論 文
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Q-매개변수화 제어를 이용한 자기축수 시스템의 불평형 보상에 대한 실험적평가

Experimental Evaluation of Q-Parameterization Control for the Imbalance Compensation of Magnetic Bearing Syatem

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Abstract - This paper utilizes the method of Q-parameterization control to design a controller which solves the problem of imbalance in magnetic bearing systems. There are two methods to solve this problem using feedback control. The first method is to compensate for the imbalance forces by generating opposing forces on the bearing surface (imbalance compensation). The second method is to make the rotor rotate around its axis of inertia (automatic balancing);in this case no imbalance forces will be generated. In this paper we deal with only imbalance compensation. The free parameter of the Q-parameterization controller is chosen such that these goals are achieved. After the introduction of a mathematical model of the magnetic bearing system, we explain the Q-parameterization controller design of the magnetic bearing system with emphasis on the rejection of sinusoidal disturbance for imbalance compensation design. The design objectives are formulated as a linear equations in the controller free parameter Q. Finally, simulation and experimental results are presented and showed the robustness and effectiveness of the proposed controllers.

Key Words : Q-parametrization, Magnetic bearings, Levitation, Vibrations, Rotor imbalance

1. INTRODUCTION

Magnetic bearings are being used today in many industrial, military, and space application. They are capable of suspending rotating shafts at high speeds without mechanical contact or lubrication. Electronically controlled active magnetic bearings offer a number of advantages: very high or low temperature operation, low power consumption, very long life, elimination of oil supply, non contamination of working fluid, vibration control, diagnostic capability.

This paper proposes a Q-parameterization control scheme for a rotating Active Magnetic Bearing(AMB) system in order to solve the problem of the imbalance. Imbalance in the rotor mass generates sinusoidal disturbance forces which cause vibration phenomena in rotating machines. Since balancing is very difficult, there is often a residual imbalance in the rotor. Moreover, the rotor sometimes becomes unbalanced while the machine is in operation.

But this imbalance problem can be overcome by using proper control scheme.There are two methods to eliminate the vibration in magnetic bearing systems. The first method is to compensate for the imbalance forces by generating electromagnetic forces that cancel these forces (imbalance compensation) [1]-[5]. The second method is to make the rotor rotates around its axis of inertia (automatic balancing) [6]-[8]. In this case no imbalance forces will be generated. In this paper we do not deal with the automatic balancing.

In this paper Q-Parameterization theory employed to design a controller for a magnetic bearing system to satabilize it and solve the problem of imbalance. Since Q-Parameterization control method formulate the design objectives as a set of linear equation, it does not have to solve the complicate optimization problem which is like H_∞ control problem. The Q-parameterization theory [9]-[11] characterizes the set of all stabilizing controllers of a given plant in terms of a free parameter Q. The controller free parameter Q is then chosen such that design specifications are achieved [7]. In imbalance compensation design, the imbalance is represented by

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sinusoidal disturbance forces, and free parameter Q of the controller is chosen such that rejection of sinusoidal disturbances is achieved. The design objectives are formulated as a set of linear equations in the free parameter Q. The free parameter Q is found by simply solving this set of linear equations.

In this paper, a four-axis controlled horizontal shaft magnetic bearing system is employed and the axial motion is not actively controlled.

2. MATHEMATICAL MODEL

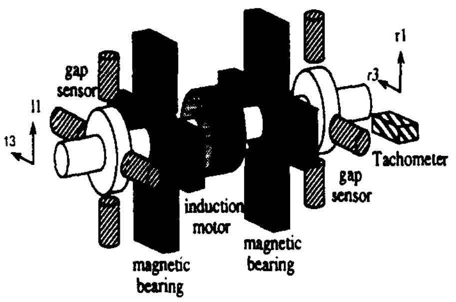

Fig. 1 Four axis magnetic bearing system

Fig. 1 shows a diagram of a four-axis controlled horizontal shaft magnetic bearing system. The diameter of the rotor is 96 mm and its span is equal 660 mm. A three-phase induction motor (1kW,four poles) is located at the center of the rotor. There are four pairs of electromagnets arranged radially on both sides of the rotor, and four pairs of eddy-current type gap sensors located on outside of the electromagnets. Further this system employs a tachometer in order to measure the rotational speed of the rotor. The parameters of the magnetic bearing system used in this research are given in Table 1.

