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ABSTRACT

Active magnetic bearings (AMBs) are a mature bearing technology that is being applied to many new turbomachines. They have become the bearing of choice for most new turboexpander applications. They are seen as a key enabling technology for compact, high speed, high power density direct-drive generators and turbomachinery. As more and more units are fielded, it will become important that end-users and analysts have the tools and background to evaluate the rotordynamics of AMB supported machinery.

This tutorial provides a basic review of AMB subsystems and key issues that need to be considered for a rotordynamic audit. Applicable API and ISO standards are reviewed from the perspective of evaluating the rotordynamic performance of AMB machinery. The tutorial concludes with an example audit of an existing AMB supported machine.

INTRODUCTION

Active magnetic bearings (AMBs) have reached a critical mass in terms of units fielded and new units in production. They have become the bearing of choice for most new turboexpander applications. They are seen as a key enabling technology for compact, high speed, high power density direct-drive generators and turbomachinery. Several of the configurations proposed for ultra-high reliability machinery for subsea processing systems use AMBs.

For critical process machines being newly manufactured or ratered, in-house and independent design audits have proven to be very effective in preventing project delays and costly downtime. This remains true for AMB-supported machinery. New versions of the various API standards are also expected to explicitly address AMB-supported machinery, possibly leading to expanded analysis requirements for these machines.

From the perspective of rotor/bearing system dynamics, AMB supported systems still have essentially the same concerns for stability, unbalance response, and critical speed margins as traditional machinery. However, the incorporation of a carefully engineered feedback control system brings new complexities. Within limits, the control system designer has considerable flexibility in specifying critical speeds and how the system will respond to unbalance and other dynamic forces. AMB system developers have developed sophisticated design tools to allow the bearing-control system-rotor structural characteristics to be evaluated, and an appropriate control system to be specified.

For refinery service, the usual American Petroleum Institute (API) standards covering machine dynamics still generally apply. However, it has been argued that some of the standard API requirements may not be completely appropriate for an AMB system. It is also clear that the API standards do not address some of the unique concerns raised by an actively controlled bearing system. Recently, the International Standards Organization (ISO) has developed a standard covering AMB system terminology, characteristics, and dynamic requirements (ISO 14839). This standard addresses a number of issues unique to an AMB system.

The goal of this tutorial is to provide the end-user, system integrator, and analyst with an overview and guide to the rotordynamic design audit process for AMB supported rotors. While axial dynamics of AMB-supported machines are an important area of concern, lateral dynamics will be the sole focus here. Where appropriate, references are provided to allow readers who need more technical background on some of the issues related to design and analysis. The tutorial concludes with an example audit.

ACTIVE MAGNETIC BEARINGS—BACKGROUND

Conceptually, an AMB system is quite simple. An AMB system uses magnetic attraction to keep the rotor located in the center of the clearance space, despite unbalance forces, aerodynamic forces, casing motion, and so forth. Such a system includes an array of powerful electromagnets that apply forces to the shaft, multiple sensors that monitor the shaft position, a controller that generates a signal that tries to move the rotor to a desired location (i.e., the center of the clearance space), and power amplifiers that convert the controller output to voltage/current suitable to drive the electromagnets.

Figure 1 shows an outline of a typical AMB system and Figure 2 provides a block diagram of the various dynamic components within the system. Each of these will be examined within this tutorial.
The appropriate forces to be applied are determined by monitoring the motion of the rotor with the sensors. If the journal is off-center, then the magnet forces are adjusted to pull it toward the center. This adjustment is done very rapidly; typically on the order of 5 to 20 thousand times per second so that the adjustments appear to be continuous. The manner in which this decision process is carried out—how the magnets are adjusted to correct the rotor position—is called the control algorithm. Typical control algorithms result in an AMB behavior that is similar to that of other bearings so that the complete unit is often described as providing stiffness and damping to the rotor. AMB control systems, however, can provide much more sophisticated control over the rotor. Quite commonly, for example, motion at one end of the rotor is linked to forces at the other end of the rotor. In addition, it is possible to allow the rotor to rotate about the center of mass, thus reducing or eliminating transmitted vibration arising from mass unbalance.

