Design of a Variable Structure Controller Based on the Force Estimator for a Single Active Magnetic Bearings Suspended Rotor System

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Abstract

This study proposes a variable structure controller with a force estimator for a single active magnetic bearing suspended rotor system to eliminate the vibration associated with unbalance of mass. The model of a single active magnetic bearing suspended rotor system is build and a displacement and unbalancing force estimator of the system is established. A variable structure controller is also established according to the estimated force to decrease the influence of the unbalance force and increase the robust of the system. The experimental results showed that thevariable structure controller based on the force estimator for a single active magnetic bearing suspended rotor system was established successfully and has good accuracy.

1 Introduction

Active magnetic bearing (AMB) is the essential part of the non-contact rotor systems and have been widely developed in industrial application, i.e., the spindle of machine tool, the flywheel energy store system, and the turbo-molecular pump, [1-4] etc. In such practical applications, they are usually seen that the unbalance forces or external forces disturbing at the magnetic suspension system; consequently, increasing the robustness of magnetic system is needed. In addition to these, magnetic suspension systems are rather complicated since their uncertainty on the physical parameters of the systems. However, so far as mathematical is concerned, the relationship between magnetic force, current and air gap of magnetic suspensions are highly nonlinear. Thus the linearized controller design technology would not work well for all of the operational range.

As mentioned above, several nonlinear control techniques have been proposed for AMB systems including sliding mode [5], feedback linearization [6], and hybrid control [7] in recent years. All designed to improve their disturbance rejection properties and robustness in terms of un-modeled dynamics and parameter uncertainties. Furthermore, there is one further problem with rotor unbalancing in an AMB system that we must not ignore, that is it appears as synchronous rotor displacement as well as synchronous transmitted force. Various methods to solve the problem of unbalanced vibration have been discussed. Chen and Lewis [8] combined an acceleration estimator with a proportional-derivative controller to suppress the vibration caused by unbalanced forces. Higuchi, Otsuka, and Mizuno [9] proposed a periodic learning control which utilized the period of oscillation and the characteristics of the system to identify the unbalancing force and reduce the vibration. Model based controllers are also sometimes used in AMBs, although a reliable model is not always known for all operating conditions [10, 11]. Methods including acceleration estimator and model based observer designs are frequency dependent. Lum, Coppola, and Bernstein [12] proposed an adaptive auto centering approach that was frequency independent, and compensated for transmitted

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forces that occurred due to unbalance in an AMB system. They showed that the adaptive auto centering control objective is equivalent to the attenuation of synchronous rotor vibration caused by mass unbalance.
In this paper, we propose a robust variable structure controller for a horizontal rotor system with active magnetic bearings to increase the robustness of system. Moreover, a model reference disturbance compensator is added to overcome the influence of the unbalance force. It should be hoped, from what has been said above, that the rotor system with active magnetic bearings will decrease the influence of unbalance or external disturbance, overcome the uncertainty on the physical parameters and work in wide operational rang with the variable structure controller. Experimental results serve as evidence of the proposed rotor system with high robustness and disturbing resistibility.

2 MODELING OF A SINGLE MAGNETIC BEARING SUSPENDED ROTOR SYSTEM

Fig.1 shows the basic structure of the horizontal rotor system used in this study. The rotor is suspended horizontally by an electric magnetic bearing on the free end and the other end is connected to an induction motor by a flexible coupling. The shaft displacements in both of the horizontal and vertical directions at the magnetic bearing location are measured by two eddy-current type sensors. The spinning shaft is driven by an induction motor. Fig.2 shows the relations between the position of the coupling, magnetic bearing and the mass center of rotor, where O is the mass center of the rotor, d is the distance of the center and a mass disk. Fig.2 also shows the directions of the four degrees of freedoms of the rotor and the action positions and directions of the magnetic forces (F_1 ~F_4) and the reacted forces (F_{xt}, F_{yt}) of the flexible coupling.

