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FLEXIBLE SHELL STRUCTURED ROTOR CONTROLLED BY DIGITAL MAGNETIC BEARINGS (TRANSPUTER CONTROL)

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Vibration control using magnetic bearings is applied to a shell structured rotor and a traveling steel sheet. It is intended that the rotor is to be light and to have a wide supporting region of magnetic flux, hence it is made of a flexible shell structure. These structures are apt to produce the shell vibrations, in addition to bending vibrations of the rotating shaft.

To improve the damping property of the structure, a new control scheme for a magnetic bearing is introduced. Each magnet is controlled individually, leading to favorable structural damping, which contrasts to the traditional controller where a pair of magnets are controlled in a push-pull operation.

An alternative similar application of this vibration control is a traveling steel sheet. The rolling process of a thin steel plate sometimes produces bending vibration, which affects adversely the surface finishing quality. Magnetic damper controlled by transputer is applied to reduce the vibration, and tested its capability.

INTRODUCTION

The use of electro-magnetic bearing systems has increased for clean or high speed rotors, mainly because of their noncontact supporting capability. The primary control for this is an analog controller [1,2].

A primary defect of magnetic bearings is poor supporting capability. Here it is intended that the rotor is to be light and to have a wide supporting region of magnetic flux. This paper introduces a flexible shell structured rotor controlled by a magnetic bearing. These structures are apt to result in the shell vibrations.

To improve the damping characteristics of the structure, a new control scheme for magnetic bearing is introduced: Four magnetic actuators are mounted on the shell structure. However, each magnetic actuator is controlled individually to achieve favorable structural damping, which is in contrast to the traditional controller where a pair of magnets are controlled in a push-pull operation.

An alternative application of this vibration control is a traveling steel sheet. The rolling process of a thin steel plate sometimes produces bending vibrations which affect adversely the surface finishing quality [3]. Magnetic damper controlled by transputer is applied to reduce the vibration, and to test its capability.

Digital control has the advantage of flexibility, including the use of higher level controls such as state feedback or adaptive control, and their use has been attempted for the control of magnetic bearings [4,5,6,7,8,9].

A theoretical treatment of digital control for radial bearings was given by Schweitzer and Ulbrich [4]. Bleuler and Schweitzer [5] analyzed the use of decentralized control for radial magnetic bearings which was prepared for the application of microprocessor-based digital control. Traxler, et al. [6] described a radial bearing using a microprocessor-based digital control. Larsonneur and Herzog [7] applied a digital output optimal control. Bleuler and Salm [8] controlled an elastic rotor with a signal processor.

One of the main purposes of this paper is to test the capability of parallel computing transputer control to the magnetic bearing. Analog control of local PD operation is also tested and compared. Then the digital PD control and the state feedback is tried. The results are compared and discussed.

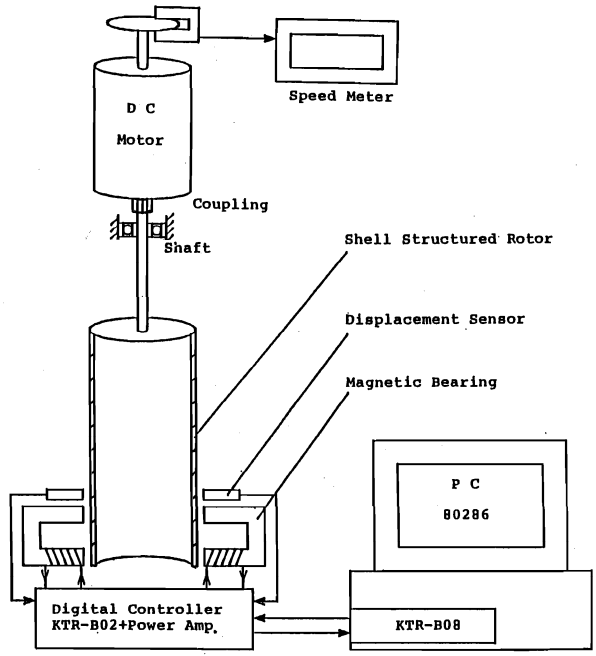


Fig. 1. The Experimental Rotor and Control System

MODELING OF THE PROPOSED ROTOR AND BEARING SYSTEM

The schematic diagram of experimental apparatus is shown in Fig. 1. The upper part of the rotor is a thin shaft supported by a ball bearing. The rotating torque is given by a DC motor through coupling. The shell structured rotor is the lower part which is controlled by a magnetic bearing.

