Lateral Vibration Control of Flexible Rotor Levitated by Magnetic Bearing Using Laminated Piezoelectric Actuator

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ABSTRACT

When the high-speed rotor suffers whirl vibration of bending mode natural frequency, the thrust disk mounted on its shaft end is subject to both lateral and rotational vibration. With the rotor riding on a magnetic bearing, there is danger that the gap between the thrust disk mounted on the rotor and the stator yoke becomes uneven in a circumferential direction so that moment may occur from circumferentially unbalanced magnetic pull between them, thus inducing bending mode natural vibration. Two methods are conceivable for solving this problem and utilizing the thrust bearing to control lateral vibration. (1) One method is by splitting the thrust coil in a circumferential direction according to the gap between the thrust disk and the stator yoke to control lateral vibration. (2)The other method is by inserting a laminated piezoelectric actuator between the thrust casing and the thrust yoke and adding moment to the thrust disk by slanting the thrust yoke by the piezoelectric actuator, thus controlling lateral vibration. This paper discusses the attempt made to control the 1st bending natural vibration of the flexible rotor by the second method. Considering that this attempt resulted in reduction of the gain of the open loop transfer function of the system at the 1st bending natural frequency, this mathod can be considered effective in lateral vibration control. The gain at the natural frequency was reduced by approximately 3dB by compensating the radial sensor signal for phase lead and adding feedback to the piezoelectric actuator.

INTRODUCTION

Magnetic bearings are superior to existing sliding bearings and roller bearings in that allow complete non-contact support. However, they pose a new problem because they have to control journal positions of levitated rotors. Above all,they can not adequately control bending natural frequencies of the rotors they support. With conventional bearings, rotors can be operated relatively easily by passing through their bending natural

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frequencies. However, with magnetic bearings, practical rotating speeds are frequently designed to be used below the bending natural frequencies of rotors. When a rotor supported by magnetic bearings is operated at a speed higher than the bending natural frequency of the rotor, either of two methods can be adopted to solve this problem: namely, mechanically balancing the rotor adequately as an elastic body, or increasing the gains or damping force of the magnetic bearings. However, they have several disadvantages, such as the need for high speed balancing equipment or the need for greater adjustment time because of the intricate control circuits. Therefore, in this paper we have tried to decrease the lateral vibration of the rotor by applying angular moment to the thrust discs tilting the thrust bearing yoke. The test results we report here confirm the good effects of this in reducing vibration.

A NEW METHOD TO CONTROL LATERAL VIBRATION

As shown in Fig.1, the thrust disc of a rotor has tilted rotational vibration when the rotor, supported by two radial magnetic bearings and one thrust bearing, is whirling in the bending natural frequency mode. The magnetic attraction force between the electro-magnetic yoke and the thrust disk increases with decreasing the gap between them. Therefore, in this example, the top edge of the thrust disc is drawn to the left, while the bottom edge is drawn to the right, and therefore the thrust disc moves in an anti-clockwise direction. The thrust disc receives a moment which increases the bending angle of the shaft, and therefore comes to have negative bending spring characteristics. However, it was found that the moment in any particular direction can be generated by controlling the circumferential clearance of the thrust discs. This bending vibration is generated mainly by unbalance of the rotor, meaning the vibration can be decreased by applying a radial force to the rotor to remove the imbalancing force[1][2]. In this vibration mode, since tilt is produced at the shaft end, the vibration can also be reduced by controlling the tilt angle at the shaft end. We adopted this method. The magnetic bearing described above is used for applying angular moment to the end of the shaft.

