Research Article

Dynamic Characteristics and Experimental Research of a Two-Span Rotor-Bearing System with Rub-Impact Fault

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Rotor rub-impact has a great influence on the stability and safety of a rotating machine. This study develops a dynamic model of a two-span rotor-bearing system with rubbing faults, and numerical simulation is carried out. Moreover, frictional screws are used to simulate a rubbing state by establishing a set of experimental devices that can simulate rotor-stator friction in the rotor system. Through the experimental platform and its analysis system, the rubbing experiment was conducted, and the vibration of the rotor-bearing system before and after the critical speed is observed. Rotors running under normal condition, local slight rubbing, and severe rubbing throughout the entire cycle are simulated. Dynamic trajectories, frequency spectrum diagrams, chart of axis track, and Poincare maps are used to analyze the features of the rotor-bearing system with rub-impact faults under various parameters. The vibration characteristics of rub impact are obtained. Results show that the dynamic characteristics of the rotor-bearing system are affected by the change in velocity and degree of impact friction. The findings are helpful in further understanding the dynamic characteristics of the rub-impact fault of the two-span rotor-bearing system and provide reference for fault diagnosis.

1. Introduction

With the requirement of high speed and high efficiency in modern rotating machines, clearance reduction between the rotor and stator has been widely used. Nevertheless, as gap decreases, rub-impact fault frequently occurs, which causes abnormal vibration that may affect the normal operation of machines. Therefore, the friction fault of the rotor-bearing system should be studied, and the vibration characteristics of the rotor-bearing system under different friction conditions before and after the critical speed should be observed to detect the machine fault and provide reference for fault diagnosis.

Scholars have conducted considerable work and obtained many valuable results on rub-impact fault. Muszynska [1, 2] discussed the chaotic responses of unbalanced rotor-bearing-stator systems with rubbing fault and presented comprehensive overviews on the dynamic analysis of rub-impact rotor-related phenomenon and the vibration response in detail, such as friction, impact, stiffness, and coupling effects. Previous research has focused on the Jeffcott-type rotor. Pavlovskaja et al. [3] studied the complex

nonlinear dynamic behavior caused by rotor-stator friction. Beatty [4] proposed a mathematical rationale of rubbing identification and presented a criteria to infer rubbing related to simple unbalanced response. Roques et al. [5, 6] combined the finite method with nonlinear contact theory to analyze the rotor-stator friction phenomenon. The governing equations embodied the contact force coupling between the rotor and stator; as a result, the friction problem became further realistic. Patel et al. [7] proposed a mathematical model consisting of interacting vibratory systems of the rotor and stator. The contact was modeled using contact stiffness, damping, and Coulomb friction. Zhang et al. [8] proposed a frictional collision microrotor model with proportional nonlinear rubbing force. The effects of rotational speed, imbalance, damping coefficient, and friction coefficient on the response of the microrotor system were evaluated. Xu et al. [9] studied the rubbing failure model of rotor system, considering the elastic deformation, contact penetration, and elastic damping support during friction. The influence of friction gap on rotor vibration was analyzed. Sun et al. [10] considered the gyro effect and

investigated the steady-state response and stability of a rotor system with frictional collisions. The complex nonlinear phenomenon during frictional collision was obtained through a numerical calculation.

