Analysis on vibration response of bearing coupling faults in rotor-bearing system

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Abstract. It is well known that the coupled fault diagnosis of rotating machinery is a challenging task, which is mainly due to the complexity of the vibration signals and the interaction of multiple fault components. In order to study the coupling vibration characteristics from unbalance and bearing fault in rotor-bearing system, a dynamic model of the coupled fault is performed to explore the phenomena of multi-fault in the rotor-bearing system in this paper. An experimental study, including seven working conditions, is also examined to further indicate the vibration response of coupling fault with unbalance and bearing fault. The influence of the rotor unbalance on the bearing vibration characteristics and the vibration rules of the coupling fault are proposed in this paper.

Keywords: Rotor-bearing system, coupling fault, fault diagnosis, vibration response.

1. Introduction

As the core of rotating machinery, the rotor system is widely used in various mechanical equipment. The vibration behavior of the rotor system plays a vital role in the performance and reliability of the equipment for fault diagnosis. When the equipment fails, multiple types of faults are usually coupled in the rotor system, especially rotor unbalance and bearing faults. Therefore, it is necessary to study the coupling vibration characteristics from unbalance and bearing fault in rotor-bearing system.

In recent years, many researches on the dynamics of the bearing-rotor system have been proposed to obtain the response of bearing coupling faults. Qin et al. [1] proposed a new dynamic model of rolling bearings to simulate the dynamic response of high-speed rolling bearings with surface defects. Yuan et al. [1] proposed a new method for modeling the vibration of the ball bearing-rotor system by analyzing the dynamic contact and path between the joint interfaces. Liang et al. [2] studied the influence mechanism of rotor offset on the nonlinear vibration response of rotating machinery under speed-increasing conditions considering the
actual working conditions of the rotor system. Yang et al. [3] studied the coupling fault impulse response characteristics and the mechanism of the abrupt change of the rotor shaft motion trajectory with the finite element model. Zhao et al. [4] discussed the influence of raceway defects on the nonlinear dynamic behavior of rolling bearings by establishing a bearing dynamics model. Patel and Naik [5] studied the bearing-rotor system and proposes a mathematical analysis model. Mereles and Cavalca [6] proposed a method alternative method based on distributed parameters or continuous models, which can be applied to the mathematical modeling of complex rotor systems. Eisa et al. [7] proposed a new analysis method that uses multi-body dynamics and finite element theory to analyze and simulate lateral and torsional vibrations. Cui et al. [8] established a nonlinear vibration model for evaluating the severity of rolling bearing faults, simulating the vibration response signals of rolling bearings with different fault levels, and performing quantitative analysis on the signals. In the above research, most of the research is focused on the dynamic model and response mechanism of the bearing-rotor system, which means that there are few studies on the vibration characteristics of coupled faults.

By considering the force analysis of the rolling bearing and the model discretization theory, this paper builds a coupled motion equation of rotor-bearing system. In this equation, the Lagrange method based on the principle of virtual displacement is used to represent the bearing-rotor system. Combined with Adams dynamic analysis and experimental simulation to study the coupling vibration characteristics from unbalance and bearing fault in rotor-bearing system.

2. System modeling and simulation
In order to better describe the complexity of the rolling bearing in the coupled situation, two mechanical analyses are set up. The force analysis of the rolling bearing with the outer ring fixed and the inner ring rotating is shown in figure 1. And the force analysis of the rolling element at the defect of the outer raceway is shown in figure 2.

![Figure 1. Bearing force analysis.](image1)

![Figure 2. Force analysis of rolling elements.](image2)

As shown in figure 1 and figure 2, \( \omega \) is the rotational angular velocity of the rolling element, \( c \) is the weight of the rolling element, \( F_n \) and \( F_r \) are the normal forces of the inner and outer rings of the bearing on the rolling element, \( F_s \) and \( F_f \) are the frictional forces of the inner and outer rings of the bearing on the rolling element, \( F_c \) and \( F_m \) are the normal force and friction force between the cage and the rolling elements.

