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DYNAMICS OF A FLEXIBLE ROTOR IN MAGNETIC BEARINGS*

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Currently, speeds in rotating machines are limited because of bearing problems, internal fluid flow forces and machine unbalances which produce large vibrations. A recent high technology development is that of magnetic bearings, which replace the usual rolling element or fluid film bearings. The new magnetic bearings can largely eliminate the wear and lubrication problems of oil lubricated bearings because the rotating component is totally supported in air by magnetic forces. Another application is as a vibration controller which is used in conjunction with existing bearings. In this case, the shaft may be made longer so that additional stages may be added or seal clearances reduced because of lower vibration levels. Thus, machine performance may be increased.

This paper discusses a magnetic bearing which was designed and tested in a flexible rotor both as support bearings and as a vibration controller. The design of the bearing is described and the effect of control circuit bandwidth determined. Both stiffness and damping coefficients were measured and calculated for the bearing with good agreement. The bearings were then placed in a single mass rotor as support bearings and the machine run through two critical speeds. Measurements were made of the vibration response in plain bushings and magnetic bearings. Comparisons were also made of the theoretical calculations with the measured peak unbalance response speeds. Finally, runs were made with the magnetic bearing used as a vibration controller.

INTRODUCTION

Magnetic bearings are beginning to be used in a wide range of flexible shaft rotating machinery. Compressors for pipeline service have recently had magnetic bearings installed to reduce vibrations and reduce the possibility of fires related to oil lubrication. They have also been tried in large pumps, turbines and other rotating machines.

A magnetic bearing consists of four or eight electromagnets arranged radially around a shaft. The dynamic properties of the bearing are controlled electronically and stiffness and damping coefficients can be chosen, within certain ranges, to minimize the rotor vibrations. Such flexibility is not possible with conventional fluid film and rolling element bearings. Other advantages include removal of oil seals,

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ability to operate in high temperature or hostile chemical environments, reduced operational costs, and significantly lower friction losses than other bearing types.

The first fully active magnetic suspension system was patented in 1957 [1]. Neiman, et. al [2], reported on the investigation of magnetic and electric forces for rotating shaft systems. Their work showed limited success with fully supporting a rotor due to axial instabilities.

The dynamics of a single mass rotor on rigid supports with control forces applied at the mass are reported in [3]. This work was then extended to include flexible supports [4]. Moore et. al [5] investigated the feasibility of active feedback control for a multimass flexible rotor with force actuators at bearing locations rather than at the shaft center. It was demonstrated that various levels of damping applied at the bearings would be a practical method of reducing vibrations.

Stanway and O'Reilly [6] presented a state space formulation for a flexible rotor with flexible supports. The method was further developed with a numerical example [7]. Forces were applied to the bearing housing to help stabilize plain oil film bearings.

Nikolajsen and Holmes [8] investigated an electromagnetic damper as applied to a transmission shaft. A test rig was built and tested, showing very good agreement with predicted amplitude through a critical speed. Gondhalekar and Holmes [9] designed and built a hybrid passive-active magnetic bearing system applied to a flexible transmission system through two critical speeds. Kaya and Roberts [10] further demonstrated the usefulness of active control flexible transmission shafts. They also developed an optimization method which would lend itself to use with a digital control approach with speed dependent characteristics.

Schweitzer and Ulbrich [11] reported on a vertical centrifuge controlled by using active magnetic bearings. Traxler and Schweitzer [12] also presented results for a rigid rotor. The emphasis was on the force and current measurements with the rotor stationary and rotating. The same authors [13] described a three level approach to designing magnetic bearings. Salm and Schweitzer [14] presented a model for a flexible rotor controlled by an active magnetic bearing at the center of a single mass rotor. Ulbrich and Anton [15] investigated the integration of displacement and velocity sensors within the magnetic bearing.

Bartlett and Taylor [16] developed a model of a solenoidal suspension and applied it to a flexible rotor. Hebbale [17] did a dissertation on the nonlinear dynamics of magnetic bearings. Yamamura [18] developed an analysis of active suspension for high speed vehicles.

Several articles have been published about the commercial possiblities of magnetic bearings [19,20]. The benefits claimed are operating speeds up to 80,000 rpm, rotor diameters of 15.2 mm to 610 mm (0.6 to 24 inches), load per bearing from 0.31 N to 2.0 E5 N (0.07 lb to 45,000 lb), ambient temperatures from -251^{o} C to 449^{o} C (-420 F to 840 F), and operating environments of vacuum, air, helium, hydrocarbons, steam, uranium hexaflouride, sea water, liquid oxygen, and liquid hydrogen.

