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ACTIVE MAGNETIC BEARING PERFORMANCE STANDARD SPECIFICATION

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Abstract

Numerous commercial turbomachinery applications utilizing active magnetic bearings are now being undertaken by several manufacturers. A minimum standardization of performance characteristics is proposed in this paper to minimize the difficulty encountered by turbomachinery manufacturers confronted with a new bearing technology. The difficulty of characterizing performance of the complete magnetic bearing system without the rotor is circumvented in this paper by suggestions for specification of bearing static and dynamic load capacity and vibration response of the completed installation.

1. Introduction

Like conventional bearings, active magnetic bearings' characteristics can be sufficiently described to allow assessments of rotor-bearing performance via rotordynamic analysis. Approximate equivalent stiffness and damping coefficients can be developed for magnetic bearings that allow conventional rotordynamic analyses including undamped and damped critical speed analyses and forced response analyses to be conducted. Although this approach allows some insight into rotor-bearing system performance, it partially masks the true nature of magnetic bearings as feedback control systems.

Since the objective of magnetic suspension applications in rotating machinery is to stably locate the rotor during machine operation, a magnetic bearing system is, in essence, a position control system that suffers some complications from rotor rotation. The accuracy of the rotor position control is determined by position sensor measurement accuracy, controller gain, controller bandwidth, and disturbance forces acting on the rotor. However, the fundamental challenge to successful magnetic bearing application is achievement of system stability. Stability of the rotor-bearing system is related to the vibration performance of the machine, an important consideration in specification of any type of bearing system.

Accordingly, it would seem that the most meaningful specification of magnetic bearing system performance would be in classical control parameters related to frequency response, relative stability, and stability robustness to parameter variations. However, this requires complete knowledge of the characteristics of the turbomachinery manufacturer's rotor which comprises the plant of the feedback control system. Furthermore, the controller actuator, the magnetic bearing, is open-loop unstable. The rotor itself may also be open-loop unstable for high-speed operation and this condition generally leads to a minimum and maximum controller bandwidth requirement for achievement of stability over the operating speed range.

The subsequent discussion is directed at an examination of the characteristics of magnetic bearing systems important to overall performance of the rotor-bearing system. This will permit development of performance criteria for inclusion in a standard specification for magnetic bearing application in turbomachinery. Not considered in the following are the physical construction features of the bearing actuators and the controller that will have important effects on system reliability.

2. Discussion

Open-Loop Behavior of Plant

For a general rotor system, a minimum of five axes of motion control are required: four axes of radial motion control, two at each of two radial bearings, and an axial axis of control provided by a thrust bearing. The basic system that is required for active control of one axis of the rotor will consist of at least one state variable sensor, a magnetic control coil, a coil driver, and the electrical network to connect it all together. These elements introduce their own individual characteristics to overall system performance including frequency response limits, eddy current losses, and saturation which must be accounted for in the overall specification of the system. In addition, mechanical resonances of the stator and rotor will interact with the performance of the control system.

Specification of magnetic bearing system performance parameters starts with a discussion of the frequency behavior of the plant of the feedback control system. The components in the upper half of Fig. 1 define the plant in one axis of control as consisting of the suspended rotor, the magnetic negative stiffness, the magnetic bearing, and the power amplifier.

Note that there are two types of disturbances acting on the plant: E_i and E_o. The first, E_i, is the external forces acting on the rotor including rotating unbalance caused by the misalignment of elastic and inertial axes and aerodynamic or hydrodynamic forces. These forces are summed with the magnetic bearing control forces before passing through the rotor to affect the rotor output state variable of position. The second type, E_o, usually ignored in conventional treatments, is the measurement disturbance created by an imperfect state variable measurement. For a position sensor looking at an imperfect shaft surface, the result is a corrupted position measurement that consists of the true center of mass position and components at harmonics of the rotational speed. This corrupted measurement is used as a feedback signal for the magnetic bearing hence producing a disturbance that excites rigid and flexible modes of the rotor system.