Magnetics

The business end of an AMB is the electromagnets, which produce the bearing forces. An electromagnet works by generating a magnetic flux distribution on a target, which, in this case, is the rotor journal.

Force Capacity

The net force on the target is found by integrating the square of the flux density, $B$, over the entire target surface. In a practical magnet, the flux density is very high in the air gap between magnet and target and is very small elsewhere so that calculating the force based only on the magnet area, as in Equation (1), is quite accurate (Marshall and Skitek, 1987).

$$ f = \sum \frac{B_i^2}{2\mu_0} A_i $$

(1)

In this equation, $\mu_0 = 4\pi \times 10^{-7}$ (Tesla m)$^2$/N is the magnetic permeability of free space, $A_i$ is the $i$-magnet pole area facing each gap and $B_i$ is the flux density (approximately uniform) in each gap.

The form of Equation (1) invites the notion of an effective magnetic pressure that is comparable to the projected area load of a conventional hydrodynamic bearing (i.e., “psi loading”):

$$ P_{\text{magnetic}} = \frac{B^2}{2\mu_0} $$

(2)

The fact that the maximum force that can be obtained from an electromagnet is limited by the magnetic saturation property of the magnet iron, $B_{\text{sat}}$, makes this approach especially useful. Thus, for materials of interest in AMB design, $B_{\text{sat}}$ ranges from about 1.2 Tesla to about 2.0 Tesla, implying a range of magnetic pressure from 0.6 to 1.6 MPa (90 to 230 psi).

For a practical AMB, the projected quadrant pole area is about 35 to 50 percent of the projected journal area. For the commercial AMB depicted in Figures 4 and 5, the projected area ratio, $r_p$, is 0.46.

$$ F_{\text{max}} = A_j r_p P_{\text{max}} $$

(4)

Given the limits on pole ratio and magnetic pressure, the maximum specific capacity of an AMB is between 0.21 to 0.80 MPa (30 to 120 psi). (Note: Specific capacity is defined here as the ratio of maximum bearing force [$F_{\text{max}}$] to projected journal area [$A_j$]: $F_{\text{max}}/A_j$.) For the commercial AMB depicted in Figure 5, the specific load capacity is 0.38 MPa (55 psi), indicating a magnetic pressure of 0.83 MPa (120 psi) and an average saturated gap flux density of 1.44 Tesla. This fact allows a relatively quick check on the force capacity of a magnetic actuator to be made at the start of the audit to ensure that the actuator is reasonably sized.

Actuator Force Linearization

As with conventional mechanical bearings, most rotordynamic analysis of an AMB system is conducted with the assumption of small motions about some steady-state location. Thus, a linearized set of equations is required to allow the linear rotordynamics model to be assembled. These equations are developed in the next few paragraphs.

It can be shown that the quadrant force for such a magnet structure can be approximated by:

$$ f = \frac{\mu_0 N^2 I^2 A_j r_p}{2g^2} $$

(5)

Of course, this relationship is inconvenient in that it is nonlinear (it depends on $I^2$ and inversely on $g^2$) and, in particular, is always...
a positive quantity pulling the rotor toward the electromagnet. However, the magnets are usually employed in opposing pairs (Schweitzer and Lange, 1976) so that:

$$f_{net} = f_1 - f_2 = \frac{\mu_0 N^2 A f_p}{2} \left( \frac{I_1^2}{g_1} - \frac{I_2^2}{g_2} \right)$$  \hspace{1cm} (6)

With this, it is now possible to generate either positive or negative forces. The choice of the two currents is aimed at generating a single force. Thus, it is reasonable to choose each current as a function of a single time varying control current, $i_c$. It is most common to select the two currents as a perturbation about a steady “bias” current $I_b$.

$$I_1 = \max(I_b + i_c, 0)$$

$$I_2 = \max(I_b - i_c, 0)$$  \hspace{1cm} (7)