This shell structured rotor has two types of motion: longitudinal bending motion and cylindrical bending vibration. If coupling between them is ignored, the two types of motion can be analyzed separately.

Longitudinal Bending Motion

The equation of motion of the longitudinal bending motion is given by

$$\begin{aligned} m\ddot{x} + kx = f_x \\ m\ddot{y} + ky = f_y \end{aligned} \quad (1)$$

where x and y are the deformations and f_x and f_y are the control forces, respectively. The m and k are the mass and stiffness matrices.

Cylindrical Bending Motion

If it is assumed that cylindrical bending vibration is not affected by the longitudinal bending motion, the equation of ring bending motion applies.

$$\begin{aligned} &\Lambda(\frac{\partial^6 w}{\partial \theta^6} + 2\frac{\partial^4 w}{\partial \theta^4} + \frac{\partial^2 w}{\partial \theta^2}) \\ &+\gamma(\frac{\partial^4 w}{\partial \theta^2 \partial t^2} + 2\Omega\frac{\partial^4 w}{\partial \theta^3 \partial t} + 2\Omega\frac{\partial^2 w}{\partial \theta \partial t}) \\ &= \frac{R^2}{D}\sum_{j=1}^{n} f_j \frac{\partial^2}{\partial \theta^2}\{\delta(\theta - \frac{2(j-1)\pi}{n})\} \end{aligned} \quad (2)$$

Here w is the bending displacements corresponding to the coordinate system which are shown in Fig. 2. Ω is the rotating speed and, Λ and γ are the stiffness coefficients determined by the shell parameters, n is the number of actuators ($n = 4$ in this case).

The solutions w can be assumed as

$$w = \sum_{i=2}^{\infty}(w_{1i}\cos i\theta + w_{2i}\sin i\theta) \quad (3)$$

Substituting eq.(3) into eq.(2) and rearranging them by means of the generalized variable $q_i = \{w_{1i}, w_{2i}\}^T$, we obtain the following equation.

$$\begin{aligned} &M_i\ddot{q}_i + C_i\dot{q}_i + K_iq_i = Q_i \\ &M_i = \begin{bmatrix} \gamma(i^2+1) & 0 \\ 0 & \gamma(i^2+1) \end{bmatrix} \\ &C_i = \begin{bmatrix} 0 & 2\gamma\Omega i(i^2-1) \\ -2\gamma\Omega i(i^2-1) & 0 \end{bmatrix} \\ &K_i = \begin{bmatrix} \Lambda i^2(i^2-1)^2 & 0 \\ 0 & \Lambda i^2(i^2-1)^2 \end{bmatrix} \\ &Q_i = \frac{R^2 i^2}{D}\begin{bmatrix} \sum_{j=1}^{n} f_j \cos\frac{2i(j-1)\pi}{n} \\ \sum_{j=1}^{n} f_j \sin\frac{2i(j-1)\pi}{n} \end{bmatrix} \end{aligned} \quad (4)$$

M_i, C_i and K_i are the generalized mass, damping and stiffness matrices and Q_i is the i-th generalized control force.

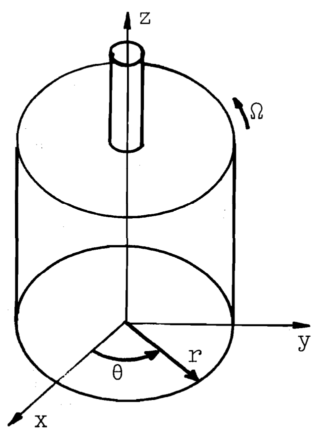


Fig. 2. The Coordinate System

CONTROL SYSTEM

Two types of control system is considered to apply to the shell structured rotor and magnetic bearings.