Table 1: Magnetic Bearing Parameters

Parameter	Symbol	Value	Unit
Mass of the Rotor	m	1.39×10^1	kg
Moment of Inertia about X	J_x	1.348×10^{-2}	$kg \cdot m^2$
Moment of Inertia about Y	J_y	2.326×10^{-1}	$kg \cdot m^2$
Distance between Center of Mass and Electromagnet	$l_{l,r}$	1.30×10^{-1}	m
Distance between Center of Mass and Motor	l_m	0	m
Steady Attractive Force	$F_{l1,r1}$	9.09×10	N
	$F_{l2\sim4,r2\sim4}$	2.20×10	N
Steady Current	$I_{l1,r1}$	6.3×10^{-1}	A
	$I_{l2\sim4,r2\sim4}$	3.1×10^{-1}	A
Steady Gap	W	5.5×10^{-4}	m
Resistance	R	1.07×10	Ω
Inductance	L	2.85×10^{-1}	H

In imbalance compensation design, a mathematical model of the magnetic bearing system has been derived in reference [6], and the obtained results are as follows:

$$\begin{bmatrix} \dot{x}_v \\ \dot{x}_h \end{bmatrix} = \begin{bmatrix} A_v & pA_{vh} \\ -pA_{vh} & A_h \end{bmatrix} \begin{bmatrix} x_v \\ x_h \end{bmatrix} + \begin{bmatrix} B_v & 0 \\ 0 & B_h \end{bmatrix} \begin{bmatrix} u_v \\ u_h \end{bmatrix} + p^2 \begin{bmatrix} E_v \\ E_h \end{bmatrix} W \tag{1}$$

$$\begin{bmatrix} y_v \\ y_h \end{bmatrix} = \begin{bmatrix} C_v & 0 \\ 0 & C_h \end{bmatrix} \begin{bmatrix} x_v \\ x_h \end{bmatrix} \tag{2}$$

Where the states x_v, x_h represent the gap displacements of the geometrical axis and imbalance is represented by sinusoidal disturbance forces p^2W. In addition, the subscript ' *vh*' stands for the interference term between the vertical motion and the horizontal motion, and p denotes the rotational speed of the rotor in [rad/s]. The vectors in equations (1), (2) are defined as

$$x_v = [g_{l1}\ g_{r1}\ \dot{g}_{l1}\ \dot{g}_{r1}\ i_{l1}\ i_{r1}]^T$$
$$x_h = [g_{l3}\ g_{r3}\ \dot{g}_{l3}\ \dot{g}_{r3}\ i_{l3}\ i_{r3}]^T$$
$$u_v = [e_{l1}\ e_{r1}]^T,\ u_h = [e_{l3}\ e_{r3}]^T$$

$$W = \begin{bmatrix} \varepsilon \sin(pt+\chi) \\ \tau \cos(pt+\lambda) \\ \varepsilon \cos(pt+\chi) \\ \tau \sin(pt+\lambda) \end{bmatrix} \tag{3}$$

where

g_j : deviations from the steady gap lengths between the electromagnets and the rotor

$\dot{g}$: differentiation of g

i_j : deviations from the steady currents of the electromagnets

e_j : deviations from the steady voltages of the electromagnets

ε, τ, χ, λ : imbalance parameters which are the error between geometrical center and mass center

(j = l1, r1, l3, r3.)

