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A CONVENTIONAL POINT OF VIEW ON ACTIVE MAGNETIC BEARINGS

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SUMMARY

Active magnetic bearings used in rotating machinery should be designed as locally controlled, independent devices similar to other types of bearings. The functions of control electronics and power amplifiers can be simply and explicitly related to general bearing properties such as load capacity, stiffness, and damping. In this paper, the dynamics of a rotor and its supporting active magnetic bearings are analyzed in a modified conventional method with an extended state vector containing the bearing state variables.

INTRODUCTION

Active Magnetic Bearings (AMBs) have been slow to show acceptance in the rotating machinery industry, and to machinery designers, they remain somewhat a mystery. AMBs have been developed over the decades by researchers in electrical engineering and systems control. Consequently, progress in their development is often presented in a language that is alien to the mechanical designers. The design engineer, whose major discipline is mechanical engineering, is likely to be unfamiliar with the concept that AMB stiffness and damping are functions of excitation frequency. In addition, the designer does not commonly use terminology such as control bandwidth, phase compensation, control spillover, etc. Not only does the AMB's electrical presentation hinder its acceptance, but also the fact that it is such a small component of a sophisticated machine; most design engineers will question whether it warrants the time and effort needed to gain familiarity with it. Consequently, greater understanding of the AMB is imperative in increasing its use in conventional rotor-bearing applications. With these observations in mind, the authors herein attempt to describe the key elements and properties of AMBs in a general engineering language that includes rotor dynamic notation which mechanical engineers are more accustomed to using.

BASIC PRINCIPLES

The principles of an AMB are well explained by Habermann and Liard (1980). The practical AMBs generally adopt an 8-pole stator configuration as shown in Figure 1. Both the stator and journal are made up of stacks of laminations of ferromagnetic material. The journal is shrunk on a shaft without windings. Laminations reduce eddy currents that not only create a power loss but also degrade the performance of the bearing. The eddy currents generate magnetic fields that oppose the flux needed for regulating the journal motion. The eight poles of the stator are separated into four quadrants (Weise, 1985). In each quadrant, the electromagnetic windings are

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wound in such a way that the magnetic flux will circulate mainly inside the quadrants (Chen and Darlow, 1987). In other words, the magnetic forces of the poles in any quadrant can be varied by changing the amount of current in the windings of that quadrant without causing force changes in other quadrants. Therefore, each quadrant of poles can be controlled independently.

It is well known that the magnetic force is proportional to the current to air gap ratio squared (see Figure 2). Engineers conscientiously try to avoid nonlinear design and analysis. The linearization of AMB dynamics is achievable by making the air gap large relative to the journal normal excursion and by providing a relatively large steady state current (called bias current hereafter) through each quadrant. The bias current produces I^2R loss, which is a major power loss in AMB. However, the total resistance including the windings and other parts in current path is not large; the AMB power loss in general is insignificant when compared to the conventional bearings, such as the hydrodynamic oil-film type.

It is noteworthy that all poles exert attractive forces on the journal. Two opposite quadrants of poles are used to center the journal in one direction. While generating cancelling forces may seem to waste electric power, the AMB stiffness and damping are directly proportional to the bias current.

The journal floating in a magnetic field that is produced only by the bias currents is not stable. This situation is analogous to supporting a vertical stick at the bottom by a hinge without controlling the hinge. To create stability at the AMB center, the journal motion must be sensed and corrected instantaneously and continuously by superimposing a small control current to each bias current. For example, when the journal moves upward off the center by a small displacement, Y, the current in the top quadrant will be reduced by a small amount, i, and the bottom quadrant increased by i. The control currents produce a net downward force (-F) that pulls the journal back to the center. From sensing Y to producing -F, a series of AMB components become involved. It is the authors' preference to separate them into two groups:

- Sensor and control electronics
- Power amplifiers and electromagnets.

Before examining them in detail, presenting basic statics and dynamics will specify the function of these components.

AMB STATICS AND DYNAMICS

To support a static load, W, in Y-direction, the net attractive force caused by the bias currents is

$$W = f(I_1^2 - I_3^2)/C^2 \qquad (1)$$

where f is a magnetic pole constant for a given number of windings.

