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Active Vibration Control for Flexible Rotor by Optimal Direct-Output Feedback Control

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ACTIVE VIBRATION CONTROL FOR FLEXIBLE ROTOR BY OPTIMAL DIRECT-OUTPUT FEEDBACK CONTROL

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ABSTRACT

Experimental research tests were performed to actively control the rotor vibrations of a flexible rotor mounted on flexible bearing supports. The active control method used in the tests is called optimal direct-output feedback control. This method uses four electrodynamic actuators to apply control forces directly to the bearing housings in order to achieve effective vibration control of the rotor. The force actuators are controlled by an analog controller that accepts rotor displacement as input. The controller is programmed with experimentally determined feedback coefficients; the output is a control signal to the force actuators. The tests showed that this active control method reduced the rotor resonance peaks due to unbalance from approximately 250 μm down to approximately 25 μm (essentially runout level). The tests were conducted over a speed range from 0 to 10 000 rpm; the rotor system had nine critical speeds within this speed range. The method was effective in significantly reducing the rotor vibration for all of the vibration modes and critical speeds.

INTRODUCTION

Active vibration control of flexible rotors can be divided into two types on the basis of the actuators. 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 and apply control forces to the 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 reduction 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. 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 at the higher order critical speeds.

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 et al. have 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). Ulbrich (1986, 1987) and Fuerst and Ulbrich (1988) have evaluated the efficiency of various control rules by using a test rotor on actively controlled supports. Palazzolo et al. (1988) 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

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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 (orthogonally). 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 bearings in general.

The ideal concept for active control is a flexible rotor supported by actively controlled magnetic bearings because this concept employs the advantages of both types. It is simpler, however, to actively control flexible shafts mounted on conventional bearings by using type 2 actuators, especially if the actuators produce a high force for a small size. In the future piezoelectric actuators may prove to be such a compact high-force actuator.

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 output feedback. The optimal direct-output feedback control method, the subject of this paper, is the easiest, simplest, and most practical method to implement.

This paper discusses the experimental active vibration control of a flexible rotor supported on two conventional ball bearings. Four electrodynamic force actuators apply control forces directly to the bearing housings. The rotor bearing system has nine critical speeds (without control) between 0 and 10 000 rpm. The optimal direct-output feedback control method was

applied to this system to effectively reduce all nine of the resonance peaks.

TEST RIG

The test rig used for the active control experiments is shown in Figs. 1 and 2. This test rig, called the rotating systems dynamics (RSD) rig, is set up at the NASA Lewis Research Center. The RSD rig is designed to simulate engine structures so that active rotor control and system dynamics interactions can be studied. The flexible steel shaft is 25.4 mm in diameter and 854 mm long and has two disks of 133-mm diameter. One disk is 27.3 mm thick and the other is 18.3 mm thick. The total mass of the flexible rotor including bearing housings is approximately 13.4 kg. The shaft is supported by 20-mm-bore, single-row, deep-groove ball bearings. The bearing span is 670 mm. Each bearing housing is supported on a squirrel cage spring (Fig. 1). The stiffness of each squirrel cage spring is 1180 N/mm. Two electrodynamic force actuators (Fig. 1) located at each bearing housing apply control forces to the bearing housings. These actuators are mounted at 90° to each other as shown in the figure. A nylon flexure is used between the bearing housing and each actuator in order to minimize coupling between the motions of the two actuators. The nylon flexures have very high longitudinal stiffness and very low lateral stiffness. The equivalent support stiffness of the squirrel cage/actuator combination is effectively the stiffness of the squirrel cage spring. Since the flexural rigidity of the nylon flexure is approximately 1 percent of the axial stiffness, we can consider the orthogonal actuator motions to be uncoupled. The electrodynamic force actuator has a frequency bandwidth of 7 kHz, a maximum exciting force of 440 N, and a maximum stroke of 6.5 mm. The rotor is

