Active Magnetic Bearings for Rolling Element Bearings Outer Race Incipient Defect Diagnosis

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***Abstract*—The active health monitoring of rotordynamic systems in the presence of bearing outer race defect is considered in this paper. The shaft is assumed to be supported by conventional mechanical bearings and an active magnetic bearing (AMB) is used in the mid of the shaft or outboard location as an actuator to apply constant force to the system. We present a nonlinear bearing-pedestal system model with the outer race defect under the AMB force. The nonlinear differential equations are integrated using the fourth-order Runge—Kutta algorithm in MATLAB and the simulation result is obtained. The simulation results show that the characteristic outer race defect signal is significantly amplified under the AMB force, which is helpful to improve the diagnosis accuracy of outer race defect.**

I. INTRODUCTION

Rolling element bearing is one of the most common components in the rotating machinery. Its health state directly determines the rotating machinery performance. Many system models [1][2][10] have been created and adopted in the research process of the rolling bearing fault characteristics.

During the bearing incipient fault state, it is difficult to extract the defect characteristic signal because the malfunction itself is not significant and the incipient fault signal is overwhelmed by the random noise and vibration caused by other moving components. For a rolling bearing with zero contact angle, the varying compliance frequency (VC) in the vibration spectrum adversely caused by the time-varying ball position equals to the outer race fault characteristic frequency, which makes it more difficult to detect the incipient small outer race defect.

Active magnetic bearing (AMB) is mainly used to support any rotating element. In addition, AMB can be used as force actuators and sensor [3]. As a force actuator, its amplitude and frequency can be easily altered and it can apply online non-contact excitation force [4]. Nordmann using AMBs as force actuators to identify dynamic characteristics of rotor components [5,6]. Kasarda [7] proposed a method using AMB for the non-destructive evaluation of manufacturing process. Xiangming Lou using AMBs as force actuators to identify the shaft unbalance [4]. Zhu et al. studied the dynamics characteristics of a rotor with crack supporting by AMBs [8]. Sawicki [9] applies a sinusoidal force from the AMB to help identify crack in the rotor. However, the application of AMB as force actuators on the rolling bearing fault diagnosis is relatively rare.

Building on the Feng et al. [11] model, this paper develops a bearing-pedestal model under the AMB force and establishes the outer race defect model. Taking SKF 61901 bearing as an example, we analyze the characteristics of fault signal under the AMB force.

The remainder of the paper is organized as follows. Section 2 presents the model of rolling element bearing with outer race defect and the equation of motion under AMB force. Section 3 describes the simulation results. Conclusions are drawn in Section 4.

II. ROLLING ELEMENT BEARING MODELING

2.1 Contact force

In the rotating state, the rolling number and radial load force of the bearing lead to the stiffness changes periodically, resulting in the varying compliance (VC) vibration. Figure 1 shows the rolling element bearing schematic diagram.

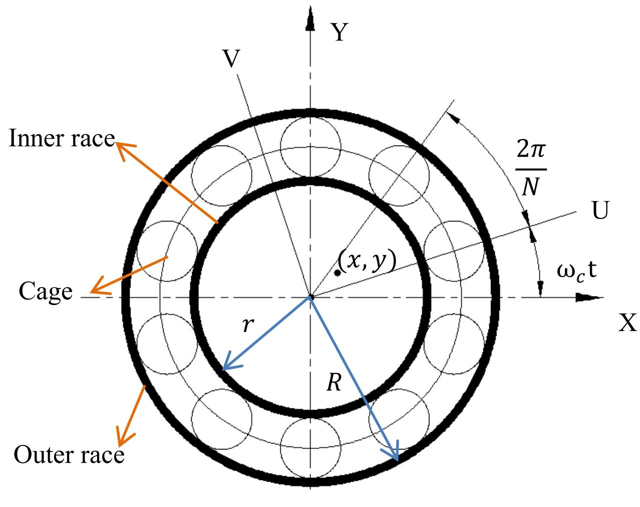

Figure 1. Schematic diagram of a rolling element bearing.

