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Improvement of Active Magnetic Bearing system using Jerk Feedback Controller for an Elastic Rotor

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Abstract

This paper presents an integrated control system performing levitation and vibration suppression for a magnetic bearing (AMB) supporting elastic rotor. Stable levitation of the elastic rotor is achieved by using PD controller. However, there is a problem with the 2nd mode of the elastic rotor appeared notably. We are aiming to introduce jerk feedback control term for this problem. In previous studies, jerk feedback controller is applied that posses difficulty in tuning. In this study, location of AMBs and the shape of the rotor are modified to solve problem as controllability and observability. Moreover, we introduce acceleration feedback controller coupling to PD controller and phase lead compensator in series. The effect of presented controller is verified by experiment.

Keywords : Vibration Control, Magnetic Bearing, Flexible Rotor, Vibration of Rotating Body

1. Introduction

Active magnetic bearings (AMBs) systems have been applied to various machines such as centrifugal compressor, vacuum pumps and energy storage flywheel systems because AMBs enable to support rotor without friction.

Naturally AMBs are also widely applied to high-speed rotors. However, as the rotation speed increases, more elastic modes of rotor and gyroscopic effect appear. Therefore suppression of rotor vibration is also an important issue for AMB systems. The aim of our study is to develop such an AMB system that can levitate an elastic rotor and suppress its vibration simultaneously.

To develop such system, authors have been developing an experimental rotor-AMB system. The rotor possesses long and thin shape so that it possesses lower natural frequencies.

In the previous studies, to levitate the rotor by using AMBs, local feedback controller for AMBs is designed. In the controller design, Local Jerk feedback controller [1] is introduced to realize stable levitation and vibration suppression of elastic rotor simultaneously. However, tuning of jerk feedback controller is difficult and sufficient damping for the 2nd mode can hardly be obtained.

To modify control performance, mechanical and control system improvement are introduced in this paper. First, to improve controllability and observability, locations of AMBs and the shape of the rotor is modified by shifting positions of sensors and actuators. Its effectiveness is examined through experimental analysis, namely comparison among modal shapes and equivalent masses [2] of the rotor before/after modification.

Moreover, an acceleration feedback controller by coupling a PD controller and a phase lead compensator in series is introduced instead of jerk feedback controller. The effect of presented controller is verified through experiments.

2. Control object

2.1 Flexible rotor

Figure 1 shows an outlook of the flexible rotor produced in this research. Table 1 shows the parameters of the rotor.

Fig. 1 Flexible rotor

Table 1 Parameters of flexible rotor

Mass	4.53 [kg]
Diameter	12 [mm]
Length	1002 [mm]

To carry out multi-mode vibration control experiment, the rotor is design so long and thin that the natural frequencies of rotor possess extremely low. The natural vibration modes of the flexible rotor obtained by suspended experiment possess 24.8[Hz] at 1st mode, 73.2[Hz] in 2nd mode, and 127[Hz] in 3rd mode, respectively. Therefore, in our research, the rotor only needed to be rotated around 4,400 rpm to reach 2nd vibration mode.

2.2 Experimental apparatus

Figure 2 shows a diagrammatical view of the AMBs-flexible rotor experimental apparatus we built. The axis of the rotor is set horizontally and two AMBs are placed near the both end of the rotor. One of the three touchdown bearings is set on the middle of the rotor, while two are placed by the AMBs. Driving motor is set at the position of one-fourth length of the rotor from the right end. Besides, in cross-sectional view, four electromagnets and four gap sensors are located in the “X” form centered with respect to the rotor axis(Fig.3). By introducing such location, stability of each AMB is much increased compared with previous “+ form” location.

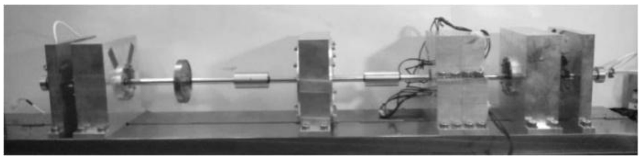


Fig.2 Outlook of experimental apparatus

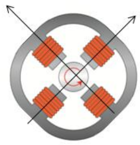


Fig.3 Radial sensor

3.Improvement of controllability and observability by changing AMB location

In case of AMBs supporting an elastic rotor, it is necessary to keep controllability and observability for elastic modes. Figure 4 shows free-free mode shape of previous rotor and locations of sensors and actuators. The actuators and sensors are located at near nodes of 2nd mode. Therefore controllability and observability for the 2nd mode are hardly kept. This fact is estimated to be the reason why the resonant peek of 2nd mode is not suppressed enough in the previous studies.