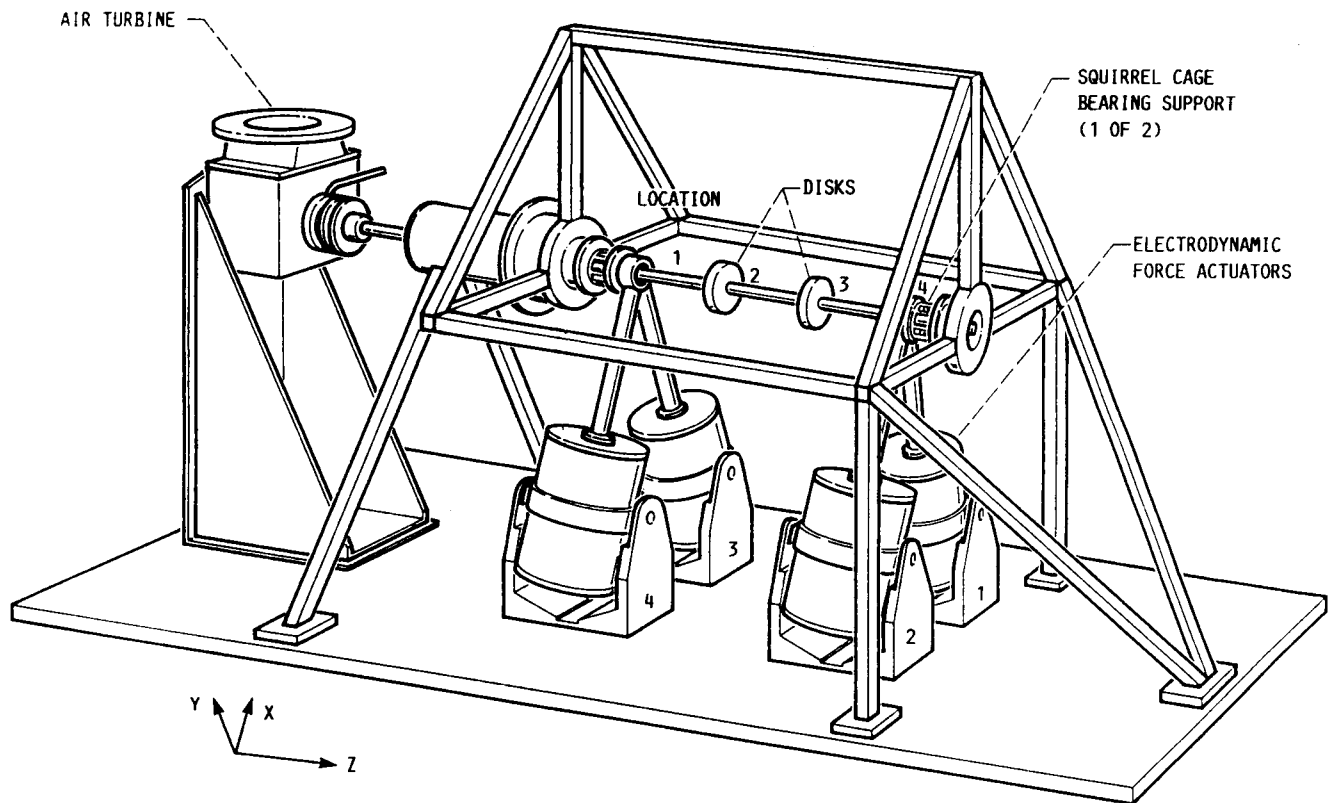


FIGURE 1. - ROTATING SYSTEMS DYNAMICS RIG. CHARACTERISTICS OF ELECTRODYNAMIC FORCE ACTUATORS: VELOCITY, 1.78 m/sec; FREQUENCY, 7000 Hz; FORCE, 445 N; DISPLACEMENT, 6.35 mm.