The outer race is fixed in the pedestal and the slippage of rolling elements is ignored. The displacement of the shaft can be divided into x and y and r_0 is the clearance between rolling elements and bearing races. The contact deformation for the ith rolling element δ_i is given by

$$\delta_i = x\cos\theta_i + y\sin\theta_i - r_0. \quad (1)$$

Here the angle position of the rolling element θ_i is a function of the time t, which is given as

$$\theta_i = \omega_c t + \frac{2\pi}{N}(i-1)$$
$$\omega_c = \frac{\omega_s r}{(R+r)}$$

where ω_s is the shaft speed; ω_c is the cage speed(can be calculated from geometry and the shaft speed ω_s assuming no slippage); r and R are the inner and outer race radius respectively; N is the number of rolling elements. The VC frequency is given as

$$\omega_{vc} = \frac{\omega_s r}{(R+r)} N = \omega_c B_N$$

The contact force F of the ball raceway is calculated using traditional Hertz theory (nonlinear stiffness) from

$$F = K\delta^n$$

where K is the contact stiffness depending on the bearing geometry and the elasticity of material; the exponent $n = 3/2$ for ball bearings. Accounting for the fact that F occurs only for positive values of δ, where " + " indicate that. The total contact forces in x and y direction can be calculated as follows:

$$F_{bx} = \sum_{i=1}^{N} K_b (x\cos\theta_i + y\sin\theta_i - r_0)_+^{3/2} \cos\theta_i,$$

$$F_{by} = \sum_{i=1}^{N} K_b (x\cos\theta_i + y\sin\theta_i - r_0)_+^{3/2} \sin\theta_i$$

2.2 *Outer race defect modeling*

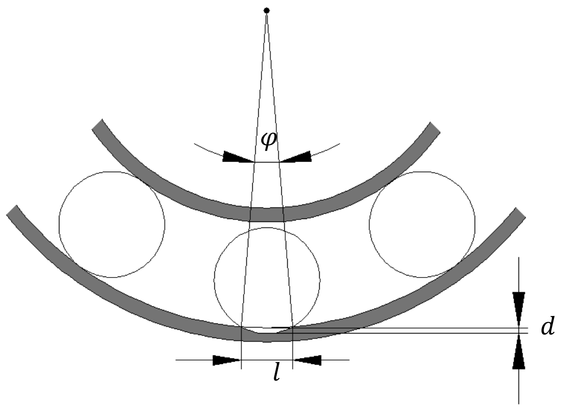

Figure 2. Defect on outer race.

The most failure feature of bearing is the local defect, such as fatigue spall and the corrosive pitting. The outer race defects can be modeled as a slight dent in Figure 2, the length of the dent is l. According to the geometry relationship

$$d = r_b - \sqrt{r_b{}^2 - (l/2)^2}, \quad (8)$$

$$\varphi = 2\arcsin(l/2R), \quad (9)$$

where r_b is the ball radius; φ is the central angle of the outer defect; d is the max increment of the radical clearance. When the ball rolling into the defect area, the radial clearance will increase suddenly, resulting in a decline of the contact force. The outer race defect angular position is assumed as $3\pi/2$ at the bottom of the outer race and the varying clearance Δd caused by the defect is modeled as a half sinusoidal wave, which is give as follows:

$$\Delta d = d\sin\left[\pi \frac{\mathrm{mod}(\theta_i,\ 2\pi) - \left(\frac{3\pi}{2} - \frac{\varphi}{2}\right)}{\varphi}\right]$$

$$\frac{3\pi}{2} - \frac{\varphi}{2} < \mathrm{mod}(\theta_i, 2\pi) < \frac{3\pi}{2} + \frac{\varphi}{2}$$

When the ball is located in the defected area, the total contact force can be calculated by

$$F_{bxd} = \sum_{i=1}^{N} K_b (x\cos\theta_i + y\sin\theta_i - (r_0 + \Delta d))_+^{3/2} \cos\theta_i$$

$$F_{byd} = \sum_{i=1}^{N} K_b (x\cos\theta_i + y\sin\theta_i - (r_0 + \Delta d))_+^{3/2} \sin\theta_i$$