Note that, because the two magnets are opposed, $g_1 = g_0 - x$ and $g_2 = g_0 + x$, where $x$ is rotor motion toward the first electromagnet and $g_0$ is the nominal air gap, corrected for iron permeability (usually on the order of 5 to 10 percent increase from the physical gap). The choice of $I_b$ will be discussed below, but it is common to scale it relative to the saturation current, $I_{sat}$, which is the highest useful or feasible coil current. The ratio of bias to saturation current is the biasing ratio, $\gamma_b$, where $I_b = \gamma_b I_{sat}$. It is common to choose $\gamma_b$ in the range from about 0.2 to about 0.5 although some AMB designs targeting very low power may push $\gamma_b$ very close to zero. If Equation (5) is combined with Equation (7) and expanded in a polynomial series about $(i_c = i_{c,0}, x = 0)$, then:

$$f \approx f_0 + K_i (i_c - i_{c,0}) - K_s x$$  \hspace{1cm} (8)

in which:

$$f_0 = \frac{\mu_0 N^2 A f_p I_b}{g_0^2}$$

$$K_i = 2\frac{\mu_0 N^2 A f_p I_b}{g_0^2}$$

$$K_s = -\frac{2\mu_0 N^2 A f_p}{g_0^3} (I_b^2 + i_{c,0}^2)$$  \hspace{1cm} (9)

The static load carried by the bearing is $f_0$, which is a function of the bias current and the steady-state component of the control signal. This force is equivalent to the steady-state load capacity of a conventional bearing. $K_i$ and $K_s$ are the actuator gain and magnetic stiffness, respectively (Humphris, et al., 1986).

Note that the magnetic stiffness $K_s$ is less than zero. This negative stiffness is essentially the increasing force felt as a magnet approaches another magnet or a magnetic surface. For a rotordynamic model, this negative magnetic stiffness is applied at each actuator location. This is the inherent source of instability in the AMB. One of the major reasons that the active control system is required is to compensate for this negative stiffness.

Equation (8) is a convenient model for the electromagnets because it is linear. For the actuator indicated in Figures 3 through 5, the actual force as a function of control current for a centered journal ($x = 0$) is plotted in Figure 6. The linear approximation is good for control currents up to 10 amps but begins to degrade after that. This nonlinear behavior is readily compensated in a digital control system by using a lookup table or curve-fitted relationship to choose the target control current, given a desired AMB force. The slope of the curve in Figure 6 is $K_i = 1.15$ kN/A at $i_c = 0$ whereas the calculation of Equation (9) predicts an actuator gain of $K_i = 0.98$ kN/A so the actual device slightly outperforms the analytic prediction.
Figure 8. Example Actuator Gain Frequency Characteristics with and without Eddy-Current Effects.

**Power Amplifiers**

Current is induced in the coils of the electromagnets by power amplifiers. These amplifiers are designed to apply large voltages to the coils and to monitor the resulting currents. The currents are typically compared to a request signal (provided by the AMB controller) and the voltages adjusted to try to force the coil current to track the request (Bardas, et al., 1990). This behavior is called transconductance. In understanding the behavior of an AMB, it is important to recognize that the power amplifier can only apply a finite voltage, fixed by the power supply ($V_{\text{max}}$), and that this limits how rapidly the AMB force can vary because of the electromagnets’ inductance. If the coil set driven by a power amplifier has inductance $L$ and resistance $R$, then the coil voltage (applied by the power amplifier) is related to the coil current by:

$$V = RI + L \frac{dI}{dt} + I \frac{dL}{dt}$$  

(10)

Exploiting the fact that there is a very simple relationship between the magnet inductance ($L$) and the actuator gain ($K_i$), the maximum rate at which the magnet’s force changes (often called maximum slew rate [Maslen, et al., 1989]) is governed by the power amplifier properties, the air gap, and the biasing ratio $\gamma$:

$$\frac{df}{dt} \leq \frac{2\gamma \beta I_{\text{sat}} V_{\text{max}}}{g_0}$$  

(11)

in which the RI term in Equation (10) has been neglected. Equation (11) is important because, in conjunction with Equation (4), it sets the dynamic capacity of the AMB. Thus,

$$|F_{\text{dyn}}(\omega)| \leq \min \left( A \rho P_{\text{max}} - F_{\text{static}}, \frac{2\gamma \beta I_{\text{sat}} V_{\text{max}}}{\omega g_0} \right)$$  

(12)

This relationship is depicted in Figure 9. As shown, the force limit is controlled by the maximum force up to some frequency, then begins to decrease due to the limit imposed by the inductance and power amplifier voltage. For the rotordynamic design audit, the results of the forced (unbalance) response calculation need to be compared with this frequency dependent characteristic to ensure that the bearing has adequate dynamic load capacity over the full operating speed range of the machine. This is especially true for very high speed machines.