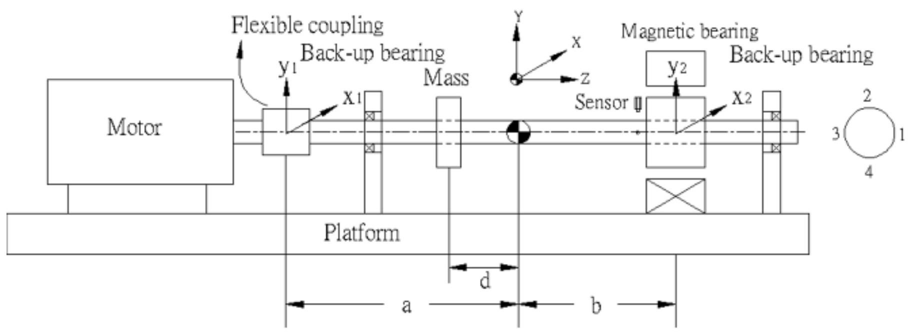


Fig.1 Structure of the single AMB system

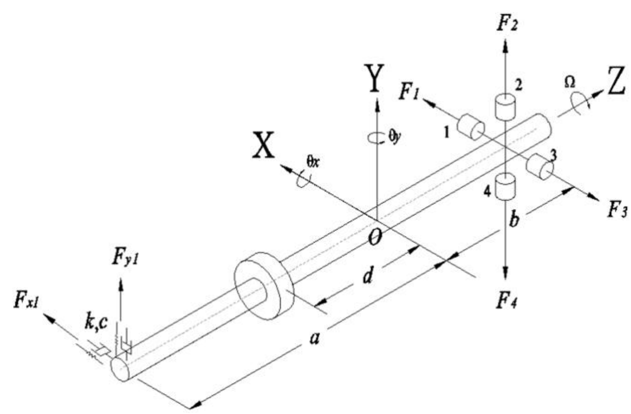


Fig.2 Geometric relationship of the single AMB system

2.1 Modeling of AMB system

The rotor is assumed to be rigid and symmetrical with uniform mass unbalance. The dynamic equations describing the rotor bearing system about the mass center are

$$
\begin{aligned}
&m\ddot{x} = m_p r\Omega^2 \cos(\Omega t + \theta) + F_{x1} + F_1 - F_3, \\
&m\ddot{y} = m_p r\Omega^2 \sin(\Omega t + \theta) - mg + F_{y1} + F_2 - F_4, \\
&I\ddot{\theta}_y - \Omega I_p \dot{\theta}_x = -dm_p r\Omega^2 \cos(\Omega t + \theta) - aF_{x1} + b(F_1 - F_3), \\
&I\ddot{\theta}_x + \Omega I_p \dot{\theta}_y = dm_p r\Omega^2 \sin(\Omega t + \theta) + aF_{y1} - b(F_2 - F_4),
\end{aligned} \tag{1}
$$

where m is the mass of the rotating shaft; $m_p$is the mass unbalance appearing at the rotating disk; θ is the initial angle of the unbalanced mass measured from the X axis; Ω is the speed of rotation around the spinning Z axis;x, y, θ_x, and θ_y are the radial displacements and rotating displacements of the mass center, respectively; I and I_p are the transverse and polar mass moment of inertia of the rotor; F_{x1} and F_{y1} are the coupling forces; x_1 and y_1 are the shaft displacements corresponding to the X and Y axesat the flexible coupling; and x_2 and y_2 are the shaft displacements at the magnetic bearing.

According to [13] and neglecting the effect of rotation, the flexible coupling forces corresponding to the X and Y axes can be expressed as follows:

$$
\begin{aligned}
&F_{x1} = -c_T \dot{x}_1 - k_T x_1, \\
&F_{y1} = -c_T \dot{y}_1 - k_T y_1,
\end{aligned} \tag{2}
$$

wherec_T is the equivalent damping and$k_T$is the equivalent stiffness of the coupling.

The magnetic forces provided by the electric magnetic bearing are functions of the width of the magnetic gap and the current driving the electromagnets [14]. Thus, the four magnetic forces can be written as a function of the driving currents and variations in the magnetic gap.

$$
F_n = f_0 + k_i i_n + k_d d_n + f(i_n, d_n), n = 1, 2, \dots, 4, \tag{3}
$$

where f_0 is the static magnetic force when the drivingcurrent is $i_n = 0$ and the magnetic gap variation is $d_n = 0$; k_d and $k_i$are the force-displacement stiffness factor and the force-current stiffness factor; and $f(i_n, d_n)$ is a high order term for the magnetic forces due to the coil currents and magnetic gap variations.

Let the four electromagnets have the same static magnetic force f_0 and the same coefficients k_d and k_i, then the four magnetic forces F_1 to F_4 can now be rewritten as

$$
\begin{aligned}
&F_1 - F_3 = k_i i_1 + k_d x_2 + f(i_1, x_2), \\
&F_2 - F_4 = k_i i_2 + k_d y_2 + f(i_2, y_2).
\end{aligned} \tag{4}
$$

For simplicity, the system equations for the designed controller are displacements in the locations of the flexible coupling and magnetic bearing. Since the rotor is assumed to be rigid and the displacement from the desired position is assumed to be small, the relationships between the shaft positions (x_1, x_2, y_1, y_2) and the mass center $(x, y, \theta_x, \theta_y)$can be shown as

$$
\begin{aligned}
&x = \frac{bx_1 + ax_2}{a + b} = \frac{bx_1 + ax_2}{L}, \\
&y = \frac{by_1 + ay_2}{a + b} = \frac{by_1 + ay_2}{L}, \\
&\theta_x = \frac{y_1 - y_2}{a + b} = \frac{y_1 - y_2}{L}, \\
&\theta_y = \frac{x_2 - x_1}{a + b} = \frac{x_2 - x_1}{L},
\end{aligned} \tag{5}
$$

where a is the distance between the flexible coupling and the mass center, b is the distance between the magnetic bearing and the mass center, and $L = a + b$.

