Application of 115kW blower supported by magnetic bearing

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Abstract

Because of its superior characteristics, magnetic bearing have been applied in many fields. The topic of this paper is the control system of magnetic bearing in blower application. Thermal expansion effects have the impact on measurement of axial displacement, we use two eddy current sensors which are in the radial symmetrical arrangement to achieve measurement of the axial displacement, and theoretical calculations and experiment verified the possibility of this method.In order to reduce vibration of the rotor, a chebyshev II notch filter is designed to suppress the special frequency, the effect of notch depth and bandwidth are discussed in the paper, through the rational allocation, the vibration of rotor are greatly suppressed; In order to improve the response of the rotor, phase compensation are used in this system, through the use of two phase compensation,the stability of the rotor rotation are improved. Simulation and experimental results show that the system has a good dynamic characteristics, the speed of the rotor is 15000rpm.

1 Introduction

In the past decade, active magnetic bearings provide unique features that are beyond the capability of any conventional bearings, as the development of this technology, magnetic bearings are used in more and more application areas, such as vacuum and cleanroom systems, machine tools, medical devices, turbo machinery, superconducting bearings [1]. In particular, with the requirement of speed and accuracy increasingly, magnetic bearing can achieve higher speed, which can offer higher power concentration. In large-scale equipment, some devices such as gearing, lubrication system can be omitted by using magnetic bearings. In blower application, by using magnetic bearings it can reduce the dimension and simplify the system equipment, so it will have important impact on miniaturization of industrial profound.

This paper focuses on the control of the magnetic bearing system, and solves the problems which emerged in the design. First, we build the model of the rotor, and use two eddy current sensors which are in the radial symmetrical arrangement to achieve measurement of axial displacement, second, design a Chebyshev II notch filter in the path through to suppress the vibration of the rotor, third, design phase compensation to improve the phase margin and the rotor response, finally, an experimental prototype are introduced, based on the simulation and experiments, the characteristics of the rotor are analyzed.

2 Model

The structure of the magnetic bearing rotor is shown in Figure 1.

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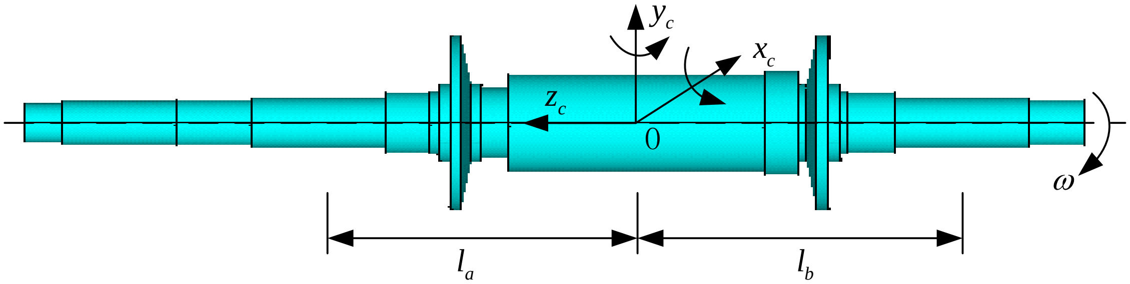

Figure 1: The structure of the magnetic bearing rotor

The assumptions are as following:
(1) The rotor is rigid and axial symmetry
(2) The structure and parameters are the same in radial
(3) Ignore the coupling of degree of freedom between radial and axial

According to the theorem of momentum and angular momentum theorem, we get the following equation of the motion of rotor.

$$\begin{cases} m\ddot{x}_c = F_{xa} + F_{xb} \\ m\ddot{y}_c = F_{ya} + F_{yb} \\ J_x\ddot{\theta}_x + \omega J_z\dot{\theta}_y = F_{ya}l_a - F_{yb}l_b \\ J_y\ddot{\theta}_y - \omega J_z\dot{\theta}_x = F_{xb}l_b - F_{xa}l_a \\ m\ddot{z}_c = F_z \end{cases} \tag{1}$$