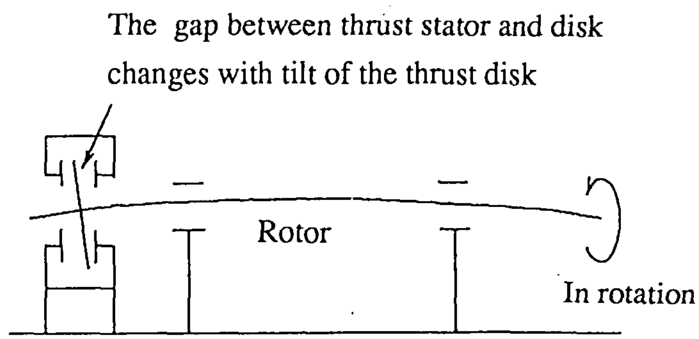

Fig.1 Vibration Mode of Rotor in Rotation

TEST ROTOR

The test rotor was supported by 5-axes controlling type magnetic bearings. We tried to verify the possibility that radial vibration in the rotor could be regulated by controlling the circumferential clearances between stator of thrust bearing and thrust discs. An apparatus shown in Fig. 2 for the test was made and vibration control tests were performed. The test rotor was equipped with an induction motor at the right end and a thrust magnetic bearing at the left end. Two radial magnetic bearings were installed between them to levitate the rotor. Proximity sensors for the radial magnetic bearings were fitted nearer the center of the rotor. Radial proximity sensors were used on the thrust bearing side for the control of circumferential clearances(i.e. bending angles). The rotor was removed from the bearings and hung in a free-free supporting condition, then the natural frequencies and the natural vibration mode of the rotor was measured. The measured data are shown in Fig. 3. In these experiments we paid attention to the first bending natural frequency.

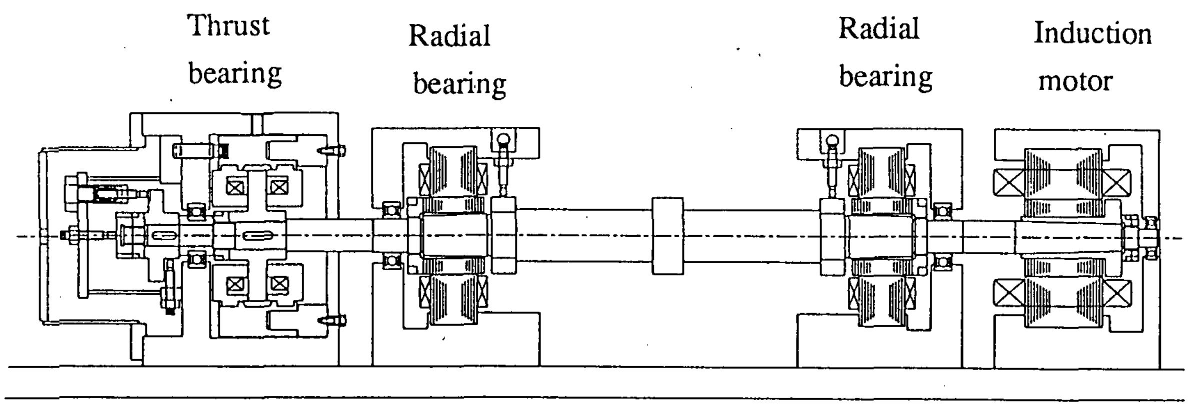

Fig.2 Test apparatus with 5-axis magnetic bearing

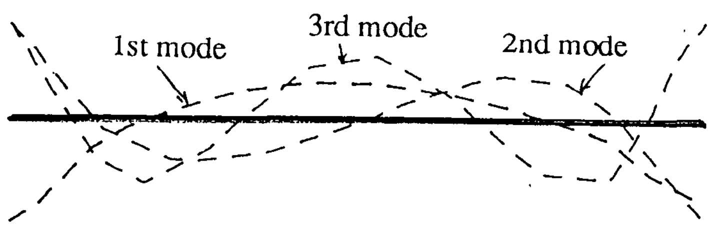

Fig.3 Natural vibration modes of the tested rotor

ESETIMATION OF MOMENT FOR LATERAL VIBRATION CONTROL

The equivalent moment, Iteq, and the equivalent spring constant of rotation of the rotating shaft at the thrust disc Keq were obtained using the two equations on the bending natural angular frequencies of the rotor, ω_n and ω_n', with and without the thrust disc respectively, as shown below:

$$\omega_n = \sqrt{K_{eq}/I_{teq}} \tag{1}$$

$$\omega_n' = \sqrt{K_{eq}/I_{teq} - \Delta I_t} \tag{2}$$

where ΔI_t is inertia moment of thrust disc. Therefore

$$K_{eq} = \omega_n'^2 \omega_n^2 \Delta I_t / (\omega_n'^2 - \omega_n^2) \tag{3}$$