The power amplifier and the magnetic bearing have been included in the definition of the plant to emphasize their influence on the control system performance. Mechanical resonances in the rotor are unavoidable and they impose constraints on the frequency dependent characteristics of the power amplifier and the magnetic bearing.

The power amplifiers and magnetic bearing contribute phase lag to the open-loop transfer function for the plant requiring stabilization at high frequencies where structural interaction can occur [1]. Core losses in the magnetic bearing contribute to phase lag by opposing flux changes in the bearing at a frequency below that normally associated with an L-R circuit. Accordingly, the bandwidth of the bearing should be as high as practical as will be shown subsequently. The magnetic bearing is represented in Fig. 1 as a single lag with a break frequency of ω_b and a roll-off thereafter which is first order (slope of -1 or less).

The power amplifier is also represented as a first order lag with a break frequency that should be as high as practical for best performance. Furthermore, the coil driver should have a characteristic which is independent of load inductance and resistance. Unfortunately, this constraint is coupled with power limitations and the complications of design offered by pulse width modulation of the current to the bearing. Preferred designs should operate as a true transconductance amplifier yielding a current output for a voltage input. The amplifier total harmonic distortion at rated output should be as small as possible to avoid nonlinear resonance effects.

Considering a simple rotor system supported by an uncompensated magnetic bearing, the equation of motion for one axis of control is:

$$m\ddot{x} + k^{-} x = f \qquad (1)$$

where m = rotor mass
k^- = magnetic bearing negative stiffness
x = rotor displacement
f = external force

Equation (1) above applies directly to the axial control axis. For a radial control direction, Equation (1) applies if k^- is taken as the effective or combined negative stiffness of all magnetic bearings in the control direction. Equation (1) assumes no cross-coupled motion between axes.

In Laplacian notation, Equation (1) can be expressed as:

$$ms^2 X(s) + k^- X(s) = F(s)$$

$$\frac{X(s)}{F(s)} = \frac{1}{ms^2 + k^-} \qquad (2)$$

where $s = j\omega$ and the capital letters in functions of s denote Laplace transforms. Accordingly, the rotor transfer function, $X(s)/F(s)$, is shown for simplicity in Fig. 1 as $1/ms^2$ with negative feedback of the magnetic bearing uncompensated negative stiffness. Thus, this representation is strictly correct for rigid body motion only but flexible mode effects are added subsequently. The $1/ms^2$ transfer function shows that the rotor acts as a low pass filter rejecting high frequency disturbances.

Flexible mode effects of cross-coupled motion of real rotor systems can be accounted for by incorporating multi degrees of freedom and including gyroscopic and internal damping parameters in the model. Using complex notation to allow a more compact form, the equation of radial motion can be written:

$$[M]\{\ddot{z}\} + [D - j\omega G]\{\dot{z}\} + [K - j\omega D]\{z\} = \{f\} \qquad (3)$$

where [M] = rotor mass matrix
[D] = rotor damping matrix
[G] = rotor gyroscopic matrix
[K] = rotor stiffness matrix
{f} = rotor external force vector
{z} = rotor displacement vector
= $[.. x_i + jy_i .. || .. \beta_i + j\alpha_i ..]^T$
x_i = translational displacement of the *ith* mass along the x- axis
β_i = angular displacement of the *ith* mass about the x- axis
y_i = translational displacement of the *ith* mass along the y- axis

α_i = angular displacement of the *ith* mass about the y- axis
ω = frequency

Following the development of Schweitzer [2], the eigenvalue problem for the above system can be formulated. Solution shows that internal damping can only destabilize if the rotor is spinning. Backward whirls cannot become unstable, but forward whirls can become unstable when the rotor speed exceeds the first flexible mode natural frequency.

