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Vibration Control of Active Magnetic Bearing Systems Using Digital Signal Processor

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Abstract

A digital signal processor(DSP) is applied to realizing a compensator of control system of active magnetic bearings, to restrict a resonance caused by the first-order bending vibration of a flexible rotor, and to run the rotor beyond the critical speed. A full-order observer is applied to the translatory rotor-motion with the first-order vibration mode. A PID control is used for the conical motion. The rotor used in the experiments is symmetric, and an electromagnet and a displacement sensor are set in collocation.

Keywords: Magnetic bearing, Vibration control, Critical speed, Digital signal processor, Observer

1. Introduction

Active magnetic bearings, which support a rotor without contact by controlling coil currents of electromagnets, are useful as a bearing for high-speed rotation. In this application, the maximum rotor-speed is usually restricted by the first flexural-critical speed of a rotor. To run the rotor at higher speeds, therefore, it is a matter of interest to surpress the bending vibration around the critical speed by controlling the rotor-supporting forces of the bearings.

For systems in which a flexible rotor is drived by a motor through a flexible coupling, suppressing several resonances was considered in [1, 2]: [1] applied a digital proportional-derivative(PD) control realized by a microprocessor for a rotor with bending natural frequnecies λ_i of 47Hz,140Hz ,...(sampling frequency 2.5kHz), and [2] applied an analog compensator with a phase-lead for a rotor of λ_i=12.5Hz, 23Hz, 84Hz,...

For systems in which a driving motor is installed in the bearing system and the rotor is drived directly, the subject is considered in [3-7]. In [3] are used an analog compensation with a three-stage phase-lead in series, and a digital compensation based on the analog control realized by a digital signal processor (sampling frequency 4kHz) for a rotor of λ_1=290Hz. In [4] for a rotor of λ_1=1490Hz, is adopted an analog compensation with a two-stage phase-lead plus three-stage band-pass filter parallel to a main compensator. In [5-6] for λ_1=185Hz, is used an analog feedback compensation based on a full-order observer.

Vibration control is also important in active magnetic bearings even at a run under the critical speed, when the variation of a load has a frequency close to the bending natural frequencies of the rotor[7, 8]. Vibration control may be necessary when the bending vibrations are stimulated by the rotor-supporting magnetic forces, for examples, [9-13]. The methods of vibration control in these cases are applicable to suppressing the resonance at critical speeds; however, the characteristics of the system in rotation of high rotor-speed, especially, around critical speeds, is much different from those at rest or in low rotation[2, 10]. Other than the control law (software), some crucial factors concerned with the setup(hardware) may have influence on the success of suppressing the resonance at the critical speeds[6].

Of various control laws for vibration control of active magnetic bearing systems, an application of observers is of much interest. Because observers are applicable in general regardless of the position of sensors and actuators, if the controlled system is observable, and this application is expected to be more robust for the noise problem than a control with a phase-lead compensation. The application of observers was considered in [5, 6, 10] with analog controls. The construction of observers, however, is so complicated that their realization with analog circuits is cumbersome.

In this paper, a digital signal processor(DSP) is used to realize a full-order observer and a PID compensator to suppress the resonance at the first critical speed for an experimental setup used in [5, 6], in which a rotor is symmetiric and an electromagnet and a displacement sensor are set in collocation. The gyroscopic effects are neglected in the design of the control system.

2. Control system of magnetic bearings

2.1 Model of dynamics

The configuration of active magnetic radial-bearings is illustrated in **Fig.1**. The rotor position is regulated through the control of magnet-coil currents supplied by power amplifires whose input signal is generated by a compensator.

