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Active Vibration Control of Flexible Rotor for High Order Critical Speeds using Magnetic Bearings

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Abstract

Experimental reseach tests were performed by means of active magnetic bearings to actively control the rotor vibrations of a very flexible rotor mounted on ball bearings. The purpose of this test is to verify the efficiency of magnetic bearing for higher order critical speeds. Therefore, we have used a slender shaft without balancing. We have located two magnetic bearings between two conventional bearings. The direct output feedback control is realized by a digital controller of DSP. Also we have used variable feedback gains and on-off control for large amplitudes. Though it was impossible to pass the some critical speeds without magnetic bearings, we can pass four critical speeds with safety when magnetic bearings are working.

1.Introduction

Active vibration control of flexible rotors can be divided into two types on the basis of the actuators as shown in Fig.1. Type 1 uses electromagnetic actuators or magnetic bearings that apply control forces to a rotating shaft and do not contact the shaft. Type 2 uses linear actuators that contact conventional bearing housings.

For type 1 controls a flexible rotor is supported by conventional bearings, and magnetic bearings or electromagnetic actuators are used only for active vibration control. Let us briefly survey the literature for type 1 controls. Burrows and Sahinkaya(1987,1988) and Redmond et al.(1985) have shown the results of active vibration control at the first critical speed with digital controllers. Allaire et al.(1986) and Kasarda et al.(1988) investigated the efficiency of active vibration control at the first critical speed by varying the location of the magnetic bearing. Bradfield et al. (1986) and Nagai et al.(1986) have tried active digital vibration control of rotating-shaft flexural vibration at the first bending mode. Bradfield et al.(1989) have done a programmable electromagnetic force control using variable feedback gains for the first critical speed. Nikolajsen et al.(1979) have presented experimental active vibration control data of a marine transmission shaft that correlate very well with theoretical data. Matsushita et al.(1987) studied liquid stabilization of unstable rotors. Most of these controls use type 1 actuators to apply control forces and are therefore limited in their ability to control higher order vibration modes. Note that all of these papers address active control at the first critical speed. It is of interest to provide active control not only at the first critical speed but also the higher order critical speeds. However we can not see the reports up to this time that the resonances at higher order critical speeds are well controlled.

Several investigators have done research in active vibration control with type 2 actuators. Stanway et al.(1981,1984) considered controllability and conducted a simulation by using state feedback. Moore et al.(1980) and Lewis et al.(1982) used transfer functions to analyze a control system and discussed the effect of velocity feedback based on an experimental study with horn speakers. Nonami has presented the experimental active control results of a Jeffcott rotor with an optimal regulator(1985) and the active control of a multibearing-disk rotor from simulations and experiments by means of quasi-modal control(1986,1988a,1988b). Also Nonami et al. have recently presented excellent data of unbalance force cancellation using feedforward control(1989a) and active vibration control of flexible rotor with eight critical speeds by direct output feedback control(1989b). Ulbrich (1986, 1987) and Fuerst et al.(1988) have evaluated the efficiency of various control rules by using a test rotor on actively controlled supports. Palazzoro et al.(1988, 1989) demonstrated active vibration control by using piezoelectric actuators to apply control forces to bearing housings.

Most papers on type 2 actuators describe the characteristics of vibration control near the second critical speed and higher order modes and show effective vibration control of the higher order modes. Type 2 actuators are more effective in controlling higher order modes because two actuators are used (one at each bearing housing) whereas only one type 1 actuator (magnetic bearing), located between the two conventional bearings, is ordinarily used. Also type 2 actuators generally have greater force capability than type 1. Therefore the higher order modes can be controlled. Generally speaking, electrodynamic force actuators(type 2 actuators) are superior to the electromagnets of magnetic bearings as actuators for vibration control because it is easier to generate the desired control force with type 2 actuators. Also, type 2 actuators are not plagued with the eddy current loss and residual magnetism inherent in magnetic bearings. This enables simpler control system design. A major drawback of the type 2 actuator is the possible coupling of motion when the actuators are mounted at 90° to each other. Hence the motion must be uncoupled by providing a link between the bearing housing and the actuator. This link must have very high longitudinal stiffness and very low lateral stiffness. Also the actuator is larger than magnetic bearing in general. The ideal concept for active control is a flexible rotor supported by only actively controlled magnetic bearings because this concept employs the advantages of both types.