The subscripts ' *l*' and ' *r*' denote the left-hand side and the right-hand side of the magnetic bearing respectively, and the subscripts '1' and '3' denote the vertical directions and the horizontal directions of the rotor respectively. The different matrices in equations (1), (2) are defined as follows [6].

$$A_d := \begin{bmatrix} 0 & I & 0 \\ A_1 + A_2A_{4d} & 0 & A_2A_{5d} \\ 0 & 0 & -(R/L)I \end{bmatrix},$$

$$A_{vh} := \begin{bmatrix} 0 & 0 & 0 \\ 0 & A_3 & 0 \\ 0 & 0 & 0 \end{bmatrix},\ B_d := \begin{bmatrix} 0 \\ 0 \\ (1/L)I \end{bmatrix},$$

$$C_d := [I\ 0\ 0],\quad E_d := \begin{bmatrix} 0 \\ 0 \\ (1/L)I \end{bmatrix},$$

$$(d = v, h).$$

$$A_1 := \frac{\alpha}{l_l + l_r}\begin{bmatrix} (l_r - l_m)(\frac{1}{m} - \frac{L_l l_m}{J_y}) & (l_l - l_m)(\frac{1}{m} - \frac{L_l l_m}{J_y}) \\ (l_r - l_m)(\frac{1}{m} + \frac{L_l l_m}{J_y}) & (l_l - l_m)(\frac{1}{m} + \frac{L_l l_m}{J_y}) \end{bmatrix},$$

$$A_2 := \begin{bmatrix} -\frac{1}{m} - \frac{l_l^2}{J_y} & -\frac{1}{m} + \frac{l_l l_r}{J_y} \\ -\frac{1}{m} + \frac{l_l l_r}{J_y} & -\frac{1}{m} - \frac{l_r^2}{J_y} \end{bmatrix},$$

$$A_3 := \frac{J_x}{J_y(l_l + l_r)}\begin{bmatrix} -l_l & l_l \\ l_r & -l_r \end{bmatrix},$$

$$A_{4v} := -\frac{2}{W} diag[F_{l1} + F_{l2},\ F_{r1} + F_{r2}],$$

$$A_{4h} := -\frac{2}{W} diag[F_{l3} + F_{l4},\ F_{r3} + F_{r4}],$$

$$A_{5v} := 2diag\left[\frac{F_{l1}}{I_{l1}} + \frac{F_{l2}}{I_{l2}},\ \frac{F_{r1}}{I_{r1}} + \frac{F_{r2}}{I_{r2}}\right],$$

$$A_{5h} := 2diag\left[\frac{F_{l3}}{I_{l3}} + \frac{F_{l4}}{I_{l4}},\ \frac{F_{r3}}{I_{r3}} + \frac{F_{r4}}{I_{r4}}\right],$$

$$E_{1v} := \begin{bmatrix} -1 & l_l(1 - \frac{J_x}{J_y}) & 0 & 0 \\ -1 & -l_r(1 - \frac{J_x}{J_y}) & 0 & 0 \end{bmatrix},$$

$$E_{1h} := \begin{bmatrix} 0 & 0 & 1 & l_l(1 - \frac{J_x}{J_y}) \\ 0 & 0 & 1 & -l_r(1 - \frac{J_x}{J_y}) \end{bmatrix}.$$

In the above equations, α denotes the coefficient of the force which occurs when the rotor eccentrically deviates, and hence we set $\alpha = 0$.

3. Q-PARAMETERIZATION THEORY

The Q-parameterization theory [9]-[11] states that the set of all stabilizing controllers for a given plant can be characterized by a free parameter Q. Consider the one-parameter-control feedback system shown in Fig. 2 to control the system described by equations (1), (2).