Choosing an I_3, I_1 can be determined by equation 1. I_3 must be larger than the control current, i, at any anticipated transient situation so that no current saturation can occur.

Assuming that

$$I_1, I_3 \gg i \quad \text{and} \quad Y \ll C$$

and that the AMB sees a dynamic mass, M, then a linearized equation of motion is

$$M \ddot{Y} - K_m Y = K_i \; i \tag{2}$$

where K_i and K_m are respectively called the current stiffness and the negative spring of the magnetic field due to the bias current. Without the small control current, i, equation 2 represents an unstable dynamic system. To make it stable, it is simple and sufficient to have

$$i = G \; E \tag{3}$$

and

$$E = -C_d \; Y - C_v \; \dot{Y} \tag{4}$$

where E is the controller output voltage signal and the input to the amplifiers, and G is the power amplifier gain (amp/volt). In other words, the small control current should be made proportional to the journal displacement and velocity. The minus sign is indicative of negative feedback, which is a common regulating mechanism. C_d and C_v are constants, and called proportional and derivative control gains.

Substituting equations 3 and 4 into 2 produces

$$M \ddot{Y} + GK_iC_v \dot{Y} + (GK_iC_d - Km) \; Y = 0 \tag{5}$$

If C_d is large enough, such that $GK_iC_d > K_m$, we achieve a stable one degree of freedom system in Y-direction which has damping coefficient GK_iC_v and stiffness coefficient $(GK_iC_d - K_m)$.

If the static load W, is miscalculated, or there is an occasional slowly varying load, (as can be caused by the gyroscopic effect of a manuvering spacecraft) the journal will sag or drift off the bearing center. To correct this potential problem, a third corrective mechanism called the integral control, as depicted in Figure 3, can be added to equation 4. That is,

$$E = -C_d \; Y - C_v \; \dot{Y} - C_e \int Ydt \tag{6}$$

As shown in Figure 3, the accumulated journal position error over a period of time will produce a part of the control voltage. The integral control does not respond to high frequency vibrations.

SENSOR AND CONTROL ELECTRONICS

For AMB journal displacement measurements, three practical sensors exist: the capacitance probe, inductance probe, and eddy current probe. Each has advantages and disadvantages, but they all relate the small distance (in mils), from the stationary sensor to the rotating shaft, to an output electrical signal in volts.

No inexpensive and reliable sensors are available yet for measuring the journal velocity ($\dot{Y}$). Consequently, different control circuit designs have been developed to produce a psuedo velocity from the displacement measurements. The feedback of velocity is in essence a corrective action of anticipation. If the vibration displacement is sinusoidal, the corresponding velocity is also sinusoidal,

$$Y = Y_o \sin(\omega t)$$

$$\dot{Y} = \omega Y_o \sin(\omega t + \pi/2)$$

but peaks 90° earlier on an oscilloscope. The logical way of obtaining the velocity from the measured displacement is by performing differentiation with an analog circuit. However, inevitable error or noise signals occur in this measurement; a small displacement error at high frequency will become a large erroneous velocity at the same frequency by differentiation, which may overhelm or bury the real velocity signals. Therefore, a practical differentiator always applies a certain "low-pass filtering", which reduces the high frequency signal content. A conflict arises between reducing the high frequency signal and keeping the lead time for the low frequency signal. The filtering aspect of circuit design may become awkward at times.

A more popular method of generating the velocity component of controllers is to use an analog circuit called phase-lead network. Each phase-lead network provides lead time or phase advance for the displacement signal in a reasonably large frequency range. The maximum phase advance per stage network is less than 70°. Two networks may be needed to cover a large spread of critical speeds. The output of a phase-lead network is a combination of displacement and velocity signals. Thus, its feedback produces the damping and contributes to the stiffness.

An alternative method, called the Velocity Observer, was recently developed and used by Chen and Darlow (1987) for producing the appropriate control signals. It integrates journal force, (equivalently, acceleration) to obtain velocity. The observer takes the displacement and the small control current as input. The Velocity Observer has an advantage over the phase-lead approach in that it covers much wider frequency range with phase advance $> 90^\circ$.

