Methodology of stiffness identification and vibration test
2.1. Construction of test rig

Micro-turbine test rig is shown in Figure 1(a), whose bearing-rotor system fulcrum construction and preload mechanism are illustrated in Figure 1(b). As we can see, the fulcrum consists of angular contact ball bearing, house and pedestal. In which, angular contact ball bearings are employed to support the rotor, which are treated as the main part of load-carrying and force-transferring, they are connected to tube-shaped house adopting end covers. The middle of the house is bolted to the pedestal, which forms an interference fit. The preload mechanism is rigid preload [14, 34], which can satisfy different tightness by the means of modifying the axial distance between end cover and house, it also alters the relative location of inner and outer rings. The relative distance between house and end cover is taken as € when the house is tight fit to the other one.

Figure 1. Test Equipment: (b)(1)Pedestal (2)House (3)End Cover 1 (4)Shaft (5)Disc 1 (6)Tested Bearing (7)Contrast Bearing (8)Disc 2 (9)End Cover 2 (10)Drum 1 (11)Drum 2 (12)Elastic Couple (13)AC Motor.

2.2. Model of bearing-rotor system support stiffness

According to Figure 1(b), the form of micro-turbine bearing-rotor system fulcrum parts is in series, therefore, the stiffness of fulcrum and each part can be equivalently calculated, respectively.

Figure 2. Fulcrum Radial Stiffness Model of Test Rig.

Ignoring damping, the radial stiffness model of fulcrum structure is shown in Figure 2(a). $K_{hi}$ ($i = 1, 2$), $K_h$ and $K_s$ is the radial stiffness of bearing, house and pedestal, respectively.
\( m_b (i=1,2), m_h \) and \( m_s \) is the mass of bearing, house and pedestal, respectively. \( F_r \) is the fulcrum radial load. The model illustrated in Figure 2(a) is further simplified as shown in Figure 2(b) by mixing the two bearings together and combining house with the pedestal, whose stiffness and mass are \( K_b, m_b \) and \( K_c, m_c \), respectively. Therefore, the motion differential equations of mass-spring system model demonstrated in Figure 2(b) is presented as follows:

\[
m_b \ddot{X}_b + K_b (X_b - X_c) = 0 \tag{1}
\]

\[
m_c \ddot{X}_c + K_b (X_c - X_b) + K_c X_c = 0 \tag{2}
\]

Owing to the stiffness of house and pedestal is so strong that their deformation is equal to zero, that is \( X_c = 0 \). Hence, the motion differential equations of mass-spring system model shown in Figure 2(b) becomes as follows:

\[
m_b \ddot{X}_b + K_b X_b = 0 \tag{3}
\]

Assuming that the fulcrum radial stiffness is \( K_r \), the integral motion differential equations of mass-spring system model is shown as follows:

\[
m_b \dot{X}_b + K_r X_b = 0 \tag{4}
\]

Comparing the Equation (3) and (4), we can conclude that the fulcrum radial stiffness is equal to the bearing radial stiffness, that is:

\[
K_r = K_b = K_{b1} + K_{b2} \tag{5}
\]

According to Equation (5), the bearing stiffness is the main contributor of micro-turbine bearing-rotor system support stiffness.

2.3. Identification method for bearing stiffness

According to Sect. 2.2, support stiffness is mainly affected by bearing stiffness in micro-turbine bearing-rotor system, hence, the bearing stiffness can be extracted from bearing-rotor system. In order to identify the bearing stiffness, the dynamics equation of micro-turbine bearing-rotor system is built. Steps as follows:

The rotor system mechanics schematic diagram of micro-turbine test rig is illustrated in Figure 3, the hypotheses is adopted as follows during modeling process:

1) The discs are rigid;
2) The shaft is rigid without mass, whose mass is lumped to discs;
3) Ignoring the torsional vibration and axial vibration, there is bending vibration in bearing-rotor system;
4) The bearing is elastic, which is modeled as the stiffness model with 4 degrees of freedom, but the house and pedestal are rigid;
5) The speed is constant.
The Fourth Chinese International Turbomachinery Conference (CITC 2020)  
IOP Conf. Series: Materials Science and Engineering  
IOP Publishing  
1081 (2021) 012012  
doi:10.1088/1757-899X/1081/1/012012

Figure 3. Bearing-rotor System Mechanics Schematic Diagram of Micro-turbine Test Rig: (b) (In which \( P_1 \) and \( D_1 \) denote the mass center and geometric center of the disc 1, respectively).