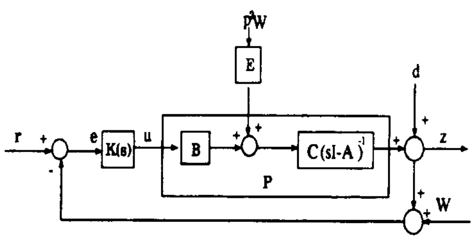

Fig. 2 One-parameter-control feedback system

Where r is the reference input signal, d is the output disturbance, W is the sensor noise, p^2W is the disturbance force, u is the controller output, z is the plant output to be regulated, and K is the stabilizing controller for P(s). Note that W, p^2W may also represent model uncertainties. In order to characterize the set of all stabilizing controllers K for P(s) using Q-parameterization theory, first we need to construct a doubly coprime factorization N, M, $\bar{N}$, $\bar{M}$, X, Y, $\bar{X}$, $\bar{Y} \in RH_\infty$ for P(s). Let (A,B,C) be system matrices of the state equations (1), (2), then such factorization is possible if the pairs (A,B) and (C,A) are stabilizable and detectable pairs, respectively. To find N, M, $\bar{N}$, $\bar{M}$, X, Y, $\bar{X}$, $\bar{Y}$ first we choose real matrices F_1, F_2 such that the eigenvalues of $A_0 = A - BF_1$ and $\bar{A}_0 = A - F_2C$ have negative real parts. Then N, M, $\bar{N}$, $\bar{M}$, X, Y, $\bar{X}$, $\bar{Y}$ can be expressed in terms A, B, C, A_0, $\bar{A}_0$, F_1, F_2 [9]. F_1, F_2 are obtained using the algebraic Riccati equation. With these choices, the set of all stabilizing controllers for P(s) is given by

$$K = \{(Y - Q\bar{N})^{-1}(X + Q\bar{M}),\ |Y - Q\bar{N}| \neq 0\} \quad (4)$$

Fig. 3 shows the block diagram representation of equation (4). Fig. 4 shows the observer-based Q-Parameterization controller with a free parameter Q. This free parameter can be chosen such that the design objectives are achieved.

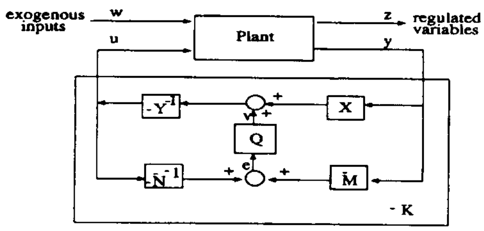

Fig. 3 Block diagram of Q-parameterization

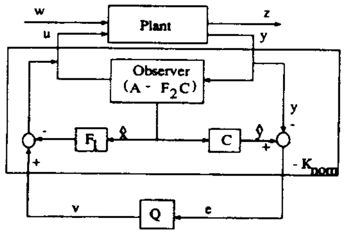

Fig. 4 Q-parameterization for estimated state feedback nominal controller

4. CONTROLLER OBJECTIVES [7], [10]

The control problem for imbalance compensation design can be defined as follows: Find a controller K such that the following requirements are satisfied:

1). The closed loop system achieves robust stability against rotor speed variation and other parameters variation, and there is fast transient response. This requirement can be satisfied if the following inequality holds.

$$Re(s_i) + \alpha_s < 0 \tag{5}$$

where s_i denote the closed loop poles,and α_s is a positive number chosen to ensure a certain degree of stability. In this paper α_s is chosen such that closed loop stability is achieved if the speed varies in the range [0,22500]rpm.

2). Damping Factor: In order to prevent undesirable high frequency oscillation and to help the magnetic bearing system step through critical speeds safely, we must put a lower limit for the damping factor. To achieve this the following inequality must hold [7]

$$Re(s_i) + \beta_d Im(s_i) \le 0 \tag{6}$$

where β_d is a positive constant chosen as a lower limit for the damping factor and $Re(s_i)$ and $Im(s_i)$ represent the real and imaginary parts of the complex number s_i..

3). Asymptotic rejection of low frequency disturbances. This can be achieved by making the sensitivity function small at low frequency. This requirement can be satisfied by choosing the controller Q-parameter Q such that

$$W_1(s=0) = 0 \tag{7}$$

where W_1 is the sensitivity transfer function i.e the transfer function from d to z [7], [10]

$$W_1 = I - N(X + Q\overline{M}) \tag{8}$$

4) Asymptotic rejection of sinusoidal disturbances (imbalance compensation): let W_2 be the transfer function from p_2W to z, then W_2 is given by [7], [10]

$$W_2 = (I - N(X + Q\overline{M}))P_1(s) \tag{9}$$

where $P_1 = C(sI-A)^{-1}E$. In order to achieve this requirement Q must be chosen such that the following identity hold

$$W_2(s = jp) = 0 \tag{10}$$

5. CONTROLLER SYNTHESIS

In this section, we design the Q-parameterization controller which achieves the above requirements. At first we assume that the speed of the nominal plant is p=0. It means that there is no coupling between the vertical motion and horizontal motion. Therefore the plant model can be separated into vertical plant and horizontal plant.