For symmetric bearing systems, we decouple the interacted conical motions of the vertical and horizontal directions, into two conical motions by neglecting the gyroscopic effect. By that we have four motions independent from each other. Then, the dynamics of a linearized motion with a bending vibration may be modeled as in Fig. 2 [6], where damping is neglected for simplicity. In the figure α and γ are constants given arbitrarily; η is a factor of the vibration mode connecting to the motion; β and ξ are given by

$$\beta = 2K_F K_y / M \alpha \quad \text{(translatory motion)} \tag{1}$$

$$= 2 \ell_1^2 K_F K_y / J \alpha \quad \text{(conical motion)} \tag{1a}$$

$$\xi = \omega_B^2 / \gamma \tag{2}$$

and where

- y : sensor output
- x_1 : rigid displacement of rotor
- x_2 : rigid velocity of rotor
- x_3 : displacement of rotor vibration
- x_4 : velocity of rotor vibration
- x_5 : variable equivalent to flux
- u : control signal
- b : gain of power amplifire
- T : time constant of magnetic force
- a_g : gap constant

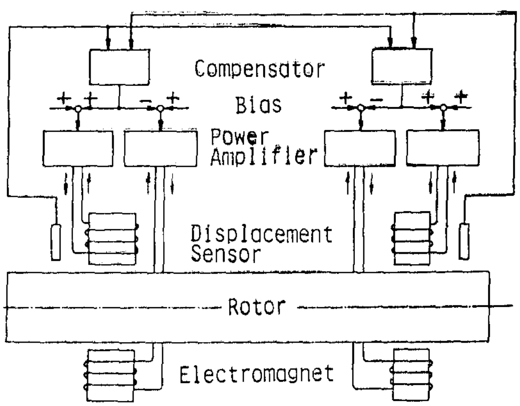


Fig. 1 Configuration of active magnetic bearings

- K_y : gain of displacement sensor
- K_F : constant of magnetic force
- M : mass of rotor
- J : moment of inertia
- ℓ_1 : distance between bearing and center of mass
- ω_B : natural frequency of bending vibration

2.2 Observer-based feedback control

The state equation of Fig. 2 is given by

$$\dot{x} = A_c x + B_c u \tag{3}$$

$$y = C x \tag{4}$$

where

$$x = [x_1 \quad x_2 \quad x_3 \quad x_4 \quad x_5]^T \tag{5}$$

$$A_c = \begin{bmatrix} 0 & \alpha & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \beta \\ 0 & 0 & 0 & \gamma & 0 \\ 0 & 0 & -\xi & 0 & \eta \\ a_g/K_y T & 0 & a_g/K_y T & 0 & -1/T \end{bmatrix} \tag{6}$$

$$B_c = [0 \quad 0 \quad 0 \quad 0 \quad b/T]^T \tag{7}$$

$$C = [1 \quad 0 \quad 1 \quad 0 \quad 0] \tag{8}$$

For the above system, a sampled-data full-order observer is constructed as follows:

$$\hat{x}(k+1) = A\hat{x}(k) + B u(k) + L[y(k) - \hat{y}(k)] \tag{9}$$

$$K = [k_1 \quad k_2 \quad k_3 \quad k_4 \quad k_5] \tag{10}$$

$$L = [\ell_1 \quad \ell_2 \quad \ell_3 \quad \ell_4 \quad \ell_5]^T \tag{11}$$

where

$$A = \exp(A_c \tau) \tag{12}$$

$$B = \int_0^{\tau} \exp(A_c t) B_c dt \tag{13}$$

and where

- $\hat{x}$:estimated valiables
- K :feedback gains
- L :parameters of observer
- τ :sampling period

Figure 3 illustrates the block diagram of the feedback control system based on the observer with an integral(I) compensation.