PD Control

Four magnetic actuators are mounted 90 deg apart on the cylindrical shell. They are controlled individually by the measured signal from four sensors. First, the analog PD controller of

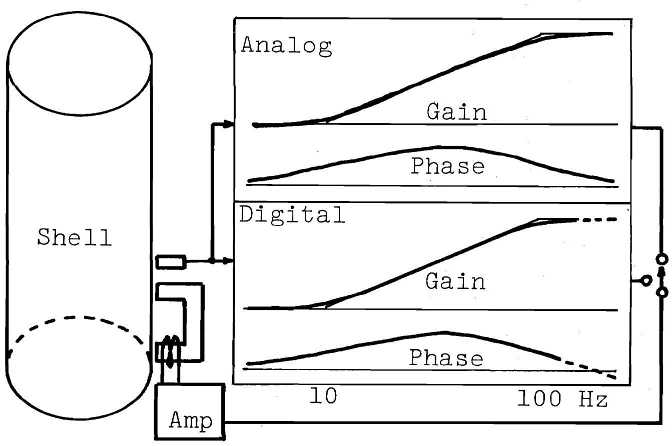


Fig. 3. The Scheme for Analog and Digital PD Controller

$$G(s) = K_P + \frac{K_D s}{T_D s + 1} \tag{5}$$

is installed in the individual actuator, with the sensor as shown in Fig. 3. The equivalent digital PD controller of

$$G(z) = K_P + \frac{K_D(z-1)}{T_D(z - \exp(-\tau/T_D))} \tag{6}$$

is realized by the transputer (KTR-B02, -B08). The frequency responses of the controller are also indicated in Fig. 3.

The State Feedback Controller

The structural vibration should be controlled softly and adding the damping, while the body motion of the rotor should be controlled rigidly. This control algorithm can be realized using state feedback. For the rotating speed of our system (up to 10,000 rpm), only the rigid motion of the longitudinal axis and the first bending ($i = 2$) vibration of the cylindrical axis should be controlled. Hence the state equation for the longitudinal bending motion

$$\begin{aligned} &x_b(k+1) = \Phi_b x_b(k) + \Gamma_b u_b(k) \\ &x_b = \{x, \dot{x}, y, \dot{y}\}^T, u_b = \{f_x, f_y\}^T \\ &\Phi_b = \exp(A_b\tau), \Gamma_b = \int_0^\tau \exp A_b(\tau - t) B_b dt \\ &A_b = \begin{bmatrix} 0 & I & 0 & 0 \\ -m^{-1}k & 0 & 0 & 0 \\ 0 & 0 & 0 & I \\ 0 & 0 & -m^{-1}k & 0 \end{bmatrix}, B_b = \begin{bmatrix} 0 & 0 \\ m^{-1} & 0 \\ 0 & 0 \\ 0 & m^{-1} \end{bmatrix} \end{aligned} \tag{7}$$

and the equation for the cylindrical bending motion

$$\begin{aligned} &x_s(k+1) = \Phi_s x_s(k) + \Gamma_s u_s(k) \\ &\Phi_s = \exp(A_s\tau), \Gamma_s = \int_0^\tau \exp A_s(\tau - t) B_s dt \\ &A_s = \begin{bmatrix} 0 & I \\ -M_s^{-1}K_s & -M_s^{-1}C_s \end{bmatrix}, B_s = \begin{bmatrix} 0 \\ M_s^{-1} \end{bmatrix} \end{aligned} \tag{8}$$

can be treated separately. The state feedback controller is simplified and shown in Fig. 4, where K_{Pb} and K_{Db} are the proportional and derivative gains for the longitudinal bending motion (rigid motion), and K_{Ps} and K_{Ds} are the gains for cylindrical bending vibration.

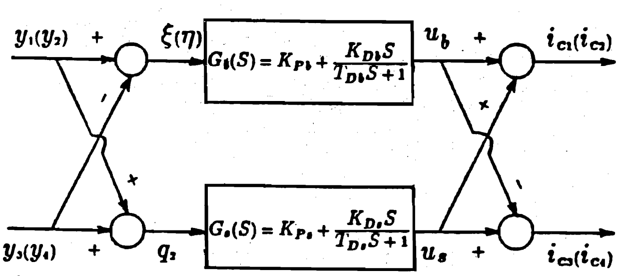


Fig. 4. The Simplified State Feedback Controller

EXPERIMENTAL RESULTS

The proposed rotor and magnetic bearing system are made as shown in Fig. 1. The cylindrical shell is made of 42 mm diameter and 0.5 mm thickness steel. The bearing actuator is diverted from a stator of a 4 phase stepping motor. The unbalance response obtained by an analog PD controller is indicated in Fig. 5. Four curves are obtained from four gap sensors. The fundamental peak near 5,000 *rpm* can be identified. However, the cylindrical bending vibration is far higher than the top speed of the system. Even without bending vibration control, it can run to the top speed. Similar results are obtained by the digital PD controller which are shown in Fig. 6. The control algorithm is realized by a parallel C program on two transputers (B02 and B08). The 4 channel PD control is realized with the sampling interval τ=0.13 ms. However the results are slightly worse than the analog ones. The state feedback has not been applied because of high bending resonant frequency. A large cylindrical shell is now being made to test the state feedback control.