$$I_{teq} = \omega_n'^2 \Delta I_t / (\omega_n'^2 - \omega_n^2) \tag{4}$$

The damping coefficient C required for the actuator to provide the rotor with damping force of resonance amplification factor 5, which is approximately equal to the factor of the sliding bearing support when the rotor passes through the bending natural frequency, is given by:

$$C/C_c = 0.1 \tag{5}$$

$$C_c = 2\sqrt{I_{teq}K_{eq}} \tag{6}$$

Assuming that a vibration with an amplitude of $100\mu m$ is generated at the natural frequency, the control moment required for suppressing the rotor vibration led by imbalance eccentricity of $20\mu m$ to $100\mu m$ is given by:

$$M_{eq} = \theta\omega C \tag{7}$$

The rotor tilt angle θ at the position of the thrust disc is obtained from the natural vibration mode of the rotor. Relations between control moments and tilt angles of the thrust disc for each bending modes are shown in Table 1 when radial vibration amplitude is $100\mu m$ at the radial sensor.

Table 1. Relation between thrust disc tilt angle and control moment

Bending mode	1	2	3	4
$\theta \times 10^4 (rad)$	13.3	11.9	10.6	9.17
$Meq(mNm)$	29.0	38.4	115	207

Assuming the rotor is in the neutral position in the thrust direction, and the same current flows through the magnetizing coils of magnets situated on both sides of the thrust disc, when the disc tilts approximately $\pm 10\mu m$ ($\theta = 2.35 \times 10^{-4}$ rad) , a relation between the moment generated by the magnetic attraction force and magnetomotive force of the electromagnet is obtained by the permeance method as shown in Fig.4. This figure shows that the rotatory spring characteristics is negative in cases where the magnetizing currents of electromagnets are not controlled.

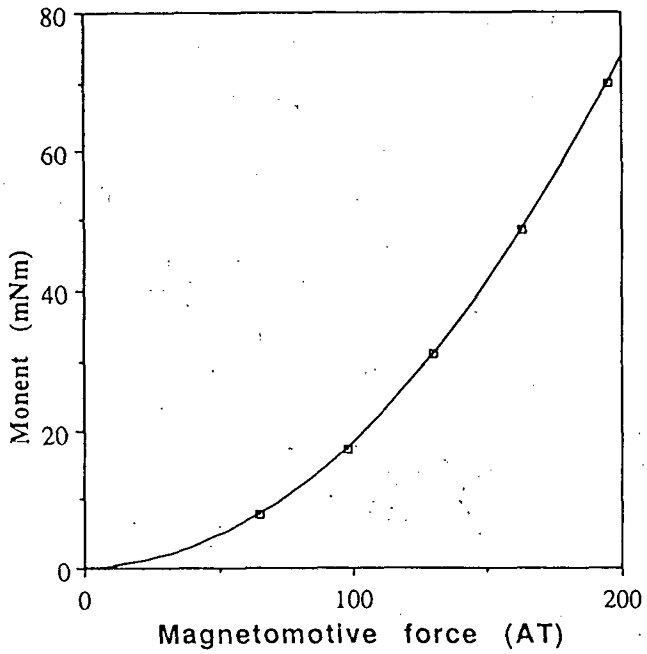

Fig.4 Relation between magnetomotive force and unbalance moment

CONSTRUCTION OF THRUST BEARING

The thrust bearing was modified to generate magnetic moment between the stator yokes of the magnetic bearing and the thrust disk, as shown in the cross-sectional view of Fig.5. Stator yokes on both sides of the thrust bearing are not fixed directly to the bearing casing, but are fixed to the intermediate casing. Certain clearances between both the stator yokes and the thrust disc were specified. The intermediate case to fix two thrust stator yokes floats on the thrust bearing casing, and is not directly fixed to it. For positioning and driving the intermediate case, four laminated piezoelectric actuators, preload bolts and coned disk springs were inserted between the intermediate casing and the bearing casing. Laminated piezoelectric actuators were push against the intermediate case by preload bolts through coned disk springs with low rigidity from the other side of the actuator, thus preventing them sliding off the casing. The dimensions and configuration of the laminated piezoelectric actuator are shown in Table 2 and Fig. 6, respectively.