The integral feedback term in equation 6 is straight forward in analog circuit design. The integral control is a time delay action. To confine its effect below 1 or 10 Hz, where system resonance seldom occurs, low-pass filtering may be included in the integrator circuit. A comparison of the mathematical expressions of the three types of circuits are presented in Figure 4.

The control electronics, which take the displacement Y as input, is generally called a Proportional-Integral-Derivative (PID) controller. There are many ways to implement the controller. Humphris et al. (1987) used a single path proportional derivative (PD) controller with two phase-advancing circuits in series. Fukata et al. (1986) used a PID controller with three parallel paths and a differentiator. Chen and Darlow (1987) used another PD controller using two parallel paths and a Velocity Observer.

The parallel-path approach is preferable because the adjustments of the stiffness and damping are less dependent on each other. A controller example is shown in Figure 5 using a single stage of phase-lead network. The transfer function of the controller can be expressed in the following closed-form formula:

$$-E/Y = C_d + C_v\,(s + a)/(s + b) + C_e/(s + \omega_1) \qquad (7)$$

From (2), the AMB dynamic force is

$$F = K_i\, i + K_m\, Y. \qquad (8)$$

Assuming the amplifiers provide control current as much as demanded with no delay, and for simplicity G = 1 (amp/volt), then

$$i = E \qquad (9)$$

Combining equations 7, 8, and 9, we have (for s = jw)

$$-F/Y = K + j\omega\ B \tag{10}$$

where

$$K = K_i[(C_d - K_m/K_i) + C_v(ab+\omega^2)/(b^2+\omega^2) + C_e\omega_1/(\omega_1{}^2+\omega^2)] \tag{11}$$

$$B = K_i[C_v(b-a)/(b^2+\omega^2) - C_e/(\omega_1{}^2+\omega^2)] \tag{12}$$

Equations (11) and (12) show that AMB stiffness and damping are functions of excitation frequency, ω. An example of the stiffness and damping using this controller is presented in Figure 6. It may appear unusual that bearing damping in Figure 6 is negative in the low frequency range. The fact is, that as long as no system vibration mode exists in that range, there should be no instability. Note that for every ω, a damped frequency value ω_n can be calculated from the bearing parameters M, K and B. A natural vibration modal frequency exists at $\omega = \omega_n$.

POWER AMPLIFIERS AND ELECTROMAGNETS

The physical size of an AMB journal and stator can be determined (Chen 1988) by first calculating the pole surface area:

$$A_p = F_{max}/250 \tag{13}$$

The formula was derived in American units for silicon steel laminations (saturation flux 110 x 10^3 lines/in.^{2}). A_p is taken as the smallest cross-sectional area (in.^{2}) along the pole. F_{max} is the maximum force (lb) that two opposite pairs of poles can take without flux saturation. Choosing the axial length L_p, the circumferential pole width is A_p/L_p. Then, the radial dimensions can be decided outward from a given shaft diameter at the AMB. The sizing guidelines are

- The cross-sectional area at any point of the flux path is not less than A_p.
- Adequate wiring space and cooling surface are provided.
- The axial length is no greater than the journal outer diameter.
- As a rule, the air gap should be ten times the expected journal excursion. The smaller the gap, the less power is required.

Converting a low power control voltage signal to a high power control current and actuating the electromagnets requires current source or power amplifiers. Two types of power amplifiers, the linear type and the pulse-width-modulation (PWM) type, have been studied and used by MTI in their magnetic bearings. The linear type applies the control signal to a power transistor in "active" mode. The transitor continuously regulates the current through the windings from a DC source. In this mode of operation, the voltage drops across the transitor, and thus the power loss is high. The PWM type applies the control signal to generate high voltage output pulses at a frequency above audible range. The on-time period of each pulse is proportional to the input signal. The high voltage pulse train produces current in the windings. The PWM amplifier is electrically noisy but much more efficient in providing the required power with low power loss. The power transitors in it operate in "saturation" mode and have small resistances. Schweitzer and Traxler (1984) had indicated that the borderline in favoring one type over the other is about 0.5 kVA.