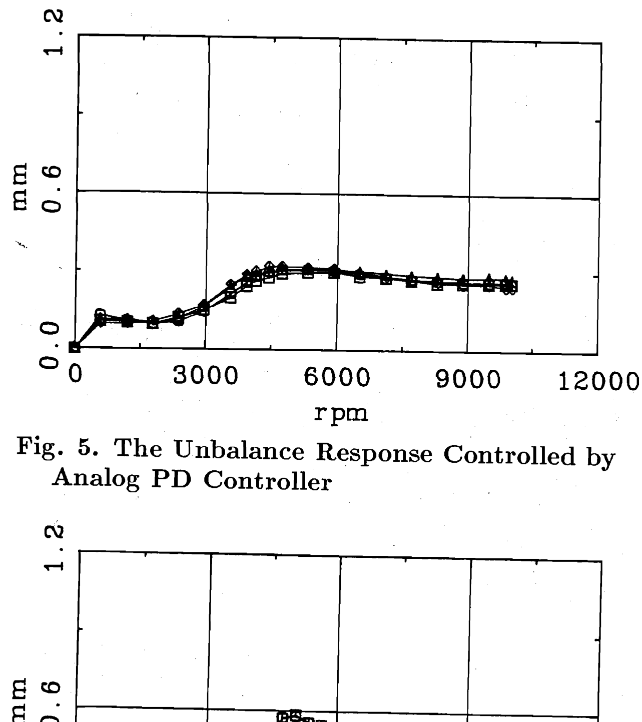


Fig. 5. The Unbalance Response Controlled by Analog PD Controller

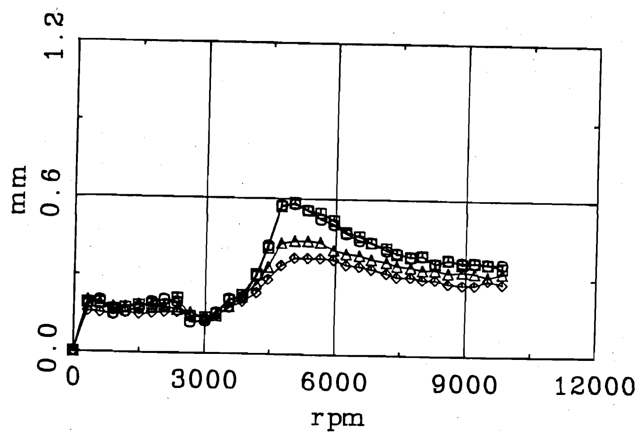


Fig. 6. The Unbalance Response Controlled by Digital PD Controller

APPLICATION TO VIBRATION CONTROL OF A TRAVELING SHEET

Similar control can be applicable to the rolling process of a thin steel sheet. Between the rollers, a sheet is apt to vibrate causing a bad surface finishing. Noncontact vibration control is highly desirable [3]. The schematic diagram is shown in Fig. 7. For experimental convenience, the plate is not moving, which is clamped in x-direction and free in y-direction. Two gap sensors are arranged on the upper side of the sheet, while two actuators are installed under the sheet.