Thus, the dynamics of the system can be rearranged in matrix form as follows:

$$
\begin{aligned}
&\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}
\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{y}_1 \\ \ddot{y}_2 \end{bmatrix}
+ \begin{bmatrix} c_T\beta_3 & 0 & \alpha_1 & -\alpha_1 \\ -c_T\beta_2 & 0 & -\alpha_2 & \alpha_2 \\ -\alpha_1 & \alpha_1 & c_T\beta_3 & 0 \\ \alpha_2 & -\alpha_2 & -c_T\beta_2 & 0 \end{bmatrix}
\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{y}_1 \\ \dot{y}_2 \end{bmatrix} + \\
&\begin{bmatrix} k_T\beta_3 & k_d\beta_2 & 0 & 0 \\ -k_T\beta_2 & -k_d\beta_1 & 0 & 0 \\ 0 & 0 & k_T\beta_3 & k_d\beta_2 \\ 0 & 0 & -k_T\beta_2 & -k_d\beta_1 \end{bmatrix}
\begin{bmatrix} x_1 \\ x_2 \\ y_1 \\ y_2 \end{bmatrix} = \\
&\begin{bmatrix} -k_i\beta_2 & 0 \\ k_i\beta_1 & 0 \\ 0 & -k_i\beta_2 \\ 0 & k_i\beta_1 \end{bmatrix}
\begin{bmatrix} i_1 \\ i_2 \end{bmatrix}
+ \begin{bmatrix} \gamma_2 & 0 & -\beta_2 & 0 \\ -\gamma_1 & 0 & \beta_1 & 0 \\ 0 & \gamma_2 & 0 & -\beta_2 \\ 0 & -\gamma_1 & 0 & \beta_1 \end{bmatrix}
\begin{bmatrix} m_p r\Omega^2 \cos(\Omega t + \theta) \\ m_p r\Omega^2 \sin(\Omega t + \theta) \\ f(i_1, x_2) \\ f(i_2, y_2) \end{bmatrix}
+ \begin{bmatrix} 0 \\ 0 \\ -1 \\ -1 \end{bmatrix} g,
\end{aligned} \tag{6a}
$$

where $\alpha_1 = \frac{I_p\Omega a}{IL}$,$\alpha_2 = \frac{I_p\Omega b}{IL}$,$\beta_1 = \frac{b^2}{I} + \frac{1}{m}$,$\beta_2 = \frac{ab}{I} - \frac{1}{m}$, $\beta_3 = \frac{a^2}{I} + \frac{1}{m}$, $\gamma_1 = \frac{bd}{I} - \frac{1}{m}$, and $\gamma_2 = \frac{ad}{I} + \frac{1}{m}$. Eq. (6) can be expressed as

$$\boldsymbol{M}\ddot{\underline{x}} + \boldsymbol{C}\dot{\underline{x}} + \boldsymbol{K}\underline{x} = \boldsymbol{B}\underline{u} + \boldsymbol{E}\underline{w} + \boldsymbol{D}g, \tag{6b}$$

where$\underline{x} = [x_1 \quad x_2 \quad y_1 \quad y_2]^T$ is the state vector, $\underline{u} = [i_1 \quad i_2]^T$ is the input vector, and $\underline{w} = [m_p r\Omega^2 \cos(\Omega t + \theta) \quad m_p r\Omega^2 \sin(\Omega t + \theta) \quad f(i_1, x_2) \quad f(i_2, y_2)]^T$ is the vector of the disturbance forces.

2.2 Model-based unbalanced forces observer

Here, we describe a decentralized force estimator for compensating disturbance forces. Considering Eq. (6a), the dynamics of the suspended magnetic part can be rearranged as follows:

$$\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}\begin{bmatrix} \ddot{x}_2 \\ \ddot{y}_2 \end{bmatrix} + \begin{bmatrix} -k_d\beta_1 & 0 \\ 0 & -k_d\beta_1 \end{bmatrix}\begin{bmatrix} x_2 \\ y_2 \end{bmatrix} = \begin{bmatrix} k_i\beta_1 & 0 \\ 0 & k_i\beta_1 \end{bmatrix}\begin{bmatrix} i_1 \\ i_2 \end{bmatrix} + \begin{bmatrix} dx_2 \\ dy_2 \end{bmatrix}, \tag{7}$$

where dx_2 and dy_2 are the sum of the disturbance forces, including unbalanced forces, force of gravity, coupling forces, and gyroscopic forces for both x and y directions.