FIGURE 2. - ROTATING SYSTEMS DYNAMICS RIG.

driven by an air turbine through a flexible coupling. The maximum rotational speed of the test rig is 10 000 rpm. The maximum radial amplitude of the bearing housing is limited to 0.5 mm in order to avoid overstressing the squirrel cage springs (this limit did not compromise the tests). Load cells between the nylon flexure and the bearing housing measure the actual active control forces of the four actuators.

From a NASTRAN analysis (Huckelbridge and Lawrence, 1987) the first free-free bending mode of the rotor is 141 Hz and the second is 304 Hz. The first five modes of the support structure are 35, 41, 61, 72, and 80 Hz. These analytically determined modal frequencies were considerably different from the experimentally determined frequencies discussed in the next section.

The shaft displacements were measured with eddy current proximity probes near the bearing housings and near each rotor disk. Four shaft displacements (two near each bearing housing) were the inputs for the analog controller. The analog controller consisted of proportional and differential control and filters and had four independent channels for displacement input and control signal output. The controller received displacement signals from the probes, differentiated the displacement, applied a feedback coefficient, and in turn outputted a control signal to a power supply that drove the electrodynamic force actuators. This feedback method is called optimal direct-output feedback control. Figure 3 shows an example of the frequency responses of the differentiator.

EXPERIMENTAL MODAL ANALYSIS

An experimental modal analysis of the flexible rotor system was performed with a structural dynamics modal analyzer. The results are shown in Fig. 4 and Table I. Figure 4 indicates that there are eight natural frequencies under 10 000 rpm. The mode shapes are complex because both the support structure and the shaft are flexible and there is a high degree of interaction between them. Therefore it was difficult to accurately identify all of the mode shapes from a mathematical model of the system as previously mentioned. The experimental modal analysis produced modal frequencies that coincided more closely with the rotating critical speeds than did the mathematical model.

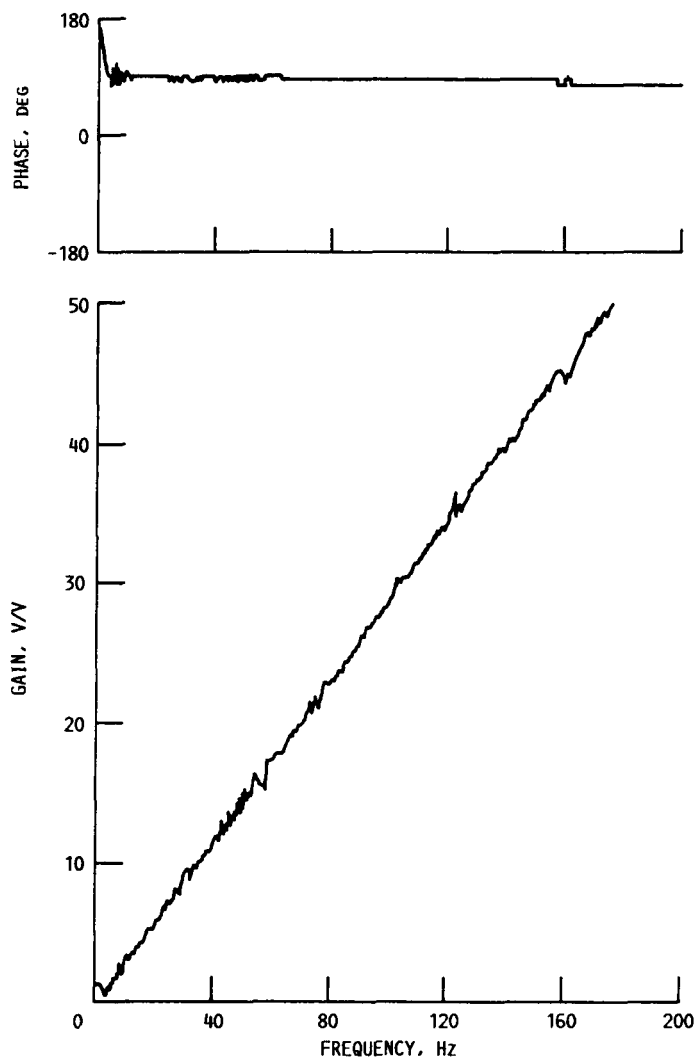


FIGURE 3. - FREQUENCY RESPONSE OF DIFFERENTIATOR.