Two previous works related to the magnetic bearing or its controls have already been published or accepted for publication. Allaire et. al [21] described the basic control scheme as applied to one pole magnetic support system. Humphris et. al [22] presented a theoretical description of the magnetic circuit and experimental measurements of the stiffness and damping coefficients. Several parameters in the controls were varied with agreement to within about 20 to 30%.

The purpose of this paper is to discuss magnetic bearings which were reported in [22] and tested in a multimass flexible rotor both as support bearings and as a vibration controller. First, a single mass rotor was supported in conventional bushings and the magnetic bearing used as a conroller near the center mass. Second, two disks for the magnetic bearings were added to the single mass rotor and the magnetic bearings used in a support mode.

MAGNETIC BEARING

The magnetic bearing [22] used in this study has four electromagnets distributed radially around a 12.7mm (0.5 inch) shaft as shown in Fig. 1. A soft iron disk 58.4 mm (2.3 inches) in diameter and 25.4 mm (1.0 inches) long was placed over the shaft to increase the area and provide a good magnetic circuit. The disks were placed either at the desired controller location or at the normal bearing support locations. Each magnet consisted of a solid soft iron core forming a horseshoe, with two pole faces cut to a diameter of 60.5 mm (2.38 inches). This gives a nominal clearance of 1.0 mm (0.040 inches). Each leg of the magnet was wound with 920 turns of wire. All magnets were the same. A rigid aluminum housing, shown in Fig. 1, provided the support for the magnets.

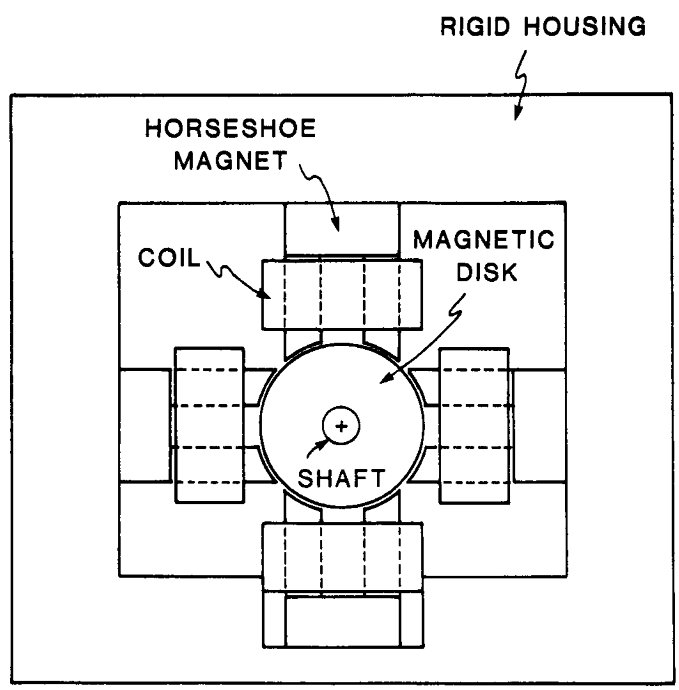


Figure 1. Magnetic Bearing Geometry

The shaft weight was supported by having a larger steady state current in the top magnet when the support mode was employed. The other three electromagnets had a steady state current in them to provide a value about which a set of linearized properties were determined.

Position sensors were located vertically and horizontally on the shaft a short axial distance from the bearing. These were standard eddy-current probes of the type normally used to monitor rotating equipment in the field. Tests showed that the magnetic fields from the bearing did not affect the probe measurements at a 25 mm (1 inch) distance.

The magnetic bearing control system is described in the Appendix. Components of the system are presented and the algorithms used to model them. Typical bearing linearized stiffness and damping coefficients are presented.

MAGNETIC CONTROLLER

In the controller mode, no rotor weight was supported. All load was carried by the conventional bearings at each end. Thus all steady state current levels were the same in all four electromagnets. Also, the required steady state currents in the control mode were much less than the current required in the support case since no load was carried. Similarly the controller dynamic coefficients (stiffness and damping) could have lower values because they were applied at the rotor center. Overall the power required for the controller mode was much less than for the support case.

The rotor was first assembled with the magnetic bearing as a controller, with the configuration illustrated in Fig. 2. The shaft had two masses on it, a center disk of weight 8.0 N (1.8 lb) and dimensions 73.15 mm (2.88 inches) in diameter by 25.4 mm (1.0 inch) thick, and the magnetic bearing disk as described in the previous section. The bearing disk weighed 4.9 N (1.1 lb). The shaft was 12.7 mm (0.5 inches) in diameter and 660 mm (26.0 inches) long. The bearing span was 508 mm (20 inches) with the center disk at midspam and the bearing disk at the one-third span. The total rotor weight for shaft plus both disks was 24.9 N (5.6 lb). As shown in Fig. 2, the rotor extended to a flexible coupling, which connected the shaft to a small electric motor drive.