The sum of disturbance forces can be expressed as

$$\begin{aligned} dx_2 &= -\gamma_1 m_p r\Omega^2 \cos(\Omega t + \theta) + c_T\beta_2\dot{x}_1 + \alpha_2\dot{y}_1 + k_T\beta_2 x_1 - \alpha_2\dot{y}_2, \\ dy_2 &= -\gamma_1 m_p r\Omega^2 \sin(\Omega t + \theta) - \alpha_2\dot{x}_1 + c_T\beta_2\dot{y}_1 + k_T\beta_2 y_1 + \alpha_2\dot{x}_2 - g. \end{aligned} \tag{8}$$

It is obvious from Eq. (7) that the system can be separated into two similar sub-systems. Hence, we design a force estimator for y_2 that is also suitable for x_2. The dynamic equation ofy_2 can be expressed as

$$\begin{aligned} \begin{bmatrix} \dot{y}_2 \\ \ddot{y}_2 \end{bmatrix} &= \begin{bmatrix} 0 & 1 \\ k_d\beta_1 & 0 \end{bmatrix}\begin{bmatrix} y_2 \\ \dot{y}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ k_i\beta_1 \end{bmatrix} i_2 + \begin{bmatrix} 0 \\ 1 \end{bmatrix} dy_2 = \bar{\boldsymbol{A}}\underline{y_2} + \bar{\boldsymbol{B}}i_2 + \bar{\boldsymbol{D}}dy_2, \\ y_{y2} &= [g_s \quad 0]\begin{bmatrix} y_2 \\ \dot{y}_2 \end{bmatrix} = \bar{\boldsymbol{C}}\underline{y_2}, \end{aligned} \tag{9}$$

Where $\underline{y_2} = [y_2 \quad \dot{y}_2]^T$ and g_s is the gain of the position sensor. We can calculate the rank of the observable matrix as

$$\operatorname{rank}(\boldsymbol{V}) = \operatorname{rank}\left(\begin{bmatrix} \bar{\boldsymbol{C}} \\ \overline{\boldsymbol{CA}} \end{bmatrix}\right) = \operatorname{rank}\left(\begin{bmatrix} g_s & 0 \\ 0 & g_s \end{bmatrix}\right) = 2. \tag{10}$$

The system is thus a fully observable system. The Luenberger state estimator is as

$$\begin{aligned} \begin{bmatrix} \dot{\hat{y}}_2 \\ \ddot{\hat{y}}_2 \end{bmatrix} &= \begin{bmatrix} 0 & 1 \\ k_d\beta_1 & 0 \end{bmatrix}\begin{bmatrix} \hat{y}_2 \\ \dot{\hat{y}}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ k_i\beta_1 \end{bmatrix} i_2 + \begin{bmatrix} l_1 \\ l_2 \end{bmatrix}(y_{y2} - \hat{y}_{y2}) \\ &= \bar{\boldsymbol{A}}\underline{\hat{y}_2} + \bar{\boldsymbol{B}}i_2 + \bar{\boldsymbol{L}}(y_{y2} - \hat{y}_{y2}), \\ \hat{y}_{y2} &= [g_s \quad 0]\begin{bmatrix} \hat{y}_2 \\ \dot{\hat{y}}_2 \end{bmatrix} = \bar{\boldsymbol{C}}\underline{\hat{y}_2}. \end{aligned} \tag{11}$$

If there is no disturbance terndy_2, the Luenberger observer makes the observed error decay to zero. In other words, we can stabilize the estimated system and make the observed displacement and velocity of the rotor approach that of a real system via suitable gains of $\bar{\boldsymbol{L}}$. Hence, the disturbance term exists all the time. The observed error is affected by the disturbance term of dy_2, leading to variation in the estimated output. In other words, the variation in theobserved output is a measurement criterion for the disturbance force dy_2.Thus if we set the error integral term $d\hat{y}_2$ to be

$$d\hat{y}_2 = g_s l_d \int (y_2 - \hat{y}_2)\, dt, \tag{12}$$

then insert this into the Luenberger observer. The observer can be rewritten as

$$\begin{aligned} \begin{bmatrix} \dot{\hat{y}}_2 \\ \ddot{\hat{y}}_2 \\ d\dot{\hat{y}}_2 \end{bmatrix} &= \begin{bmatrix} 0 & 1 & 0 \\ k_d\beta_1 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix}\begin{bmatrix} \hat{y}_2 \\ \dot{\hat{y}}_2 \\ d\hat{y}_2 \end{bmatrix} + \begin{bmatrix} 0 \\ k_i\beta_1 \\ 0 \end{bmatrix} i_2 + \begin{bmatrix} l_1 \\ l_2 \\ l_d \end{bmatrix}(y_{y2} - \hat{y}_{y2}), \\ \hat{y}_{y2} &= [g_s \quad 0 \quad 0]\begin{bmatrix} \hat{y}_2 \\ \dot{\hat{y}}_2 \\ d\hat{y}_2 \end{bmatrix}. \end{aligned} \tag{13}$$

The dynamic equation for error can now be expressed as

$$= \begin{bmatrix} -g_s l_1 & 1 & 0 \\ -g_s l_2 + k_d \beta_1 & 0 & 1 \\ -g_s l_d & 0 & 0 \end{bmatrix} \underline{e} + \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} d\dot{y}_2, \tag{14}$$

where $\underline{\dot{e}} = [e_{y2} \quad \dot{e}_{y2} \quad dy_2 - d\hat{y}_2]^T$ and $e_{y2} = y_2 - \hat{y}_2$.