EXPERIMENTS USING OPTIMAL DIRECT-OUTPUT FEEDBACK CONTROL METHOD

In order to provide damping for the various vibration modes, feedback coefficients must be set in the controller. These coefficients may be calculated from mathematical models for the mode shapes of the rotor system, or they may be determined experimentally. Experimental determination of the coefficients consists of operating the rotor system and setting the controller feedback coefficients such that the vibration amplitudes are minimized at the various critical speeds. Since the mode shapes for the rotor were very complex and could not be accurately predicted by the mathematical model, the feedback coefficients were determined experimentally. This method is called optimal direct-output feedback control.

During the experimental tests it was found that the electrodynamic force actuators (Fig. 1) produced a high degree of passive damping when their power amplifiers were turned on even with the controller off (i.e., with no active control signals being sent to the power amplifiers). This passive damping must be

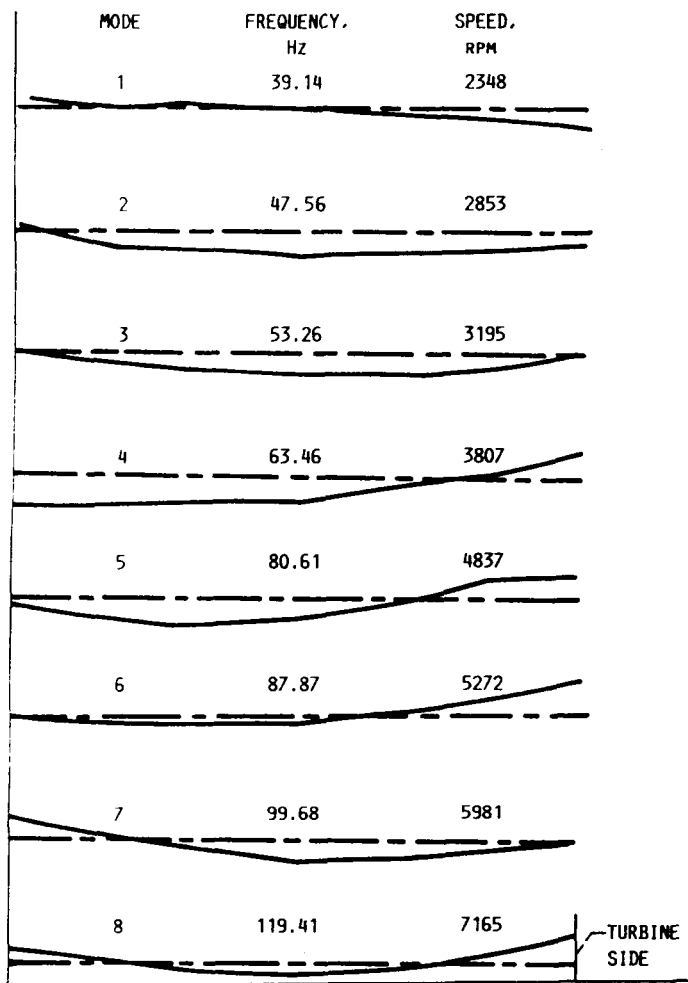


FIGURE 4. - EXPERIMENTAL ROTOR MODE SHAPES DETERMINED WITH STRUCTURAL DYNAMICS MODAL ANALYZER.