The transfer functions for the flexible mode resonances can be expressed as [3]:

$$R(s) = \prod_{i=1}^{n} \left\{ \frac{\left(\frac{s}{\omega_{zi}}\right)^2 + 2\,\xi\left(\frac{s}{\omega_{zi}}\right) + 1}{\left(\frac{s}{\omega_{pi}}\right)^2 + 2\,\xi\left(\frac{s}{\omega_{pi}}\right) + 1} \right\} \quad (4)$$

where ω_{zi} is the zero of the *ith* resonance
ω_{pi} is the pole of the *ith* resonance
ξ is the damping ratio including external as well as internal effects

At each resonance there is a gain increase of 40 dB per decade over a frequency range corresponding to the pole-zero separation, and a maximum phase shift of 180° depending on the damping and the pole-zero separation. This damping can easily result in a narrow band gain increase of 40 dB or more.

The complex S plane can be used to describe the relative stability of the open- and closed-loop plant, i.e., before and after compensation. The poles of the rotor transfer function correspond to the eigenvalues of the related characteristic equation for Equation (3); the real part is the damping factor and the imaginary part is the damped natural frequency. Stability requires all poles to be located in the left-half plane (LHP) where the negative real part of the eigenvalue ensures sufficient damping.

The rigid body transfer function of $1/ms^2$ results in two poles at the origin of the complex S plane. These rigid body poles may migrate from the origin due to gyroscopic effects as rotational speed is increased. Generally, the rotor transfer function will also contain a number of complex conjugate poles and zeros as described by Equation (4). These poles and zeros tend to migrate toward the right-half (RHP) of the complex S plane with increasing rotor speed due to the rotor cross-coupled damping effects. Thus, they can easily contribute to rotor instabilities. Fig. 2 shows a representation of the poles and zeros for a typical open-loop rotor system and their direction of travel with increasing speed.

Another important instability in the open-loop plant is that caused by the uncompensated, negative stiffness of the magnetic bearing itself. This aspect of magnetic bearings has been well documented in the literature (e.g., [4]).

The force developed by a magnetic bearing operating on a rotor through an air gap is:

$$F = \frac{\partial F}{\partial I}\,\partial I + \frac{\partial F}{\partial x}\,\partial x \quad (5)$$

where x = rotor displacement.
I = bearing current

or expressed as a stiffness,

$$K = K^{+} + K^{-} \quad (6)$$

where the first term is the bearing positive "current" stiffness and the second term is the bearing negative "position" stiffness caused by the increasing attractive force as the gap is reduced. This must be overcome for stable suspension since the homogeneous solution to Equation (1) is in the form of hyperbolic functions indicating that x grows with time. This can be accomplished for a given rotor position in the air gap by ensuring that the first term in Equation (6) is larger, i.e., $K^{+} > K^{-}$.

A Bode plot of the above defined plant for a typical rotor system at zero speed neglecting the bearing negative stiffness is illustrated in Fig. 3. These plots are the steady state magnitude and phase ratios of output position, as produced by an input sinusoidal voltage, versus the frequency of input excitation, ω. It is easy to show that inclusion of the bearing negative stiffness yields undamped poles in the complex S plane. Means of negating this effect will be discussed subsequently.

At low frequencies, the zero speed plant is dominated by the rigid body poles with a second order magnitude roll-off (slope of -2) until the first flexible mode resonance shown arbitrarily at 10^3 rad/sec. The trough of the resonance is caused by the zero of the resonance transfer function and the peak of the resonance is caused by the pole of the resonance transfer function, Equation (4). The pole-zero combination of the flexible mode resonance causes a 180° maximum phase shift. The magnitude slope steepens and the phase falls off from -180° with increasing frequency due to the simple poles of the amplifier and bearing which have been placed arbitrarily in Fig. 3 at 10^4 rad/sec and 10^3 rad/sec, respectively. These characteristics imply that the addition of phase lead is required to stabilize the system. The occurrence of additional flexible mode resonances further complicates the problem of plant stabilization as will be shown subsequently.

The magnitude plot will be somewhat modified with increasing speed if cross-coupled damping or large gyroscopic effects cause migration of the poles and zeros in the complex S plane. Gyroscopic effects may cause the system poles and zeros to move either toward or away from the RHP with increasing speed indicating a destabilizing or stabilizing influence, respectively, according to the values of rotor polar and transverse inertias, rotor flexibility, and geometric parameters. As the rotational speed is increased, cross-coupled damping may cause some forward whirl poles and zeros of the rotor transfer function to migrate toward the RHP and the rotor can become open-loop unstable when the rotational speed equals a rotor flexible natural frequency. The pole and zero positions in the complex S plane tend to restrict the minimum and maximum gain and bandwidth of a compensating controller as shown by Johnson [5]. Also, the amplifier and bearing bandwidths introduce further gain and bandwidth restrictions to the controller.