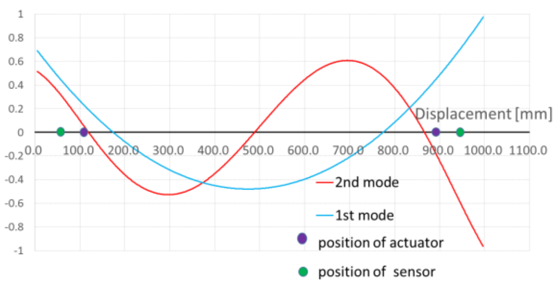


Fig. 4 Free-free mode shape of rotor

In order to modify controllability and observability, locations of sensors and actuators must be separated from the nodes of 2nd mode. To realize this, sensor and actuator are shifted 19mm closer to the ends of the shaft. Figure 5 shows a schematic diagram of this modification, while Figure 6 shows free-free 2nd mode shape before and after shifting actuator. It is shown that actuator is moved away from the nodes of 2nd mode and problem of controllability is modified by moving actuator. As sensors are also shifted in the same manner, the observability is also expected to be modified.

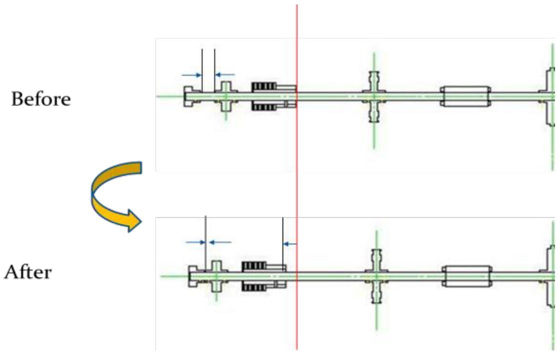


Fig.5 Moving positions of sensors and actuators

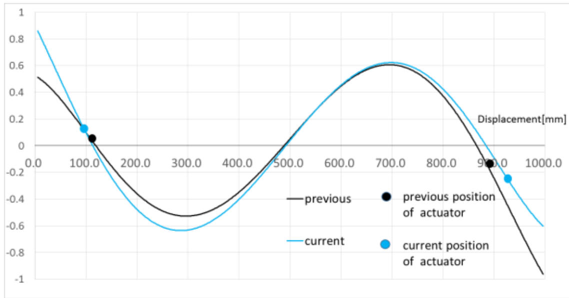


Fig. 6 mode shape before and after shifting

To confirm the effectiveness of shifting, equivalent mass is identified at actuator position of 2nd mode through experimental identification. Table 2 shows the result of identification. Both equivalent masses after shifting become about 0.6 times smaller than before shifting. Beside, equivalent stiffness obtained by using natural frequency of 2nd mode and identified equivalent masses are shown in the same table. Equivalent mass and stiffness are fictitious physical quantities identified onto each mode at specific location. Therefore, it can be considered that the less equivalent mass or stiffness denote higher controllability and observability, because lighter weight and weaker spring means easier control for the same control force, and bigger response for the same excitation force.

Table 2 Equivalent mass and stiffness

	Equivalent mass [kg]		Equivalent stiffness[MN/m]	
	Left	Right	Left	Right
Before shifting	13.5	14.4	2.72	2.91
After shifting	7.50	8.43	1.37	1.54
Rate of change	0.56	0.58	0.50	0.53

4.Controller design

In the experimental apparatus, the observation outputs are four gap distances (namely two directions of two AMBs), while controllable inputs are four voltages for four pairs of electromagnetic actuators (See Fig.3), respectively. Our strategy is to introduce four local feedback controllers for each sensor-actuator pairs (namely two directions of two AMBs).

As stated in “Introduction,” we succeeded stable levitation by using a jerk FB controller in the previous studyt. However, as physical meaning of the jerk gain was not intuitively interpreted, tuning of the jerk controller is difficult especially in the 2nd mode.

To overcome this, an acceleration feedback controller by coupling a PD controller for gap distance and a phase lead compensator in series is introduced instead of jerk feedback controller in this study.

4.1 Problems in the jerk FB controller design

Figure 7 shows block a diagram of the Jerk feedback controller we introduced as local feedback controller. The input for the controller is gap distance, while the output of the controller is control voltage for actuator. ”D” denotes derivative while “K” denotes feedback gains. Gain “Kaa” denotes the gain for the three-order derivative of the gap distance, namely “jerk feedback gain.”