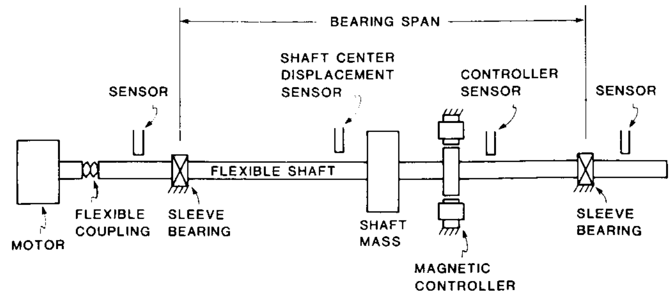


Figure 2. Diagram of Flexible Rotor Supported in Conventional Bearings With Magnetic Controller

At each end of the rotor were conventional sleeve bearings. They are of oil impregnated bronze construction backed by a rubber O ring. This type of bearing is common in many small laboratory rotor kits and is considered sufficient to demonstrate the principles of vibration reduction by magnetic controllers. The stiffness of the sleeve bearing and associated housing is approximately 3.5 E5 N/m (2000 lb/in). Also, the effective stiffness of the coupling is estimated to be about 8.8 E4 N/m (500 lb/in).

The rotor was run up in speed through one critical speed with the controller turned off. Figure 3 shows the horizontal peak to peak response. The peak response occurs at approximately 2845 rpm at 0.52 mm (20.5 mils). With the controller on, the critical speed increases to approximately 3610 rpm and the amplitude reduces to about 0.063 mm (2.5 mils). For this case, the magnetic controller gains were set to have a calculated stiffness of 1.2 E5 N/m (700 lb/in).

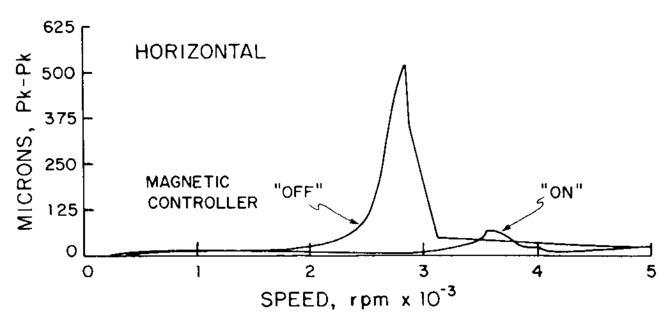


Figure 3. Flexible Rotor Response at Quarter Span, with and without Magnetic Controller

The critical speed increased 28% and the vibration level dropped to 12% of the original value. In this particular experiment, no efforts were made to optimize the vibration reduction by modifying the magnetic bearing control settings. In general, it would be expected that the introduction of stiffness and damping at the center of the rotor should greatly reduce the vibration levels. The point being made here is that the magnetic damper works and the measured results can be compared to predicted rotor behavior.

Figure 4 shows the rotor calculated first critical speed and mode shape, without the magnetic controller, determined by a standard transfer matrix method. The calculated value is 2766 rpm or only about 3% below the measured value. This indicates that the rotor model is a good one. Figure 5 gives the calculated critical speed and mode shape with the controller turned on. Again the calculated value of 3556 rpm is very close to the measured value of 3610 rpm.