In summary, plant variations with speed, open-loop instabilities caused by unfavorable gyroscopic and internal damping effects, component frequency response limitations, and structural interactions combine to make control synthesis of a stabilizing compensator for a typical rotor system a difficult proposition.

Controller Requirements

Since the plant comprises an open-loop system that may have one or more zeros and poles in the RHP, feedback alone is not sufficient for stabilization. Some form of series compensation with lead networks is required for stability. We are interested in stability at nonsynchronous frequencies as well as synchronous frequencies. Furthermore, parameter variations and nonlinear effects within the system components can be expected. Therefore, the controller for a magnetic bearing system has to possess robustness properties that ensure stability under these conditions.

Another requirement of the controller is that the closed-loop system must accommodate disturbances, E_i in Fig. 1, at both synchronous and nonsynchronous frequencies. Nonsynchronous disturbances can come from a variety of sources including coupling misalignment, seal and impeller hydrodynamic effects, rubs, etc. The controller must possess good synchronous response characteristics since the primary exciting mechanism in most turbomachinery applications will be mass unbalance. The synchronous response must be limited by the controller to rotor whirl amplitudes that are less than the magnetic bearing air gap. The corresponding synchronous bearing forces must be consistent with the dynamic load capacity of the magnetic bearings. Accordingly, a relatively flat synchronous response is desirable.

As in other position control systems, optimal control for a magnetic bearing system can generally be achieved with full state feedback, Fig. 4. However, the rotor state variables of velocity and acceleration are not naturally accessible in a magnetic bearing suspension and must be generated with more hardware than a displacement feedback system. The result is additional complexity and cost of the total system. This problem can be circumvented in digital control systems by generating state variables with software.

The negative stiffness of the bearing is a challenging control problem. Here again, state variable feedback and digital technology offers a solution. As shown by Chen [6], a velocity observer implemented into a model based compensator can recover rotor velocity without differentiation of the rotor position signal and ensure system stability despite the magnetic bearing negative stiffness.

Alternatively, flux feedback can minimize the negative stiffness with less control complexity. Since the magnetic force of attraction is given by:

$$F = \frac{B^2 S}{2\mu_0} \tag{7}$$

where B = magnetic induction
S = bearing projected surface area
μ_0 = permeability of free space

it is evident that a linear force vs. rotor displacement is possible if $B \sim \sqrt{x}$. This latter relationship is possible by measuring B. The result is a bearing force characteristic which is independent of the bearing air gap; the unstable effect largely disappears. In fact, reductions in the negative stiffness on the order of five to ten times are achievable.

Some bearing designs utilize linearization of the bearing stiffness by the use of large DC current biases alone to minimize the effect of negative stiffness. This scheme does not eliminate the unstable effect of the negative stiffness and the remaining nonlinearity introduces resonant frequencies into the dynamic system which are subharmonic and ultraharmonic to the main resonance (pole) predicted on the basis of linear theory. These nonlinear resonances can cause further restrictions on stabilizing controllers particularly when the bearing is operating near saturation.

A review of the optimal control technologies seems to indicate that the linear quadratic regulator (LQR) offers everything desirable in a magnetic suspension: good stability, robustness, and flat frequency response properties. Essentially, this design methodology is similar to shaping of the open-loop Bode magnitude plots for performance enhancement. Selection is made of the weighting matrices used to express the relative cost assigned to deviations of the control state from equilibrium and the relative cost assigned to control effort. This technique can be applied to both full state feedback controllers and output controllers using model based compensators. The velocity observer mentioned above is an example of the latter.