The Vibrating Modes of the Sheet

The equation of plate motion is written by

$$\rho h\frac{\partial^2 w(x,y,t)}{\partial t^2} + D\nabla^4 w(x,y,t) - Th\frac{\partial^2 w(x,y,t)}{\partial x^2} = \sum_{n=1}^{2} f_n(t)\delta(x-x_n)\delta(y-y_n) \quad (9)$$

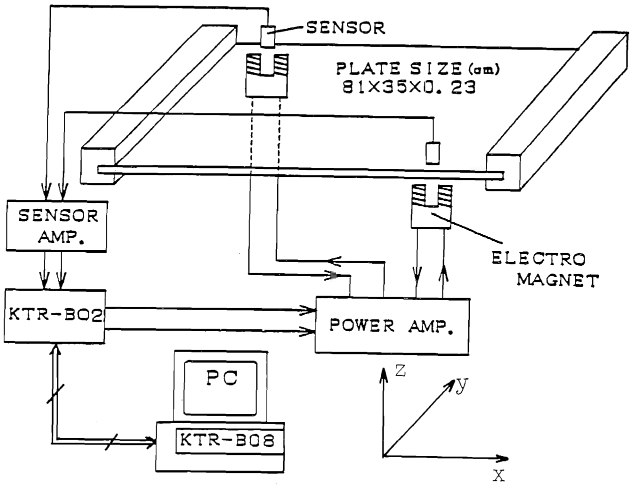
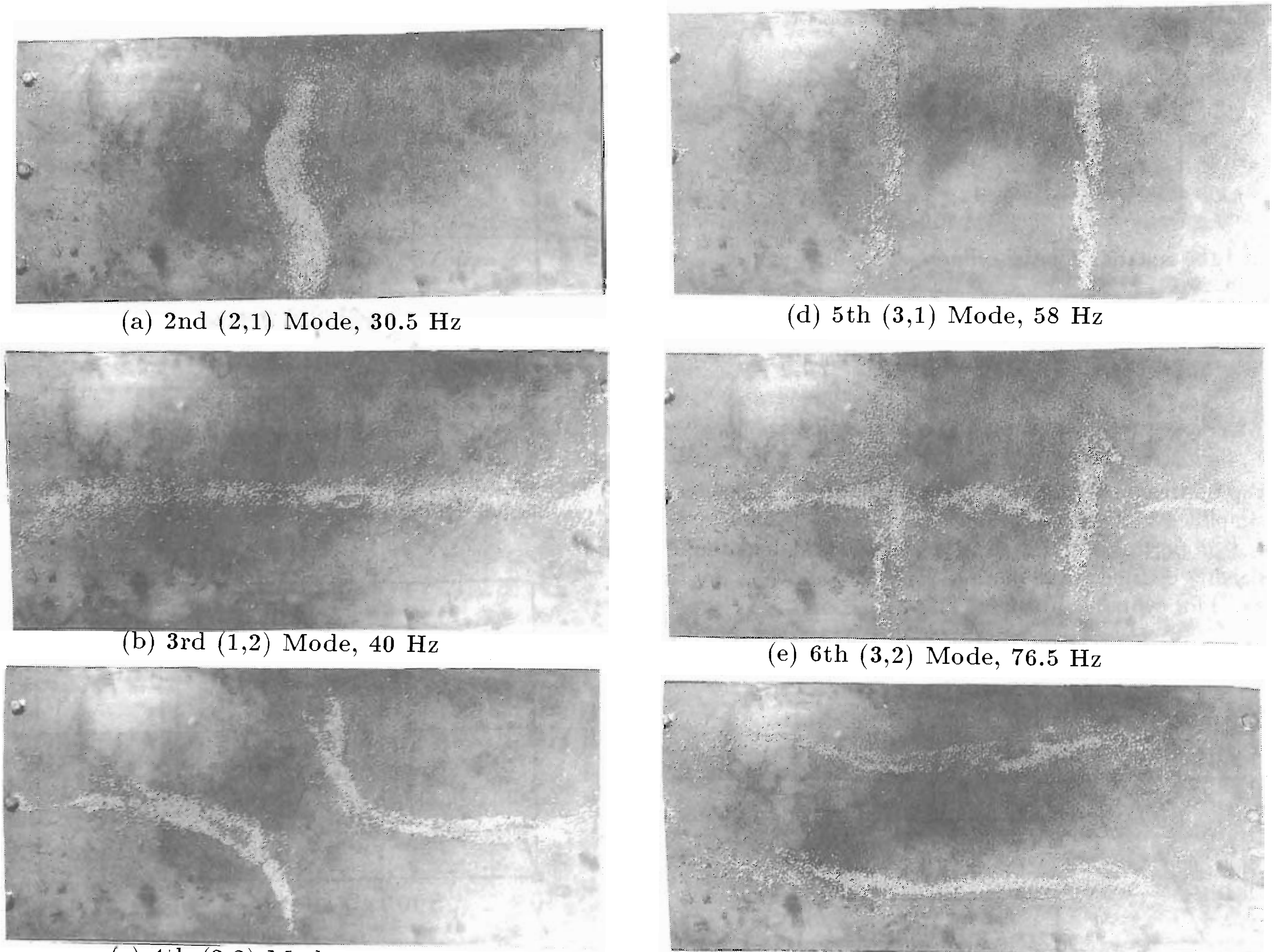