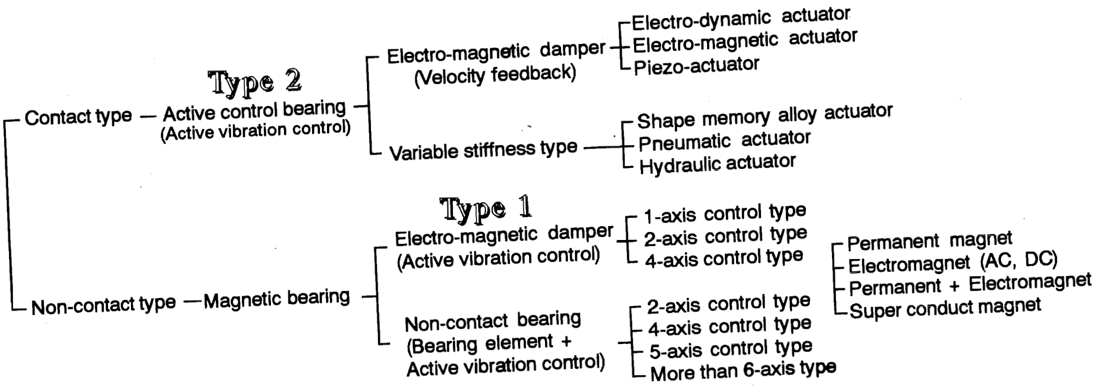

Fig.1 Classification of rotor control method based on control device

The purpose of this study is to verify the capability of magnetic bearing as active control actuator for higher order bending modes. Therefore, we have used a slender and very flexible shaft without balancing supported by ball bearings which has four bending critical speeds under 100 Hz. We have located two magnetic bearings between the two conventional ball bearings. It is difficult to realize a control system with full-state feedback for a flexible rotor that takes into account the higher order modes. Therefore we must utilize a control system with direct output feedback for magnetic bearings. The direct output feedback control is realized by a digital controller of DSP. Also we have used variable feedback gains depending on rotating speeds and have used on-off control algorithm for very large amplitudes. As the results, the active vibration control efficiency has been experimentally verified.

2. Test Rig

The test rig used for the active control experiments is schematically shown in Fig.2. The flexible steel shaft is 10 mm in diameter and 940 mm long and has nine disks of 67 mm diameter. Each disk is 20 mm thick and each mass is 0.5 kg. The total mass of the flexible rotor is approximately 5 kg. The shaft is supported by 10 mm single row, deep-groove ball bearings. The bearing span is 800 mm. The shaft is connected to the high frequency driving motor of 1.5 kW by a flexible coupling. Two active magnetic bearings located between two ball bearings apply control forces to the shaft. The gap between the electromagnets and the disk is about 1.5 mm. The touch down bearings protect in an emergency when the amplitudes exceed about 1.2 mm. The shaft displacements measured by optical sensors are measured as close as possible to the magnetic bearings. These signals are the inputs for the digital controller consisted of DSP through 8 channel analog-digital converter. After the computation of control signals based on the algorithm of Fig.5, the output signals from the DSP is supplied to the magnetic bearings through 8 channel digital-analog converter and power amplifiers.

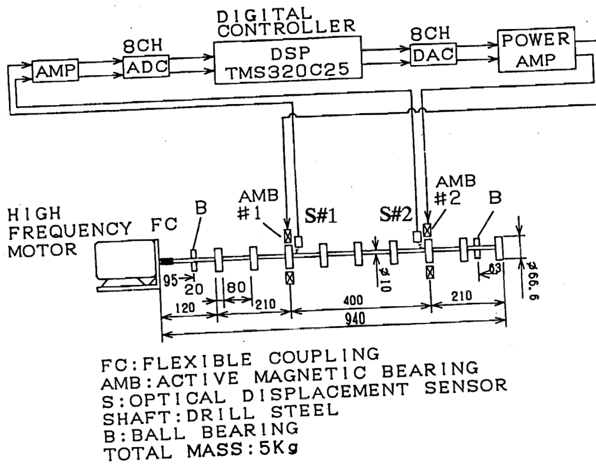

Fig.2 Schematic diagram of test rig

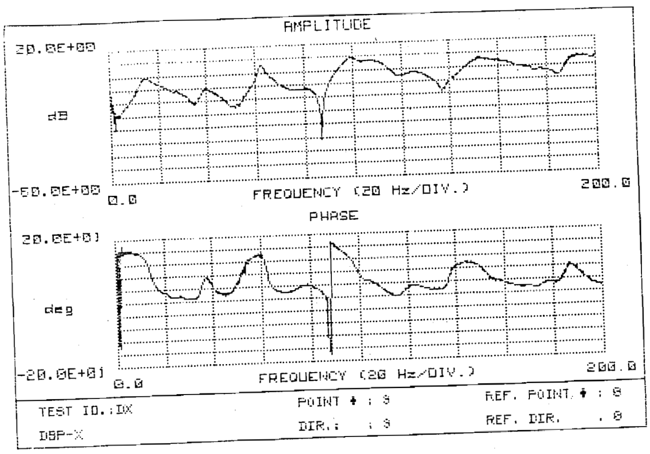

Fig.3 One example of transfer fuction of test rotor with control using AMB #1 and #2 by FFT analyzer