The optimal performance of the full state feedback controller using variable gain to account for variation in plant with rotational speed appears ideally suited to magnetic bearing applications. Nevertheless, as shown by Johnson's analysis [5] for his two mass rotor system, output state feedback using LQR design techniques failed to provide any real advantage over simple lead-lag controllers in terms of nominal stability, synchronous response to measurement error, synchronous response to mass unbalance, stability robustness to speed variation, and stability robustness to variations in structural characteristics of the rotor.

For all the designs studied by Johnson, the open- loop RHP poles and zeros placed minimum and maximum bandwidth restrictions on stabilizing compensators. The above results can be expected to apply as well to general rotor systems with the additional characteristics of variation in pole frequencies with rotation speed and additional bandwidth and gain restrictions caused by higher order structural resonances. Some rotor systems with large disks can also be expected to exhibit unstable gyroscopic modes. Rotors with unsymmetric cross section could be subject to parametric self-excited resonance vibration. However, these are all characteristics that have been successfully addressed in industrial turbomachinery with lead-lag controllers for both direct and cross axis compensation.

Compensation of the typical plant defined in Fig.s 1 and 3 can be readily achieved with a lead-lag controller provided special attention is made to the flexible mode resonances that fall within the bandwidth. To provide stability, the controller gain must be increased to provide a gain crossover in a region where phase is greater than -180°. However, the location of the flexible mode resonances with respect to the low frequency gain crossover dictates the minimum amount of phase necessary for plant stabilization.

This condition can be clearly depicted on a Nichols plot of open-loop magnitude and phase vs. frequency. In this plot, the critical points where instability occurs are the multiple origins of -180° ± n 360° phase and 0 dB magnitude. The gain and phase margins for relative stability are measured, respectively, from the phase crossover at -180° ± n 360° and gain crossover at 0 dB. There may be multiple crossovers at a single critical point due to flexible mode resonances.

Each mode affects the magnitude-phase plot by contributing a counterclockwise loop to the loop transfer function [3], Fig. 4 . A phase stabilized mode is one which for arbitrarily small structural damping ratio is closed-loop stable regardless of how high the loop gain is raised; the mode "loop" never crosses the -180° ± n 360° line, even for zero damping. A gain stabilized mode is one which is closed-loop stable for the actual loop gain and damping ratio, but can become unstable if the gain is raised or the damping ratio is lowered; the mode loop crosses the -180° ± n 360° line if damping ratio is sufficiently small.

The Nichols plot of Fig. 4a was developed for a typical rotor such as shown in Fig. 3 after simple lead-lag compensation. Several flexible modes are depicted. The plot shows the need to stabilize mode 2 by providing additional phase lead to compensate for the amplifier and bearing phase lag thereby avoiding the phase crossover at the mode resonance frequency. The result after higher order lead compensation is shown in Fig. 4b. A phase crossover with reasonable gain margin has been provided between resonances (after mode 2 in Fig. 4b). Higher order modes of large magnitude also have sufficient phase margin (e.g., mode 4 in Fig. 4b).

As discussed above, a simple lead cell is sometimes insufficient to provide acceptable stability margin and higher order lead compensation is necessary. This results in greater loop gain which can destabilize gain stable modes (e.g., mode 3 in Fig. 4b with a lower damping ratio). There is a ready solution to this problem which is used all the time by the authors' company and affiliates: a notch filter to remove the gain at the offending frequency with a minimal increase in the overall gain. Mechanical dampers mounted to the shaft have also been successful in attenuating these resonances at high frequencies thereby stabilizing the offending mode.

The decision to employ an output controller, be it a lead-lag or a model based compensator using a velocity observer, still requires consideration of the feedback measurement system--the position sensor. The sensor must have good sensitivity, temperature stability, high signal to noise and harmonic rejection characteristics, and high linearity with respect to changes in the magnetic bearing air gap. Magnetic bearing systems have been built with capacitive, eddy-current, optical, and inductive devices, but the success of these systems in industrial environments varies in part with the extent to which the above criteria are satisfied.