TABLE I. - SUMMARY OF EXPERIMENTAL MODAL ANALYSIS

Mode	Frequency, Hz	Damping, percent	Modal mass, kg	Modal damping, N sec/m	Modal stiffness, N/m
1	39.1	1.60	0.71	5.59	43.07
2	47.6	2.11	.61	7.38	52.35
3	53.3	1.53	.52	5.36	58.61
4	63.5	1.16	.44	4.07	69.83
5	80.6	.82	.35	2.88	88.70
6	87.9	.94	.32	3.28	96.69
7	99.7	2.40	.28	8.40	109.74
8	119.4	1.67	.23	5.85	131.43

accounted for in evaluating the total active control damping. The tests consisted of the following:

- (1) Determining the baseline uncontrolled unbalance response for all critical speeds between 0 and 10 000 rpm
- (2) Determining the passive damping effect of the electrodynamic force actuator/power amplifier combination
- (3) Determining the total damping effect of the active control system

Two types of optimal direct-output feedback control were tested. Active control 1 uses velocity feedback only, and active control 2 uses combined displacement and velocity feedback. Separate active control tests were

performed with all four actuators (two per bearing housing), then with only two actuators (one at each bearing housing), and finally with only two actuators (both at one bearing housing).

From the baseline undamped unbalance response test, resonance peaks were observed near 2400, 2780, 3030, 3150, 3870, 4780, 6200, 6700, and 7030 rpm in either the x-direction or the y-direction. Most of these modal frequencies agreed reasonably well with the experimental modal analysis. The peak amplitudes near 2400, 6200, 6700, and 7030 rpm were considerably less than those at the other speeds. Although it was not necessary to actively control the resonances at these speeds, the active control was still applied so that the uncontrolled amplitudes could be compared with the controlled amplitudes in this speed range.

Case When All Four Actuators Were Used for Control

Figure 5 shows the time history of the experimental data at disk 2 (Fig. 1) from 2700 to 3200 rpm. These data show the amplitudes in the x_2 and y_2 directions (orthogonal) for three cases: no control, passive control, and active control 1. It was found that the passive electromagnetic damping effect contributed considerable amplitude reduction and that the resonance peaks disappeared entirely when active control 1 was applied.

Figure 6 shows the unbalance response at the four measuring stations (Fig. 1) from 2500 to 5000 rpm for the x and y directions. The data are for no control, passive damping, and active control 1. The damping characteristics of active control 1 are excellent; the unbalance response amplitudes were reduced to less than 25 μm at every measuring station. The amplitudes were reduced to runout level and dynamic deflections were reduced to zero. The unbalance amplitude response of active control 2 (velocity and displacement feed-