Nowadays, in order to improve energy efficiency and higher thrust-to-weight ratio, double-disc rotor systems are widely used in rotating machinery such as aeroengines. Therefore, the study of the rotor-bearing system rubbing test is necessary. At the same time, the double-disc rotor system exhibits more complex dynamic characteristics than the single disc rotor system. In the above literature, the research focus is on the numerical simulation of a single rotor system, which is obviously not satisfactory for the actual rotating machinery. Through a numerical model and by building an experimental platform, Torkhani et al. [11] considered light, medium, and heavy friction on a rubbing experiment of a rotor-bearing system and discussed the experimental results of the frictional collision of the rotor system. Chen et al. [12] examined the influence of the geometric parameters of the shell and the position of the deformation angle on contact stiffness. A new model was developed to simulate the frictional collision contact force between the rotating shell and stator in the rotor system, and the frictional collision contact model was verified. Cong et al. [13] proposed an impact energy model for assessing the probability or severity of a rubbing fault. The frictional screw was used to simulate the local rubbing fault, and a rotor-stator friction experiment in the rotor system was conducted. Chu and Lu [14] utilized a special stator structure to simulate the state of complete friction, investigated the nonlinear vibration characteristics of the rubbing rotor system, and observed various periodic and chaotic vibrations. On the basis of the structural characteristics of an aeroengine turbine casing, Wang et al. [15] simulated local frictional impact stiffness on a test bench and analyzed the vibration characteristics of the local frictional collision. Ma et al. [16] analyzed two types of rubbing faults of crack and frictional collision coupling, oil film instability, and bearing frictional collision in the rotor system and obtained the vibration system of the rotor with cracks and frictional collisions. Yang et al. [17] established a two-rotor system dynamic model for analyzing unbalanced-fixed frictional coupling faults. They investigated the dynamic characteristics of a dual rotor system and the influence of model parameters on the basis of the Lankarani–Nikravesh model. Lu and Chu [18] examined the dynamics of a vertical Jeffcott rotor under frictionless, partial friction, and full ring friction conditions. Roques et al. [5]

introduced the rotor-stator model of the turbine generator for studying the speed transients of the rotor-stator friction caused by accidental shedding of the blades. Hua et al. [19] established a mathematical model to evaluate the dynamic characteristics of the rotor-rubber bearing system. The frictional force was generated on the basis of the relative speed, and the response of the rotor-rubber bearing system was analyzed in detail via numerical calculation. Wang et al. [20] proposed a demodulation technique based on improved local mean decomposition. The effectiveness of this technique was demonstrated in the presence of frictional collision faults in the rotor system of a gas turbine.

In the existing research, the analysis of the rubbing failure of the two-span rotor rotating machine before and after the critical speed is insufficient. So in the present study, the dynamic model of the two-span rotor-bearing system with rubbing faults is developed, and the numerical simulation of changing the speed and rubbing condition is carried out. Meanwhile, an experimental device for simulating rotor-stator friction in the rotor system is established. The vibration characteristics are numerically studied through an experimental research on the three stages of the double-span rotor system (normal operation, slight rubbing, and severe rubbing) at different speeds to provide reference for fault diagnosis.

2. Mathematical Model of the Two-Span Rotor-Bearing System

In order to study dynamic behavior efficiently, a simplified mathematical model of the double-disc rotor-bearing system with rub-impact fault was developed, as shown in Figure 1. The two ends of the rotor are rigidly supported, and there is a massless elastic shaft between the two supports, k is the rigidity of the rotating shaft, the masses of the discs are m_1, m_2 , and the damping is C , which is installed between the rotating shafts. The picture on the right shows the rubbing diagram, where e is the eccentricity of the disc. The system normal collision force is F_N , and the tangential friction is F_T .

2.1. System Equation without Rubbing Fault. The double-disc rotor model studied in this section is shown in Figure 1. The radial displacement at the turntable is X and Y . Only the lateral vibration of the rotor is considered, and the influence of rubbing on the system quality and damping is ignored. Establishing the system of differential equations of motion based on Lagrange equation

$$\begin{cases} m_1 \ddot{X}_1 + C \dot{X}_1 + K_2 (X_1 - X_2) + K_1 X_1 = m_1 e_1 \omega^2 \cos \omega t, \\ m_1 \ddot{Y}_1 + C \dot{Y}_1 + K_2 (Y_1 - Y_2) + K_1 Y_1 = m_1 e_1 \omega^2 \sin \omega t - m_1 g, \\ m_2 \ddot{X}_2 + C \dot{X}_2 + K_2 (X_2 - X_1) + K_3 X_2 = m_2 e_2 \omega^2 \cos(\omega t + \varphi), \\ m_2 \ddot{Y}_2 + C \dot{Y}_2 + K_2 (Y_2 - Y_1) + K_3 Y_2 = m_2 e_2 \omega^2 \sin(\omega t + \varphi) - m_2 g, \end{cases} \quad (1)$$

where m_1 is the quality of the left disk; m_2 denotes the quality of the right disk; K_1, K_2, K_3 represent the stiffness matrices of the system; X_1, Y_1 refer to the left

disk center displacement; X_2, Y_2 indicate the right disk center displacement; e_1, e_2 refer to the rotor eccentricity; ω indicates the angular velocity of the rotor; and φ

FIGURE 1: Two-span rotor-bearing system.

is the vector angle between the unbalanced mass eccentricities.