3. Experimental Modal Analysis and FEM Analysis

An experimental modal analysis of the flexible rotor system was performed with a two channel FFT analyzer. Figure 3 shows one of transfer function when two magnetic bearings are working. Figure 4 shows the mode shapes based on experimental FFT analysis. Figure 3 and 4 indicate that there are four natural frequencies under 100Hz. The mode shapes are very simple because the laboratory test rotor is supported as the simple support or hinged support at near both end in addition to two magnetic bearings. From FEM analysis using a mathematical model of Fig.2, the first bending mode of the flexible rotor supported by only conventional bearings is 10.0 Hz, the second is 37.0 Hz, the third is 66.2 Hz, the fourth is 85.5 Hz and the fifth is 117.0 Hz. On the other hand, the natural frequencies in the case of the rotor controlled by two magnetic bearings are shown in Fig.4. Two cases with and without magnetic bearings are almost similar concerning their natural frequencies except the first mode. Also these mode shapes are shown in Fig.4. These analytically determined natural frequencies and mode shapes are very close to the experimentally determined modal frequencies and mode shapes.

4. Control Strategy

It is predicted that the unbalance responses become large because the test rotor is very slender and long shaft. Of course, it is difficult to take a fine balancing for this shaft on account of same reasons. Therefore, we have to think the controller limitter which means the following devise. The moment the maximum deflection in magnetic bearings exceeds some threshold of amplitudes, one of two face to face electromagnets is shut down because the other side of electromagnet immediately recovers the shaft to the stable equilibrium. As soon as the shaft deflection becomes smaller than the threshold amplitude, the electromagnet recovers as the actuator. This concept is based on that the large amplitudes have a strong nonlinearity in magnetic bearings. So, not only the digital linear controller can not carry out such nonlinear control, but also the threat remains as a possibility of a generation for a self-excited vibration caused by the contact between the shaft and the touch down bearing. Figure 5 shows the above mentioned algorithm of a judge by comparison with the threshold amplitude.

One more important thing is to realize the variable feedback gains depending rotating speed. The test rotor has four bending critical speeds under 100 Hz. This gives a hint that four kinds of optimal feedback gains exist for each mode taking into account the saturation of digital controller. It is so called an adaptive algorithm. We have experimentally searched for the optimal feedback gains by means of the experimental FFT analysis and the test run. We selected the three turning speeds for four group of gains and their gains

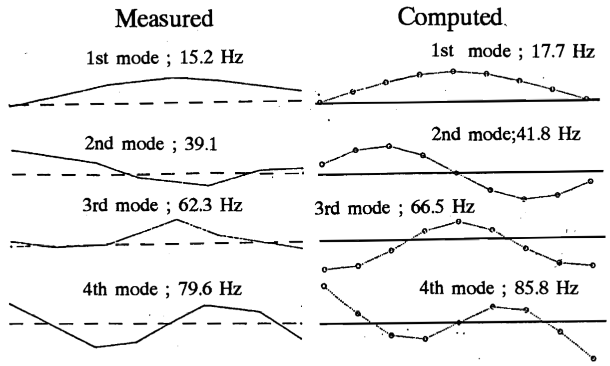

Fig.4 Measured mode shapes and computed mode shapes of of test rotor with control using AMB #1 and #2

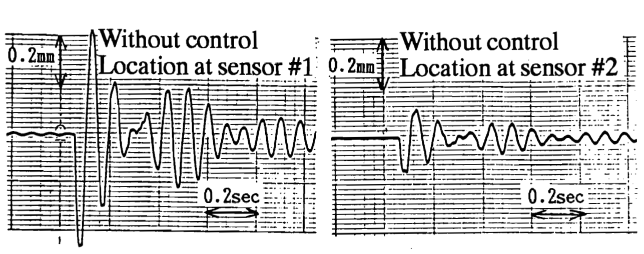

Fig.6 Impulse responses with and without control at sensor locations for the horizontal direction