Fig. 7. The Scheme of the Vibration Control of Plate System



(a) 2nd (2,1) Mode, 30.5 Hz

(d) 5th (3,1) Mode, 58 Hz

(b) 3rd (1,2) Mode, 40 Hz

(e) 6th (3,2) Mode, 76.5 Hz

(c) 4th (2,2) Mode, 43.5 Hz

(f) 7th (1,3) Mode, 133 Hz

Fig. 8. The Experimental Modal Shapes of the Clamped-Free Plate

If the vibrating mode is approximated by a product of x and y directional beam modes, then

$$w(x,y,t) = \sum_{i=1}^{\infty} X_n(x) Y_m(y) w_i(t) \qquad (10)$$

can be used. Where $X_n(x)$ is the n-th mode of clamped beam and $Y_m(y)$ is the m-th mode of free beam.

The experimental vibrating modes are obtained by forcing the plate with a sinusoidal signal and taking the picture of the modal shape. The modes of $2.3 \times 810 \times 350mm$ plate are shown in Fig. 8. Unfortunately, the higher mode is not clear. Especially, the mode of thinner plate could not be measured. Hence the actuator location method which eliminate a specific resonances, i.e. pole/zero cancellation, can not be applicable. In this paper a local PD control is used to reduce the plate vibration.

Experimental Results

First, the impulse response test is conducted by forcing the plate with a dropping elastic ball and measuring the response. They are shown in Fig. 9. Where, (a) indicates the response without feedback, (b) is the response with analog PD control, while (c) is the response obtained with digital PD control. The frequency response test is also conducted and shown in Fig. 10. The peaks between 10 - 200 Hz are reduced significantly with the feedback. However, the peak near 500 Hz becomes unstable with a high feedback gain. The previously mentioned digital PD controller is also applied. The results are similar, but not as good as for the analog controller.

CONCLUSIONS

For the purpose of light weight and high load capability, a shell structured rotor is proposed. To reduce the structural vibration, four actuators are arranged and controlled individually from the measured shell displacements. Digital transputer control as well as an analog PD are applied to the magnetic bearing. The control algorithm is easily implemented by a parallel C program. However, the test shell rotor is so small that the resonant frequency of structural vibration is several times higher than the top speed of the system and could not be recognized.

Similar vibration control can be applied to the rolling process of the thin steel plate. Plate vibration occurs between the rollers. The resonances between 10 - 100 Hz are reduced significantly by the PD controller. However, the peak near 500 Hz becomes unstable with a high feedback gain. An improved control algorithm is needed for this system.

REFERENCES

1) Humphris, R. R., Kelm, R. D., Lewis, D. W. and Allaire, P. E., "Effect of Control Algorithms on Magnetic Journal Bearing Properties", J. of Eng. for Gas Turbines and Power, Trans. ASME, Vol.108, Oct. 1986, p.624-632