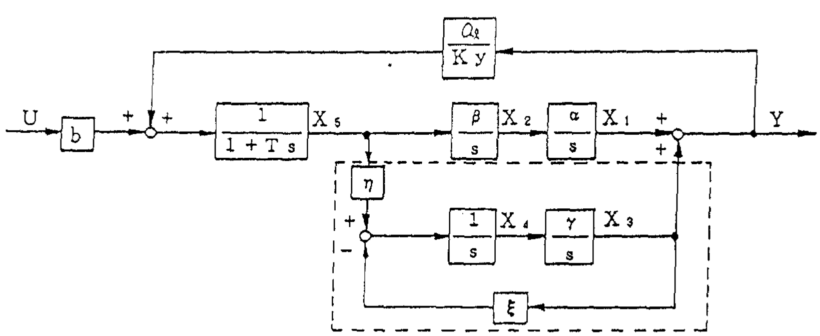


Fig. 2 Model of rotor motion with a bending vibration

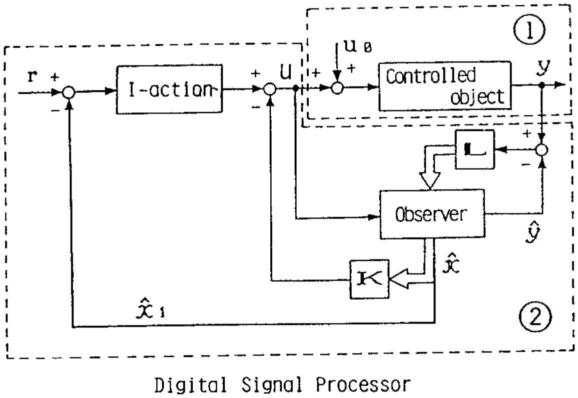


Fig. 3 Control system based on observer

2.3 PID control

We use a digital PID action of the form written by

$$u(k) = -[K_P\, y(k) + u_I(k) + u_D(k)] \tag{14}$$

where

$$u_I(k) = u_I(k-1) + K_I\, y(k) \tag{15}$$

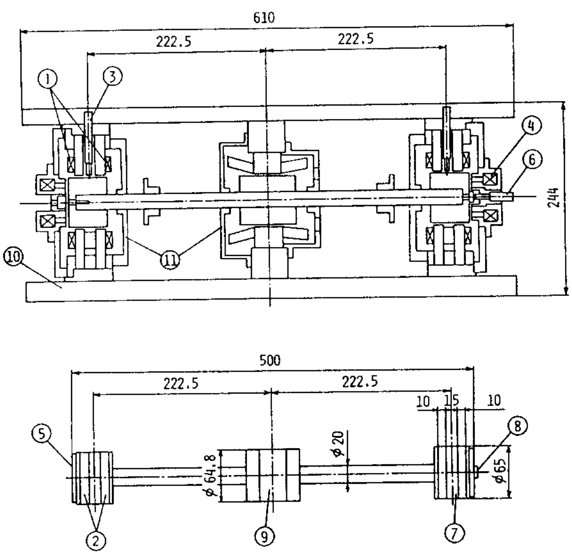
$$u_D(k) = K_D[y(k) - y(k-1)] + K_D'\, u_D(k-1) \tag{16}$$

and where K_P, K_I, K_D and K_D' are gains. The D-action of eqn. (16) is derived from the first-order approximation of an analog pseud-derivative action.

3. Experiments

3.1 Experimental setup

Figure 4 shows the experimental setup. The flexible rotor is symmetric with respect to the center of mass. The characteristics of the two radial bearing systems are almost the same. The displacement sensors are collocated to the position of the electromagnets, each of which has one coil and has separated two magnetic circuits. Natural bending frequencies of the rotor are 185Hz



Radial bearings:
- ① Electromagnet
- ② Core of rotor
- ③ Displacement sensor (eddy current type)
- ⑦ Target ring of sensor

Thrust bearing:
- ④ Electromagnet
- ⑤ Core of rotor
- ⑥ Displacement sensor
- ⑧ Target ring of sensor