The inductive sensor with individual coil elements bridged around the rotor for rotor surface harmonic rejection has performed admirably in turbomachinery applications. This sensor intrinsically minimizes the E_0 disturbance in Fig. 1 due to an imperfect shaft surface. Output disturbances, due to misalignment of the sensor measurement axis and the bearing axis must be minimized by good machine design practice. This generally requires direct mounting of the sensor to the bearing assembly. Also, where electromagnetic interference is a concern such as in motors and generators, shielding can be implemented to ensure a clean position sensor signal.

Minimum acceptable performance of sensors for both radial and axial position measurement are listed in the summary section which follows. The phase lag characteristics of sensors, particularly inductive types, do not compromise the design of stabilizing controllers since these are low power devices that can be readily compensated.

Previous considerations of the characteristics of stabilizing controllers assumed colocation of the bearing and sensor. Most practical system will have sensors and bearings separated by an axial dimension along the length of the rotor. Depending on node locations of resonant frequencies relative to the bearing and the sensor, the sensor gain may be affected. Maslen and Bielk [7] show that the effect of sensor noncolocation is generally to reduce the stability of some higher order modes and increase the stability of lower order modes.

Bearing System Load Capacity

The magnetic bearing actuator design must be consistent with frequency and voltage characteristics of the controller in order to develop peak dynamic forces as required over the control bandwidth to maintain contact free suspension. Furthermore, this must be done within current limitations without excessive bearing inductance.

For a four quadrant bearing system, two different operating modes are possible: Class A and Class B. Class A takes advantage of opposing quadrants to react the dynamic load.

As shown by Bornstein [8], neglecting the reluctance of the magnetic core , this configuration can develop a peak dynamic force given by:

$$F = \frac{I_{max} V_{max} S_i N_{pp}}{2 S \varepsilon_0 \omega} \qquad (8)$$

where
- I_{max} = maximum current available from amplifier
- V_{max} = maximum voltage available from amplifier
- S_i = inductive area of a single magnetic bearing pole
- N_{pp} = number of pole pairs in the bearing quadrant
- S = projected area of magnetic bearing quadrant
- ε_0 = magnetic bearing air gap
- ω = frequency of disturbance

and all variables are in consistent MKS units.

Implicit in the development of the above is that the saturation current of the bearing coincides with the amplifier current rating; a usual condition for cost effective designs.

Class B control utilizes current modulation to only one quadrant to react to the dynamically varying load while the opposing quadrant is energized with a small bearing DC current. For horizontal installation, only the upper quadrants modulate the load. The maximum dynamic force developed in a Class B configuration neglecting magnetic core reluctance is [8]:

$$F = \frac{I_{max} V_{max} S_i N_{pp}}{2.83 S \varepsilon_0 \omega} \frac{1}{X_s^{1/2}} \qquad (9)$$

where the factor of safety over the static load rating, X_s, has been purposely introduced to illustrate the reduction in dynamic load capacity incurred by overspecification of X_s.

Recommendations for minimum load capacity specifications are contained in the summary.

Overall System Performance Specification Using Vibration Standards

Ideally, performance criteria for a magnetic bearing controller could be specified by the classical control system specifications of gain and phase margin (relative stability), synchronous frequency response, and stability robustness to plant parameter variations. However, as can be concluded from the above discussion, there are a multitude of options for stabilizing controllers and actuator designs for a given rotor system. Identification of suitable criteria is difficult in rotor- bearing systems where rotor open-loop behavior will vary widely across the class of turbomachinery and instability can arise from various sources. This condition makes specification of controller performance requirements difficult.

The problem of practical system performance specifications can be resolved by resorting to the traditional frequency response parameter used for performance evaluation of turbomachinery rotor bearing systems: vibration amplitude. Although vibration standards are often invoked to minimize damaging dynamic loading of conventional bearing systems, satisfaction of appropriate vibration standards for magnetic bearing systems ensures that the previously mentioned attributes of closed-loop stability at synchronous and nonsynchronous frequencies have been met, and that the system is relatively insensitive to disturbances. However, vibration standards do not explicitly indicate relative stability margins and they do not ensure stability robustness. They also unfairly burden the magnetic bearing manufacturer with the task of demonstrating acceptable performance over a rotor system for which he is not completely responsible. Nevertheless, with proper rotor design, manufacture, and balancing, the authors' company has found it possible to meet rigid vibration requirements.