2.2. Mathematical Model of Rubbing Force. The normal and tangential impact forces of the rotor during rubbing can be expressed as

$$\begin{cases} F_N = \begin{cases} 0, & r_c < \delta, \\ (r_c - \delta)k_c, & r_c \geq \delta, \end{cases} \\ F_T = fF_N, \end{cases} \quad (2)$$

where δ is the distance between the rotor and stator, k_c denotes the radial stiffness of the stator, r_c indicates the radial displacement of the rotor, and f refers to the friction coefficient between the rotor and stator.

The rubbing force in the x and y directions is as follows:

$$\begin{cases} \begin{bmatrix} F_X \\ F_Y \end{bmatrix} = -\frac{r_c - \delta}{r_c} k_c \times \begin{pmatrix} 1 & -f \\ f & 1 \end{pmatrix} \begin{bmatrix} X \\ Y \end{bmatrix}, & r_c > \delta, \\ F_X = F_Y = 0, & r_c < \delta. \end{cases} \quad (3)$$

Combined with these two cases, the previous formula can be expressed as

$$\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = -\frac{1 + \text{sign}(r_c - \delta)}{2} \frac{r_c - \delta}{r_c} k_c \times \begin{pmatrix} 1 & -f \\ f & 1 \end{pmatrix} \begin{bmatrix} X \\ Y \end{bmatrix}. \quad (4)$$

Therefore, the impact forces on the left and right disks are as follows:

$$\begin{cases} \begin{bmatrix} F_{1X} \\ F_{1Y} \end{bmatrix} = -\frac{1 + \text{sign}(r_{c1} - \delta_1)}{2} \frac{r_{c1} - \delta_1}{r_{c1}} k_c \times \begin{pmatrix} 1 & -f \\ f & 1 \end{pmatrix} \begin{bmatrix} X_1 \\ Y_1 \end{bmatrix}, \\ \begin{bmatrix} F_{2X} \\ F_{2Y} \end{bmatrix} = -\frac{1 + \text{sign}(r_{c2} - \delta_2)}{2} \frac{r_{c2} - \delta_2}{r_{c2}} k_c \times \begin{pmatrix} 1 & -f \\ f & 1 \end{pmatrix} \begin{bmatrix} X_2 \\ Y_2 \end{bmatrix}, \end{cases} \quad (5)$$

where δ_1 and δ_2 denote the clearance between the left and right disks and the stator and r_{c1} and r_{c2} represent the radial displacement of the rotor left and right disks.

2.3. System Dynamics Equation. On the basis of the motion theorem, the differential equations of motion of the rub-impact rotor are obtained. Only the rubbing fault of a single disc is considered here:

$$\begin{cases} m_1 \ddot{X}_1 + C\dot{X}_1 + K_2(X_1 - X_2) + K_1X_1 = F_{X1} + m_1 e_1 \omega^2 \cos \omega t, \\ m_1 \ddot{Y}_1 + C\dot{Y}_1 + K_2(Y_1 - Y_2) + K_1Y_1 = F_{Y1} + m_1 e_1 \omega^2 \sin \omega t - m_1 g, \\ m_2 \ddot{X}_2 + C\dot{X}_2 + K_2(X_2 - X_1) + K_3X_2 = m_2 e_2 \omega^2 \cos(\omega t + \varphi), \\ m_2 \ddot{Y}_2 + C\dot{Y}_2 + K_2(Y_2 - Y_1) + K_3Y_2 = m_2 e_2 \omega^2 \sin(\omega t + \varphi) - m_2 g, \end{cases} \quad (6)$$

where F_{X1} and F_{Y1} represent the rubbing forces of the left disk in the x and y directions, respectively.