$$P = \begin{bmatrix} P_v & 0 \\ 0 & P_h \end{bmatrix} \tag{11}$$

Then, a controller will be designed for each plant. The final controller K for the entire plant P is constructed with the combination of these controllers as follows:

$$K = \begin{bmatrix} K_v & 0 \\ 0 & K_h \end{bmatrix} = \begin{bmatrix} A_{kv} & 0 & B_{kv} & 0 \\ 0 & A_{kh} & 0 & B_{kh} \\ C_{kv} & 0 & D_{kv} & 0 \\ 0 & C_{kh} & 0 & D_{kh} \end{bmatrix} \tag{12}$$

K_v denotes the controller for the vertical plant and K_h denotes the controller for the horizontal plant. In order to satisfy controller requirements (1), (2) we choose the matrices F_1, F_2 such that eigenvalues of A_0,, $\overline{A}_0$ lie in the domain D shown in Fig.5. [9]

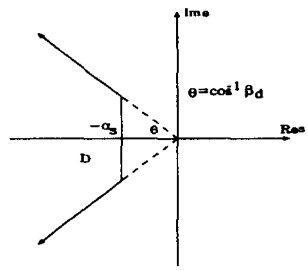

Fig.5 Generalized Region of Stability

In order to satisfy requirements (3), (4) we choose the vertical controller Q-parameter Q_v and horizontal controller Q-parameter Q_h such that equations (7), (10) are satisfied. From (7) we have

$$N_x(0)\,(X_x(0) + Q_x(0)\overline{M}_x\,(0)) = I, \;\; x = v, h \tag{13}$$

from (10) we have

$$I - N_x(jp)(X_x(jp) + Q_x(jp)\overline{M}_x\,(jp)) = 0, x = v, h \tag{14}$$

Equation (14) is complex equation and this equation is in fact two equations, one for the real part and one for the imaginary part. This means that we have three equations in the unknown Q_x. Since we need to satisfy three equations, we should allow three variable coefficients. So we take the vertical controller Q-parameter Q_v and the horizontal controller Q-parameter Q_h in the form

$$q_{vij}(s) = \frac{a_{vij}s^2 + b_{vij}s + c_{vij}}{(s + p_{s1v})(s + P_{s2v})} \tag{15}$$

$$q_{hij}(s) = \frac{a_{hij}s^2 + b_{hij}s + c_{hij}}{(s + p_{s1h})(s + P_{s2h})} \tag{16}$$

where a_{vij}, b_{vij}, c_{vij}, a_{hij}, b_{hij}, c_{hij} (i=1,2, j=1,2) are design parameters for the vertical motion and horizontal motions, respectively, and p_{s1v}, p_{s2v}, p_{s1h}, p_{s2h} > α_s are fixed.

Then we have the following linear equations.
For imbalance compensation

$$\begin{array}{l} A_{v1}X_{v1}=B_1,\ \ A_{v2}X_{v2}=B_2 \\ A_{h1}X_{h1}=B_1,\ \ A_{h2}X_{h2}=B_2 \end{array} \quad (17)$$

where

$$\begin{array}{l} A_{v1}=N_v(0),\ \ A_{v2}=N_v(jp) \\ A_{h1}=N_h(0),\ \ A_{h2}=N_h(jp) \\ X_{v1}=Q_v(0),\ \ X_{v2}=Q_v(jp) \\ X_h=Q_h(0),\ \ X_{h2}=Q_h(jp) \end{array} \quad (18)$$

$$B_1=(B_{11}\ \ B_{12})=(I-N_x(0)X_x(0))\overline{M}_x(0)$$

$$B_2=(B_{21}\ \ B_{212})=(I-N_x(jp)X_x(jp))\overline{M}_x\,(jP) \quad (19)$$

where x=v,h. Solving equations (17) for X_{v1}, X_{v2}, X_{h1}, X_{h2} we can easily find the design parameters a_{vij}, b_{vij}, c_{vij}, a_{hij}, b_{hij}, c_{hij}(i=1,2,j=1,2) as follows:

$$\begin{array}{lll} c_{vij} &=& X_{v1ij}p_{s1v}p_{s2v} \\ b_{vij} &=& Im\{X_{v2ij}(p_{s1v}+jp)(p_{s2v}+jp)\}/p \\ a_{vij} &=& (c_{vij}-Re\{X_{v2ij}(p_{s1v}+jp)(p_{s2v}+jp)\})/p^2 \end{array} \quad (20)$$