In fact, the proposed vibration standard is the same to which a wide class of turbomachinery are now accepted--the American Petroleum Institute (API) Standards. Fig. 5 shows the various standards for motors (API 541), pumps (API 610), centrifugal compressors (API 617), gas turbines (API 616), and steam turbines (API 611). The herein proposed standard corresponds to API 617 and is cross-hatched in the figure. This standard requires that vibration be limited to $2780/\sqrt{f}$ μm or

50 µm, whichever is less, where f is the frequency in cpm. The authors' company has been able to meet these requirements in all of its installations provided the customer's rotor is multiplane balanced to a minimum of the ISO 1940 standard for the appropriate class of machinery.

3. Summary

The unique performance aspects of magnetic bearing applications to turbomachinery have been reviewed in the context of actively controlled rotors to identify the salient performance requirements that need to be identified by machinery manufacturers in the specification of magnetic bearing systems for their use. While these specifications cannot be totally definitive, they do provide the genesis of a common specification that will help ensure satisfactory performance of the rotor-bearing system.

The results of this review and recommended specification criteria can be summarized as follows:

The bearing system shall permit the driven equipment to run continuously at all speeds within the operating speed range at the maximum unbalance permitted by ISO 1940 for the appropriate class of rotor without exceeding a double amplitude of vibration measured adjacent to the individual bearings of $2780/\sqrt{f}$ µm or 50 µm, whichever is less, and where f is the frequency of vibration in cpm. The controller may use analog or digital technology of any of a variety of algorithms provided it meets the above requirements for vibration and is stable (repeatable) with time. Output or full state feedback using position, velocity, current, flux, or force is permissible provided it meets the above requirements for vibration and is stable with time. Power amplifiers shall be low distortion, linear or switching type with a bandwidth consistent with the above vibration requirements. Bearing bandwidth shall be consistent with the above vibration requirements.

The load capacity of the bearing system shall be a minimum of the total of static plus dynamic loads for the application where static and dynamic loads are as defined below. Static loads consist of weight and steady state aerodynamic, hydrodynamic, or electromagnetic loads of the driven equipment and dynamic loads consist of varying components of the above, plus unbalance loads of the driven equipment. The required dynamic capacity of radial bearings is specified at synchronous frequency and shall be a minimum of 2.5 times the appropriate ISO grade to which the rotor is balanced. The required capacity of axial bearings shall be a minimum of the total of specified static and dynamic loads as indicated above.

The radial position sensors shall feature a linearity and sensitivity that is independent of environmental temperature and that is consistent with stable operation of the magnetic bearing-rotor system. This generally requires linearity for radial sensors to within 2% over the magnetic air gaps and a minimum sensitivity of 10 V over the air gap involved. These requirements may be relaxed for axial sensors. To reduce plant measurement disturbance, the mounting of the radial position sensors shall ensure a concentricity with respect to the radial bearing bore of 50 µm per meter of bearing diameter with a lower limit of 15 µm.

Not previously addressed in this paper, but also important:

All magnetic bearing components shall be compatible with the specified application operating environment of temperature, pressure, humidity, and process fluid. Temperature increase due to joule, hysteresis, eddy current, and windage losses within the magnetic bearing components shall not unduly compromise the performance or the life of the bearing system. Also, operating stress levels due to centrifugal, thermal, electromagnetic, and mechanical loads shall not unduly compromise the performance or the life of the bearing system.

Before design approval, the magnetic bearing manufacturer should be required to conduct and present sufficient analyses to demonstrate that the above requirements are met. However, the equipment manufacturer is ultimately responsible for satisfactory performance of the complete machine with the magnetic bearing system. To this end, the magnetic bearing manufacturer should supply this necessary performance data to enable the equipment manufacturer to make this determination and release the magnetic bearing vendor to proceed with the manufacture and delivery of the complete magnetic bearing system.