Figure 9. Dynamic Capacity as a Function of Frequency.

The power amplifier is characterized by its gain, bandwidth, voltage limit, and maximum current. For a given electromagnet set, the maximum current $I_{\text{max}}$ should be matched to the saturation current for the electromagnets, $I_{\text{sat}}$. Then the dynamic capacity of the AMB will be controlled by the maximum amplifier voltage ($V_{\text{max}}$) and by the ratio between bias current and maximum control current ($\gamma_b$). Making the bias ratio high increases the nominal electrical power dissipation in the magnet coils (this scales roughly as the square of the bias ratio) so there is an explicit tradeoff between power dissipation in the AMB and dynamic capacity.

Since the amplifier is part of the control feedback loop, the frequency response of the amplifier $G_{\text{amp}}(s)$ and actuator combination needs to be included in the model. As shown in Figure 10, $G_{\text{amp}}(s)$ could be modeled as a second order low pass filter. In other cases, the model’s frequency response characteristics could be substantially more complex.

Figure 10. Frequency Response Characteristics of an Example Amplifier.
Eddy-Currents

Eddy-currents are induced in any conductive material that is exposed to a changing magnetic flux (Stoll, 1974). These eddy-currents tend to counteract the flux induction of the coil currents at high frequencies and, so, act to limit the magnetic bandwidth of the electromagnets. For radial actuators, finely laminated radial actuators are used. They should exhibit a magnetic bandwidth on the order of 2 kHz, so that this effect can either be neglected or lumped into power amplifier dynamics (Sun and Yu, 2002, Zmood, et al., 1987, Meeker and Maslen, 1998, Kasarda, et al., 1998).

However, for the axial (thrust) actuator, solid pole actuator designs are generally used. Solid pole actuators will substantially reduce the magnetic bandwidth. It is difficult to build a conventional axial AMB system with more than 10 to 40 Hz bandwidth (Zhu et al., 2005). Thus, eddy-current effects should be considered in an axial analysis.

Sensors

AMB systems have been built using a number of different noncontact sensor technologies. These include eddy-current probes, variable reluctance probes, capacitance probes, optical probes, and Hall-effect probes. Each has advantages in specific applications, but the general properties that need to be assessed are:

- Bandwidth
- Electrical noise and magnetic field rejection
- Robustness to environment
- Compatibility with canning in harsh service environments
- Reliability
- Cost

As with the amplifier/actuator, sensors have some finite bandwidth. In many cases, a first-order low pass characteristic with a bandwidth on the order of 4 kHz to 15 kHz is appropriate. An example is shown in Figure 11. Most AMB vendors can supply an appropriate transfer function $G_s(s)$ for the sensors used.

$$G_s(s) = \frac{\text{Gain} \cdot \omega_{BW}}{s + \omega_{BW}} [\frac{V}{m}]$$

Figure 11. Example Frequency Response Characteristics for an Eddy-Current Sensor.