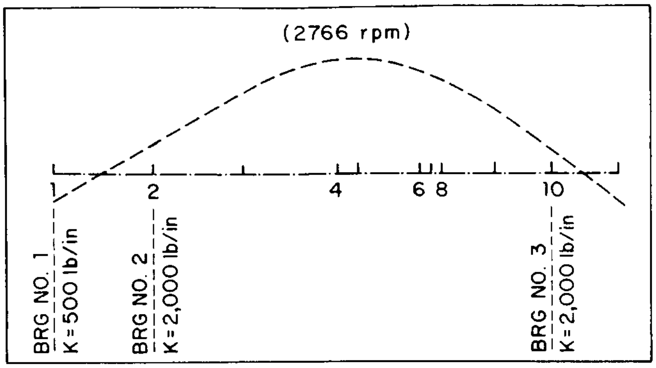


Figure 4. Computed Critical Speed and Mode Shape Without Magnetic Controller

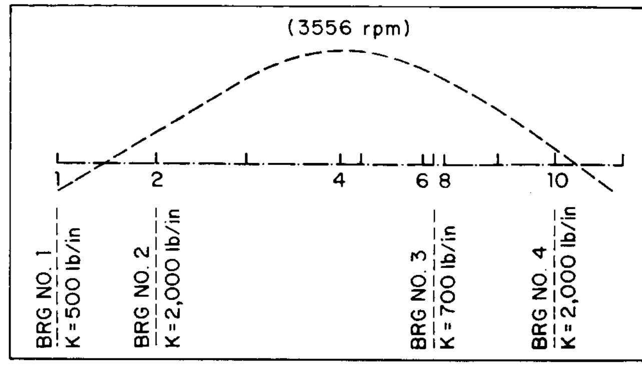


Figure 5. Computed Critical Speed and Mode Shape With Magnetic Controller

A series of spectrum analysis plots versus time, illustrating the effect of the magnetic controller, is shown in Fig. 6. The rotor was run at a constant speed of 2600 rpm or just below the first critical speed. Initially the controller was "off", with no current in any of the magnets. The vibration amplitude is large and constant over time. After approximately 32 seconds, the current was turned "on" in all four magnets. The amplitude of vibration immediately dropped to a very low level and remained at that level with no observable transient response.

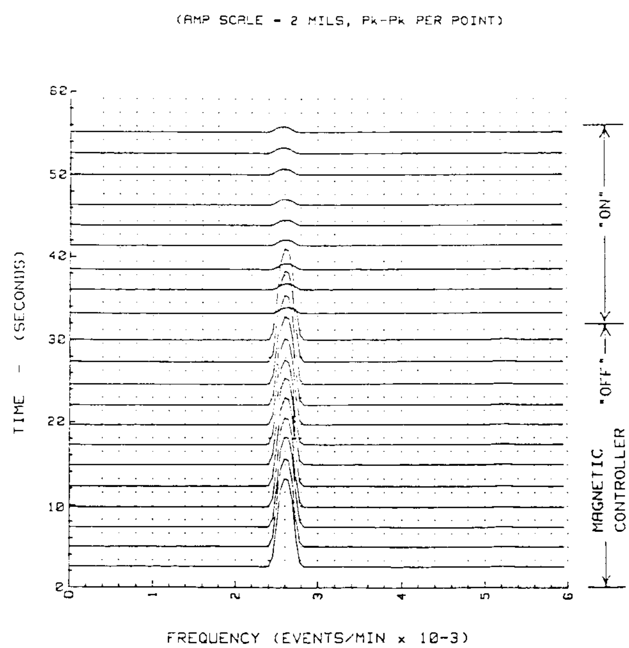


Figure 6. Spectrum Analysis Plots showing the effect of Magnetic Controller "off" and "on"

MAGNETIC SUPPORT BEARING

The rotor-bearing configuration was changed so that two identical magnetic bearings were used to replace the conventional bearings. Figure 7 illustrates the geometry. It had three masses on it - a center disk of weight 8.0 N (1.8 lb) and two magnetic bearing disks weighing 4.9 N (1.1 lb) each. Disk and shaft geometries were identical to those used in the controller mode. The total rotor weight was 24.9 N (5.6 lb). Again the bearing span was 508 mm (20 inches).

Backup bearings were placed inboard of each magnetic bearing. Each was a conventional sleeve bearing of the type described in the previous section except that they were bored out to a clearance of 0.51 mm (0.020 mils) radial. The magnetic bearing radial clearance was much larger at 0.76 mm (0.030 inches). The rotor amplitude of vibration near the bearings was always well below this value. Thus

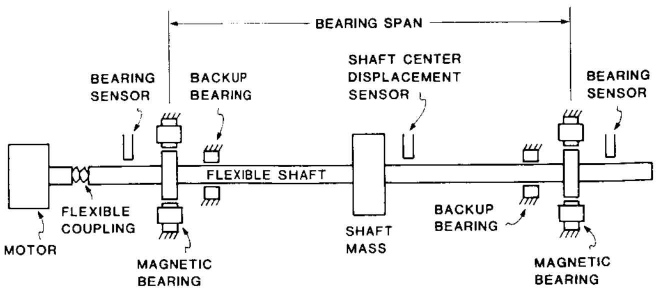


Figure 7. Diagram of Flexible Rotor Rig Supported in Magnetic Bearings

these bearings were never in contact when the magnetic bearings were activated. This included start-up conditions (no shaft rotation) or running through critical speeds.

The same type of noncontact induction probes were employed for the rotor. Two sets were used at the magnetic bearing disks for both feedback control and monitoring. A third was placed adjacent to the center mounted disk for monitoring.