The bearing-rotor system motion differential equation with 4 degrees of freedom established by employing the Lagrange energy method is shown as follows:

\[
\begin{align*}
(m_1 + m_2)\ddot{x}_i + (K_{h1} + K_{b2})x_i + m_2a\ddot{\theta}_j + (K_{h1}a_1 + K_{b2}a_2)\dot{\theta}_j &= (m_1 + m_2)e_i\Omega^2 \cos(\Omega t + \phi_1) \\
(m_1 + m_2)\ddot{y}_i + (K_{h1} + K_{b2})y_i - m_2a\ddot{\theta}_j - (K_{h1}a_1 + K_{b2}a_2)\theta_j &= (m_1 + m_2)e_i\Omega^2 \sin(\Omega t + \phi_1) \\
m_2a\ddot{x}_i + (K_{h1}a_1 + K_{b2}a_2)x_i - (m_2a^2 + J_{d1} + J_{d2})\ddot{\theta}_j - (J_{p1} + J_{p2})\Omega^2 \dot{\theta}_j &= m_2e_i\Omega^2 a \cos(\Omega t + \phi_1) \\
-K_{h1}a_1^2 + K_{b2}a_2^2 + K_{b1} + K_{b2}\dot{\theta}_j &= m_2e_i\Omega^2 a \cos(\Omega t + \phi_1) \\
-m_2a\ddot{y}_i - (K_{h1}a_1 + K_{b2}a_2)y_i + (m_2a^2 + J_{d1} + J_{d2})\ddot{\theta}_j - (J_{p1} + J_{p2})\Omega^2 \dot{\theta}_j &= m_2e_i\Omega^2 a \sin(\Omega t + \phi_1) \\
+K_{h1}a_1^2 + K_{b2}a_2^2 + K_{b1} + K_{b2}\dot{\theta}_j &= m_2e_i\Omega^2 a \sin(\Omega t + \phi_1)
\end{align*}
\]

where \( m_i, J_{pi}, J_{ph} \) (\( i = 1, 2 \)) is the mass, polar inertia moment and radial inertia moment of rotor system lumped to the disk \( D_i \) (\( i = 1, 2 \)), respectively. \( x_i, y_i \) (\( i = 1, 2 \)) is the displacement of disc \( D_i \) (\( i = 1, 2 \)) along X and Y direction, respectively. \( \theta_{x_i}, \theta_{y_i} \) (\( i = 1, 2 \)) is the angular displacement of disc \( D_i \) (\( i = 1, 2 \)) along X and Y direction, respectively. Moreover, \( (m_1 + m_2)e_i\Omega^2 \) is time-varying amplitude of force induced by rotor unbalance, \( m_2e_i\Omega^2 a \) is time-varying moment amplitude of \( D_2 \) acting on the disc \( D_1 \), which is originated from rotor unbalance.

Ignoring \( \theta_{x_i}, \theta_{y_i} \) (\( i = 1, 2 \)), the Equation (6) becomes as follows:
\[(m_1 + m_2)\ddot{x} + (K_{b1} + K_{b2})x = (m_1 + m_2)e_i \Omega_i^2 \cos(\Omega_i t + \varphi_1)\]
\[(m_1 + m_2)\ddot{y} + (K_{b1} + K_{b2})y = (m_1 + m_2)e_i \Omega_i^2 \sin(\Omega_i t + \varphi_1)\]
\[m_2 \ddot{a}x + (K_{b1}a_i + K_{b2}a_2)x = m_2 e_i \Omega_i^2 a \cos(\Omega_i t + \varphi_1)\]
\[-m_2 \ddot{a}y - (K_{b1}a_i + K_{b2}a_2)y = m_2 e_i \Omega_i^2 a \sin(\Omega_i t + \varphi_1)\]

According to modal analysis theory, the Equation (7) is further simplified as follows:
\[m\ddot{x} + K_x = m e_i \Omega^2 \cos \Omega t\]
\[m\ddot{y} + K_y = m e_i \Omega^2 \sin \Omega t\]

where \(m = m_1 + m_2\) is the mass of system, \(K = K_{b1} + K_{b2}\) is the support stiffness, \(e_i\) is the geometric eccentricity and \(\Omega\) is the rotational speed.