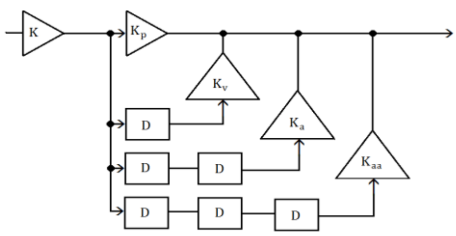


Fig.7 Block diagram of jerk FB Controller

In our previous study, transfer function of local jerk feedback controller shall be described as equation (1).

$$\begin{aligned} G(s) &= K_{aa}s^3 + K_a s^2 + K_v s + K_p \\ &= K_{aa}\left(s+\alpha\right)\left(s^2 + 2\varsigma\omega_n s + {\omega_n}^2\right) \end{aligned} \tag{1}$$

In the equation (1), the natural frequency ω_n and damping ratio ζ are tuned to the 2nd mode of the rotor to realize pole-zero cancellation, while the term "s+α" denotes so-called PD controller. According to this tuning method, we aim at the levitation of the rotor and the suppression of the elastic second mode of the rotor simultaneously.
However, according to this tuning method, the ratio of K_{aa} to K_p is fixed to a specific number as shown in Eq.(2) naturally.

$$K_p \big/ K_{aa} = \alpha{\omega_n}^2 \tag{2}$$

This constraint gives us a trade-off in control performance. To realize stable and stiff levitation, gain K_p shall be as high as possible. Besides, to give sufficient damping effect onto the 2nd mode of the rotor, gain K_{aa} shall be increased. However, due to the constraint shown in Eq.(2) and the fact that α and ω_n are almost fixed, the gain K_{aa} cannot be increased so high. Therefore, using jerk feedback controller, high stiffness and strong damping cannot be realized simultaneously. This property is trade-off.

4.2 Acceleration FB controller

To overcome the trade-off limit, acceleration FB controller by coupling a PD controller and phase lead compensator in series is introduced in this study.

PD controller is introduced to levitate the rotor. Transfer function of PD controller is

$$G_{pd}(s) = K_p + K_v s \tag{3}$$

Besides, phase lead compensator is introduced to add damping to the resonance peak of 2nd mode. Transfer function of Phase lead is as follows:

$$G_d(s) = \frac{\alpha_d T_d s + 1}{T_d s + 1} \quad \left(\alpha_d > 1\right) \tag{4}$$

By connecting these in series, the transfer function of the acceleration FB is as follows.

$$G_c = K\left(K_p + K_d s\right)\frac{\alpha_d T_d s + 1}{T_d s + 1} \quad \left(\alpha_d > 1\right) \tag{5}$$

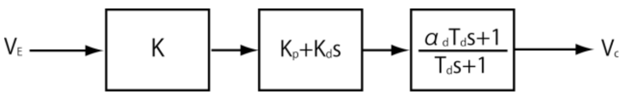


Fig.8 Block diagram of Acceleration FB controller

As shown in Eq.(5), the order of the presented PD-phase lead compensator is the 2nd. This means that the highest gain is multiplied to 2nd-order derivative of gap distance, namely gap acceleration. Therefore we call this presented controller as "acceleration feedback controller."

5.Simple model for controller design

To investigate the controller design, analytical model of control object, namely elastic rotor-AMB apparatus is needed for computational discussion. In this paper, simple model made of electromagnet and a solid levitated object including one degree-of-freedom vibration system is adopted [1].

Transfer function of control object is

$$G_o(s) = \frac{K_s IH}{\left(Ms^2 - K_m\right)\left(Ts+1\right)} \tag{6}$$

where

M : Mass of rotor
K_m : Rigidity value for negative spring
K_s : Coefficient of the electric current for the control power of the electromagnet
I : Amplification rate of the amplifier
H : Amplification rate of the sensor
T : Time constant

By coupling acceleration feedback controller and control object, closed-loop evaluation and control system design can be discussed over computational analysis. The closed-loop transfer function of local feedback control with control object is as (7) by connecting (5) and (6).

$$G(s) = K\frac{\left(K_p + K_d s\right)\left(\alpha_d T_d s + 1\right)K_s IH}{\left(T_d s+1\right)\left(Ms^2 - K_m\right)\left(Ts+1\right)} \quad \left(\alpha_d > 1\right) \tag{7}$$

To evaluate closed-loop property of the presented acceleration feedback controller, closed-loop transfer function is calculated by using MATLAB. Figure 9 shows the result.

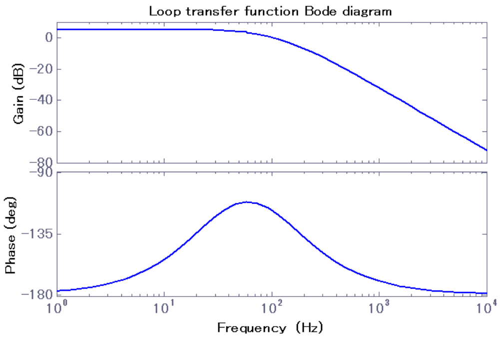


Fig.9 Bode plot of a loop transfer function

In an upper figure, namely gain property, it is clearly shown that the dynamical peaks due to electromagnet and mechanical vibration are well suppressed. Of course this result is just a case with simple model. Precise evaluation using experimental apparatus should be carried out.