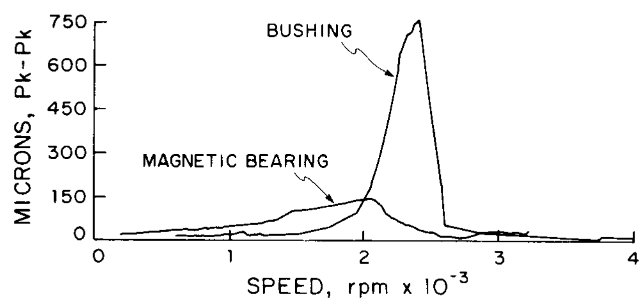
Supporting the rotor weight required the use of higher currents in the top magnets than in the sides and bottom magnets. Typical values are reported in [22]. Both bearings had essentially the same operating conditions.

A preliminary run was made with the rotor in conventional bearings (without the magnetic controller). Then, the conventional bearings were immediately replaced by the magnetic bearings in exactly the same locations. Thus the shaft had the same unbalance level in the center disk. The state of unbalance of the added bearing disks was unknown.

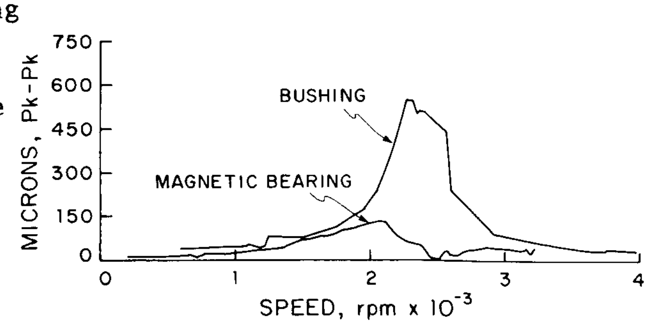
Figure 8 gives the results plotted on the same axes. In the vertical direction, the peak amplitude was 0.72 mm (30 mils) at about 2,400 rpm with sleeve bearings. When the magnetic bearings were used, the vibration level dropped to about 0.14 mm (5.5 mils) at approximately 2,000 rpm. Thus the vertical vibration level was reduced by a factor of greater than 5. It had gain values of K_g = 1.75 and K_r = 20. In the horizontal direction, the vibration level was reduced from 0.56 mm (22 mils) with the sleeve bearing to about 0.13 mm (5 mils) with the magnetic bearing. The vibration level was reduced by factor of 4. Also, K_g = 1.0 and K_r = 20 were the gain settings for the horizontal direction. For this preliminary run, no attempt was made to "optimize" the magnetic bearing settings to reduce vibrations.

Two additional cases of runs were made with the magnetic bearings in the support mode. The cases are

	Vertical Gain		Horizontal Gain	
Case	K_g	K_r	K_g	K_r
1	1	2	4	10
2	4	8	4	10



a) vibration level in vert. direction



b) vibration level in horiz. direction

Figure 8. Comparison of Flexible Rotor Vibrations at Shaft Center For Conventional Bearings vs. Magnetic Bearings

Again the primary objective is to compare the measured first critical speeds to the calculated values based upon the magnetic bearing stiffness as evaluated by the theory in [22].

Figure 9 shows the vibration amplitude and phase angle vs. speed at the rotor midplane (near the center mass) for Case 1. Results from probes near the bearings are similar but with lower amplitudes. Figure 10 gives the frequency spectrum for the vertical direction at the midspan. The vibrations are nearly all synchronous.

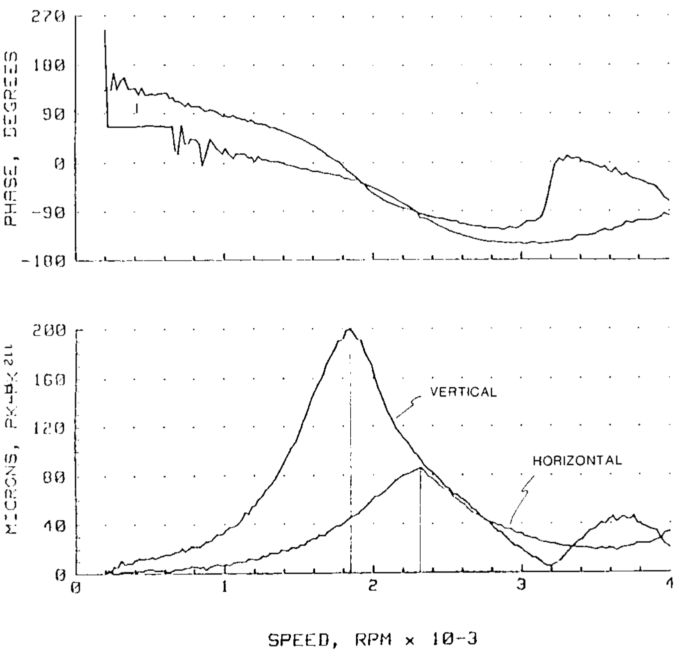


Figure 9. Horizontal and Vertical Midspan Vibration Plots for Rotor in Magnetic Support Bearing - Case 1