First, a compensation circuit leads the phase of the output signal from the proximity sensors of the radial magnetic bearing on the thrust bearing side. Then, push-pull signals are generated by the driving amplifier of power amplifier from the compensated signals. Finally, the piezoelectric actuators are driven by the power amplifier. By adjusting the

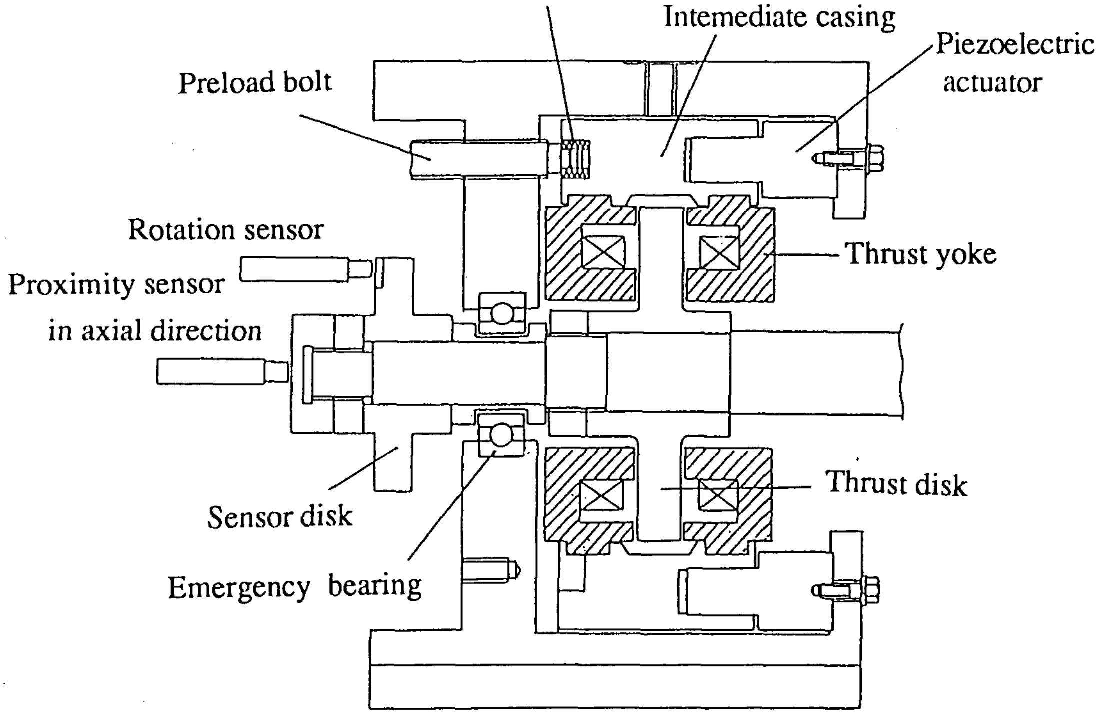

Fig.5 Sectional view of the thrust bearing with piezoelectric actuators

voltage of each of the actuators, situated on the up, down, left and right side of the shaft, the intermediate casing and the thrust yoke are tilted and aligned. The block diagrams of the control system and the transfer function of the phase compensation circuit are shown in Fig.7 and 8.

Table 2 Specifications of laminated piezoelectric actuator

Outer diameter	$11.5mm$
Length	$38.4mm$
Maximum deflection	$17\mu m$
Compliance	$0.017\mu m/N$
Resonance frequency	$14kHz$