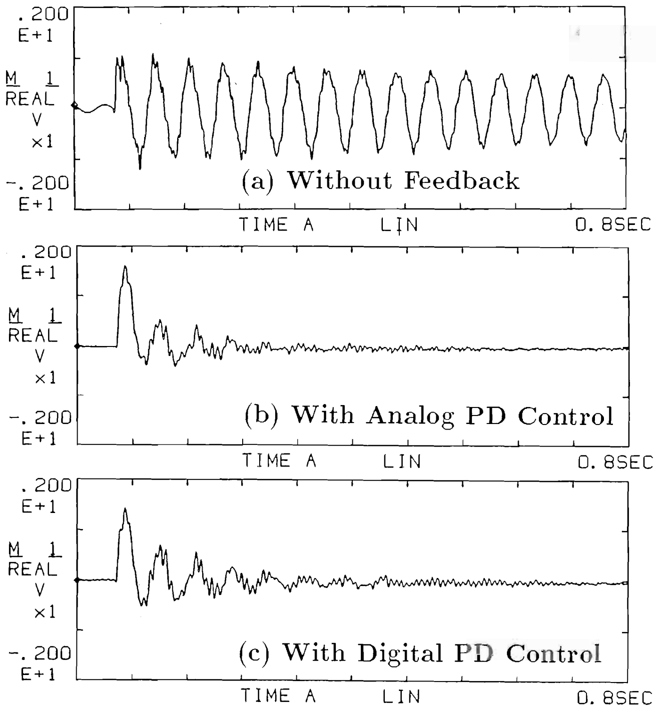
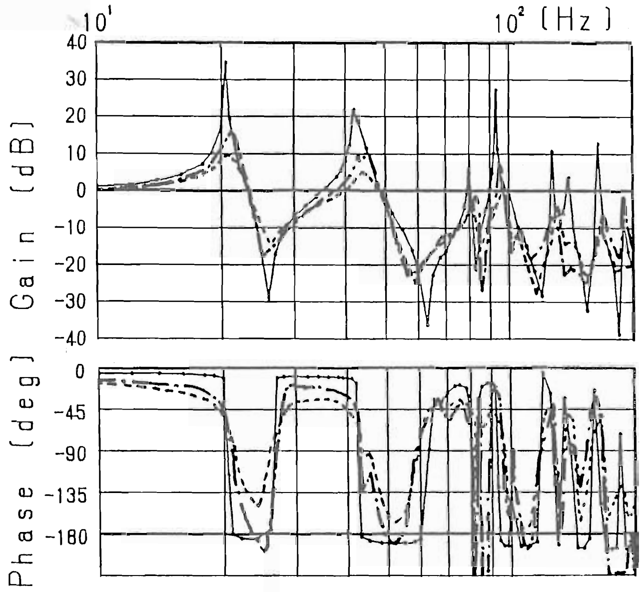


Fig. 9. The Comparison of Impulse Responses Without Feedback, With Analog PD and Digital PD Control



(a) Without Feedback
(b) With Analog PD Control
(c) With Digital PD Control

Fig. 10. The Comparison of Frequency Responses Without Feedback, With Analog PD and Digital PD Control

2) Matsushita, O., Takahagi, M., Tsumaki, N., Yoneyama, M., Sugaya, T., Bleuler, H., "Flexible Rotor Vibration Analysis Combined with Active Magnetic Bearing Control", Int. Conf. on Rotordynamics, IFToMM & JSME, Tokyo, Japan, Sept. 1986

3) Y. Oshima and T. Shimogo, "Electro-Magnetic Levitation Control of a Travelling Elastic Plate", Proc. of The Int. Conf. on Advanced Mechatronics, May 21-24, 1989, Tokyo, Japan.

4) Schweitzer, G. and Ulbrich, H., "Magnetic Bearings - A Novel Type of Suspension", Institution of Mechanical Engineers, Second Int. Conf. on Vibrations in Rotating Machinery, Cambridge, U. K., Sept. 1980, Paper C273/80

5) Bleuler, H., Schweitzer, G., "Dynamics of a Magnetically Suspended Rotor with Decentralized Control", First IASTED Int. Sym. on Applied Control and Identification, Copenhagen, Denmark, June 28-July 1, 1983, p.17-22

6) Traxler, A., Meyer, F. and Murbach, H. P., "Fast Digital Control of a Magnetic Bearing with a Microprocessor", Int. Kongress Mikroelektonik, Munich, Germany, Nov. 13-15, 1984

7) Larsonneur, R. and Herzog, R., "Optimal Design of Structure Predefined Discrete Control for Magnetic Bearings (SPOC-D)", Proc. of First Int. Sym. on Magnetic Bearings, June 6-8, 1988, Zurich, Switherland.

8) Bleuler, H. and Salm, J., "Active Electro-Magnetic Suspension and Vibration Control of an Elastic Rotor with a Signal Processor", 4th Int. Conf. on Vibrations in Rotating Machinery, Sep. 1988, Edinburgh, Scotland.

9) Okada, Y., Nagai, B. and Shimane, T., "Digital Control of Magnetic Bearing with Rotationally Synchronized Interruption", Proc. of First Int. Sym. on Magnetic Bearings, June 6-8, 1988, Zurich, Switzerland.