Controller

As indicated in Figure 1, the controller inputs are the rotor motion (and, typically, rotor speed and phase). The controller processes these signals through a carefully designed control algorithm to determine the signals to send to the power amplifiers, which in turn apply forces to the shaft. Modern AMB systems nearly always use digital controllers. These controllers are described in terms of their:

- Sampling rate—how often they measure the sensor signals and update the amplifier signals
- Delay—how long it takes to measure the signals coupled with how long the control computations take
- Algorithm—how the sensor information is used to produce the amplifier commands

Together, these properties lead to a physical description of the controller in terms of a set of frequency domain transfer functions within $G_c(s)$:

$$\begin{bmatrix} \dot{V}_{r,1} \\ \vdots \\ \dot{V}_{r,n} \end{bmatrix} = \begin{bmatrix} G_{r,1}(s)_{1,1} & \cdots & G_{r,1}(s)_{1,n} \\ \vdots & \ddots & \vdots \\ G_{r,n}(s)_{n,1} & \cdots & G_{r,n}(s)_{n,n} \end{bmatrix} \begin{bmatrix} V_{r,1} \\ \vdots \\ V_{r,n} \end{bmatrix} = [G_r(s)] \{\dot{V}_r\}$$ (13)

In its most conceptually simple form, the controller implements only the transfer functions on the main diagonal ($G_r(s)_{i,i}$) and the others are zero. In this case, each sensor signal is used only for control of the adjacent actuator axis. This would be called decentralized control (Bleuler, 1984) and would tend to mimic the behavior of mechanical bearings, assuming that the sensor locations are sufficiently close to the electromagnets.

More commonly, however, the off-diagonal elements are also used, leading to coupled control. The widely used tilt and translate approach or center-of-gravity control produce such a controller. In general, this approach provides better overall system performance. The individual transfer functions could also change as a function of speed. This is known as “gain scheduling.” Software used for rotodynamic design audits of AMB systems needs to be capable of accepting these multi-input, multi-output (MIMO) control system models.

The classic proportional-integral-derivative (PID) algorithm is a common place to start when designing an AMB control system. A typical PID frequency response is shown in Figure 12.

$$G_{\text{PID}}(s) = \frac{K_p + K_i}{s + sK_D} \left[\frac{\omega_{BW}^2}{s^2 + 2\zeta\omega_{BW}s + \omega_{BW}^2}\right] [\frac{V}{V}]$$

Figure 12. Example Frequency Response for Proportional, Integral, Derivative Control.
COMPARISON TO MECHANICAL BEARINGS

Because the ultimate function of an AMB system is to support the rotating shaft, it is useful to compare their properties to conventional mechanical bearings as they affect the overall system.

Dynamics

Like a conventional bearing, the AMB system resists forces that try to push a shaft away from the reference (center) position, limits displacement amplitudes when passing through critical speeds, and prevents rotor instability in the presence of destabilizing forces such as aerodynamic cross-coupled stiffness. Due to the strong connection with “stiffness” and “damping” in achieving these goals, it is tempting to reduce the AMB component to an equivalent bearing stiffness and damping. In particular, doing so would permit analysis using conventional rotordynamics codes and thereby avoid investment in new computational machinery. However, this is nearly always a bad idea because the dynamics of the AMB are only very crudely approximated by equivalent stiffness and damping. There are several features of AMBs that challenge this approximation:

1. The sensor and magnet (actuation) locations are almost never identical. For higher flexible modes, this noncollocation can be a significant effect even to the extent that the motions of the sensor and actuator nodes are out of phase.

2. The transfer functions of AMB controllers are often much more complex (and intentionally so) than a control algorithm that would correspond to stiffness and damping. The simplest example is the use of integral terms (discussed below), which lead to very large differences between the AMB dynamics and an equivalent stiffness and damping. In addition, notch filters, high frequency roll-off, and sampling/computational delay alter the bearing dynamics in ways that are not well approximated by stiffness and damping.

3. In many AMB applications, the controller mixes axes so that motion at one end of the rotor may affect the control applied at the other end or there may be mixing between planes to control highly gyroscopic rotors. In either case, modeling the AMB as equivalent stiffness and damping effects at each journal location clearly misses a crucial feature of the bearing’s dynamic behavior.

To illustrate some of these issues, consider a simple mass supported in a single axis AMB, and the same mass supported by a spring and damper. The simplest, most common AMB controller structure is a bandwidth-limited PID controller with some embellishments added to deal with rotor or structure flexible modes. In the simpler case of a rigid mass, a simple bandwidth-limited PID controller is sufficient.