6. SIMULATION AND EXPERIMENTAL RESULTS

We design the Q-parameterization controllers by the method discussed in the previous section. The controller K(s) is designed at p=0 and the operating speed for imbalance compensation design is assumed to be p=2π20[rad/s] (=1200[rpm]). The μ synthesis Toolbox [12] with Simulink were used for the controller design and simulation.

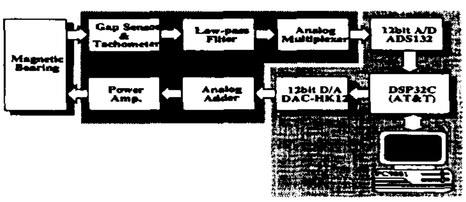

Fig. 6 Digital control system

Fig. 6 shows the digital control system for the real time implementation. The experimental machine is controlled by a digital control system that consists of a 32-bit floating point Digital Signal Processor (DSP) DSP32C(AT&T), 12-bit A/D converters and 12-bit D/A converters. In order to make the imbalance we added a small weight (20[g]) to the left side of the rotor shaft. The designed continuous-time controllers are discretized via the well-known Tustin transformation at a sampling rate of 128μs.

6.1 Simulation Results

We choose p_{s1v}=30, p_{s2v}=62, p_{s1h}=30, p_{s2h}=60, for automatic balancing, and F_{1v}, F_{2v}, F_{1h}, F_{2h} were obtained using the algebraic Riccati equation and are given by

$$F_{1v}^T=\begin{bmatrix} -7.9191e+06 & 3.2873e+02 \\ 3.2873e+02 & -7.9191e+06 \\ -3.2553e+04 & -7.8825e+01 \\ -7.8825e+01 & -3.2553e+04 \\ 4.2109e+03 & -3.4471e-01 \\ -3.4471e-01 & 4.2109e+03 \end{bmatrix}$$

$$F_{2v}=\begin{bmatrix} 1.8898e+03 & -3.1067e-01 \\ -3.1067e-01 & 1.8898e+03 \\ 1.1897e+05 & -5.8710e+02 \\ -5.8710e+02 & 1.1897e+05 \\ -1.0464e+02 & 2.7447e-01 \\ 2.7447e-01 & -1.0464e+02 \end{bmatrix}$$

$$F_{1h}^T=\begin{bmatrix} -2.1074e+06 & 1.2380e+02 \\ 1.2380e+02 & -2.1074e+06 \\ -1.3975e+04 & -3.3130e+01 \\ -3.3130e+01 & -1.3975e+04 \\ 1.9025e+03 & -2.1965e-01 \\ -2.1965e-01 & 1.9025e+03 \end{bmatrix}$$

$$F_{2h}=\begin{bmatrix} 1.8510e+03 & -1.2346e-01 \\ -1.2346e-01 & 1.8510e+03 \\ 4.6408e+04 & -2.2852e+02 \\ -2.2852e+02 & 4.6408e+04 \\ -1.9347e+01 & 7.1165e-02 \\ 7.1165e-02 & -1.9347e+01 \end{bmatrix}$$

The controllers Q-parameter Q_v, Q_h that can satisfy equations (13), and (14) for imbalance compensation design were found to be

$$\begin{array}{l} Q_{v11}=\dfrac{-1.069e+08s^2-1.431e+10s-7.642e+11}{s^2+210s+10400} \\ Q_{v12}=\dfrac{-2.791e+05s^2-2.431e+07s-1.483e+09}{s^2+210s+10400} \\ Q_{v21}=\dfrac{-2.791e+05s^2-2.431e+07s-1.483e+09}{s^2+210s+10400} \\ Q_{v22}=\dfrac{-1.069e+08s^2-1.431e+10s-7.642e+11}{s^2+210s+10400} \end{array} \quad (21)$$

$$\begin{array}{l} Q_{h11}=\dfrac{-4.98e+07s^2-5.364e+09s-3.022e+11}{s^2+210s+10400} \\ Q_{h12}=\dfrac{-1.509e+05s^2-9.127e+06s-6.176e+08}{s^2+210s+10400} \\ Q_{h21}=\dfrac{-1.509e+05s^2-9.127e+06s-6.176e+08}{s^2+210s+10400} \\ Q_{h22}=\dfrac{-4.98e+07s^2-5.364e+09s-3.022e+11}{s^2+210s+10400} \end{array} \quad (22)$$