6.Experimental analysis

Figure 10 shows a block diagram of Local feedback controller coupled with real control object. In this system, low-pass filter is inserted to eliminate observation noise is modeled as 1st-order low-pass filter. Its cut-off frequency is around 70 Hz.

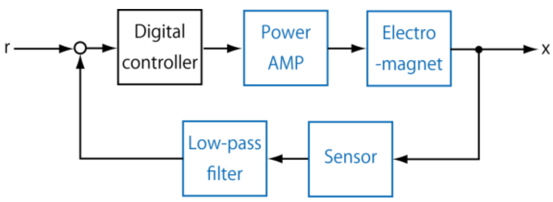
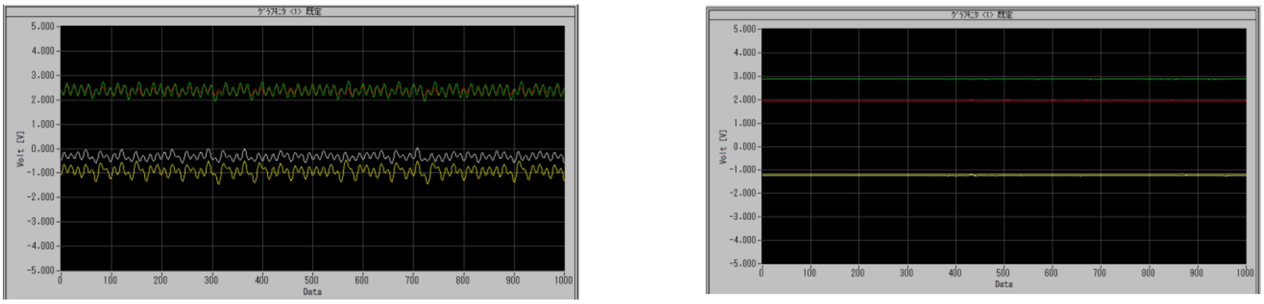


Fig .10 Block diagram of closed-loop system

In the tuning of the feedback gains for acceleration feedback controller, P gains are first tuned according to previous results to realize levitation. Then D gains are tuned so that 2nd mode vibration of the rotor is suppressed. Through this procedure, it is found that there exists the optimal ratio of K_p/K_v. So in the latter tuning process, the ratio is fixed to the optimal value and tuned.

Figure 11 shows time responses of four gap distances of four AMBs with previous jerk controller (a) or present acceleration feedback controller. The horizontal axis denotes time (0-1000 ms) while vertical axis denotes gap sensor output. Voltage 4 (or -4) shows touchdown. The rotor is stably levitated in (a) and (b). Besides, persistent vibration remains in (a) while smooth levitation is realized in (b). This result clearly shows the superiority of presented acceleration feedback controller.



(a) Jerk FB control (b) Acceleration FB control

Fig.11 Time response in levitation

Figure 12 shows frequency responses. The input is impulse excitation force and the output is displacement of the rotor. The horizontal axis denotes frequency (0-160 Hz), while vertical axis denotes gain (gap distance/impulse force, -50 to -150 dB). Blue line denotes the case with previous jerk controller, while red line denotes the case with presented acceleration controller.

The resonance peak of 2nd mode is suppressed in red line, while keen peak remains in blue line. This also supports the effectiveness of the presented acceleration feedback controller.

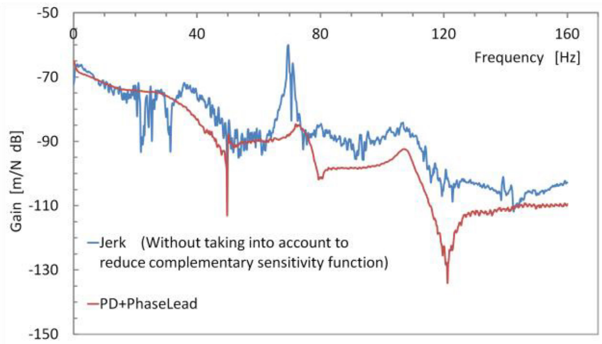


Fig.12 Frequency response in levitation

7.Conclusions

To improve controllability and observability, locations of AMBs and the shape of the rotor is modified to shift position of sensors and actuators. The effectiveness is validated by comparison of mode shape and equivalent mass. Damping performance to 2nd mode is expected to be modified

Moreover, we introduce acceleration feedback controller by coupling a PD controller and a phase lead compensator in series. The effect of presented controller is verified through experiment.

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