- ⑨ Rotor of induction motor
- ⑩ Frame
- ⑪ Emergency bearing

Fig. 4 Experimental setup

Table 1 Data of experimetal setup

Rotor:				
Mass	$M =$	4.4		kg
Moment of inertia	$J =$	0.092		kgm^2
	$J_P =$	0.019		kgm^2
Distance between bwaring and center of mass	$l_1 =$	222.5×10^{-3}		m
Radial bearings:				
Coil: Turns	$N =$	200		
Inductance	$L =$	6.0×10^{-3}		H
Resistance	$R =$	3.2		Ω
Air gap	$x_0 =$	0.5×10^{-3}		m
Area of magnet	$A =$	2.4×10^{-4}		m^2
Bias current	$I_{10} =$	2.0		A
	$I_{20} =$	2.4	(Vert.)	A
	$=$	2.0	(Hor.)	A
Constant of force	$K_F =$	85	(Theoret.)	N/A
	$=$	96	(Exp.)	N/A
Bias force(2.0A)	$F_{10} =$	42.7	(Theoret.)	N
Gain of power amplifire	$b =$	1.14	(Vert.)	A/V
	$=$	1.00	(Hor.)	A/V
Gain of sensor	$K_y =$	10.6×10^3		V/m
Time constant	$T =$	0.14×10^{-3}		s
Constant	$a_0 =$	4.3×10^3	(Vert.)	A/m
	$=$	3.5×10^3	(Hor.)	A/m
Constants:				
	$\alpha =$	1.0×10^3		
	$\beta =$	4.12×10^2	(Trans.)	
	$=$	9.41×10^2	(Con.)	
	$\gamma =$	1.0×10^3		
	$\eta =$	2.54×10^2	(Trans.)	
	$=$	0.62×10^2	(Con.)	
	$\xi =$	1.20×10^3	(Trans.)	
	$=$	1.70×10^4	(Con.)	

for the first mode, 655Hz for the second-mode, and 1240Hz and 1890Hz for the third- and the forth-order, respectively.

An induction motor drived by a VVVF(variable voltage variable frequency) inverter is installed in the centor of the bearing system. The axial direction is supported by a similar magnetic bearing at the ends of the rotor. The data of the set-up and of the experiments are given in **Table 1**.

Two DSPs(μPD77230) are applied to the four control systems of the radial bearings. One is for the two observers of the translatory motions with sampling frequency 6.1kHz(0.164msec). The other is for two PID controls of the conical motions with sampling frequency 23.2kHz (0.043msec). Several bending vibration modes higher than the second were passively suppressed by notch filters.

3.2 Dynamics in non-rotation

The control parameters including the observer's were tuned by try and error in the state of non-rotation for the frequency response of the loop transfer function. The parameters selected are given in **Table 2** and **Table 3**.

Experimental data will be shown for the horizontal motion. **Figures** 5 and 6 give the frequency characteristics of the loop transfer function which is measured through the frequency response of the control signal u to the input signal u_0 (see Fig. 3). Figure 5 is for the translatory control system with the observer, and Fig. 6 is for the conical control system with the PID compensator.

Table 2 Parameters of observer-based compensator

Control		Observer	
k_1	2.0	ℓ_1	0.7
k_2	10.0	ℓ_2	0.5
k_3	0	ℓ_3	0
k_4	5.0	ℓ_4	0.5
k_5	1.0	ℓ_5	10.0
K_I	30.0		
T_I	0.53 s		
Poles			
0.991 0.983±0.156j 0.978 0.587 −0.620		0.971±0.130j 0.716±0.393j 0.090	

Table 3 Parameters of PID

K_P	0.70	
K_I	2.2×10^{-3}	1/s
K_D	32	s
K_D'	6.5×10^{-2}	s

The frequency response of the rotor displacement y for a disterbance u_0 is shown in Fig. 7 for the translatory motion. The first bending vibration is suppressed sufficiently small around the critical frequency, but anther vibration appears at frequency 590 Hz. This vibration is considered to be stimulated by the interaction of the rotor motion with the stator vibration. This resonance was small in an analog control with similar compensators. The difference between the two controls lies in the phase-lag of the loop transfer function; this lag is larger in the digital control than in the analog control. It is necessary to compensate this phase-lag to suppress this resonance.