The inductance of electromagnet windings can cause a problem of control current delay. The inductance, L, is proportional to N^2A_p/C. A time constant, L/R, is associated with the power amplifier-electromagnet system. A controller signal at ω >

R/L will have a phase delay > 45°, which is one-fifth of a cycle. Also, the inductance limits the permissable rate of change of current, dI/dt (called slew rate) to V_S/L (V_S = DC supply in volts). The delay problem is more severe in systems with PWM amplifiers. An experience was recently reported and well explained by Bradfield, et al. (1987). To take into account this effect of inductive phase lag on the system dynamics, an approximate linear transfer function is

$$G = I/E = 1/[(L/R)S + 1] \tag{14}$$

In the above, we neglected the component of back emf, due to variations of inductance as a result of changes in the air gap. For clarity, the maximum amplitude of the transfer function G has been assumed to be one. According to equation 14, it is only close to one at $\omega < R/L$. Above R/L, the control current is attenuated and delayed. The phase advance by the controller is cancelled by the phase lag in the amplifier system. This means negative AMB damping can occur in the high frequency range, and the control of flexible rotor criticals can be in jeopardy.

ROTOR-AMB SYSTEM DYNAMICS

AMB dynamic stiffnesses are generally lower than ball bearings or oil-film bearings. On a typical critical speed map, as shown in Figure 7, one can expect that the first two criticals will have rigid mode shapes, and they can be easily controlled. The higher modes (particularly the third and the fourth modes) with bending mode shapes must be given careful consideration in the design of the AMB controls. The first requirement is to assure the existence of positive damping for these modes, if the AMBs are responsive to the modal frequencies. Although oil-film bearings always provide positive damping, there is no guarantee of this for AMBs. Because of the inductance mentioned above, the control current at the high frequencies may lag behind the displacement measurement. If the operating speed is well below the bending critical, one can reduce the lagging current to a minimum at that frequency through electronic means. Then, the AMB would not be responsive to the modal resonance.

If the operating speed is above the bending critical, it is imperative to provide enough leading current at the appropriate frequency. Even once the proper control response has been achieved, the ability of the AMBs to control rotordynamic response will be significantly affected by rotor mechanical design. Care must be taken in mechanical design to ensure that the AMB locations are not at nodes of the vibrational modes of interest.

This system design challenge can be aided by performing conventional rotor-bearing dynamic analyses, such as calculating undamped critical speeds, unbalance response, and stability, which have been performed by MTI as part of the magnetic bearing design analysis. Hustak, et al. (1985) have performed similar analyses for two compressor rotors supported by AMBs. They have shown measured AMB stiffness and damping coefficients as functions of excitation frequency. A modified but more rigorous rotordynamics approach, which has the inherent advantage of integration of controller dynamics into the system equations, is described below.

The state vector of the conventional rotor model is extended to include the state variables of the AMBs. The controller dynamics of each AMB axis, such as shown in Figure 5, and the amplifier dynamics (equation 14) can be represented to be a set of first-order linear differential equations in terms of the extended states. The coupling terms between the rotor model and the AMB model exist in the AMB dynamic force, which is represented by equation 8. The resultant electomechanical model using finite element formulation for the rotor part, can be used for eigenvalue/eigenvector and force response analyses. The rotor model input is the same as conventional model including sections of shaft with specified inner diameter, outer diameter, length, and concentrated mass and inertias. The input for each bearing would be the bearing station number, the measurement station number, and the parameters K_i, K_m, a, b, C_d, C_v, C_e, ω_1, and L/R. For properly sized AMBs, the feedback gains, C_d and C_v, are the key parameters that can be optimally determined by performing the eigen and response analyses. This approach to integrated rotordynamic analysis has been applied to typical magnetic bearings developed at MTI.

CONCLUSIONS

The essence of control and the properties of active magnetic bearings have been described and quantified in a language that is more adaptable to mechanical engineers. In doing so, the following viewpoints on AMB development have been emphasized:

1. AMB should be treated as a locally controlled and independent device similar to other types of bearings.

2. AMB control axes should be made independent from each other, with identical but tunable electronics.

3. Parallel circuits for stiffness and damping control should be implemented to permit more versatility in the application of these bearings.

4. Pitfalls exist in controlling the rotor bending critical modes; the reasoning, and design guidelines can be explained easily in mechanical terminology.

5. Modified rotordynamics analysis method with a state vector extended to include the AMB state variables are needed for rigorous system performance prediction.