The characteristic equation of Eq. (19) is

$$s^3 + g_s l_1 s^2 + (g_s l_2 - k_d \beta_1) s + g_s l_d = 0. \tag{15}$$

By the Routh-Hurwitz stability criterion, the stability conditions are

$$\begin{aligned} & g_s l_1 > 0, \\ & g_s l_2 - k_d \beta_1 > 0, \\ & g_s l_1 (g_s l_2 - k_d \beta_1) > g_s l_d > 0. \end{aligned} \tag{16}$$

Hence, suitable gains are selected according to Eq. (16) to make the observer stable. We can also design an observer for $x_2$to make both observers stable and feedback to the system.

Since the force observer design is based on the dynamic equations, and there are some model uncertainties of the flexible coupling and the nonlinearities of the magnetic bearing system, the observer would not work well for all the operation speed. And the phase of estimated force signal may be lag and the amplitude may be smaller than exact disturbance. But the estimated force is nearly. Thus, we can design a variable structure controller with a disturbance compensator to reduce the influence of the unbalance force by a small discontinuous input to reduce the chatter of the variable structure control system.

3 DESIGN OF the VARIABLE STRUCTURE CONTROLLER

In this paper, a variable structure controller is proposed to increase the robustness and to overcome the influence of uncertainty on magnetic bearings. Furthermore, a force and disturbance compensator was added to the system to lower the magnitude of the switching force in the VSC and then the chatter will be reduced. Because of the purpose of controller was hoped to keep motion in the center of the active magnetic bearing for a horizontal rotor, so the controlling aim was holding the location of the system (12) with a feed forward compensated force in the set point. The system can be expressed as

$$\begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \ddot{x}_2 \\ \ddot{y}_2 \end{bmatrix} + \begin{bmatrix} -k_d \beta_1 & 0 \\ 0 & -k_d \beta_1 \end{bmatrix} \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} = \begin{bmatrix} k_i \beta_1 & 0 \\ 0 & k_i \beta_1 \end{bmatrix} \begin{bmatrix} i_1 \\ i_2 \end{bmatrix} + \begin{bmatrix} dx_2 \\ dy_2 \end{bmatrix}. \tag{17}$$

Since the system independent, the sliding surface could be choose as an independent type as

$$s = \begin{bmatrix} s_1 \\ s_2 \end{bmatrix} = \begin{bmatrix} h_1 & 0 \\ 0 & h_2 \end{bmatrix} \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} + \begin{bmatrix} \dot{x}_2 \\ \dot{y}_2 \end{bmatrix} = H \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} + \begin{bmatrix} \dot{x}_2 \\ \dot{y}_2 \end{bmatrix} \tag{18}$$

where h_1 and h_2 are positive number.

Choose Lyapunov function V as

$$V = \frac{1}{2} s^2 \tag{19}$$

The differentiation of the Lyapunov function is

$$\begin{aligned} \dot{V} &= s^T \dot{s} = s^T \left(H \begin{bmatrix} \dot{x}_2 \\ \dot{y}_2 \end{bmatrix} + \begin{bmatrix} \ddot{x}_2 \\ \ddot{y}_2 \end{bmatrix} \right) \\ &= s^T \left(\begin{bmatrix} h_1 & 0 \\ 0 & h_2 \end{bmatrix} \begin{bmatrix} \dot{x}_2 \\ \dot{y}_2 \end{bmatrix} - \begin{bmatrix} -k_d \beta_1 & 0 \\ 0 & -k_d \beta_1 \end{bmatrix} \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} \right. \\ &\qquad \left. + \begin{bmatrix} k_i \beta_1 & 0 \\ 0 & k_i \beta_1 \end{bmatrix} \begin{bmatrix} i_1 \\ i_2 \end{bmatrix} + \begin{bmatrix} dx_2 - \widehat{dx_2} \\ dy_2 - \widehat{dy_2} \end{bmatrix} \right). \end{aligned} \tag{20}$$

Chosen control input

$$\begin{bmatrix} i_1 \\ i_2 \end{bmatrix} = -\frac{1}{k_i \beta_1} \left(\begin{bmatrix} h_1 & 0 \\ 0 & h_2 \end{bmatrix} \begin{bmatrix} \dot{x}_2 \\ \dot{y}_2 \end{bmatrix} - \begin{bmatrix} -k_d \beta_1 & 0 \\ 0 & -k_d \beta_1 \end{bmatrix} \begin{bmatrix} x_2 \\ y_2 \end{bmatrix} + \begin{bmatrix} w_1 sign(s_1) \\ w_2 sign(s_2) \end{bmatrix} \right. \tag{21}$$

where $w_1 > \left|dx_2 - \widehat{dx_2}\right|$ and $w_2 > \left|dy_2 - \widehat{dy_2}\right|$, then $\dot{V}$ can be rearranged as

$$\dot{V} = -|s_1|w_1\left\{1 - \frac{s_1(dx_2 - \widehat{dx_2})}{|s_1|w_1}\right\} - |s_2|w_2\left\{1 - \frac{s_2(\,dy_2 - \widehat{dy_2})}{|s_2|w_2}\right\} < 0. \tag{22}$$

Therefore, the adaptive variable structure controller for the horizontal rotor with an active magnetic bearing had been developed. By Lyapunov criterion, the stability conditions were confirmed.