The equation becomes as follow:
\[\ddot{x} + \omega_n^2 x = e_i \Omega^2 \cos \Omega t\]
\[\ddot{y} + \omega_n^2 y = e_i \Omega^2 \sin \Omega t\]

where \(\omega_n = \left(K_e/m\right)^{1/2}\), is the system natural frequency.

Marking \(z\) as \(z = x + iy\), the above equation can be written as follows:
\[\ddot{z} + \omega_n^2 z = e_i \Omega^2 e^{i\Omega t}\]

where the particular solution is \(z = A e^{i\Omega t}\), then, the solution of equation (10) is obtained as follows:
\[z = \frac{e_i (\Omega / \omega_n)^2}{1 - (\Omega / \omega_n)^2} e^{i\Omega t}\]

Therefore, the unbalance response of rotor system is shown as follows:
\[x = \frac{e_i (\Omega / \omega_n)^2}{1 - (\Omega / \omega_n)^2} \cos(\Omega t)\]
\[y = \frac{e_i (\Omega / \omega_n)^2}{1 - (\Omega / \omega_n)^2} \sin(\Omega t)\]

The corresponding vibration amplitude is written as follows:
\[|A| = \frac{e_i \Omega^2}{\omega_n^2 - \Omega^2} = \frac{e_i (\Omega / \omega_n)^2}{1 - (\Omega / \omega_n)^2}\]

According to Equation (13), the amplitude of vibration \(|A|\) is greater than zero, the expression of support stiffness along \(x\) or \(y\) direction is calculated as follows:
\[K_i = \frac{m\Omega^2 * (e_i + A)}{A}\]

As shown in Equation (14), the support stiffness of micro-turbine bearing-rotor system is dependent on the unbalance, geometry eccentricity and vibration amplitude. Comparing Equation (5) and Equation (14), the relationship of each bearing stiffness and displacement is presented as follows:
\[K_{b1} = K_{b2} = \frac{m\Omega^2 * (e_i + A)}{2A}\]

where geometry eccentricity \(e_i\) is determined by modal test.
In a word, the stiffness revolution rule of bearing under excessive preload is achieved by adopting Equation (15). Firstly, the eccentricity \( e_1 \) of bearing-rotor system is obtained through modal test. Then, the micro-turbine bearing-rotor system test rig is employed to perform bearing run-to-failure test, at the same time, the shaft vibration of the whole life is recorded. Finally, the bearing stiffness identification and evolution rule are implemented based on Equation (15). Specific framework is explained in Figure 4.

![Figure 4. Framework of Stiffness Identification.](image)

### 2.4. Method for vibration test

Micro-turbine bearing-rotor system test rig is illustrated in Figure 1(a), the collected vibration data is engaged to bearing stiffness identification over the service process. The rotor is supported by two angular contact ball bearings, which is coupled to AC motor for different revolutions. The preload mechanism is rigid preload as shown in Figure 1(b), the preload grade is realized with the help of modifying the spin depth of bolt to acquire different axial relative displacement \( e \). The bearings are located in E, F of test rig, thereinto, the bearing in position E is tested bearing, the one in position F is contrast bearing. The run-to-failure test is last until the occurrence of fault or significant increment of vibration, which means bearing becomes invalid. In the meantime, test is completed. The total failure time (the whole life period) is calculated from bearing is running after installed.

The vibration of bearing and shaft is collected by acceleration sensors CAYD115V-100A mounted on house and eddy current sensors RP6606XL fixed near the shaft, respectively. As shown in Figure 5, sensor 1 is utilized to record vertical vibration of shaft, whose horizontal vibration is collected by sensor 2. Sensor 3 and 4 is employed to obtain vertical vibration of tested bearing and contrast bearing, respectively. Moreover, data acquisition equipment is composed of hardware NI Cdaq-9174 and software programmed based on NI LabVIEW, which can cover the command of real-time acquisition and demonstration. Speed is 2400rpm, sample frequency is 25.6Hz, sample duration is 20s, the data is recorded every 1 min. Tested bearing is angular contact ball bearing 7208AC.
3. Methodology of stiffness identification and vibration test

3.1. Comparison of braces stiffness under different preload
Micro-turbine bearing-rotor system test rig consists of angular contact ball bearing, house and pedestal, whose stiffness is arranged in tandem. In order to prove that the main contributor of support stiffness is bearing stiffness, the stiffness of braces under different preload is achieved adopting LMS to perform modal test of braces. Preload grade is modified by swiveling bolt to changing axial relative displacement. As depicted in Figure 3, three eddy current sensors are arranged evenly along circumference of end cover near tested bearing. The mean of displacements of three points is treated as $e$, which is shown in Table 1, preload grades are divided into loose, tight, normal and overtight according to $e$.