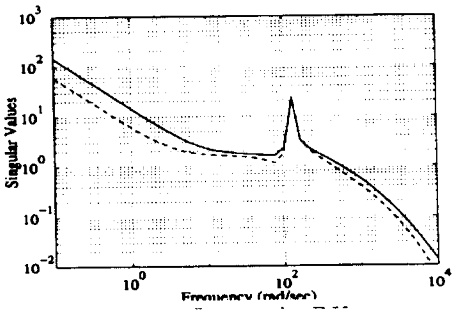

Fig. 7 Loop gain PK

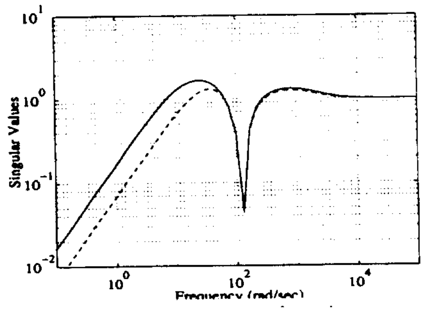

Fig. 8 Sensitivity function

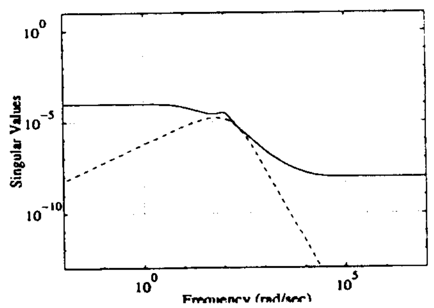

Fig. 9 Robust Stability Test (p=2 π375 [rad/s])

Fig. 7 shows the singular values of the loop gain PK. We can see from this figure that high values of loop gain is achieved at low frequency and low values of loop gain is achieved at high frequency. Fig. 8 shows the singular values of the sensitivity function. From this figure we can see that the sensitivity function is very small at the operating speed $2\pi 20$[rad/s](=1200[rpm]). This means good suppression of the imbalance forces is achieved. In the design of Q-parameterization controllers, we ignored the interference terms, which express the gyroscopic effect of the plant, when p=0. We therefore verify the robust stability of this system against changes in the rotational speed of the rotor. Let the perturbed plant (p$\neq$0) be denoted by P_p and the additive perturbation Δ_p from P is as follows:

$$\Delta_p = P_p - p \qquad (23)$$

Then the robust stability is guaranteed within the the following inequality:

$$\bar{\sigma}(\Delta_p) < \frac{1}{\bar{\sigma}(K(I-PK)^{-1})} \qquad (24)$$

Fig. 9 shows the robust stability test for the imbalance compensation design. For the test of the robust stability we used the following equations $1/\bar{\sigma}(K(I+GK)^{-1})(-)$ and $\bar{\sigma}(\Delta_p)(--)$. From this figure we can see that robust stability is achieved up to p=2π375[rad/s] (=22500[rpm]).

6.2 Experimental Results

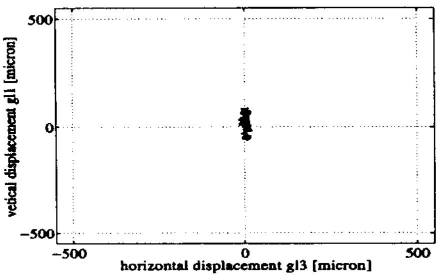

Fig. 10 Geometrical axis displacements(left side, p=1050[rpm])