4 Experimental results

4.1 Experimental setup

A photograph of the experimental setup used in this paper is shown in Fig. 3. The experimental setup is a two-axis controlled horizontal shaft magnetic bearing with symmetric structure. The magnetic bearing has four identical electromagnets equally spaced radially around a rotor disk which is made of laminated stainless steel. Each electromagnet consists of a coil and a laminated core which is made of silicon steel. The system is driven by an AC motor through a flexible coupling in order to isolate the vibration originated from the motor. A pair of eddy current type proximity probes is placed outside the shaft near the electromagnets for measuring the horizontal and vertical displacements of the geometric center of the shaft.

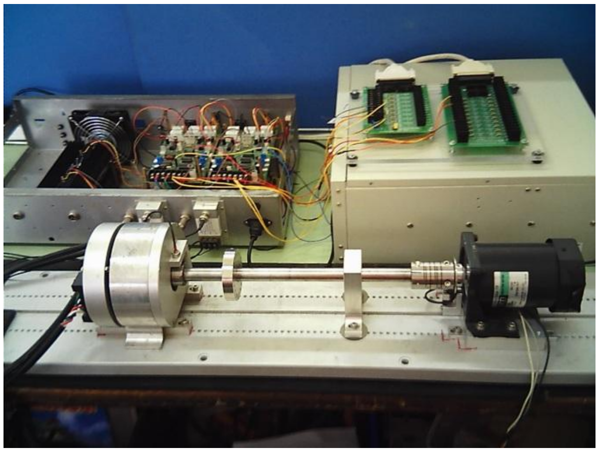


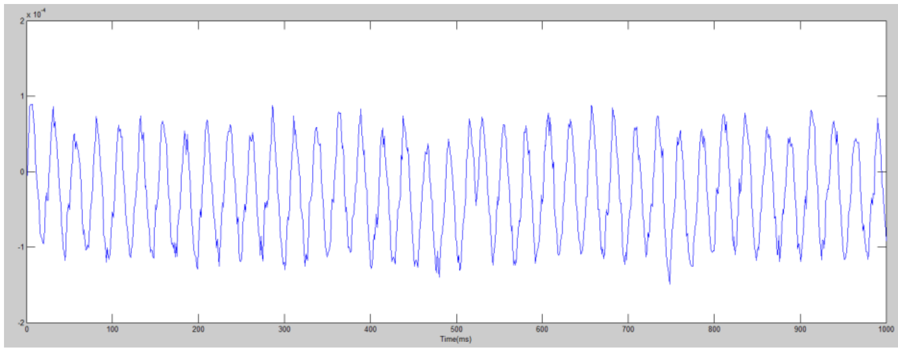
Fig. 3.A photograph of the experimental setup.

4.2 Results

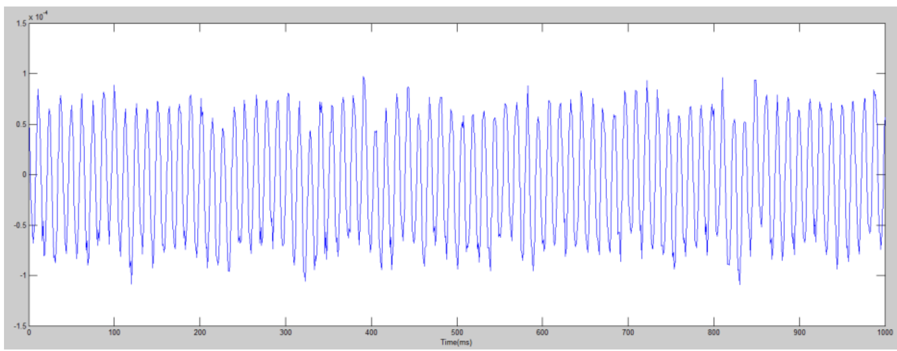
There are two pairs of electromagnets in this AMB system: on the X and Y axes. The two pairs of electromagnets are controlled simultaneously by the VSC controllers with or without the disturbance compensator. In general, the shaft displacement of the rotor center in the horizontal direction is smaller than that in the vertical direction because the vertical direction is affected by gravity. Hence, in this paper, we only show the shaft displacement on the Y axis. Fig. 4 shows the shaft displacement on the Y axis and the orbits of the rotor center when only using a VSC controllers without the disturbance compensator. Fig. 5 shows the measured shaft displacement and shaft force on Y axis using VSC controllers without disturbance compensator. From the results shown in Fig. 5 we can see that the observed shaft displacement obtained from the model-based observer are very close to the measured shaft displacement obtained from the position sensors. Fig. 6 shows the shaft displacements on the Y axis and the orbits of the rotor center using the VSC controllers with the disturbance compensator when the rotating speeds changes from 40 and 80Hz. By observing the difference between Fig. 4 and Fig. 6, the chatter phenomenon in both systems, the VSC controllers with the disturbance compensator is better. It can be seen that the proposed scheme can reduce the shaft displacement noticeably. The orbits around the rotor center become obviously smaller.