These mechanical engineering viewpoints are long overdue in gaining prominence and influencing the development of AMB technology.

NOMENCLATURE

AMB	Active magnetic bearings
A_p	Pole cross-sectional area
a	Phase-lead zero parameter
B	AMB damping coefficient
b	Phase-lead pole parameter
C	Radial air gap
C_d	Proportional feedback gain
C_v	Derivative control gain; Phase-lead feedback gain
C_e	Integral feedback gain
E	AMB controller output voltage signal
F	AMB perturbed magnetic force
G	Power amplifier gain
L	Self-inductance of electromagnets
L_p	AMB axial length
I	Bias current
i	Control current

j	$\sqrt{-1}$
K	AMB stiffness coefficient
K_i	Current stiffness
K_m	Magnetic stiffness due to bias currents
M	Dynamic mass
N	Number of winding turns
Q	Integrator output
R	Resistance in current path
s	Laplace variable
t	Time
W	AMB static load
Y	AMB journal displacement in Y-direction
Z	Phase-lead circuit output
ω	Excitation frequency
ω_n	AMB critical frequency
ω_1	integrator cut-off frequency
$\dot{}$	d/dt
$\ddot{}$	d^2/dt^2

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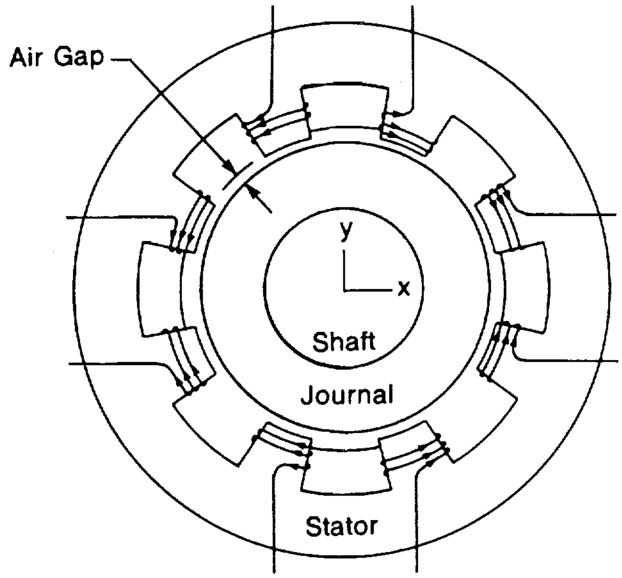