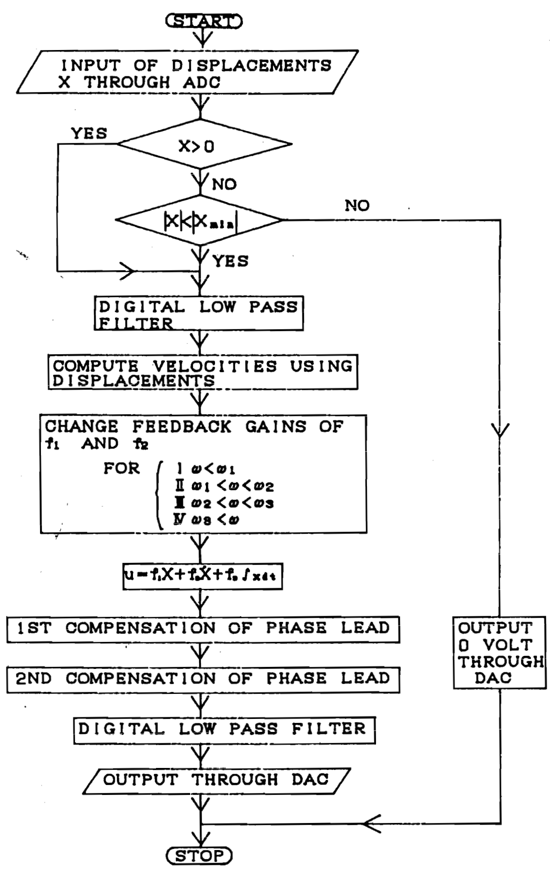

Fig.5 Flow chart of control algoritm to execute in DSP controller

are constant up to the next turning speed. The whole algorithm consists of some digital filters based on PID and phase compensations. This is shown in Fig.5.

We used DSP control to realize the algorithm shown in Fig.5. This DSP chip is TMS320C25 and 40 MHz for clock. The analog-digital converter of 12 bit has a capability of parallel eight channels samples and holds at the same time. The sampling frequency is 200 kHz for each channel. On the other hand, the digital -analog converter of 14 bit has a capability of eight channels outputs simultaneously. This sampling time is 5 μsec. As the result, it is very faster than another DSP chip. In our experiments, the sampling time is 250 μsec to execute the algorithm in Fig.5 for eight channels.

5. Experiments by Means of DSP

Figure 6 shows the impulse responses at locations of two sensors without control and with control using active magnetic bearings #1 and #2. In the case of no control, the responses at two locations are modulated and so complicated. This is caused by the beat of two adjacent natural frequencies between the first bending for the horizontal direction and the first bending for the vertical direction. In the case of control, the amplitudes are immediately reduced to zero within half period. However the small vibrations can be observed near the equilibrium because the velocity feedback gains are so high. The responses are very well in general.

Figure 7 shows the unbalance responses at the two measuring stations, namely two sensor locations of Fig.2 from 0 to 100 Hz for the horizontal and vertical directions. The data are for no control, active control using only AMB #2 and active control using AMB #1 & #2. The data were measured on the condition that the rotating speed was automatically increased with 1 Hz/sec for the acceleration rate. So, it seems that theses data are the unbalance responses in the case of steady state. The unbalance responses without control are big amplitudes because the shaft has big unbalances. As the necessary consequency, the shaft contacts with the touch down bearings. We observed the violent vibration and the rubbing vibration caused by hitting against the touch down bearings at all critical speeds up to 100 Hz. In particular, when the shaft passes through the first critical speed, it was impossible to