5 Conclusion

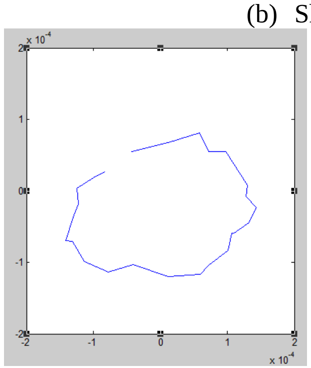
In this paper, a variable structure controller is proposed with a model-based unbalanced forces observer for suppressing unbalanced vibration in an AMB system. First, a model-based unbalanced forces estimator for the observation of unbalanced forces is described. The experimental results show that the observed shaft displacements obtained with the observer are very close to the measured shaft. Secondly, we designed the variable structure controller with the compensator to suppress the chatter. The experimental results also show that the scheme has the ability to improve the performance for the AMB system.



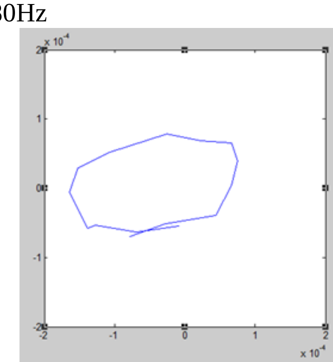
(a) Shaft displacement on 40Hz



(b) Shaft displacement on 80Hz

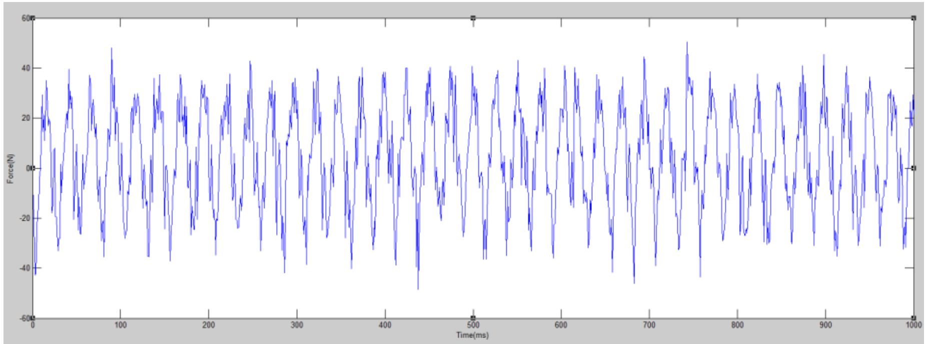


(c) Shaft orbit on 40Hz

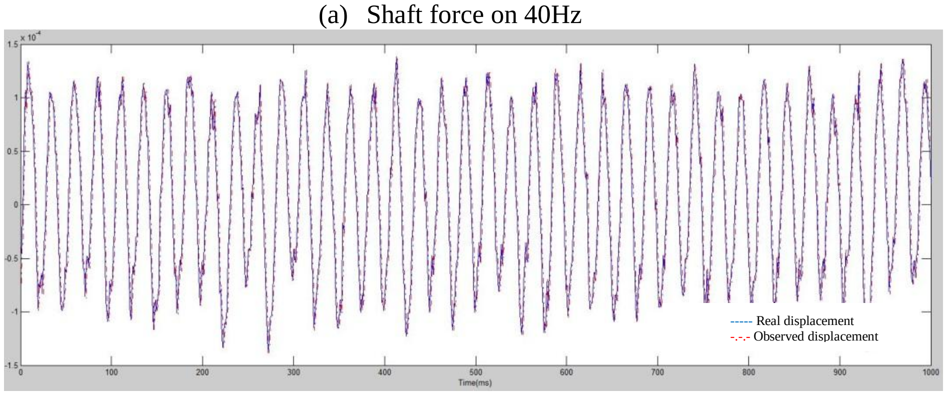


(d) Shaft orbit on 80Hz

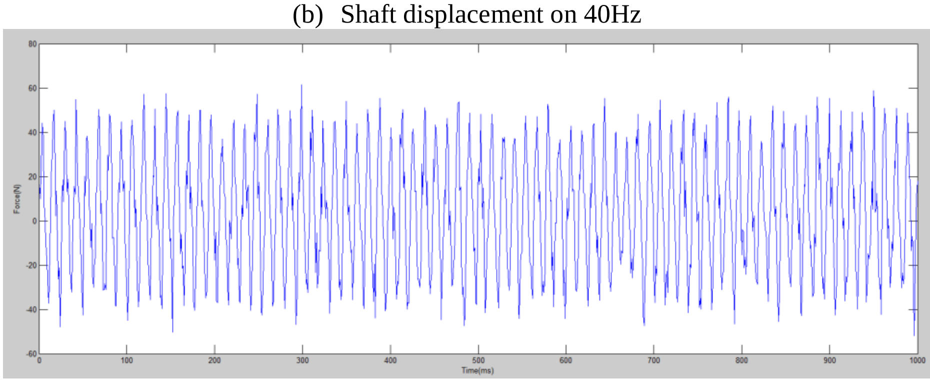
Fig. 4 Shaft displacements on Y axis and orbits of rotor center using VSC controllers without disturbance compensator