![Figure 5. Location of Sensors.](image_url)

![Figure 6. The Relative Distance Test.](image_url)

| Grade  | $e$/mm | Tightness | $e$/mm |
|--------|--------|-----------|--------|
| Loose  | 0.818  | Normal    | 0.764  |
| Tight  | 0.772  | Overtight | 0.682  |

According to literature[3], the stiffness of braces under different preload is achieved by modal test. From the results are illustrated in Table 2, there differs 1-2 order of magnitude between stiffness of bearing and that of house or pedestal. Consequently, bearing stiffness contributes mostly to support stiffness in micro-turbine bearing-rotor system test rig, namely, $X_s=0$. 

---

8
Table 2. Comparison of Braces under Different Preload

| Grade  | Loose   | Tight   | Normal  | Overtight |
|--------|---------|---------|---------|-----------|
| Pedestal | 1.25e8N/m | 1.25e8N/m | 1.25e8N/m | 1.25e8N/m |
| House   | 3.74e7N/m | 3.74e7N/m | 3.74e7N/m | 3.74e7N/m |
| Bearing | 2.02e6N/m | 2.58e6N/m | 4.96e6N/m | 9.08e6N/m |

The curves of bearing stiffness under different preload are described in Figure 7. The bearing stiffness along vertical radial direction is equal to that of horizontal radial direction. They are increasing sharply with the increment of preload. However, the stiffness along horizontal radial direction raises faster than that of vertical radial direction, the maximum difference is 15.65%.

![Figure 7. Bearing Stiffness of Different Preload.](image)

3.2. Bearing stiffness evolution rule under excessive preload

The run-to-failure test is invalid under loose preload owing to roller has tendency to skid earlier, which results in instability between roller and cage. Nevertheless, it is time-consuming and costly to perform run-to-failure test due to the actual service life is too long under normal preload. Hence, the excessive preload is adopted in this test, which is corresponding to the grade of overtight shown in Table 1.

The test lasts 32 min, bearing loses efficacy due to seizure. As stiffness is the natural property of bearing-rotor system, in other words, the support stiffness is constant when bearing-rotor system is changeless. Therefore, the eccentricity $e_1$ induced by machining and installation is abstracted employing initial stiffness collected in Sect.3.1 during the process of bearing stiffness identification over the whole life. Next, In order to acquire bearing stiffness evolvement curve along vertical and horizontal direction during the whole life, the displacement every 2 min is introduced into the identification method of bearing stiffness explained in Sect.2.3. The result is illustrated in Figure 8. The development tendency of stiffness between vertical radial direction and horizontal radial direction is similar, which is approximately divided into three stages: first fluctuation (0-6min), stabilization (6-16min) and second fluctuation (16-32min). In stage one, the curve increases firstly and then decreases. The stiffness in stage two is almost stable. In stage three, the stiffness curve oscillates frequently and notably, after one sharp increment (16-18min), the curve keeps steady for a moment, then comes into volatility period (22-32min), finally, the stiffness drops down owing to bearing seizure. Comparing stiffness evolution between 0-4min and 28-32min, the trend of them is homology,
which indicates that bearing is inclined to seizure in primary stage, the reason leading to this phenomenon is bearing friction moment enlarges due to bearing parts expand induced by violent rise of temperature, however, the amount of heat is not enough to produce abundant expansion at present, thus, the seizure is absent. The missing legend of Figure 9, Figure10 and Figure 11 is same to that of Figure 8.

![Figure 8. Bearing Stiffness Evolution Curve.](image)