Fig. 1 8-Pole Active Magnetic Bearing Configuration

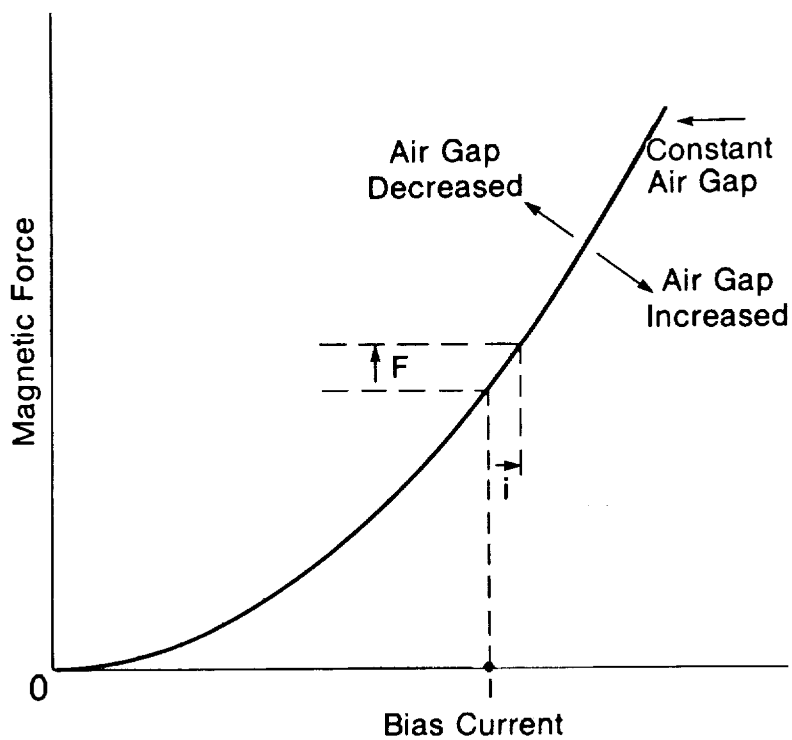

Fig. 2 Nonlinearity of Magnetic Force

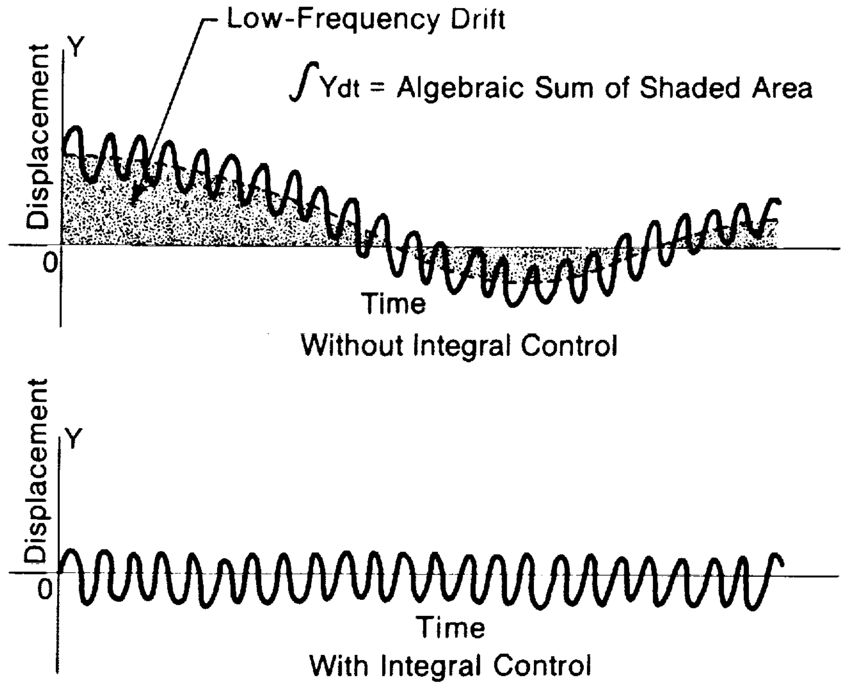