For this example, a stable control system that is tuned to match an assumed stiffness and damping from about 1 Hz to 100 Hz was implemented. Figure 13 compares the dynamic stiffness of the spring-mass-damper system to the mass-controller system. Over the specified bandwidth, the systems are seen to agree very well. However, at low and high frequencies, the characteristics are quite different. The difference at low frequencies is due to the integrator (“I” part of “PID”) and is very valuable in terms of static load rejection: this is very intentional. The difference at high frequencies reflects the fact that the AMB has finite bandwidth and is implemented digitally on a computer with finite sampling rate: not design “features” but concessions to reality.

Figure 13. Magnitude of the AMB Support Stiffness Compared to that of a Comparable Spring/Damper Combination (K = 50 kN/mm and C = 1.0 kN-sec/mm).

Do these discrepancies lead to stability problems? No. The system is perfectly stable and even robustly so. Do these discrepancies lead to dynamic behavior different from that provided by an “equivalent” stiffness and damping? Certainly.

This example demonstrates one of the reasons it is quite important in doing a rotordynamic design audit to implement the actual AMB control system, including the sensor and amplifier/actuator dynamics.

Finally, a common practice in modeling rotors on fluid dynamic bearings is to compute a “synchronously reduced” bearing stiffness and damping, and then use these to model the machine dynamics at a given rotor speed—both for forced response and for stability analysis. This leads to the idea of doing a similar thing with AMBs. That is, to compute:

\[
\begin{align*}
K_{\text{eqv}}(\omega) &= \text{Real}(\text{Total Transfer Function}) \\
C_{\text{eqv}}(\omega) &= \frac{\text{Imag}(\text{Total Transfer Function})}{j\omega}
\end{align*}
\]  

(14)

It is easy to show that this approach will work perfectly to compute forced response and the resulting numbers will be correct. However, it is absolutely not correct to compute stability in this manner. Figure 14 plots the equivalent stiffness and damping as a function of frequency using this approach.

Figure 14. “Synchronously Reduced” Equivalent Stiffness and Damping, Normalized by the Nominal K = 50 kN/mm and C = 1.0 kN-sec/mm.

As Figure 14 clearly shows, the synchronously reduced damping of this AMB - which is known to be stabilizing—is nonetheless negative over wide spans of the frequency range. Computing stability of the system at these frequencies using the synchronously reduced coefficients would certainly predict instability.
In the end, the key notion to carry away from this discussion is that the dynamics of the AMB cannot be usefully simplified by substituting equivalent stiffness and damping coefficients except when doing forced response analysis. As will be discussed below, this will generally mean that stability and sensitivity analysis for a rotor equipped with magnetic bearings will require computational machinery that is not part of the normal set of rotordynamics tools.

**Capacity**

As discussed previously, the nominal static load capacity of a magnetic bearing is controlled by the magnetic flux saturation density of the magnet iron—which produces a maximum effective magnetic pressure—combined with the projected area of the electromagnets. For comparison to fluid film bearings, it is useful to speak in terms of **specific capacity**, which is maximum load capacity divided by journal projected area (length times diameter). For an AMB, this number is somewhere between 0.2 to 0.8 MPa (30 to 120 psi). Importantly, loads that exceed this limit will always lead to contact with the auxiliary bearing. Typically, the auxiliary bearing is designed to carry (briefly!) a load on the order of three times this level. For comparison, a fluid film bearing typically is expected to have a load capacity of about 2 to 4 MPa (300 to 600 psi). This means that the journal of the highest capacity AMB will have about 2.5 times the journal area that is required for a comparable fluid film bearing; more commonly, expect a factor of 4 or 5.

Furthermore, the load capacity of the fluid film bearing is typically the expected operating load for which the bearing has an effectively infinite life. A fluid film bearing can usually carry a much larger load (four or five times larger) for very short periods of time, so it has a high overload capacity. The auxiliary bearing of the AMB system provides a comparable overload capacity, but with a relatively limited life and complicated dynamics in the transition from magnetic support to auxiliary contact. Consequently, AMBs are usually not suitable for applications with frequent and large impact loads.

These rules of thumb and characteristics provide some initial sanity checks when evaluating an AMB system.