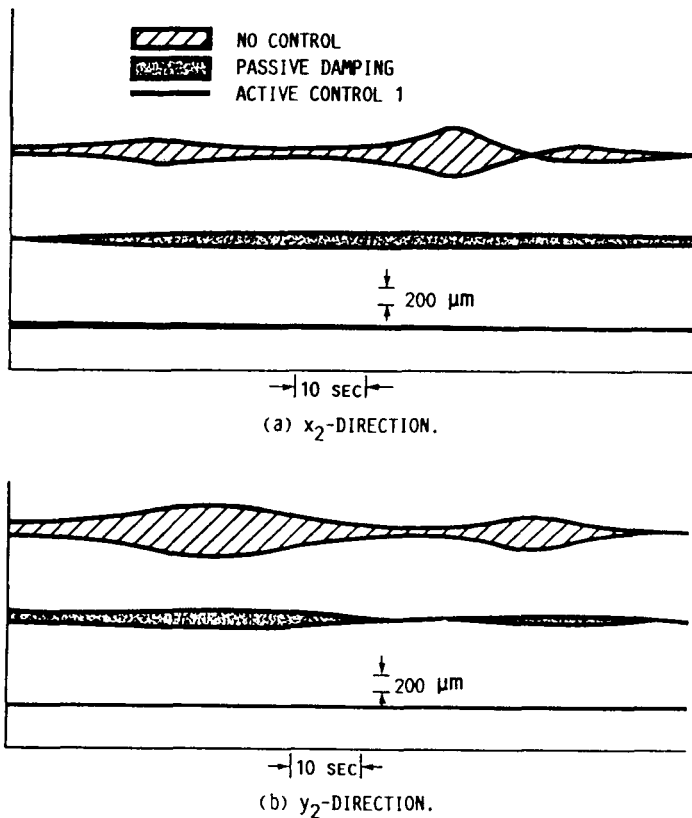


FIGURE 5. - EXPERIMENTAL AMPLITUDE TIME HISTORY FOR APPROXIMATELY 2700 TO 3200 RPM.