2.3 *AMB force modeling*

The rotor AMB system contains a PID controller, sensors and power amplifiers. If only power amplifiers are activated and a specific signal is imported, the AMB can be converted to a force actuator. Ignoring the hysteresis loss and edge effect, the AMB force can be calculated by

$$f_{AMB} = \frac{1}{4}\mu_0 n^2 A \frac{i^2}{s^2}\cos\beta$$

Where μ_0 stands for the magnetic permeability of a vacuum; n is the windings of a coil; A is the cross section of the air gap; i is the current value; s is the length of air gap and β is the an angle between forces and magnetic poles.

2.4 *Bearing-pedestal modeling*

Here we adopt the model of Feng et al. [12] for studying the dynamics of rolling element bearings, which is symbolically represented in Figure 3. The model has four DOF, including two DOF of inner race (x_s, y_s) and two DOF of pedestal (x_p, y_p).

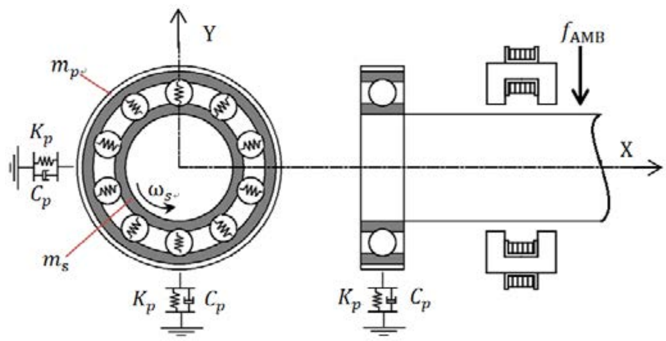

Figure 3. Four-DOF bearing-pedestal model.

If the rolling element is not located in the defect region, the actual radical clearance r_a is given by

$$r_a = r_0$$

If the rolling element is located in the defect region, the actual radical clearance r_a is given by

$$r_a = r_0 + \Delta d. \quad (16)$$

The total contact forces of the model can be calculated as follows:

$$f_{bx}=\sum_{i=1}^{N}K_b\left[\left(x_s-x_p\right)\cos\theta_i+\left(y_s-y_p\right)\sin\theta_i - r_a\right]_+^{3/2}\cos\theta_i$$

$$f_{by}=\sum_{i=1}^{N}K_b\left[\left(x_s-x_p\right)\cos\theta_i+\left(y_s-y_p\right)\sin\theta_i - r_a\right]_+^{3/2}\sin\theta_i$$

Taking into the nonlinear contact force, the governing equations of motion can be written as follows:

$$m_s\ddot{x}_s+c_s\dot{x}_s+f_{bx}=F_u\cos\omega_s t, \quad (19)$$

$$m_s\ddot{y}_s+c_s\dot{y}_s+f_{by}=F_u\sin\omega_s t-m_s g-f_{AMB}, \quad (20)$$

$$m_p\ddot{x}_p+c_p\dot{x}_p+K_p x_p-f_{bx}=0, \quad (21)$$

$$m_p\ddot{y}_p+c_p\dot{y}_p+K_p y_p-f_{by}=-m_p g, \quad (22)$$

where m_s is the mass of the shaft and the inner race; c_s is the bearing contact damping; m_p is the mass of the pedestal; K_p and c_p are the stiffness and damping of the pedestal. Constant AMB force f_{AMB} is applied in the middle of the shaft or outboard location in the y direction. F_u is the unbalance force which is zero for a balanced rotor.

III. RESULT

3.1 Inputs for the model

The nonlinear differential equations are evaluated numerically using the fourth-order Runge-Kutta algorithm in MATLAB and the transient response is obtained. In order to get the desired results, the inputs used are shown in Table I.

The dynamic response of a health bearing, fault bearing, health bearing under AMB force and fault bearing under AMB force are calculated respectively. And the response is analyzed between 0.7 and 0.8s.