### 3.3. Rotor vibration characteristic during the whole life under excessive preload

During the running process of bearing, the variation of its stiffness has directly effect on rotor vibration and stability. As shown in Figure 9, there are system vibration responses in time domain of main infection moments during the whole bearing life. The corresponding responses in frequency domain are introduced in Figure 10. It is obvious that the rotor displacements are same along horizontal and vertical direction, the rotational frequency \( f_r \) evoked by unbalance of rotor system is dominant. In first fluctuation stage (0-6min), the amplitude of rotor displacement firstly decreases and then increases, the chief frequency is rotational frequency \( f_r \), whose amplitude has the same tendency with that in time domain, however, the amplitude of fundamental train frequency \( f_c \) slightly go up in 2min. In stable running stage (6-16min), the amplitude of rotor displacement remains unchanged, the dominant frequency component is rotational frequency \( f_r \), whose amplitude raises lightly. In second fluctuation stage (16-32min), the amplitude of displacement cuts down initially, subsequently, it keeps stable for a period until the occurrence of impulse signal, only in seizure stage, fundamental train frequency \( f_c \), 2-times \( f_c \) and \( f_r \) are governing, whose amplitudes are abruptly increasing.
Figure 9. Time Responses of Various Moments.
As shown in Figure 11, the rotor axial orbit has the same variation tendency with displacement, which performs regular, stable and variable size “rhombus” whirling motion. Rotor radial runout changes in different degrees along different directions: the horizontal radial runout of rotor increases lightly in first increment stage (0-2min), the vertical radial runout of rotor enlarges in second addition stage (16-18min), besides, the radial runout has the notable expansion in third increment stage (26-30min) in both directions. In the moment of seizure, the axial orbit of rotor is complicated, which is caused by fundamental train frequency $f_c$ and 2-times $f_c$. 

Figure 10. Frequency Responses of Various Moments.
4. Conclusions
In this paper, the bearing stiffness identification method is promoted based on bearing-rotor system dynamics, the study on bearing stiffness under different preload and its evolution rule under excessive preload is carried out, the results are explained as follows:
(1) Preload has obvious effect on bearing stiffness of micro-turbine. With the increment of preload, bearing stiffness goes up rapidly.
(2) The bearing stiffness curve performs the inclination of “fluctuation-stability-significant fluctuation”, on the basis of the curve, the bearing life is divided into three stages.
(3) With the development of bearing stiffness, the corresponding responses in time and frequency domain are varying in different extent. In first fluctuation stage, the amplitude of rotor displacement firstly cuts down and then increases, the chief frequency is rotational frequency $f_r$. In stabilization stage, the amplitude of rotor displacement remains steady, the dominant frequency component is rotational frequency $f_r$. In second fluctuation stage, the amplitude of displacement reduces initially, subsequently, it keeps unchanged for a period until impulse signal appears, only in seizure phase, fundamental train frequency $f_c$ and 2-times $f_c$ are dominant.