In order to facilitate calculation and avoid excessive truncation errors, the dimensionless transformations are

given as follows: $x_1 = X_1/\delta$, $y_1 = Y_1/\delta$, $x_2 = X_2/\delta$, $y_2 = Y_2/\delta$, $\omega t = \tau$, $\omega(d/dt) = d/d\tau$, $\omega^2(d^2/dt^2) = d^2/d\tau^2$, $\dot{x} = dx/d\tau$, $\dot{y} = dy/d\tau$, $\ddot{x} = d^2x/d\tau^2$, and $\ddot{y} = d^2y/d\tau^2$, and the dimensionless equations are carried out:

$$\begin{cases}
\ddot{x}_1 + \frac{C}{m_1\omega}\dot{x}_1 + \frac{K_2}{m_1\omega^2}(x_1 - x_2) + \frac{K_1}{m_1\omega^2}x_1 = \frac{F_{x1}}{m_1\delta\omega^2} + \frac{e_1}{\delta}\cos\tau, \\
\ddot{y}_1 + \frac{C}{m_1\omega}\dot{y}_1 + \frac{K_2}{m_1\omega^2}(y_1 - y_2) + \frac{K_1}{m_1\omega^2}y_1 = \frac{F_{y1}}{m_1\delta\omega^2} \\
+ \frac{e_1}{\delta}\sin\tau - \frac{g}{\delta\omega^2}, \\
\ddot{x}_2 + \frac{C}{m_2\omega}\dot{x}_2 + \frac{K_2}{m_1\omega^2}(x_2 - x_1) + \frac{K_3}{m_2\omega^2}x_2 = \frac{e_2}{\delta}\cos(\tau + \varphi), \\
\ddot{y}_2 + \frac{C}{m_2\omega}\dot{y}_2 + \frac{K_2}{m_1\omega^2}(y_2 - y_1) + \frac{K_3}{m_2\omega^2}y_2 \\
= \frac{e_2}{\delta}\sin(\tau + \varphi) - \frac{g}{\delta\omega^2}.
\end{cases} \quad (7)$$

3. Numerical Results and Discussion

Changes in rotational speed and rubbing conditions have a significant effect on the motion of the rotating machine. In the simulation, by changing the speed and the stiffness of the rubbing, the rotor is numerically simulated before the critical speed, near the critical speed and after the critical speed. The motion characteristics of the rotor under different rubbing conditions are also studied. The main parameters of the system are shown in Table 1.

According to equation (1) in Section 2.1 and equation (7) in Section 2.3, the equation of motion is reduced from the second order to the first order, substituted into the corresponding system parameters, and the numerical integration is solved by the fourth-order Runge-Kutta method; the effect of speed and rubbing stiffness variation on the system is obtained. The dynamic trajectories, frequency spectrum diagrams, and chart of axis track are used to reflect the rotor motion characteristics.

3.1. Speed Effect. Figure 2 shows the dynamic response of the system to the change of the rotational speed. It can be seen from the figure that the change of the rotational speed has a great influence on the displacement response of the rotor-bearing system, and the stability of the system is different in distinct speed segments. Before the critical speed, the system displacement response is relatively small. When the speed is close to the critical point, the displacement of dynamic trajectory, frequency spectrum diagram, and chart of axis track is obviously increased, and the system displacement response is severe. After the critical speed, the rotational motion displacement response gradually decreases.

In summary, the response of the rotor-bearing system is affected by the speed. When the speed exceeds the critical speed, an increase in the speed range will further stabilize the system.

TABLE 1: Main parameters of the system.