$$\begin{cases} F_{xa} = K_{xx}{}^{a}\cdot x_a + K_{xi}{}^{a}\cdot i_{xa} \\ F_{xb} = K_{xx}{}^{b}\cdot x_b + K_{xi}{}^{b}\cdot i_{xb} \\ F_{ya} = K_{yy}{}^{a}\cdot y_a + K_{yi}{}^{a}\cdot i_{ya} \\ F_{yb} = K_{yy}{}^{b}\cdot y_b + K_{yi}{}^{b}\cdot i_{yb} \\ F_{zc} = K_{zz}\cdot z_c + K_{zi}\cdot i_{zc} \end{cases} \qquad \begin{cases} x_c = \dfrac{l_b}{l}x_a + \dfrac{l_a}{l}x_b \\ y_c = \dfrac{l_b}{l}y_a + \dfrac{l_a}{l}y_b \\ \theta_x = \dfrac{1}{l}y_a - \dfrac{1}{l}y_b \\ \theta_y = -\dfrac{1}{l}x_a + \dfrac{1}{l}x_b \\ l = l_a + l_b \end{cases}$$

Where m is the rotor mass, F_{xa}, F_{xb}, F_{ya}, F_{yb} and F_z are the bearing forces in X, Y and Z directions, i_{xa}, i_{xb}, i_{ya}, i_{yb} and i_{zc} are the control currents, K_{xx}^{a}, K_{xx}^{b}, K_{yy}^{a}, K_{yy}^{b} and K_{zz} are the force-displacement coefficients, x_a, x_b, y_a and y_b are the radial displacement, l is the distance between the two radial bearings, ω is the rotational speed, θ_x and θ_y are the angular motion around the Y and X axis, and O is the mass center of the rotor,

3 Measurement of axial displacement

Generally, magnetic bearings are consist of sensors, controller, power amplifier and electromagnetic coil, in order to maintain a rotor stable, it needs five degrees of freedom to be constrained, and another freedom is constrained by the motor, so five displacement sensors are needed to measure the axial displacement to achieve the active control. But due to the eddy current, harmonics, and other factors [2-3], it can easily lead to the thermal expansion for the rotor , this will lead to the deviation of the axial displacement measurement, and make the control system unstable. So we install two eddy current sensors in radial direction, and arrang them symmetrically, the installation diagram of the eddy current sensors are shown in Figure 2.

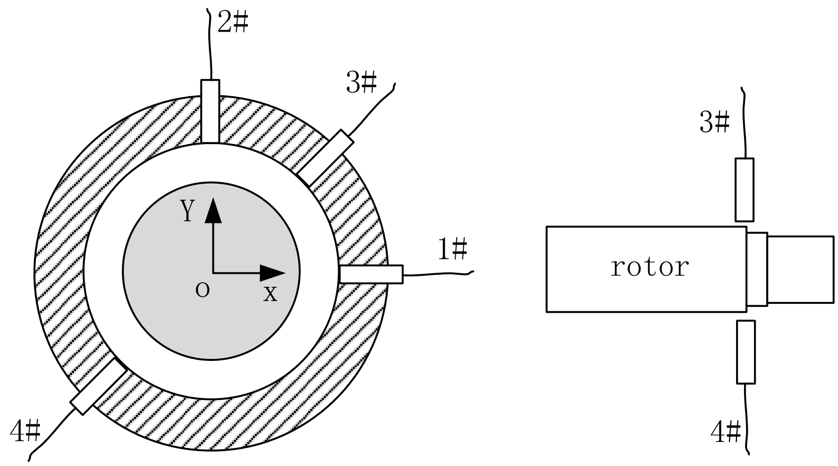

Figure 2: Axial displacement measurement of magnetic bearing

1# and 2# are the probes of the eddy current sensors which measure X and Y direction separately, 3# and 4# are the probes which arrange symmetrically in the radial. The relationship between output voltage and displacement can be written as:

$$U = k \cdot d + U_0 \tag{2}$$

Where U is the output voltage of the sensor, k is the sensitivity of the sensor, d is the distance between the probe and the object, U_0 is the constant bias.