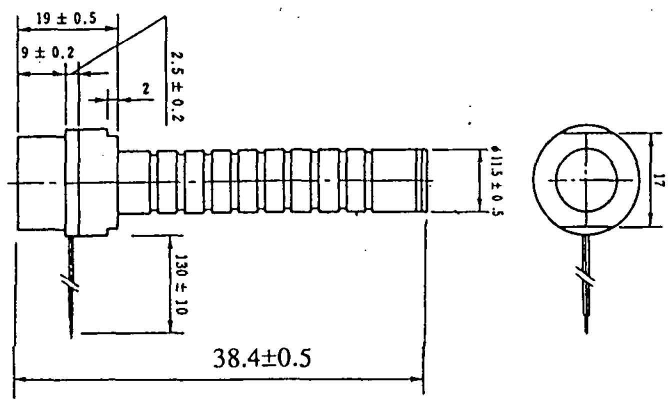

Fig.6 Piezoelectric actuator

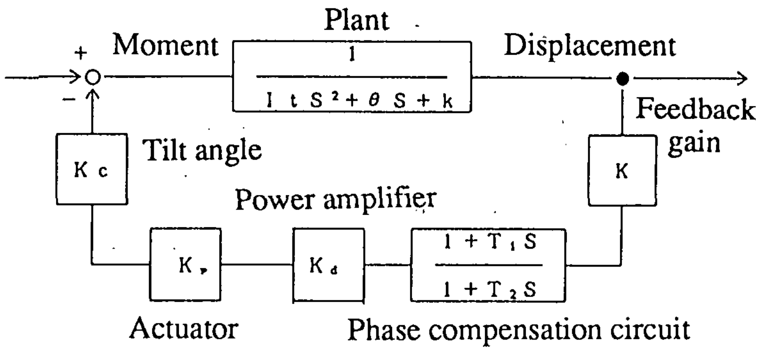

Fig.7 Block diagram of the control system of intermediate casing tilt

RESULTS

AT NON-FLOATING

At first, only thrust magnetic bearings were operated by supporting the rotor with emergency bearings. The effect of the excitation of the intermediate casing of the thrust bearing on the rotor could be directly observed because it was not subject to the vibration control of the radial magnetic bearings. Frequencies were swept at approximately 200Hz by adding 150V to piezoelectric actuators. The resonance frequency rose to 225Hz with the change of natural frequency of the rotor because the rotor was in contact with the emergency bearings. The waveforms of the displacement signals from the acceleration sensors attached to the intermediate case (in the axial direction) and the rotor (in the radial direction) are shown in Fig.9. The rotor could be excited by 9.6μm by the excitation of 17.5μm of the intermediate case, as shown in Fig. 9.