During the rotation of the bearing, the rolling element are subjected to the frictional force of the inner ring and the radial force of the inner and outer rings, thereby exerting force on the cage and driving the cage to rotate together. The process of bearing entry and exit defects is shown in figure 2: The rolling element enters the defect will lose contact with the outer raceway, and instantaneous impact occurs when the rolling element rolls out of the defect.
2.1. Theoretical model

The equations for describing the motion model of the rotor-bearing system is shown in the following manner. In this work, equations (1) to (4) are are used as the solution method.

\[ M\ddot{x}(t + \Delta t) + C\dot{x}(t + \Delta t) + Kx = F_e(t + \Delta t) + F_e(t) - F(t) \]  \hspace{1cm} (1)

Where \( M \) is the element mass matrix, \( x \) is the nodal acceleration vector, \( C \) is the element damping matrix, \( \dot{x} \) is the node velocity vector, \( K \) is the element stiffness matrix, \( \Delta t \) is the displacement increment vector, \( t \) is time, \( F_e \) is the equivalent nodal contact force, \( F \) is the force of the internal nodes.

Solve the equation of motion by explicit central difference method,

\[ x(t + \Delta t) = x(t) + \dot{x}(t)\Delta t + \frac{1}{2}\ddot{x}(t)\Delta t^2 \]  \hspace{1cm} (2)

\[ \ddot{x}(t + \Delta t) = \ddot{x}(t) + \frac{1}{2}(\ddot{x}(t) + \ddot{x}(t + \Delta t))\Delta t \] \hspace{1cm} (3)

Substituting Eq. (2) and Eq. (3) into Eq. (1),

\[ \ddot{x}(t + \Delta t) = [F_e(t + \Delta t) + F_e(t + \Delta t) - F(t) - K(t)x - (\ddot{x}(t) + \frac{1}{2}\ddot{x}(t))\Delta t]C \left( M + \frac{1}{2}\Delta t C \right)^{-1} \] \hspace{1cm} (4)

Then substituting Eq. (2) and Eq. (4) into Eq. (3), solving Eq. (3) to get all the parameters. (The force in the equation is obtained by the penalty function method.)

2.2. Simulation result analysis

In this part, adams software is used to simulate the dynamics response of ball bearing 6204. The simulation model of the rotor - bearing system is shown in figure 3(a), including bearings, bearing seats, shaft and unbalanced rotors. The bearing model is shown in figure 3(b), the fault degree of the inner and outer rings in the model is simulated by a pit with the diameter of 2mm and the depth of 1mm. The rolling elements fault is simulated by a cutting surface with the width of 2mm.

![Figure 3. Schematic diagram of simulation model.](image)

(a) Rotor-bearing system model  \hspace{1cm} (b) Bearing fault model

![Figure 4. Simulation results of different unbalance.](image)

(a)10% \( U_{peq} \) time domain graph; spectrogram; envelope spectrum.  \hspace{1cm} (b)20% \( U_{peq} \) time domain graph; spectrogram; envelope spectrum.

As shown in the figure 4, \( F_i \) is the fault frequency of bearing inner ring, \( U_{peq} \) is the max
allowable unbalance. Through the rolling bearing 6204 with two unbalance degrees, the influence of unbalance degree on bearing vibration response is studied. It can be seen that the amplitude of the unbalanced fault frequency increases with the increase of the unbalanced degree, and the unbalanced degree has no obvious influence on the bearing fault frequency.

3. Experimental data analysis

In this part, a deep groove ball bearing of ER-12K is used for simulation and analysis, and the bearing parameters is listed in table 1. Via the fault simulation test bench of SQ Company, seven types of faults including one normal bearing, three bearing faults and three coupled faults are simulated respectively.