Parameter	Value
Quality, m_1, m_2 (kg)	32, 30
Equivalent stiffness, k_i ($i = 1, 2, 3$) (N/m)	$2.5e7, 1e8, 2.5e7$
Damping, C ($N \cdot s \cdot m^{-1}$)	2100
Rubbing clearance, δ (mm)	0.11
Friction coefficient, f	0.1
Rotor eccentricity, e_i ($i = 1, 2$) (mm)	$1.1e-2, 1.2e-2$

3.2. Impact of Rubbing Stiffness. The dynamic response of the system is shown in Figure 3. The dynamic and static stiffness ratio reflects the rubbing force of the system. When $k_1 = 0$, the dynamic response of the rotor system is no fault. As shown in (1), the axis trajectory is a single circle, only 1X component exists in the spectrum diagram, and the system only has unbalance excitation. (2) and (3) are the response of the system when $k_1 = 0.5$ and $k_1 = 1$ respectively. Comparing (1), as the dynamic and static stiffness ratio increases, the displacement amplitude of the time-domain diagram gradually becomes smaller. The axis trajectory gradually changes to "8" shape, which is the typical axis of the rubbing fault. The amplitude of the 1X component on the spectrogram gradually becomes smaller and lower than the amplitude of the 2X component, which means that the rubbing force suppresses the unbalanced fault feature to a certain extent. In addition, 3X, 4X, and other high-frequency components have appeared. The rubbing characteristics are obvious.

In summary, the rubbing stiffness has a great influence on the motion state of the rotor. As the rubbing stiffness increases, the motion state of the system will change.

4. Experimental Setup

The rotor system is a core component of aeroengines and gas turbines. The two-span rotor-bearing system is a form of construction often used in modern aeroengine and gas turbine rotor systems. In this paper, the two-span rotor test bench model is taken as the research object, and the dynamic characteristics of the rubbing response of the model are studied.

4.1. Experiment Arrangement of the Rotor System. As shown in Figure 4, the rotor test rig is composed of a motor, a flexible coupling, a sliding bearing, three graphite bearings, two shafts, and two disks. A flexible coupling for reducing the effect of the motor on the rotor system connected the motor and the rotor system. In the rub-impact experiment, we installed several movable sensor brackets on which two eddy current proximity probes are mounted to measure the vibration displacements of the shaft in the horizontal x and vertical y directions. A speed sensor is installed at one end to measure the rotor speed. Table 2 shows the main component parameters of the two-span rotor-bearing system.

The entire rotor test bed is based on digital speed governor, sensors, multifunction filter amplifier and acquisition instrument, and computer software as auxiliary

FIGURE 2: Dynamic response of the speed change to the system.

equipment. The working mechanism is that the digital speed governor adjusts the rotating speed of the test bed, the sensors measure the various signals produced during the vibration of the test bed, and the acquisition instrument collects all types of signals. The computer software displays the collected data to the terminal screen. Figure 5 shows the concrete process in an experimental system diagram of the flexible rotor test bed.

4.2. Rub-Impact Device. To simulate the rub-impact fault that often occurs in practical rotor systems, several movable sensor brackets are designed, as shown in Figure 6. A radial rubbing fault is simulated by using a rub screw to hit the radial surface of the shaft. The rub screw is fastened to the mounting block with a lock nut and adjusted on the basis of the different positions of the friction failure. The screw can be moved radially from the inside to the outside to adjust the rotor-to-stator clearance in accordance with the experimental requirements.

4.3. Experimental Description of Different Rubbing Faults. The rotor vibrates sharply at a resonant speed. Therefore, this experiment simulated the rubbing fault of the actual rotor system before and after the critical speed. This study verified

that the first critical speed of the test bench is approximately 3100 rev/min. For the rub-impact experiment, the following tests were performed: under normal conditions, local slight rubbing, and severe rubbing throughout the week at speeds of 2000, 3000, and 4000, respectively. In the first experiment, the actual rotor speed-up process was simulated and the system stability from the speed before and after the critical speed was observed without any rubbing conditions. In the second experiment, a partial rubbing test was conducted and the thimble was gently pushed to the rotor surface, resulting in a partial slight rubbing effect. In the third experiment, the thimble will be immediately pushed into a tight coupling with the rotor, which will cause rubbing during the entire week.

5. Analysis of Experimental Results

Rotational speed has a significant influence on rotary movement. Rotors generally start with slow rotations under low rotational speed. In the experiment, the rotor was simulated under normal working conditions, local slight rubbing, and full-cycle severe rubbing by changing the rotational speed, and the motion characteristics of the rubbing rotor were analyzed. Through multiple simulation experiments, representative data of the rubbing experiment

FIGURE 3: Dynamic response of rubbing stiffness change to the system.