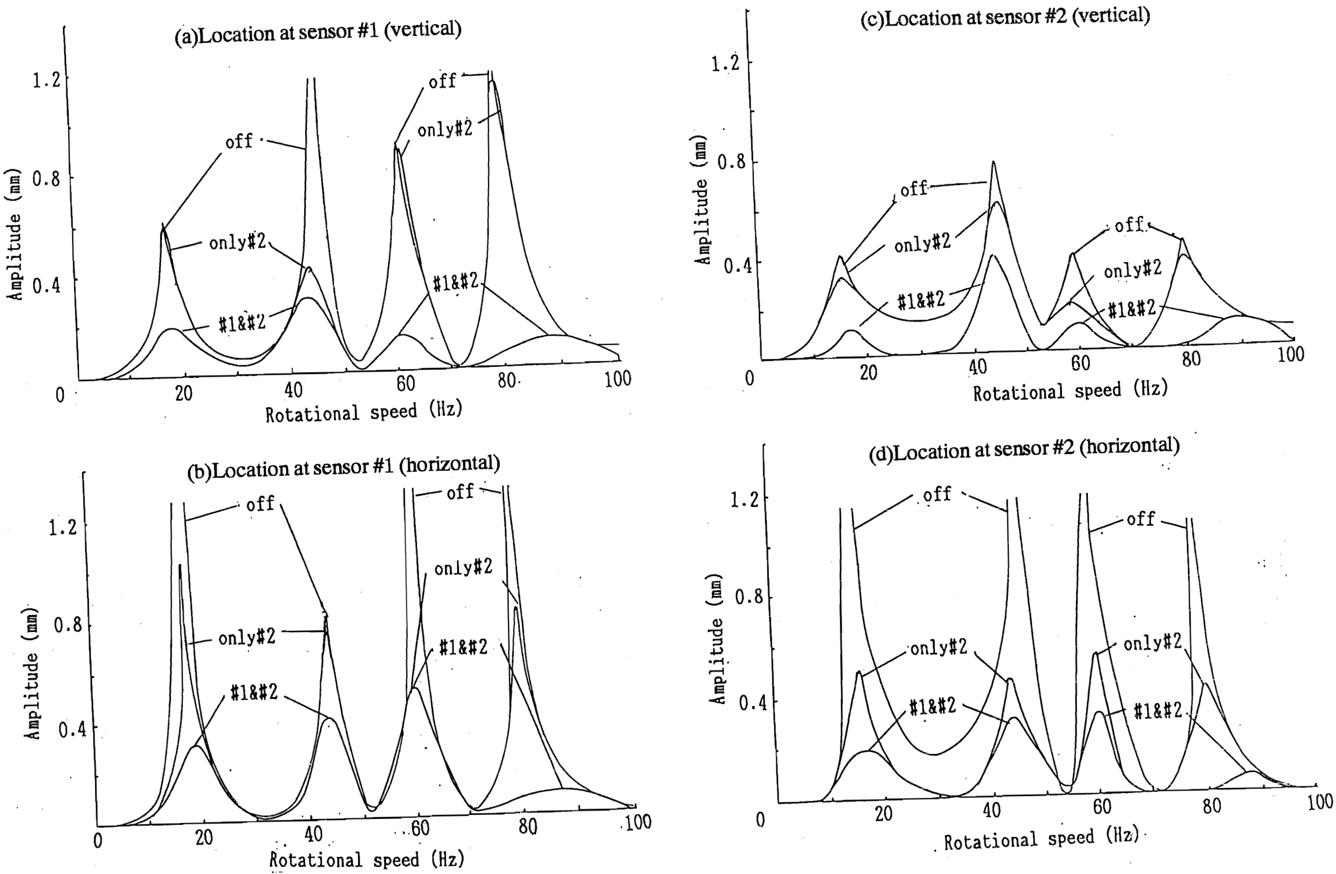

Fig.7 Measured unbalance responses at sensor locations in three cases, namely without control, with control using only AMB #2 and with control using AMB #1 and #2

automatically increase the rotating speed in many cases. Also it was sometimes impossible to pass the fourth critical speed.

In contrast to the case of no control, it is very easy to pass with safety at the every critical speed and it is possible to reduce under the considerable small amplitude in transition through critical speeds in the case of active control using AMB #1 & #2. It was confirmed varying the threshold amplitude of X_{min} that the efficiency of the controller limiter which means on-off switching. This control strategy is so useful for active vibration control of large amplitudes like a nonlinear behavior. Also variable feedback gains are better than constant feedback gains on the point of view for optimization of each mode. The resonance peaks still exist and do not disappear entirely until now. These results do not signify the limitation of AMB. It is sure that we can get more good data near future because the total maximum current of the power source is at the highest 1.5 A including bias currents when total eight electromagnets are working. It is considered that electromagnetic forces are essentially effective for small vibration near the equilibrium. Therefore, the control efficiency might be increased for a fine balanced rotor when the same digital algorithm would be applied.

In the previous discussion, two AMB(#1 & #2) were used for active vibration control. Now, we examine the vibration control effect when only AMB #2 was used for control. Also Fig.7 shows the experimental data. The responses in this case are of course worse than that in the case that AMB #1 & #2 are working regarding vibration control efficiency. This is due to the simple reason of the insufficiency of control forces comparing with strong unbalance forces. Although AMB #2 was working in cases of the previous and the present, the resonance amplitudes are influenced by the control of AMB #1 as shown in Fig.7 (c),(d). This means that the dynamic characteristics with active control in the left side of AMB #1 and the right side of AMB #2 interact with each other. and that vibration control is improved by the superposition.

6. Conclusions

The active vibration control for higher order bending modes, namely up to fourth bending mode, have been tried by means of DSP control. The control strategy with variable feedback gains for small amplitudes and on-off control for large amplitudes have been applied to very slender shaft without balancing. From the experiments, it was impossible for the test rotor to pass the first and the fourth critical speeds without magnetic bearing. By contrast, we can pass four critical speeds with safety using magnetic bearing.

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