Acknowledgement
This research work was supported by the National Key R&D Program of China (No.: 2018YFB2000300) and the Natural Science Foundation of Liaoning Province (No.: 20180550792) and the High-level Talents of Dalian City (No.: 2018RQ18)
References
[1] Tandon N and Choudhury A 1999 A review of vibration and acoustic measurement methods for the detection of defects in rolling element bearings Tribol Int 32 469-80
[2] Aktürk N, Uneeb M and Gohar R 1997 The Effects of Number of Balls and Preload on Vibrations Associated With Ball Bearings Journal of Tribology 119 747-53
[3] Tiwari M, Gupta K and Prakash O 2000 Effect of Radial Internal Clearance of a Ball Bearing on the Dynamics of a Balanced Horizontal Rotor Journal of Sound and Vibration 238 723-56
[4] Alfares M A and Elsharkawy A A 2003 Effects of axial preloading of angular contact ball bearings on the dynamics of a grinding machine spindle system Journal of Materials Processing Technology 136 48-59
[5] Harsha S P, Sandeep K and Prakash R 2003 Effects of preload and number of balls on nonlinear dynamic behavior of ball bearing system International Journal of Nonlinear Sciences and Numerical Simulation 4 265-78
[6] Cao H, Li Y, He Z and Zhu Y 2014 Time Varying Bearing Stiffness and Vibration Response Analysis of High Speed Rolling Bearing-rotor Systems Journal of Mechanical Engineering 50 73-81
[7] Jin Y, Yang R, Hou L, Chen Y and Zhang Z 2017 Experiments and Numerical Results for Varying Compliance Contact Resonance in a Rigid Rotor–Ball Bearing System Journal of Tribology 139 1-17
[8] Wardle F and Poon S Y 1983 Rolling bearing noise-cause and cure Chartered mechanical engineer 30 36-40
[9] Than V and Huang J 2016 Nonlinear thermal effects on high-speed spindle bearings subjected to preload Tribol Int 96 361-72
[10] Hwang Y and Lee C 2010 A review on the preload technology of the rolling bearing for the spindle of machine tools International Journal of Precision Engineering and Manufacturing 11 491-8
[11] Harris T A 1965 How to compute the effects of preloaded bearings Production Engineering 19 84-93
[12] James G, Katter J and Tu J F 1996 Bearing Condition Monitoring for Preventive Maintenance in a Production Environment Tribol T 39 936-42
[13] Gunduz A, Dreyer J T and Singh R 2012 Effect of bearing preloads on the modal characteristics of a shaft-bearing assembly: Experiments on double row angular contact ball bearings Mechanical Systems and Signal Processing 31 176-95
[14] Cao H, Holkup T and Altintas Y 2011 A comparative study on the dynamics of spindles with respect to different preload mechanisms The International Journal of Advanced Manufacturing Technology 57 871-83
[15] Zhang J, Fang B, Zhu Y and Hong J 2017 A comparative study and stiffness analysis of angular contact ball bearings under different preload mechanisms Mech Mach Theory 115 1-17
[16] Lundberg G and Palmgren A 1944 Dynamic capacity of rolling bearings Acta Polytechnica Scandinavica. 1 7
[17] Hagiu G D and Gafiganu M D 1994 Preload-service life correlation for ball bearings on machine tool main spindles Wear 172 79-83
[18] G.Hagiu 2003 Reliable high speed spindles by optimum bearings preload International Journal of Applied Mechanics and Engineering 8 57-70
[19] Kaszor J and Raczyński A 2014 The effect of preload of angular contact ball bearings on durability of bearing system Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology 229 723-32
[20] Zhang J, Fang B, Hong J and Zhu Y 2017 Effect of preload on ball-raceway contact state and fatigue life of angular contact ball bearing Tribol Int 114 365-72
[21] Wu P and Zhao X 2019 Incipient Fault Detection of Rolling Bearing Based on Fuzzy Entropy and Fractal Dimension Chinese Journal of Turbomachinery 61 71-9
[22] Zhang C 2020 The Life Prediction of Bearing for Metro Fan Based on EMD Denoise and LSTM Network Chinese Journal of Turbomachinery 62 77-82
[23] Harsha S P 2005 Nonlinear dynamic analysis of an unbalanced rotor supported by roller bearing Chaos, Solitons & Fractals 26 47-66
[24] Harsha S P 2005 Non-linear dynamic response of a balanced rotor supported on rolling element bearings Mechanical Systems and Signal Processing 19 551-78
[25] Bai C and Xu Q 2006 Dynamic model of ball bearings with internal clearance and waviness Journal of Sound & Vibration 294 23-48
[26] Bai C, Xu Q and Zhang X 2006 Nonlinear stability of balanced rotor due to effect of ball bearing internal clearance Applied Mathematics and Mechanics (English Edition) 27 175–86
[27] Bai C, Zhang H and Xu Q 2007 Effects of axial preload of ball bearing on the nonlinear dynamic characteristics of a rotor-bearing system Nonlinear Dynamics 53 173-90
[28] Bai C, Zhang H and Xu Q 2013 Subharmonic resonance of a symmetric ball bearing–rotor system International Journal of Non-Linear Mechanics 50 1-10
[29] Zhang X, Han Q, Peng Z and Chu F 2015 A new nonlinear dynamic model of the rotor-bearing system considering preload and varying contact angle of the bearing Commun Nonlinear Sci 22 821-41
[30] Ashtekar A and Sadeghi F 2011 Experimental and Analytical Investigation of High Speed Turbocharger Ball Bearings Journal of Engineering for Gas Turbines and Power 133 1-14
[31] Allaire P, Dimond T and Cao J 2019 Bearing Fix for Large Sub-synchronous Vibration in Steam Turbine-gearbox-generator System Chinese Journal of Turbomachinery 61 34-38
[32] Xiao P, Chen Y, Li Q and Wang W 2015 Effects of Tilting Pad Bearing Preload on the Rotor Stability Chinese Journal of Turbomachinery 57 31-36
[33] Long L, Lv B and Yuan W 2018 Influence of the Support Dynamic Stiffness on Critical Speeds of a Gas Generator Rotor of a Turbine Engine Chinese Journal of Turbomachinery 60 66-70
[34] Li H and Shin Y C 2004 Analysis of bearing configuration effects on high speed spindles using an integrated dynamic thermo-mechanical spindle model International Journal of Machine Tools and Manufacture 44 347-64