FIGURE 4: Rotor test rig.

TABLE 2: Main parameters of the two-span rotor-bearing system.

Component name	Value	Unit
DC motor	0–10000	rad/min
Sliding bearing	24×30 (diameter $\times$ length)	mm
Graphite bearing	21×14 (diameter $\times$ length)	mm
Long shaft	10×500 (diameter $\times$ length)	mm
Short shaft	10×400 (diameter $\times$ length)	mm
Large disk	80×16 (diameter $\times$ length)	mm
Small disk	80×20 (diameter $\times$ length)	mm
Flexible coupling	24×30 (diameter $\times$ length)	mm
Sampling frequency	4000	Hz
Lubrication oil	30

FIGURE 5: Schematic diagram of the experimental data acquisition system.

FIGURE 6: Rub-impact device.

were collected, and dynamic trajectories, frequency spectrum diagrams, chart of axis track, and Poincare maps were used to reflect the rotor motion characteristics.

To facilitate comparison of the features, the experiment provides the vibration characteristics of the two-span rotor under normal operation of the power frequency, as shown in Figures 7–9. The rotor vibration waveforms are relatively stable when the power frequency is running normally. The time-domain images are standard sinusoidal and cosine signals, and the frequency on the spectrum is relatively simple, mostly based on the fundamental frequency. The axis track of the rotor is close to a circle, and the Poincare response of the system is presented as an isolated point. These results show that the two-span rotor test rig is relatively standard; thus, the rub-impact fault signals obtained from the two-span test bed are reliable.

Figures 10–12 present the vibration characteristics of the local slight rubbing of the rotor at different speeds before and after critical speed. When the rotor is running at a low speed and slight local rub, the vibration waveform presents two large and small peaks and valleys in each cycle, that is, a rub-impact phenomenon occurs in each cycle. In the vibration spectrum, the fundamental frequency, $2X$, and $3X$ are found. The $2X$ amplitude is higher than the fundamental frequency in the spectrum. When the velocity increases to 3000 and 4000 r/min, because the impact velocity is high, the impact is mainly reflected on the elastic impact and the rotor speed instantly changes the direction and size. Then, the “peak value” of the vibration waveform of the displacement signal will exhibit a sharp point. Furthermore, the fundamental frequency is dominant in the rotor vibration at a high

rotational speed. High-order frequency doubling, mainly double frequency, third harmonic frequency, and fourth harmonic frequency, is also observed, and all have high-frequency harmonic envelope. By observing the axis locus of the rotor signal at various rotational speeds and at low speed, the chart of axis track presents a “crescent” shape. When approaching critical speed, the axis path presents a sharp angle at the impact point. Across the critical speed, the vibration is reduced and the axis path flattens at the friction point. The comparison of the axis trajectory without rubbing shows that slight rubbing will cause nonlinear pulsation in the impact point, and the orbit of the axis center will become irregular and distorted. At different rotational speeds, the Poincare response of the system is concentrated near one point; thus, local slight rubbing does not change the system periodicity.

When the thimble is pushed immediately into the rotor, severe impact throughout the entire cycle will occur. In the entire cycle under severe rub impact, frictional vibration will cause large system stiffness changes. The stable full-cycle rub impact will not last for a long time, but it will immediately damage the structural surface, change the rub-impact state, and have transient response characteristics. Figures 13–15 present the vibration characteristics of the severe rubbing of the rotor at different speeds before and after critical speed.