Fig. 3 Elimination of Low-Frequency Drift through Integral Control

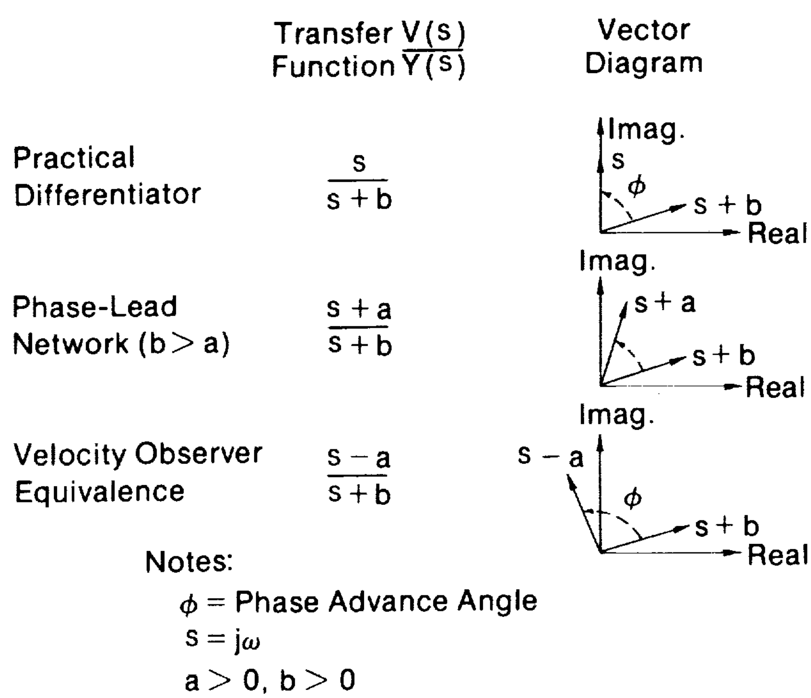

Fig. 4 Comparison of Three-Phase Advancing Circuits

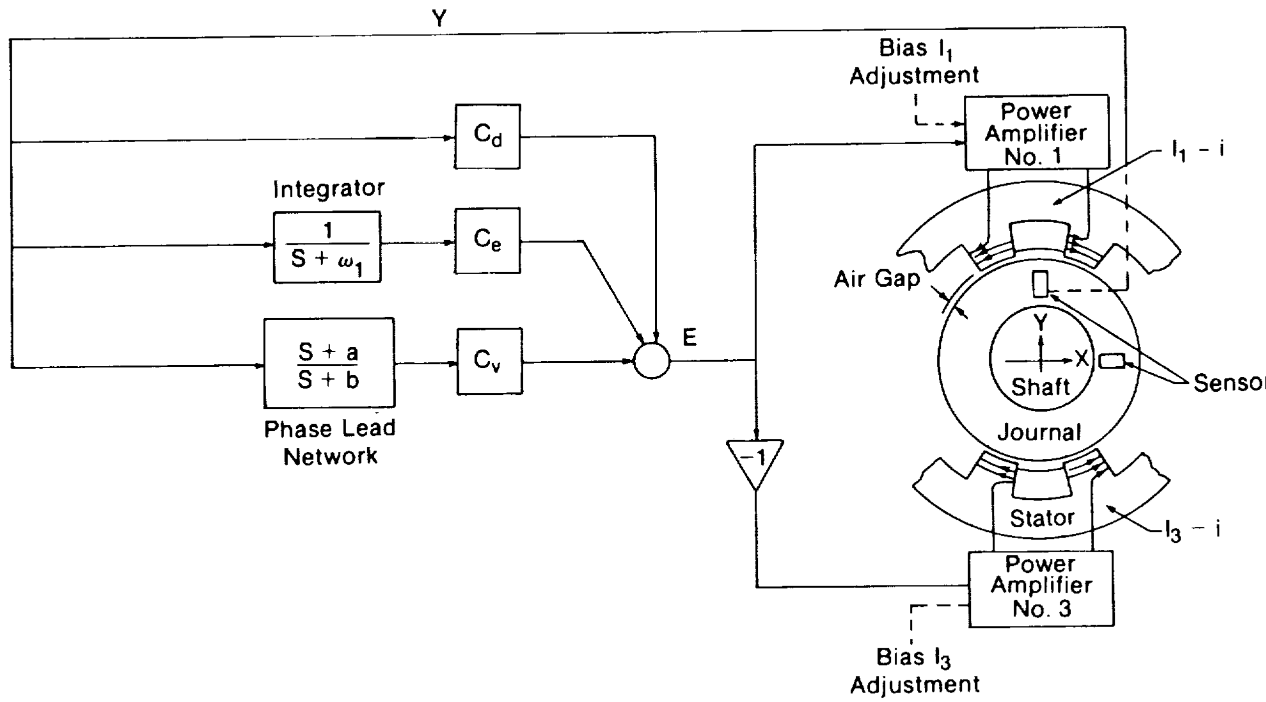

Fig. 5 Proportional-Integral-Derivative (PID) Controller