When the rotor deviates from the equilibrium position, the output voltage of 3# and 4# can be expressed as [3]

$$\begin{cases} U_3 = \left[(c_z + c_x) d_1 + k_{x0} \right] x + k_{z0} d_1 + U_0 \\ U_4 = \left[(c_z + c_x) d_2 + k_{x0} \right] x + k_{z0} d_2 + U_0 \end{cases} \tag{3}$$

Where U_3 and U_4 are the output voltage of the 3# and 4# eddy current sensors separately, c_x is the couple coefficient of radial sensitivity and axial displacement, c_z is the couple coefficient of axial sensitivity and radial displacement, d_1 and d_2 are the displacement between the probe of sensors and rotor's surface, x is the displacement of axial.

So the axial displacement can be written as:

$$U = U_3 + U_4 = C_1 x + C_2 \tag{4}$$

Where $C_1 = 2\left[(c_z + c_x) d + k_{x0} \right]$, $C_2 = 2k_{z0} d + 2U_0$, $d = d_1 + d_2 =$ constant

By decoupling, the output voltage of the two sensors are become one order function for the axial displacement, and don’t consist radial displacement, which realize the measurement of axial displacement.

In this section, experimental results are presented to verify the measurement of axial displacement. First, have a stable rotor suspension after different value of axial position is set. Then record the axial displacement directly by dial gauge, meanwhile, record the output voltage of two axial eddy current sensors. The method of axial displacement decoupling is used as mention above. The relationship between output voltage and the axial displacement are shown in Figure 3. After decoupling, the output voltage and the axial displacement are linear relationship, which verify the correction of Equation (4). Figure 4 is the step response of axial sensor.

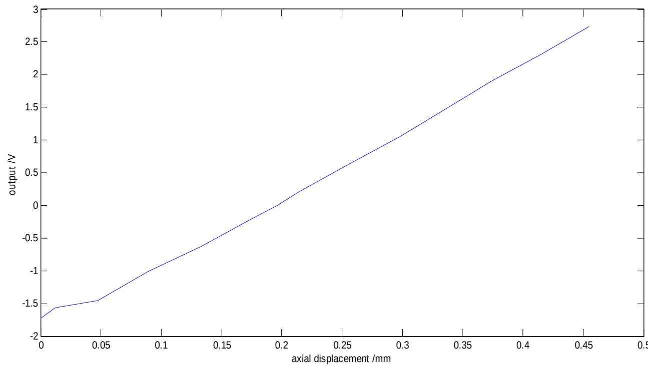

Figure 3: Curve of output voltage and axial displacement

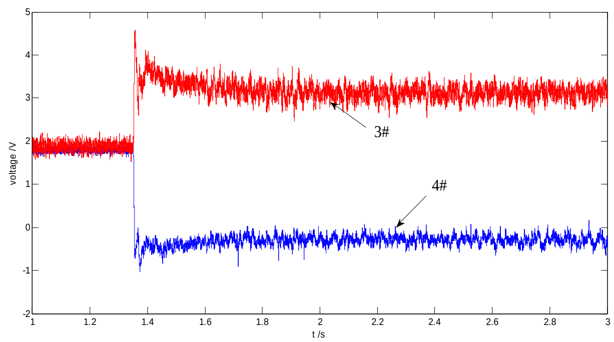

Figure 4: Step response curve of the rotor

4 Notch filter and phase compensation

As the factor of rotor manufacturing and assembly, it is unavoidable to encounter the vibration of the rotor in the natural frequency. Vibration in the natural frequency will increase the amplitude of the displacement, and will affect control accuracy of the rotor. When the vibration amplitude exceeds a certain value, it will lead to saturation of power amplifier, and cause nonlinear phenomenon, even worse, it may cause serious unstable in the rotating at high speed, so it is necessary to suppress the vibration of the rotor. A notch filter is designed to the control loop in series, it can reduce the amplitude of specific frequencies which can reduce the vibration of the rotor. We use a chebyshev II notch filter, as it doesn't have fluctuation in the passband and has a high decay rate. Discrete chebyshev type II notch filter can be expressed as

$$H(z)=\frac{B(z)}{A(z)}=\frac{b_1+b_2z^{-1}+...+b_{n+1}z^{-n}}{a_1+a_2z^{-1}+...+a_{n+1}z^{-n}} \tag{5}$$

The parameters of notch filter are notch depth and notch frequency, notch depth mainly determined by the transfer function poles. If the poles get closer to the unit circle, the amplitude of the frequency response will get deeper. Figure 5 shows a second order notch filter with different notch depth, when the depth of the notch filter is deeper, the effect of the suppress is better, but this will have a great impact on the phase which is near the notch frequency, that is, it will decrease the stability of the control system, and lead to unstable when the notch frequency is not set correctly..