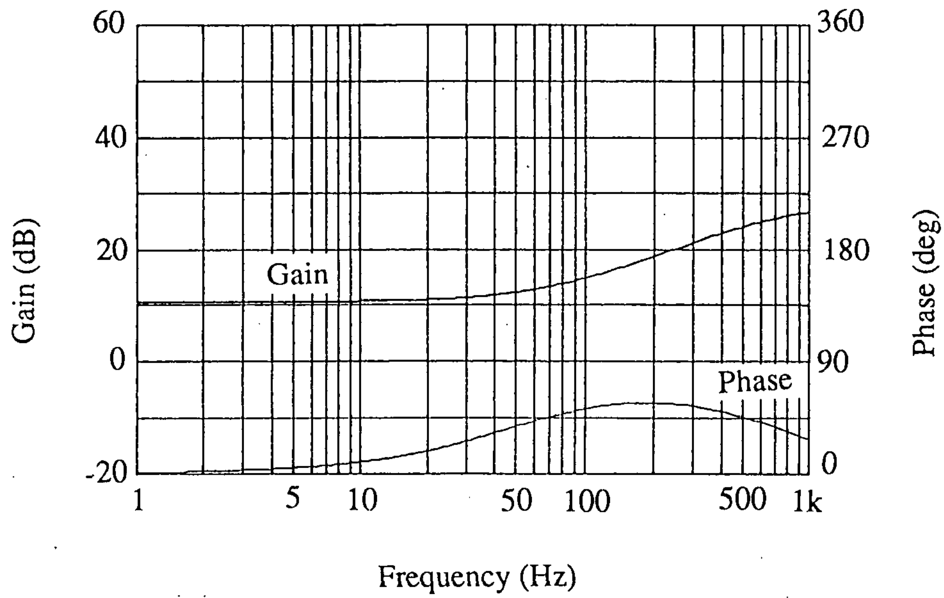

Fig.8 Thransfer function of the phase compensation circuit

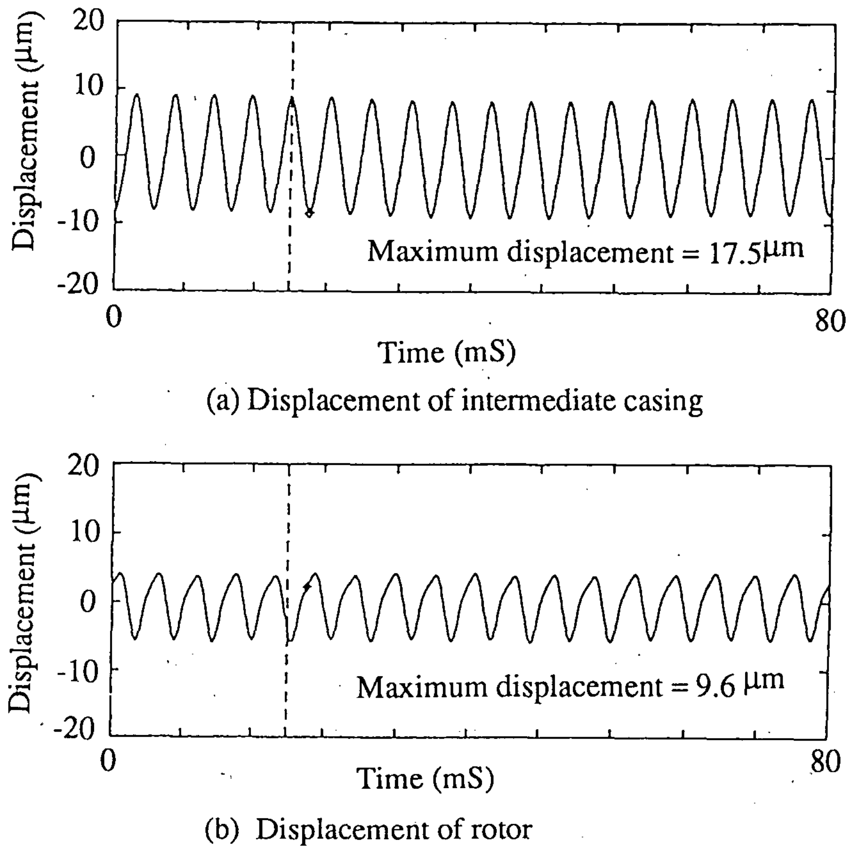

Fig.9 Displacement of the intermediate casing and the rotor on excitaion

AT FLOATING

In case of non-contact support of the rotor by using the magnetic bearing in all directions, signals of the proximity sensors at the radial magnetic bearing nearer the thrust bearing were fed back to piezoelectric actuators. Control moments produced by the control system change as the result of the bias currents of the thrust bearings, while the rigidity of the radial bearing change according to the bias currents of the radial magnetic bearings. Therefore, here we measured the open loop transfer function of the controller in the radial direction with the parameters of both bias currents. Then, we evaluated the effect of this controlling method from the gains in the objective 1st order natural bending frequency. An example of the transfer functions measured at that time is shown in Fig.10. Gains at the peak frequency of 200Hz as seen in Fig.10 were obtained both in the case of control and no control, as shown in Fig.11. The bias currents of the thrust magnetic bearing are indicated on the abscissa, and the gain levels of the peak frequency are indicated on the ordinate. As you can see in Fig. 11, the difference in gain between when the piezoelectric actuator is switched on and off increases as the bias current of the thrust magnetic bearing increases, and as the bias current of the radial magnetic bearing decreases. With the current of the thrust magnetic bearing,4(A), and the current of the radial magnetic bearing,1.5(A), the gain decreased by 3.5(dB) with the controlling effect. When the bias current of the thrust magnetic bearing increased the controlling effect increased due to the increase in the magnetic attraction force between the stator yoke and the thrust disc. On the other hand, when the bias current of the radial magnetic bearing increased the controlling effect of the proposed method decreased.