TABLE I. INPUTS FOR THE MODEL

1. Parameters of SKF 61901 ball bearings	
Outer race radius R	9 mm
Inner race radius r	mm
Number of ball N	
Contact stiffness K_b	7×10^9 N/m$^{3/2}$
Ball diameter d	mm
Pitch diameter D	mm
Contact angle α	0°
Radical clearance r_0	µm
B_N	
2. Other inputs	
Mass of shaft/inner race m_s	1.2kg
Bearing contact damping c_s	3000N · s/m
Mass of pedestal m_p	0.2kg
Pedestal stiffness K_p	1.5×10^7N/m
Damping in pedestal c_p	3000N · s/m
AMB force f_{AMB}	60N

3.2 Outer race defect frequency under AMB force

Actually, even a perfect bearing will generate VC vibrations due to the time-varying distribution of the balls relative to the inner and outer races, which is shown in Figure 4. For a bearing with a stationary outer race, the outer race defect frequency is given by

$$f_o=\frac{1}{2}f_s\left(1-\frac{d}{D}cos\alpha\right)N$$

For a deep groove ball bearing, $f_o=f_{vc}$ accounting the zero contact angle α, which makes it more difficult to diagnose the outer race defect.

Figure 4 shows the simulation acceleration response of a healthy bearing at 3600 rpm in the y direction. It can be seen that even a perfect bearing will generate VC vibrations due to the time-varying distribution of the balls relative to the inner and outer races. But the AMB force has little influence on the vibration response for a healthy bearing.

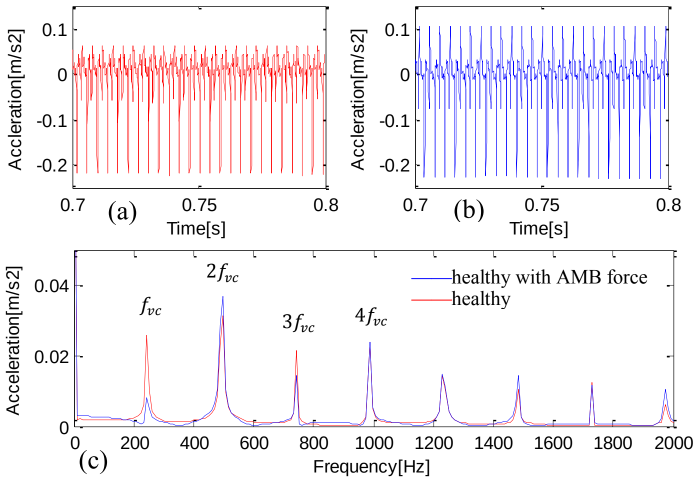

Figure 4. Vibration response of healthy bearing, at 3600 rpm: (a)healthy beaing without AMB force, (b) healthy bearing with AMB force and (c) Frequency spectrum of the envelope between 0.7 and 0.8s.

Figure 5 is the simulation results of a fault bearing with outer race defect at 3600rpm. It can be seen that for an incipient small defect, the amplitudes of defect characteristic frequency f_o and multiples are small in the frequency spectrum. However, there is significant increase in the amplitude at these frequencies under AMB force. The same conclusion can be obtained at 100 Hz (6000 rpm) in Figure 6.

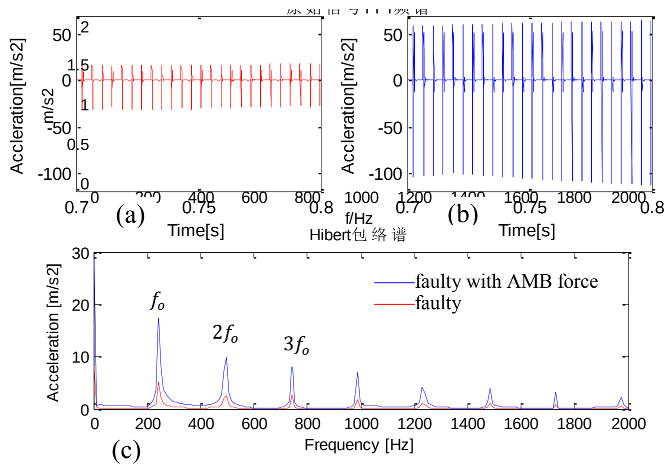

Figure 5. Vibration response of bearing with the outer race small defect, $l=0.2$mm, at 3600 rpm: (a)defect bearing without AMB force, (b) defetc

bearing with AMB force and (c) Frequency spectrum of the envelope between 0.7 and 0.8s.