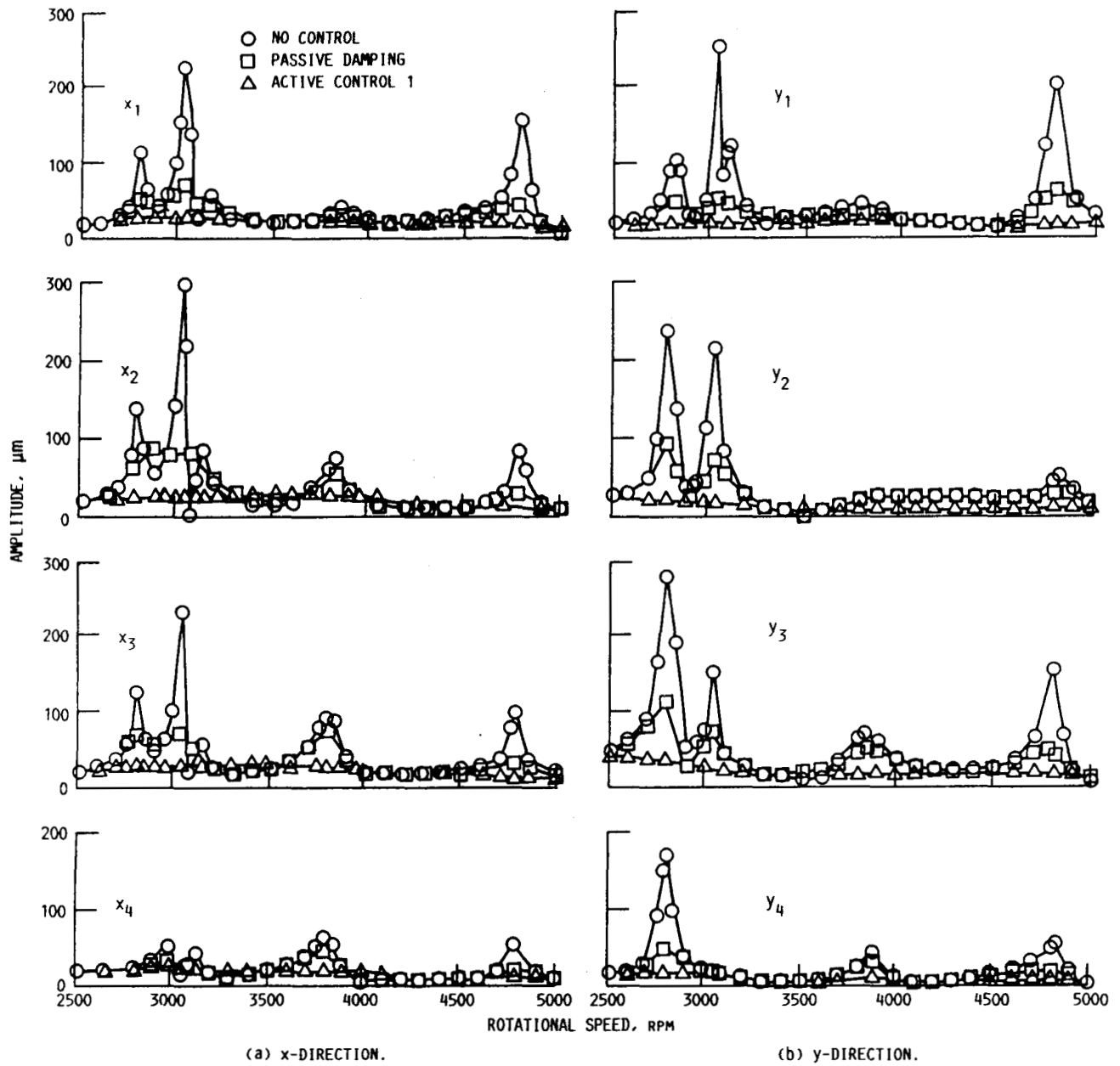


FIGURE 6. - EXPERIMENTAL UNBALANCE RESPONSE FOR NO CONTROL, PASSIVE DAMPING, AND ACTIVE CONTROL 1.

back) was comparable to that of active control 1 (velocity feedback only). Hence active control 1 was adequate for active control of this test rotor.

Figure 7 shows experimentally measured control forces for active control 1 associated with the response of Fig. 6. As previously mentioned, the active control forces were measured by load cells located between the nylon flexures and the bearing housing. Figure 7 shows that the active control forces increased near critical speeds. Note that the control forces near the critical speed of 3050 rpm were smaller than those near the critical speed of 3800 rpm. This is due to the differences in the mode shapes. The maximum control force required was approximately 27 N, and the average control force was approximately 13 N. Only 6 percent of the maximum available actuator power and only 3 percent of the available average power were used

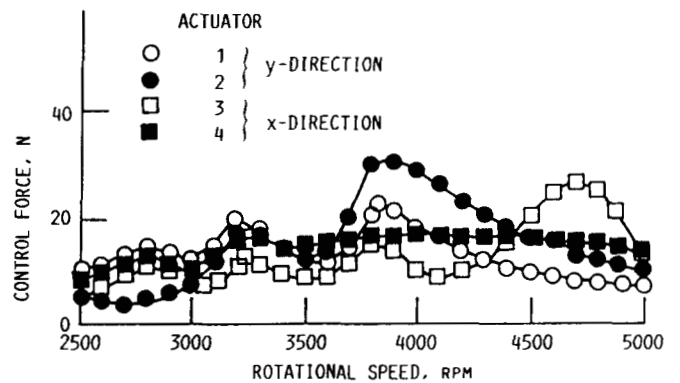


FIGURE 7. - EXPERIMENTAL CONTROL FORCES FOR ACTIVE CONTROL 1.

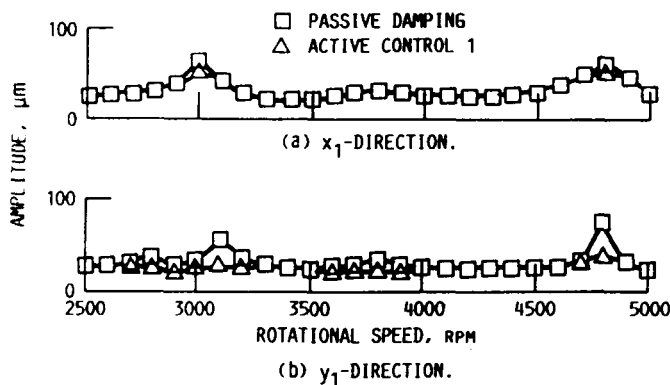


FIGURE 8. - EXPERIMENTAL UNBALANCE RESPONSE WITH CONTROL ONLY IN y_1 -DIRECTION (ACTUATORS 1 AND 3).

in these tests. The required actuator power, of course, depends on the amount of unbalance. For the same critical speeds and the same unbalance forces, we would be able to control a rotor of approximately 200 kg by using the actuators in this test rig.