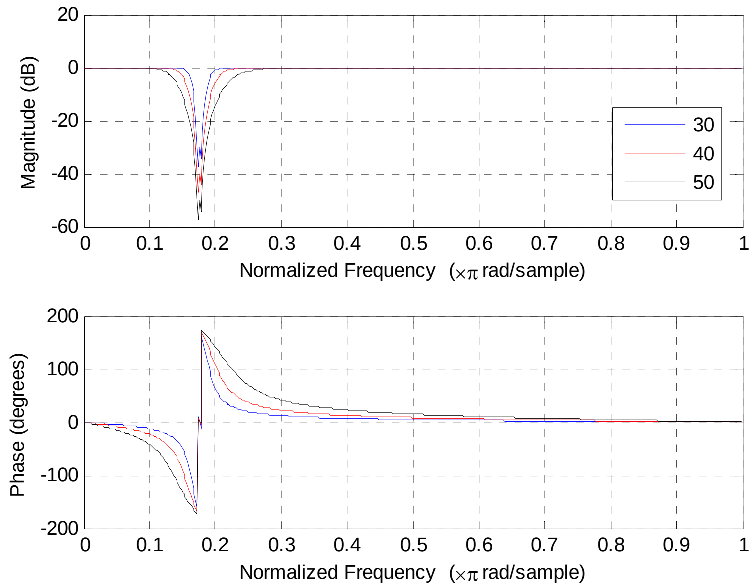

Figure 5: Bode plot with different notch depth

Figure 6 represents performance of notch filter on vibration suppression, blue line is the curve without notch filter, it can be seen that a vibration frequency is at 830Hz, in the time domain, it increases the amplitude of the dispacement, after using notch filter, the frequency component in 830Hz is greatly suppressed.

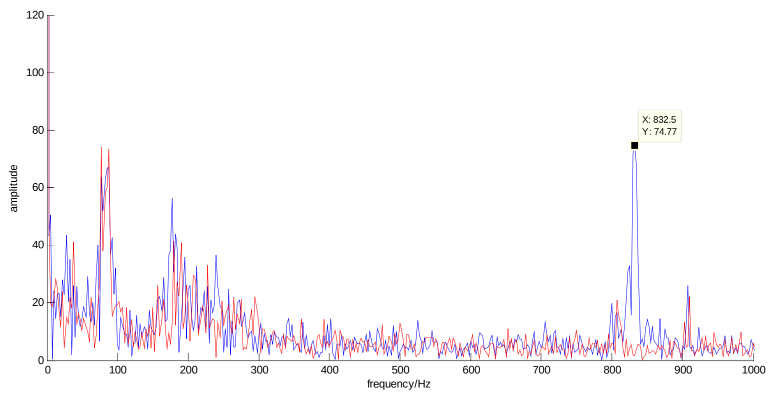

Figure 6: Frequency spectrum of compensation of notch filter

Additionally, in order to improve the stability of the control system, the phase compensation is designed on the critical frequency, the phase compensation can supply a 90 degree lead. Figure 7 shows the bode plot of phase compensation. By using phase compensation, the gain before the frequency compensation is lower, so when using the phase compensation, the gain of the system should be increased correspondingly.