When the speed is 2000 r/min, the rotor moves from a normal power frequency to severe rubbing and time-domain waveform irregularity, and the rotor amplitude decreases consistently. Hence, the thimble remains close to the outer edge of the rotor in the entire process, which limits the amplitude pulsation. The reverse friction of the thimble reduces the main vibration frequency of the rotor, and the frequency component is different from that of local rubbing. In comparison with local rub impact, no pulsation of the axis track of the rotor during the entire cycle of rub impact is observed. As the thimble is sticking closely to the rotor surface, although the figure becomes irregular, the trajectory is relatively stable and the amplitude decreases. When the rotational speed is 3000 r/min, which is close to the critical speed, the vibration is intense. Serious rubbing fault occurs in the rotor system, and the phenomenon of clipping wave peak can be seen from the waveform. As shown in the spectrogram, in addition to the fundamental frequency components, significant half-frequency, $3/2X$ and $5/2X$ fractional octave, and high-frequency components, such as $2X$, $3X$, $4X$, and $5X$, also appear. The axis track is relatively

FIGURE 7: Normal state at 2000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 8: Continued.

FIGURE 8: Normal state at 3000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 9: Normal state at 4000 r/min. (a) time-domain waveform. (b) spectrum graph. (c) axis orbit. (d) Poincare orbit.

disordered, and the system response Poincare is concentrated around two points. Hence, when the critical speed is reached, the serious rubbing fault will be that the motion

state of the system is multiperiodic. The rotational speed is increased to 4000 r/min, which is larger than the critical speed and vibration decreases. At this time, the time-domain

FIGURE 10: Local slight rubbing at 2000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 11: Continued.

FIGURE 11: Local slight rubbing at 3000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 12: Local slight rubbing at 4000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) axis orbit. (d) Poincare orbit.

waveform is irregular due to severe rubbing, and the waveform “clipping” phenomenon occurs. The amplitude of the second octave in the spectrum is higher than the

fundamental frequency, and the high-power spectrum, such as $3X$ and $4X$, appears. The axis orbit presents a sharp angle; the rotor is in a slight bending state. The Poincaré of the

FIGURE 13: Severe rubbing at 2000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 14: Continued.

FIGURE 14: Severe rubbing at 3000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

FIGURE 15: Severe rubbing at 4000 r/min. (a) Time-domain waveform. (b) Spectrum graph. (c) Axis orbit. (d) Poincare orbit.

system response is concentrated near 1 point, and its motion can still be regarded as a single cycle.

6. Conclusions

- (1) The dynamic equations of the two-span rotor-bearing system with rubbing faults are established and analyzed, and the numerical simulation of changing the speed and rubbing condition are carried out. The numerical analysis indicates that changes in speed and the interference of the rubbing force increase system instability, and the dynamic characteristics become further complex. Moreover, experimental investigations are conducted intensively on the basis of the numerical simulation in this study.
- (2) The experimental analysis indicates that rotational speed has a significant influence on rotary movement. Under similar rubbing conditions, the dynamic characteristics of system stability are dissimilar at different speed stages. The speed is low at 2000 r/min before the critical speed, rubbing fault occurs at this time, and the system has a relatively small degree of response. When the rotational speed is 3000 r/min near the critical point, the system responds violently once the rubbing fault occurs, and when rubbing is severe, the system will be in a multiperiodic motion. However, after the first-order critical speed is reached and the speed continues to increase to 4000 r/min, the rotational motion will gradually change from a nonstationary phase to a relatively stationary phase. This situation reflects the principle of supercritical unit operation: when the speed exceeds the critical speed, an increase in a certain speed range will further stabilize the system.
- (3) The response of the rotor-bearing system is affected by rubbing conditions. The rotor's various dynamic characteristics are stable when no rubbing fault appears. When the system is subjected to a local slight friction, with the increase in rotational speed, waves, troughs, and sharp corners appear in the vibration waveforms, and the high-frequency harmonic envelope appears. Slight friction causes a nonlinear runout at the impact position, causing the axis trajectory to become distorted and irregular, and the Poincare response is a single point, which is a single-cycle motion. Under severe rubbing, the amplitude fluctuation is limited and the amplitude decreases. In comparison with slight rubbing, as the rotational speed increases close to the critical speed, severe rubbing will cause the occurrence of a fractional frequency multiplication component, and the axis trajectory becomes abnormally disordered. Moreover, the Poincare response is near two points, indicating that it is in a multicycle motion state at this time. The analysis results verify the accuracy of the numerical simulation.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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