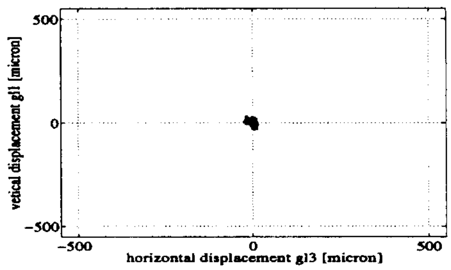

Fig. 11 Geometrical axis displacements(left side, p=1200[rpm])

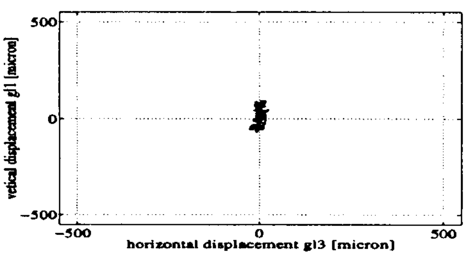

Fig. 12 Geometrical axis displacements(left side, p=1350[rpm])

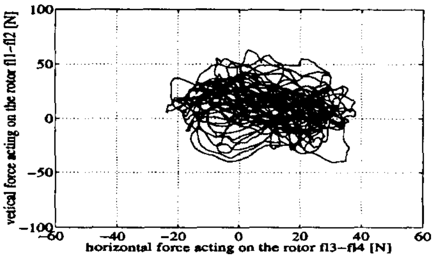

Fig. 13 Forces acting on the rotor(left side, p=1050[rpm])

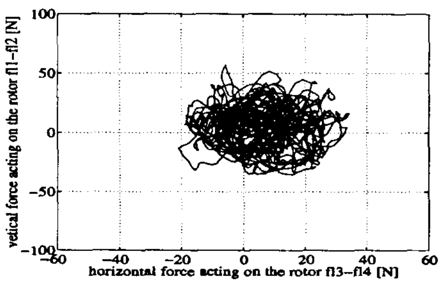

Fig. 14 Forces acting on the rotor(left side, p=1200[rpm])

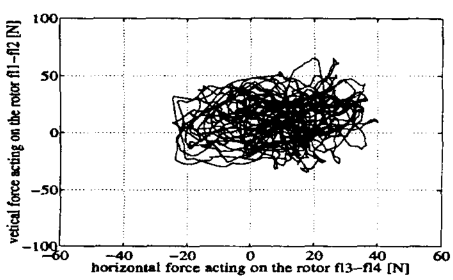

Fig. 15 Forces acting on the rotor(left side, p=1350[rpm])

Figs. 10, 11, and 12 show the experimental results of the geometrical axis displacements at different speeds for the imbalance compensation design. In Fig. 11 we can see that the geometrical axis vibration is completely eliminated at the design speed $p=2\pi 20$[rad/s] and is very small at other speeds ($p=2\pi 17.5$[rad/s], $p=2\pi 22.5$[rad/s]). This means that the magnetic bearing generates sinusoidal disturbance forces which cancel the imbalance forces. Fig. 13, 14 and 15 show forces acting on the rotor at different speeds for the imbalance compensation design. In these figures we can see no big difference of forces acting on the rotor. This is because electromagnets makes some forces to compensate for the imbalance forces. In other words, at operating speed(1200[rpm]) electromagnets are generating opposing forces on the bearing surface so as to compensate imbalance forces.

7. CONCLUSIONS

This paper presents the method to control the vibration caused by imbalance in the rotor of a four-axis magnetic bearing systems. To overcome the imbalance in the rotor of the magnetic bearing system, we used imbalance compensation method. The imbalance is modeled as sinusoidal disturbance forces. The Q-parameterization theory has been employed to design a controller which stabilizes the system and achieves the desired goals. The controller Q-parameter is found simply by solving a set of linear equations. The controllers that were obtained have 20 states, with 4 inputs and 4 outputs. The controller is designed at speed p=0 (nominal plant) and experimental results were obtained at three different speed $p=2\pi 17.5$[rad/s], $p=2\pi 20$[rad/s], $p=2\pi 22.5$[rad/s] for imbalance compensation design. The results show good robustness to model uncertainties and show that the magnetic bearing systems can be used to control vibrations in rotating machinery by compensating for the imbalance forces (imbalance compensation).

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