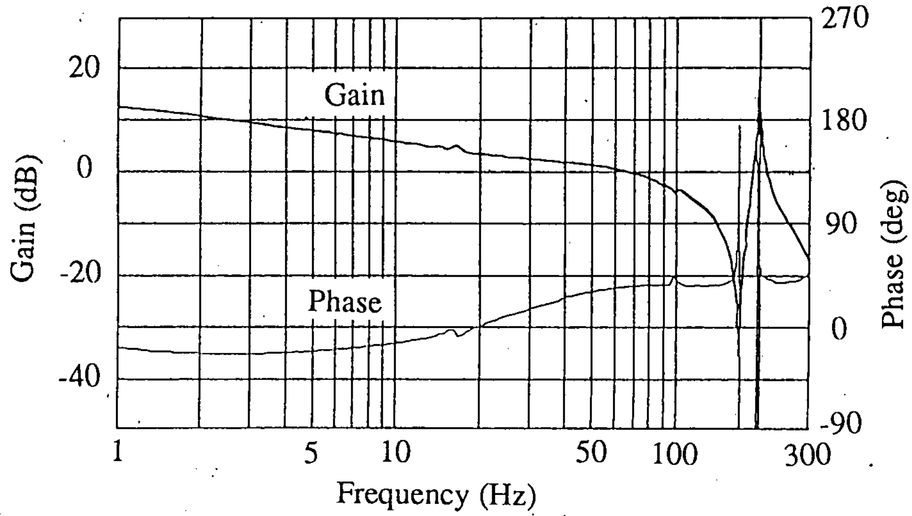

Fig.10 Open loop transfer function of radial magnetic bearing without control of thrust yoke tilt

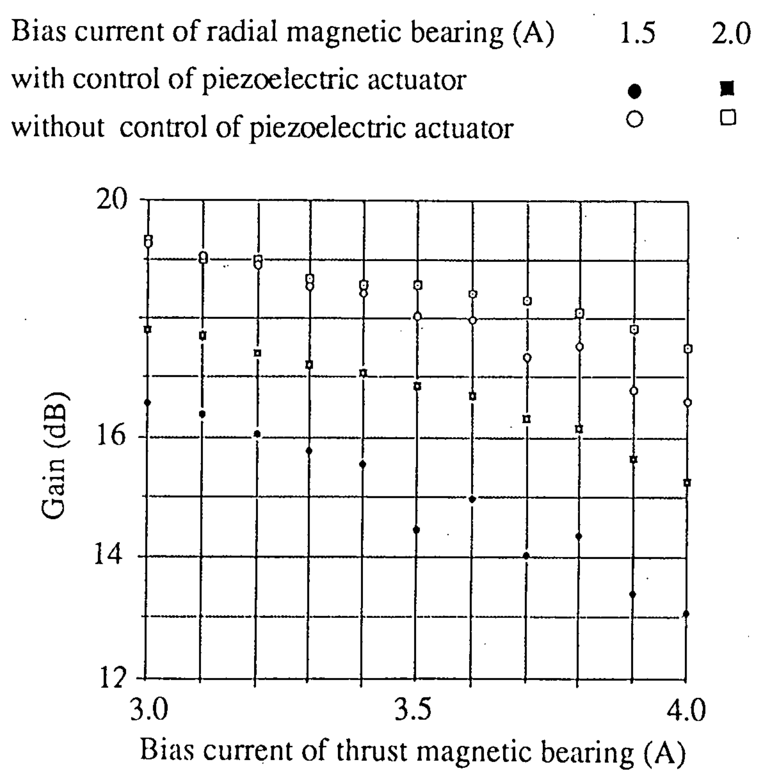

Fig.11 Gain of bending mode natural frequency of rotor

CONCLUSIONS

We propose a method for controlling the radial vibration of a rotor supported by 5-axes control type magnetic bearing, by controlling the circumferential clearance between the stator yokes of the thrust magnetic bearing and the thrust disc. We controlled the circumferential clearance between them by the signals from the proximity sensors of the radial bearings and by the newly adopted laminated piezoelectric actuators. Using this method we confirmed experimentally that the response amplification factor of the bending natural frequency of the rotor can be reduced by 3.5dB, which showed that the proposed controlling method was effective.

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