**Diagnostics**

Although not a part of a rotordynamic design audit, it is useful to point out that one big advantage of using AMBs at the system level is that they can provide a large amount of information about the machine’s condition. One obvious source of information is the position sensors, which report rotor vibration that is commonly used in health monitoring and diagnosis. Indeed, it is common practice to install eddy-current probes in high capital value machinery for continuous vibration monitoring. One element of an API rotordynamic audit concerns how an unbalanced rotor would be predicted to perform relative to typical sensor vibration limits.

However, AMBs provide a very important second level of information not ordinarily available with mechanical bearings; because the coil currents are known, it is relatively easy to estimate the actual bearing forces. This gives a much more direct evaluation of operating loads such as mass unbalance or aerodynamic side loads. Monitoring of axial AMB bearing forces can be very useful in detecting incipient surge. Since the AMB controller is digital, all of this information is automatically available in a digital form so it is easily accessed and interpreted by a higher level health monitoring system.

**MAJOR AREAS OF CONCERN FOR ROTORDYNAMIC DESIGN AUDIT**

The objective of a rotordynamic design audit is to determine whether the rotor/AMB system can be expected to perform “well” under “expected” operating conditions. The major areas of concern are the same as for a rotor supported by a conventional mechanical bearing, with several additional concerns for the AMB system. Table 1 compares the analysis aspects of AMB supported machines to those supported by conventional bearings.

**Table 1. Rotordynamic Analysis Aspects of Different Bearing Technologies.**

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<tr>
<th>Analysis</th>
<th>Conventional Bearing Supported</th>
<th>AMB Supported</th>
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<tr>
<td>Undamped Critical Speed</td>
<td>Critical speed map</td>
<td>Critical speed map</td>
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<td>Mode shapes near expected</td>
<td>Free-free modes’</td>
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<td>bearing K</td>
<td>frequencies and shapes</td>
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<td>Unbalance Response</td>
<td>Probe Bode plots versus</td>
<td>Dynamic load capacity</td>
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<td>unbalance at various</td>
<td>Probe Bode plots versus</td>
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<td>clearances</td>
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<td>Rub limits with unbalance at</td>
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<td>various clearances</td>
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</table>

**Stability Analysis**

A critical property of any engineered system is stability. Once a rotor/bearing system becomes unstable, prediction of rotor vibration can only be done by complicated nonlinear analyses and, in general, the only thing limiting the rotor motion is contact in the critical clearances. Therefore, instability of the machine is to be avoided. A stability analysis is accomplished by constructing a model of the system under specific operating conditions (rotor speed, pressure, etc.) and computing the eigenvalues of the system. These eigenvalues are complex numbers; the system is unstable if the real part of any eigenvalue is nonnegative.

For stability analysis of an AMB supported system, it is always necessary to consider all modes, both forward and backward over the full range of rotor speeds, because the AMB system presents a potential destabilizing mechanism regardless of rotor speed. For example, unlike conventional bearings, an AMB system can be unstable at 0 rpm. The full frequency range must also be considered. This is in contrast to a fluid film bearing supported system where instability primarily affects synchronous or subsynchronous modes.

One caveat to this relates to eigenvalues associated with high frequency rotor modes. Obviously, the rotor by itself—the free-free condition—is stable and has positive damping (typically, modal damping levels are on the order of 0.1 percent). That is, rotor vibrations once initiated will always die out over time. However, the system model of the rotor with AMB may indicate that some of the higher frequency rotor modes are unstable. This condition requires some care in assessment, as it may not actually be of concern. For instance, if the frequency of the mode is higher than half the sampling rate of the digital controller, then the AMB system is not capable of driving the mode unstable, so the instability is an artifact of modeling assumptions. Furthermore, the amount of energy that an AMB system can add to the rotor at high frequencies is very limited. Consequently, even a truly unstable mode may exhibit only a very small limit cycle—well within critical clearances—if the frequency is high enough.
As a result, it is common to truncate all of the rotor flexible modes except the first two or three when constructing a system model for stability and forced response analysis.