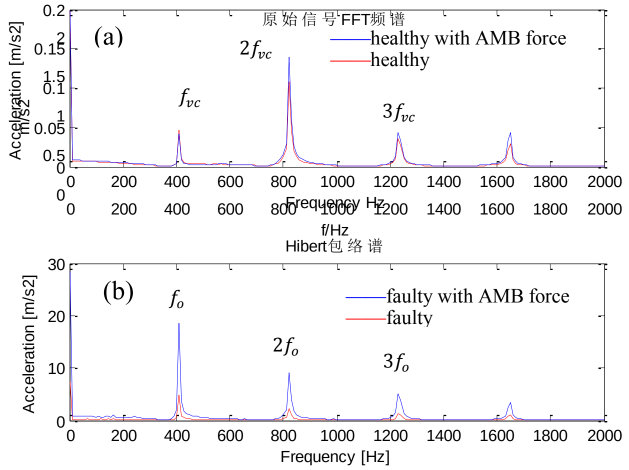

Figure 6. Frequency spectrum of the envelope between 0.7 and 0.8s, at 6000 rpm: (a) healthy bearing and (b) bearing with outer race defect; small defect $l = 0.2$mm.

3.3 *Effect of AMB force on the dynamic response*

The amplitude of AMB force has an effect on the fault characteristic frequency. Figure 7 shows the effect of the AMB force amplitude on the dynamic response of a bearing with the outer race defect. Figure 8 shows the defect characteristic frequency f_o trend along with AMB force. It can be seen that the amplitude of the characteristics defect frequency increases with the AMB force rising.

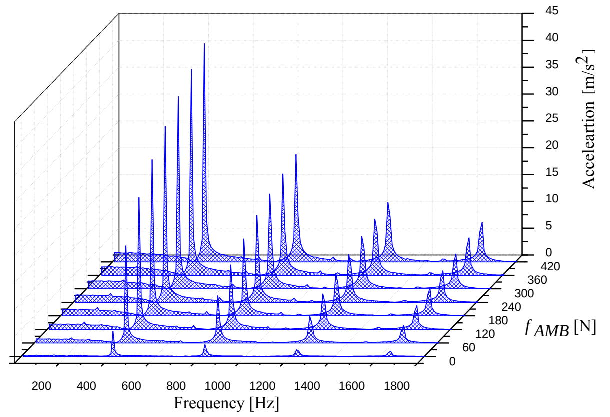

Figure 7. The effect of the AMB force amplitude on the dynamic response of bearing with the outer race defect, at 6000 rpm, small defect $l = 0.2$mm.

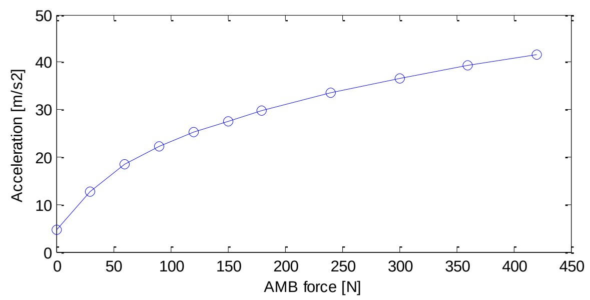

Figure 8. The effect of the AMB force amplitude on the outer race defect amplitude

IV. Conclusions

The simulation results show that AMBs can be used as an online force actuator to diagnose the bearing incipient outer race small defect. There is significant increase in the characteristic frequency amplitude with the increase in the AMB force, which is very helpful to improve the diagnosis accuracy. Furthermore, the effect of adding an extra force to the system might urge the defect growth, which is obviously a disadvantage. Current efforts are directed toward experimental verification of this work and the application of other bearing and shaft damage.

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