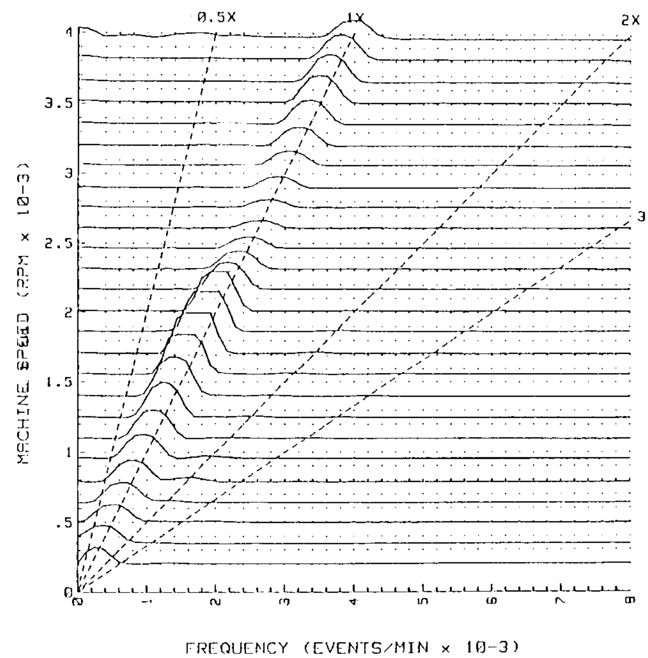


Figure 10. Frequency Spectrum for Vertical Direction at Midspan--Case 1

For Case 1, peak vibrations occur at 1860 rpm in the vertical direction and 2320 rpm in the horizontal direction. Again using the theory from [22] the calculated bearing stiffnesses are

Case 1

Direction	Stiffness N/mm (lb/in) Inboard	Outboard
Vertical	33 (187)	65 (370)
Horizontal	411 (2335)	333 (1890)

Figure 11 shows calculated critical speeds and the mode shapes for the above stiffnesses. In the vertical direction, the calculated value is 1803 rpm. This is only 3% below the actual value. The horizontal calculated critical is 2650 rpm which is 14% above the measured value.

For Case 2, the vertical gains were increased. Figure 12 gives the resulting vibration plot. The vertical critical speed increased to 2720 rpm (1860 rpm for Case 1) due to the higher bearing stiffness. Calculated bearing stiffnesses are shown on the following page. The calculated vertical critical speed is 2672 rpm. This is below the measured value by 3%. In the horizontal direction, the measured peak was 2260 rpm while the calculated value is 2625 rpm. The mode shapes for Case 2 were similar to those for Case 1.