The frequency responce of the displacement of the conical motion for a disturbance is shown in **Fig. 8**, where the second bending vibration of 655 Hz cannot be suppressed. It seems difficult to restrict this mode in this experimental setup, because the actuators are near to the node of the vibration mode. However, it was necessary to give a enough phase-margin around this frequency to

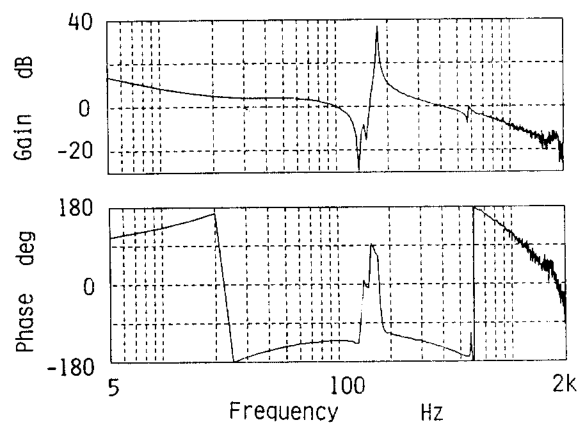


Fig. 5 Loop transfer-function of translatory motion (observer-based control)

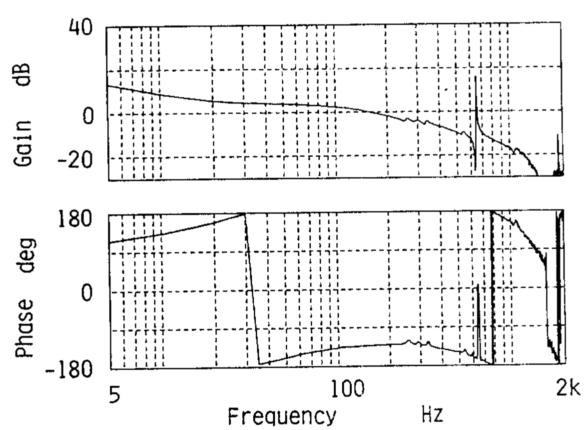


Fig. 6 Loop transfer-function of conical motion (PID control)