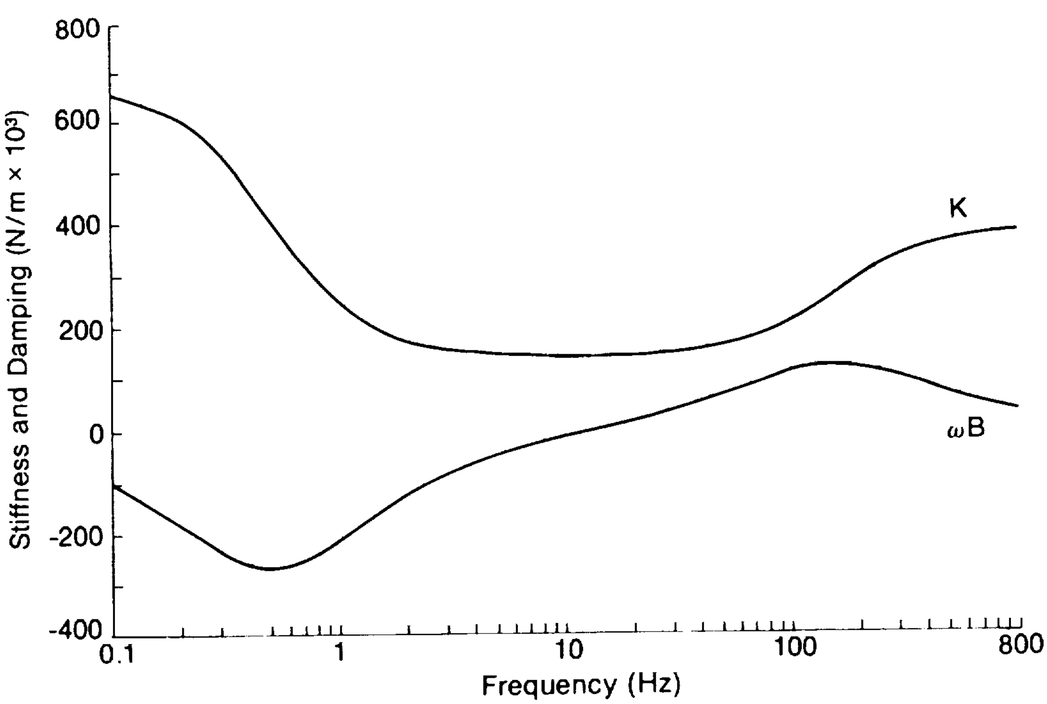

Fig. 6 Stiffness and Damping of a Typical Active Magnetic Bearing

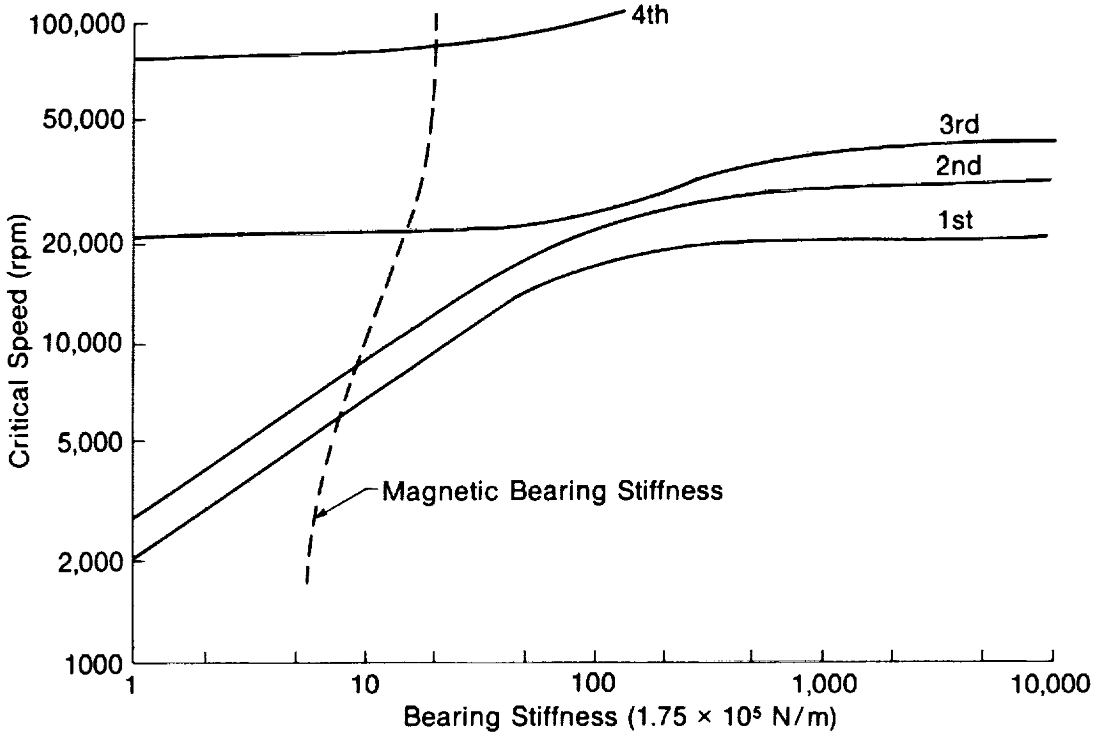

Fig. 7 Rotor Critical Speed Map