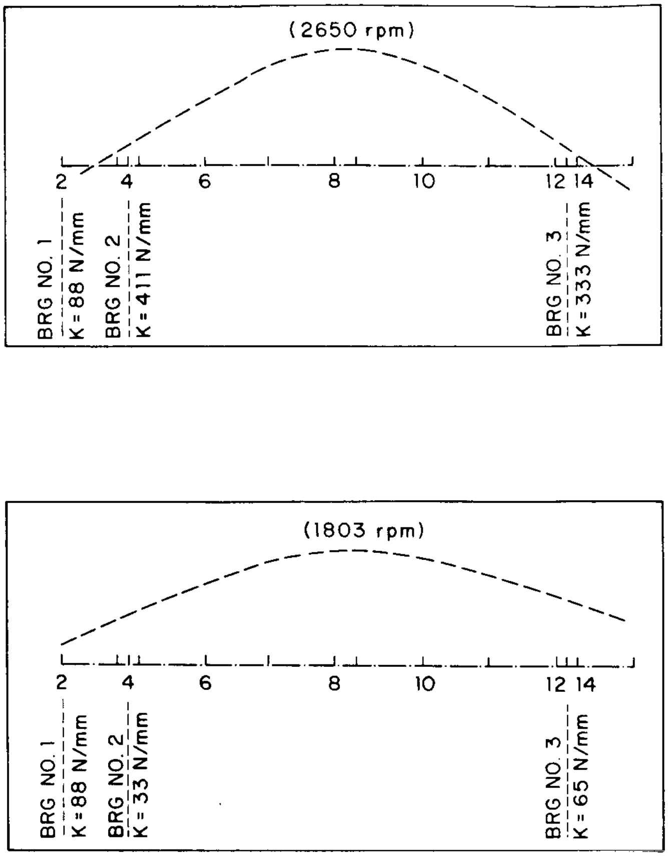


Figure 11. Calculated Mode Shapes and Critical Speeds for Case 1

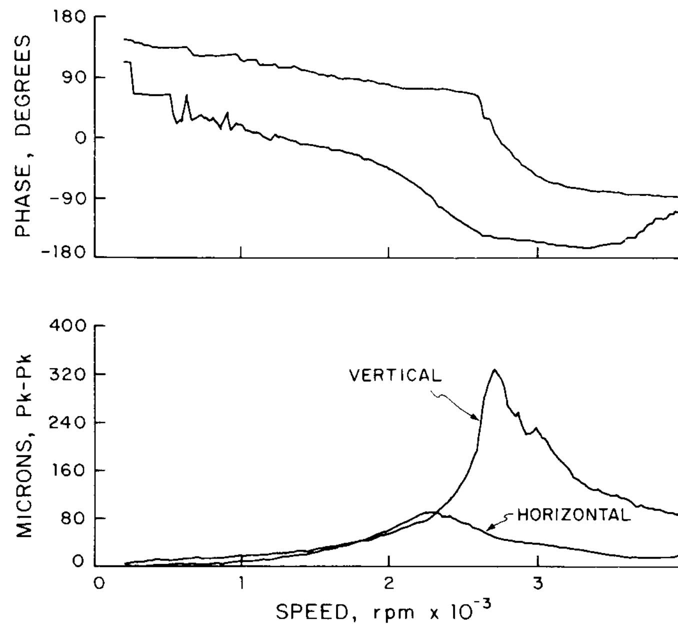


Figure 12. Horizontal and Vertical Midspan Vibration Plots For Rotor In Magnetic Support Bearing Case 2 (Vertical K_g=4, K_r=8 Horizontal K_g=4, K_r=10)

Case 2

Direction	Stiffness N/mm (lb/in) Inboard	Outboard
Vertical	372 (2125)	543 (3100)
Horizontal	420 (2400)	337 (1925)

CONCLUSIONS

This paper reports the effect of a magnetic bearing used in both controller and support modes. Generally the conclusion is that the magnetic bearing strongly affects the vibration levels in the rotor. Using the magnetic controller reduced the vibration level to 12% of the original value in the vertical direction. In the support mode, the vibration level decreased by about the same factor in the vertical direction as compared to the rotor in conventional support bearings. This reduction of vibration levels was achieved without optimizing the settings of the control parameters.

Another result of this work is a comparison of the measured and calculated critical speeds. Table 1 gives the summary for the cases considered in this work. In each case the stiffness values were evaluated using the method in [22]. The largest error for the calculated critical speeds is 16%. Overall, this indicates that machine undamped critical speeds can probably be accurately determined theoretically before magnetic bearings are installed in a machine.

APPENDIX - MAGNETIC BEARING PROPERTIES

A block diagram of the control circuit for each magnet in each bearing is shown in Fig. A.1. It operates as follows. The eddy-current induction probe senses the position of the shaft. The signal goes to the sensor amplifier which has a fixed gain, low pass filter and reference adjustment. A compensator has an adjustable position gain K_g and rate gain K_r. These are used to vary the bearing stiffness and damping. The next components are the summer and lead network. Finally the signal passes through the position amplifier which determines the steady state current provided to the electromagnet and hence the operating position of the shaft in the bearing.

Specific numerical values for the magnetic bearings are given in [22]. They are omitted here due to length considerations. Also the theoretical model of the bearing properties is developed and presented in [22] with comparisons to measured results.

One of the important characteristics of any bearing is the stiffness. An advantage of the magnetic bearing is that the stiffness can be varied simply by changing the gain in the control circuit. Figure A.2 shows the theoretical and measured static stiffness of one magnet of the bearing used in a support mode for two control circuits, A and B, with different bandwidths as a function of proportional control circuit gain. The agreement is within about 20%, with the theoretical values being higher than the measured stiffnesses.

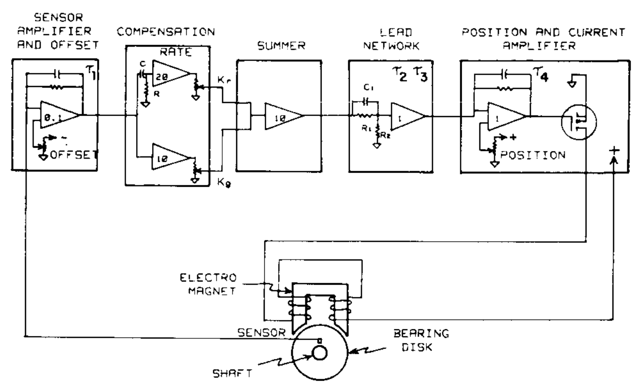


Figure A.1. Block Diagram of Magnetic Support System

Another important characteristic of a bearing is the damping coefficient. Often the primary vibration reduction in rotating machines comes from the oil lubricated bearings. Magnetic bearings should be at least as effective as conventional fluid film or rolling element bearings with squeeze film dampers. Figure A.3 gives a comparison between the theoretically calculated damping coefficient and measured values obtained from system responses to step inputs. There is some scatter in the experimental data but the agreement is within about 30% for both of the proportional gain cases considered. Note that for the damping, the coefficient is a function of both the proportional and rate gains due to the control circuit used. It may also be noted that both the stiffness and damping coefficients are reasonably linear over the ranges measured.