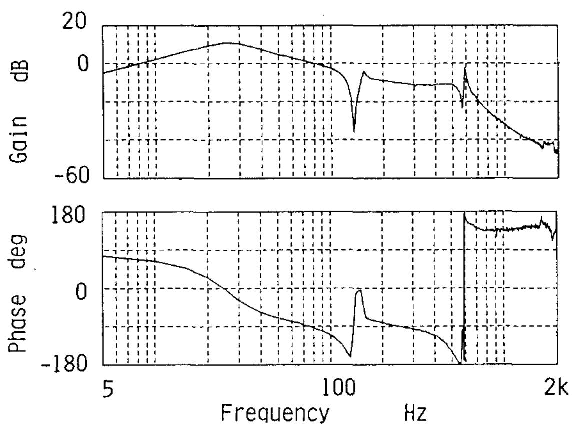


Fig. 7 Frequency response of translatory motion for disturbance

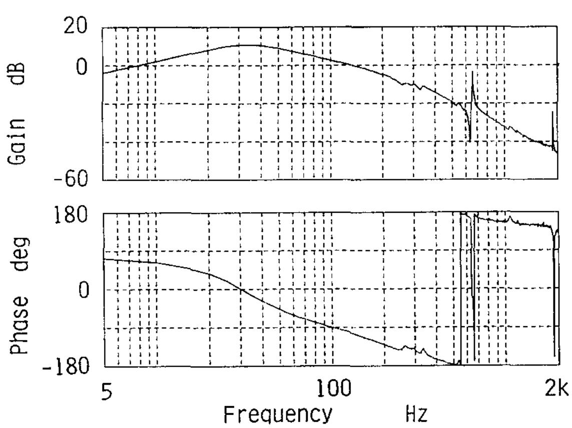


Fig. 8 Frequency response of conical motion for disturbance

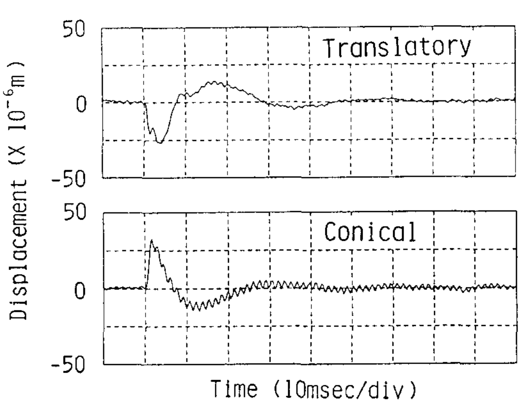


Fig. 9 Impulse responses

have a stable control system. To so do, we set the sampling period sufficiently small, 0.043msec.

Figure 9 gives responses for an impulse added to the rotor near to the bearing. The motion with the peak of about 30μm recovers to the steady state in about 30 msec. A small vibration appears in the transient motion, especially, in the conical motion. This is the second bending vibration which could not be suppressed.

3.3 Whirling of rotor

The whirling of the rotor is shown in **Fig. 10** with the amplitude of the translatory motion in the horizontal direction (the mean of the whirlings in the two bearings). This curve was written down in the free rotation after the driving of the rotor, to remove the influence of the motor driving force[6]. The whirling increases around the first-bending critical speed of about 9,000rpm with the restricted amplitude of 5μm ; after that the whirling increases gradually with rotor speed. until the run comes into instable beyond about 25,000rpm. The gyroscopic effects did not act on the first critical speed; this seems to be due to the symmetricity of the rotor.

4. Conclusions

For an active magnetic bearing system which is symmetiric and an electromagnet and a displacement sensor are set in collocation, a digital signal processor is applied to realizing compensators of control system to suppress the resonance around the first-order bending critical speed. A full-order observer is applied to the control of the translatory motion with a first bending vibration motion, and a parallel PID to the control of the conical motion neglecting a bending motion. In the experiments, the whirling at the first bending critical speed(about 9,000rpm) was suppressed, and the rotor was drived beyond the critical speed through the maximum speed of about 25,000 rpm.

The maximum run-speed is about 60 % of the case of analog control[5, 6] with similar compensators, in which the rotor speed can be raised close to the second bending criticalspeed (about 40,000rpm). Sampling period determines the performance of the vibration control. The primary problem of digital signal processing seems to be the

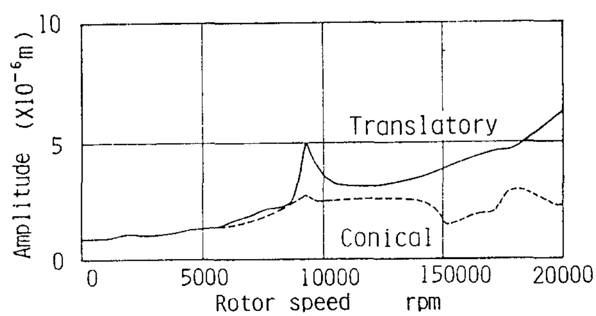


Fig. 10 Whirling of rotor in rotation

phase-lag of signals in high frequencies[3]. We can improve this problem by taking a smaller sampling period, or by introducing a phase-lead compensation in the control signal. By that we may be able to rise the maximum run-speed. From the experimental results, we conclude that this lag may stimulate the response between the stator and the rotor: the resonance cannot be supressed when the phase-lag is large.

Another problem was a noise problem caused by digital processing, the problem that the coil currents fluctuate periodically, when the sampling period is not sufficiently small. The crucial problem of the application of DSPs, however, might be the high cost for the present.

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