The maximum uncontrolled amplitude at the critical speed near 2400 rpm was only 40 μm . The maximum amplitude at the critical speed near 6700 rpm was about 80 μm . Applying active control 1 reduced all of the resonance peaks between 0 and 10 000 rpm to less than 25 μm .

Active Control When Only Two Actuators Were Used for Control

Case when only one direction was controlled. In the previous discussion, four actuators (two at each bearing) were used for active vibration control. Now we examine the vibration control effect when only the two actuators for the y -direction (actuators 1 and 3) were used for control.

Figure 8 shows the experimental unbalance response at location 1 for the x_1 and y_1 directions when actuators 1 and 3, both in the y -direction, were used for control. The characteristics for the y -direction, which was actively controlled, almost coincided with the responses of y_1 shown in Fig. 6 (case for four actuators). Although the y -direction should have no effect on the x -direction, the x -direction was influenced by the control in the y -direction and the unbalance response amplitudes in the x -direction were also reduced. This means that the dynamic characteristics with active control in the x - and y -directions interact with each other and that vibration control is improved by the superposition. This verifies that actuators are required at right angles (orthogonal) to each other at least for the active vibration control of a rotor system.

Case when only the turbine-side actuators or opposite-turbine-side actuators were controlled. Figure 9 shows the experimental unbalance response for the x_1 direction when only actuators 3 and 4 (Fig. 4) were used for control. Figure 10 shows the experimental unbalance response for the y_1 direction when only actuators 1 and 2 were used for control. The response of Fig. 9 is better than that of Fig. 10 regarding vibration control efficiency. This is due to differences in mode shapes. The turbine side had modal loops at the second, fourth, and fifth modes, and the other side had modal nodes at the second, third, and fifth modes except for the fourth mode at 3800 rpm.

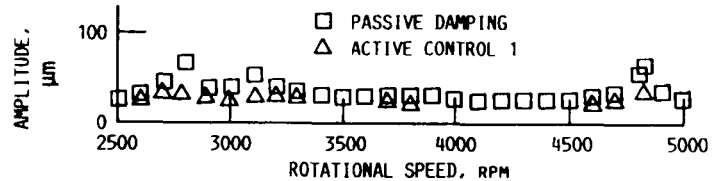


FIGURE 9. - EXPERIMENTAL UNBALANCE RESPONSE FOR x_1 -DIRECTION WITH ONLY TURBINE-SIDE ACTUATORS (3 AND 4) USED FOR CONTROL.

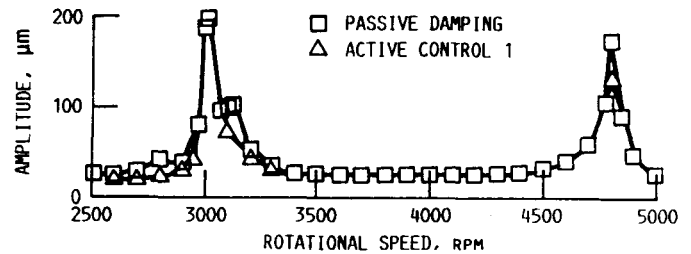


FIGURE 10. - EXPERIMENTAL UNBALANCE RESPONSE FOR y_1 -DIRECTION WITH ONLY ACTUATORS 1 AND 2 (OPPOSITE TURBINE SIDE) USED FOR CONTROL.

CONCLUSIONS

A great deal of effort is required to generate a mathematical model for a flexible rotor supported on a flexible bearing housing as in turbomachinery. Often the model does not predict all of the vibration modes accurately enough to be used for active control. Because it achieves active control without the aid of a mathematical model the optimal direct-output feedback control method discussed in this paper is desirable for cases when the mathematical models are deficient. Using this control method with optimal feedback gains, we reduced all of the resonance amplitude peaks between 0 and 10 000 rpm to less than 25 μm , close to the run-out level.

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