Table 1. Bearing parameters.

| Bearing brand | Bearing type | Number of balls | Ball diameter (in) | Pitch diameter (in) |
|---------------|--------------|-----------------|-------------------|-------------------|
| Rexnord       | ER-12K       | 8               | 0.3125            | 1.318             |

In this experiment, the rotor frequency and signal cut-off frequency are set as 40 Hz and 5000 HZ, and the number of spectral lines taken as 12800. The time-domain waveforms of three different coupled faults are shown in figure 5.

Figure 5. Coupling fault time domain waveform: (a) Unbalanced - bearing outer ring fault; (b) Unbalanced - bearing inner ring fault; (c) Unbalance-bearing rolling element fault.

Predictably, the vibration characteristic values of different fault types in the same bearing will be different. Calculate the root mean square value and kurtosis value of seven kinds of experimental data, and list the point line graph and histogram are shown in figure 6.

Figure 6. Comparison chart of characteristic values of various fault types.

Where NOR means normal, OUT means outer ring fault, INN means inner ring fault, ROLL means rolling element fault, UN-OUT means unbalance-outer ring coupling fault, UN-INN means unbalance-inner ring Coupling fault, UN-ROLL means unbalance-rolling element coupling fault.

It can be seen from figure 6(b)-(c) that with the coupling of unbalanced fault, the coupled RMS value is greater than the RMS value of bearing fault and the coupled kurtosis value is less
than the bearing fault kurtosis value.

3.1 Envelope analysis

Further study the mutual influence and vibration characteristics of unbalanced-bearing coupling faults, envelope analysis is used to process the experimental signals, and the single and coupled bearing fault envelope spectrum are shown in figure 7. In the test, the characteristic frequencies of bearing inner ring fault, outer ring fault and rolling element fault are calculated, and the results are as follows: 198Hz, 121.92Hz and 79.68Hz [9].

![Figure 7](image)

Figure 7. Single and coupled bearing fault envelope spectra: (a) Single and coupled bearing outer ring fault envelope spectra; (b) Single and coupled bearing inner ring fault envelope spectra; (c) Single and coupled bearing rolling element fault envelope spectra.

In figure 7, \( f_i \) is the fault frequency of the bearing inner ring, \( f_o \) is the fault frequency of the bearing outer ring, and \( f_r \) is the fault frequency of the bearing rolling elements. Comparing figure 7(a), figure 7(b) and figure 7(c), it can be seen that the unbalanced fault frequency (rotational frequency) is clearly visible. The coupled bearing fault frequency amplitude is higher than the amplitude in the single bearing fault envelope spectrum, and the sidebands of the unbalanced fault frequency are more obvious than the single bearing fault envelope spectrum.

3.2 Cepstrum analysis

In order to observe the periodic frequency components in the fault data more clearly, cepstrum analysis is further used to process the experimental signals, and the single and coupled bearing fault cepstrum are shown in figure 8.

![Figure 8](image)

Figure 8. Single and coupled bearing fault cepstrum: (a) Single and coupled bearing outer ring fault cepstrum; (b) Single and coupled bearing inner ring cepstrum; (c) Single and coupled bearing rolling element fault cepstrum.

As shown in figure 8, it can be clearly seen that the rotation frequency conversion component is 0.025s (40 Hz). Comparing and analyzing the cepstrum transformation of the experimental data, it is found that the bearing fault frequency in the coupled fault cepstrum is more obvious.
4. Conclusions
In order to reveal the coupling vibration characteristics from unbalance and bearing fault in rotor-bearing system, A dynamic model of the coupled fault is built, and it is further study by the vibration signals from experimental simulation. The conclusions of this paper include three aspects: 1) The change of the unbalance in the rotor unbalance-bearing coupling fault has little effect on the magnitude of the bearing fault frequency; 2) With the coupling of unbalanced fault, the kurtosis value of the vibration response decreases and the root mean square value increases; 3) With the coupling of unbalanced fault, both the bearing fault frequency amplitude and the sideband amplitude in the envelope spectrum increase, and the bearing fault frequency in the cepstrum is easier to find.

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