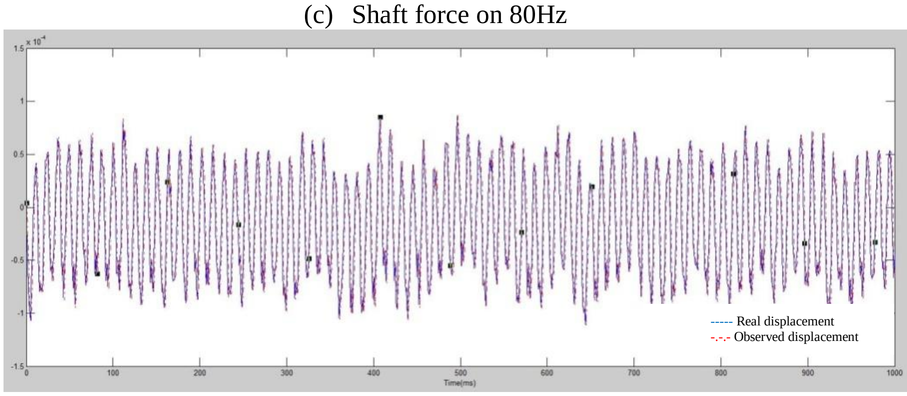
(a) Shaft force on 40Hz



(b) Shaft displacement on 40Hz

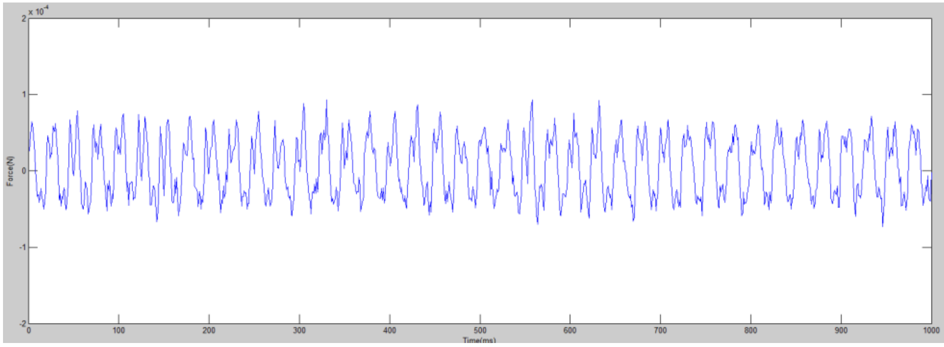


(c) Shaft force on 80Hz

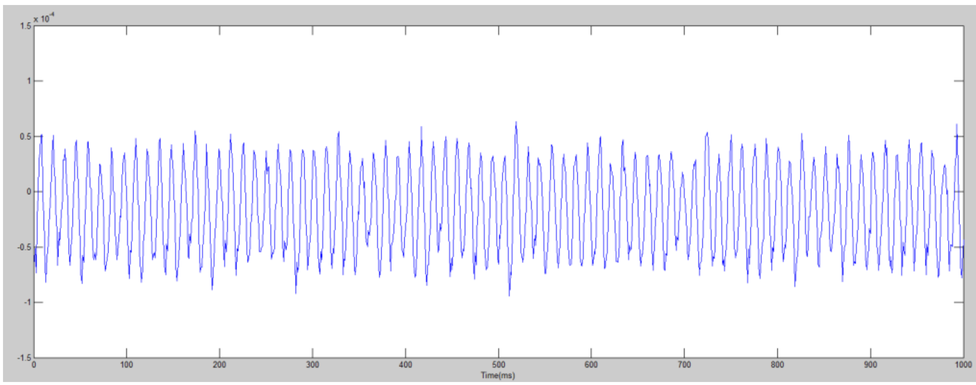


(d) Shaft displacement on 80Hz

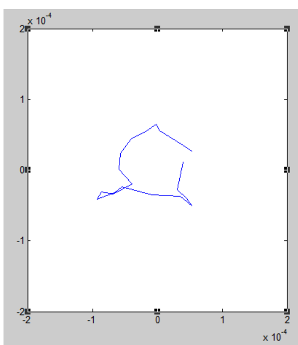
Fig. 5 Measured shaft displacement and force on Y axis using VSC controllers without disturbance compensator



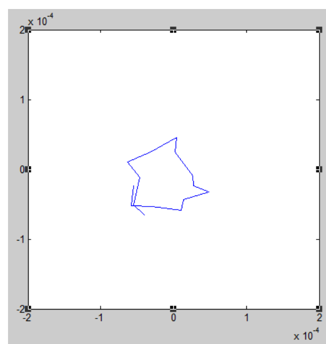
(a) Shaft displacement on 40Hz



(b) Shaft displacement on 80Hz



(c) Shaft orbit on 40Hz



(d) Shaft orbit on 80Hz

Fig. 6 Shaft displacements on Y axis and orbits of rotor center using VSC controllers with disturbance compensator

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