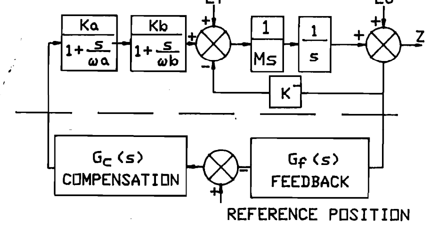

FIGURE 1
MAGNETIC BEARING SYSTEM FOR ONE AXIS CONTROL

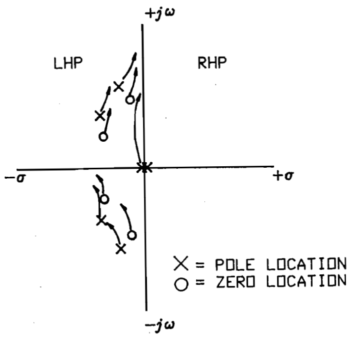

FIGURE 2.
COMPLEX S PLANE. ROOT LOCUS VS. ROTATIONAL SPEED FOR TYPICAL ROTOR SYSTEM-TWO RIGID BODY MODES AND TWO FLEXIBLE MODES (COMPLEX CONJUGATE PAIRS) SHOWN.

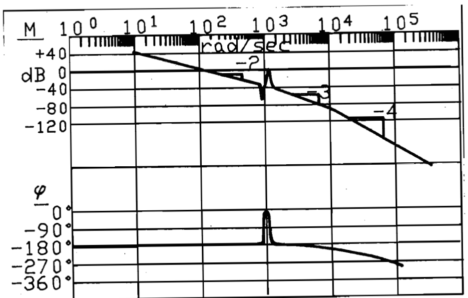

FIGURE 3.
TYPICAL PLANT OPEN LOOP BODE PLOT ONE FLEXIBLE MODE RESONANCE

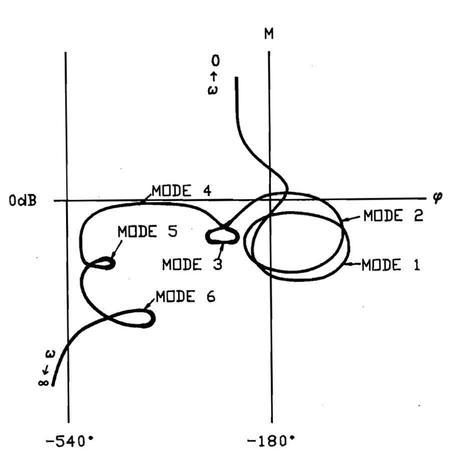

FIGURE 4a
LEAD-LAG COMPENSATED PLANT BEFORE FLEXIBLE MODE COMPENSATION.

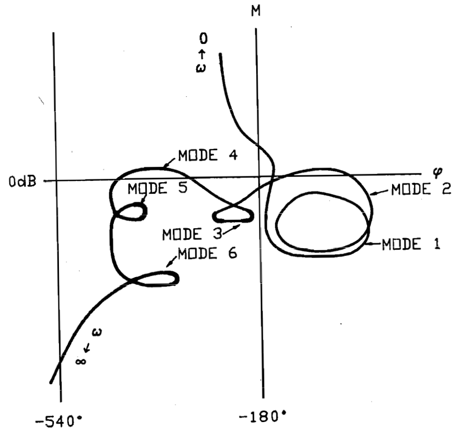

FIGURE 4b
PLANT AFTER FLEXIBLE MODE COMPENSATION.

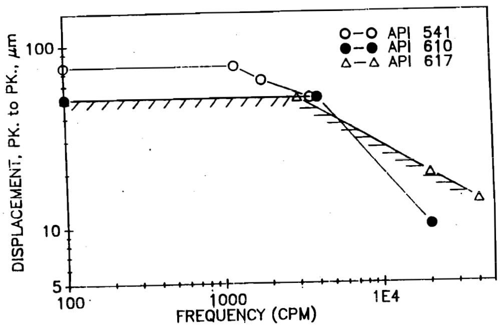

FIGURE 5. PROPOSED VIBRATION STANDARD

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