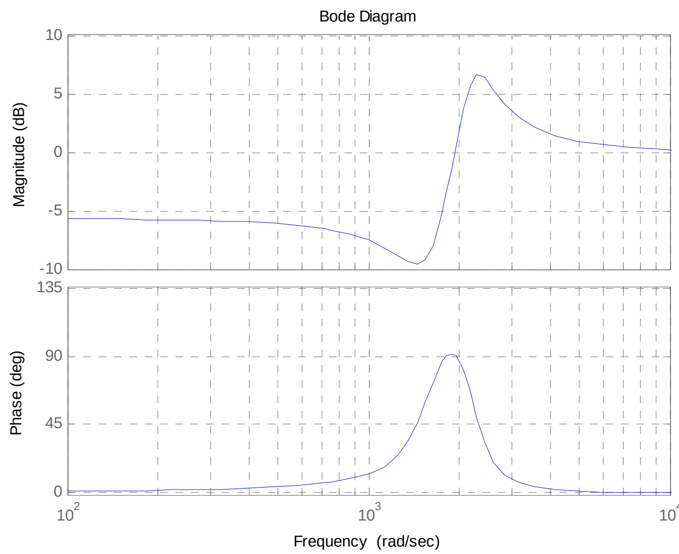

Figure 7: Bode plot of phase compensation

5 Prototype test

Figure 8 shows the prototype of magnetic bearing blower, and the relevant parameters are shown in Table 1.

parameters	radial	axial
Air gap [mm]	0.3	1.0
Bias current[A]	3.5	2.0
Force-current-coefficients[N/A]	397.93	437.51
Force-displacement-coefficients [N/m]	-4.64×10^{6}	-8.75×10^{5}
Mass of rotor[kg]	45	

Table 1: Related parameters of magnetic suspension blower

Figure 8: View of magnetic suspension blower prototype

In the system, there are six eddy current displacement sensors, with the sensitivity coefficient of 10V/mm, power amplifier provids ten channel output current with the maximum current of 8 A, the bus voltage is 311V, the DC gain of the switch power amplifier is 0.8A/V. The system uses DSP as a master controller with a sampling rate of 10KHz, a decentralized PID control strategy is applied in the displacement loop, before the PID controller, a notch filter is added, with a notch frequency of 830Hz, and two phase compensations are applied, the frequency of phase compensation are 220Hz and 460Hz. Asynchronous motor is applied to the magnetic bearing, which is driven by inverter. One degree of freedom model is validated via MATLAB. Figure 9 shows the simulation results of the closed-loop system for one axis.

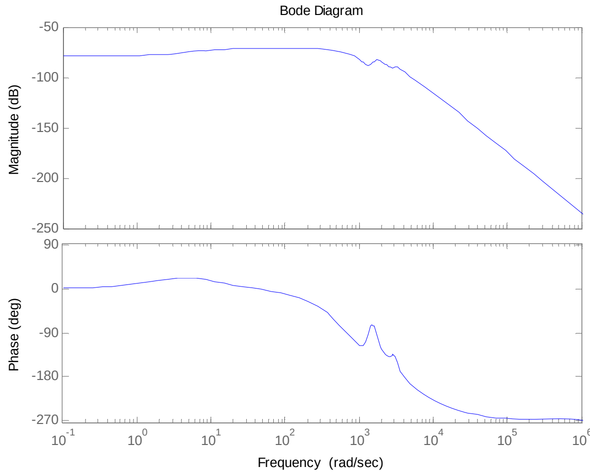

Figure 9: Bode plot for one axis

We carried out the speed up experiment, and measured four radial displacement of the rotor, Figure 10 shows the spectrum analysis of 1# sensor output voltage at 250Hz.

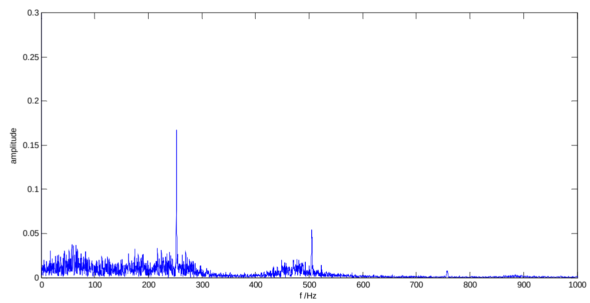

Figure 10: Spectrum of rotor displacement (250Hz)

6 Conclusion

This paper describes the control system of magnetic bearing blower. Axial displacement is measured by using two eddy current sensors which arrange symmetrically, experimental results show that this method can achieve good performance in certain distance.Then a notch filter and two phase compensation are designed to suppress the rotor vibration, which improve the control accuracy. Finally, the rotation of the rotor is up to 15000rpm and has a good performance. However, noise is a problem which can not be ignored. In the future, we will focus on suppressing noise of the control system.

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