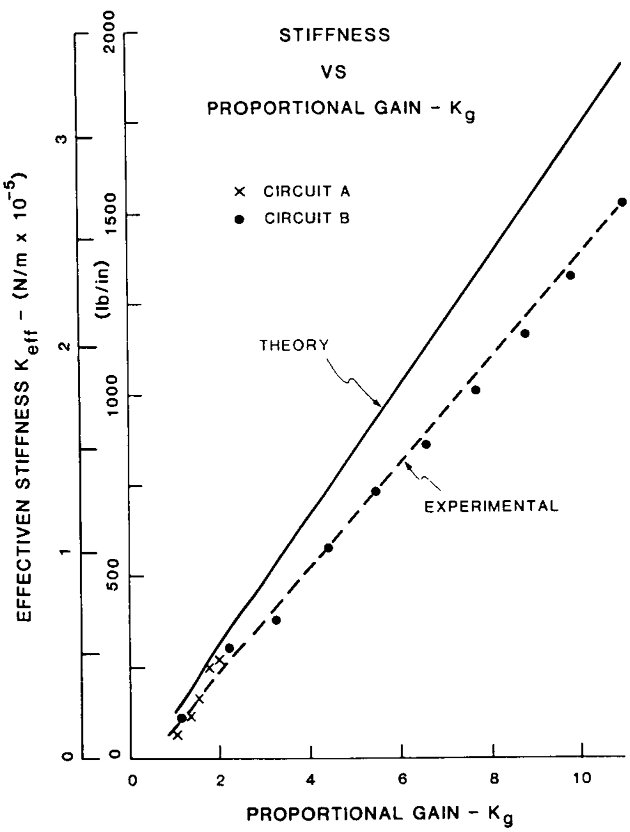


Figure A.2. Theoretical and Measured Stiffness for Magnetic Bearing

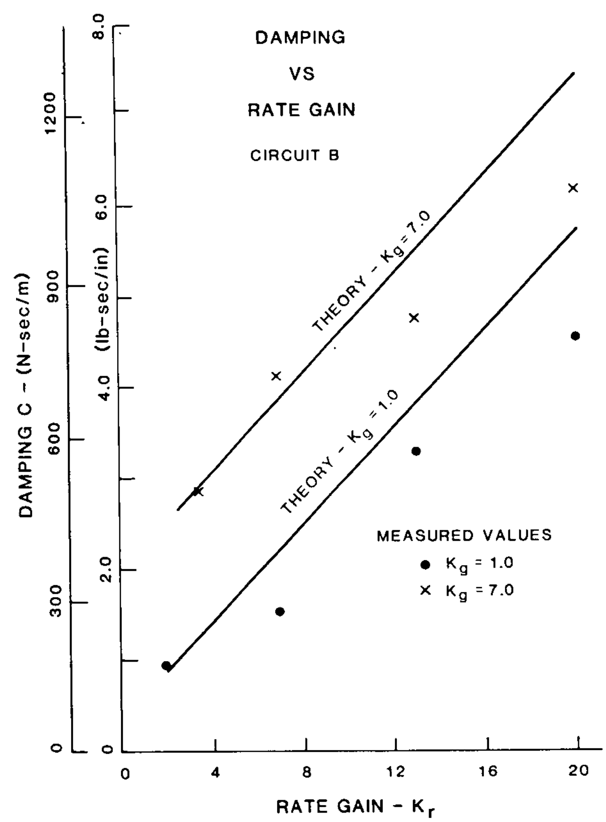


Figure A.3. Theoretical and Measured Damping for Magnetic Bearing

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Table 1
Critical Speeds

Mode	Case	Measured (rpm)	Calculated (rpm)	Error (%)
Controller Mode	No Controller	2845	2766	-3%
	With Controller	3610	3556	-1%
Magnetic Bearing Support Mode	Case 1/Vertical	1860	1803	-3%
	Case 1/Horizontal	2320	2650	+14%
	Case 2/Vertical	2720	2672	-2%
	Case 2/Horizontal	2260	2625	+16%