**Stability Sensitivity Analysis**

A rotor/AMB system that is stable as modeled may not be stable in the field. The reason for this is usually that some system parameter is substantially different in the field from the modeled value—a problem referred to as parameter uncertainty. In addition, a system that is stable when installed may become unstable at some later time due to changes in operating conditions. In this case, some system parameter changes and a stability model of the modified system would indicate the instability. This problem is referred to as parameter variability. Both parameter uncertainty and variability are facts of life for the engineer/analyst and should be accounted for in assessing system stability.

The process of considering parameter uncertainty and variability in a stability analysis is called sensitivity analysis. The goal of stability sensitivity analysis is to determine whether or not the system can be expected to be stable for the likely range of parameter values. Thus, for instance, if the system includes a balance piston seal and this seal has a nominal cross coupled stiffness of $K_{xy,0}$, then nominal stability would be required; the system should be stable with this value of $K_{xy,0}$ included. But because this value is probably uncertain (hard to model) and also variable (dependent on gas pressure, gas mole weight, and rotor speed), it is also important that the system remain stable for a wide range of actual $K_{xy}$, which might be as wide as, for instance, $0 < K_{xy} < 2 K_{xy,0}$. Therefore, a sensitivity analysis would examine system stability for selected values of $K_{xy}$ in this range; acceptable sensitivity would imply stability over the whole range. This analysis is an integral part of an API analysis for many machines.

Obviously, this assessment can get complicated when the system includes numerous uncertain parameters that can vary independently. Consequently, one common approach to sensitivity analysis is to carry out a Monte Carlo analysis in which random values of each of the uncertain parameters are chosen within the likely ranges, a stability model is constructed, and stability is assessed. This is done many, many times until either a target level of confidence is reached or an unstable instance is discovered.

**Capacity Analysis**

Because it is well known that load capacity is a critical limiting factor in the design and performance of AMB systems, a crucial element of any system audit is capacity analysis. The conceptual underpinnings of this analysis were presented earlier in this tutorial. While capacity analysis can be done at the preliminary level using the analytic expressions developed in those sections, it will always be better to use finite element analysis or, if available, test data. Figure 15 shows the force generated by the electromagnets described in Figure 4 when the coils of a single quadrant are driven to specific current levels.

This finite element analysis is used to determine the ultimate static load capacity. Looking at Figure 15, it is clear that the force conforms to the quadratic dependence on current predicted by Equation (5) out to a current of about 27 amps and then begins to diverge, producing less force than the simple analysis predicts. This reduction in force is due to progressive saturation of the magnet iron at high magnetic flux densities. Deciding when to call the magnet “saturated” is not completely obvious; to produce very high forces will require very high currents and this becomes thermally prohibitive. Generally, some rule of thumb is followed; when the actual force is less than some fixed percentage of the quadratic (analytic) prediction, the magnet is said to be saturated. For the present example, this ratio is 0.5 at 50 amps and the vendor has selected this point as “saturation.” It would be unusual to accept a significantly lower ratio as the saturation point.

Once the quadrant capacity is determined, this number is then combined with the known voltage and current limits of the power amplifiers, the nominal electromagnet air gap, and the vendor specified bias current according to Equation (12) to determine the dynamic capacity of the magnet array as a function of frequency, as depicted in Figure 9.

**Forced Response Analysis**

Perhaps the simplest element of a rotor/AMB system audit is forced response analysis. Once the system model is constructed for stability/sensitivity analysis, forced response analysis is a relatively simple process. Generally the input force is steady-state response to unbalance, but other inputs could be examined.

When the objective of this study is to evaluate the worst case response, a technique based on the maximum singular value of a particular set of system matrices has been shown to be very effective (Cloud, et al., 2002). Although not in common use, the technique referenced has the advantage of guaranteeing that the response to the worst case distribution has been found (subject to some fairly reasonable assumptions).

**Transient Response Analysis**

For certain specific situations, a transient response analysis might also be performed. Examples of such situations include blade loss, earthquake loading, and maneuver loads. Transient response analysis has also been used to study the behavior of a rotor drop onto the auxiliary bearings. This point will be discussed in a later section.

Otherwise, transient response calculations are not typically part of the standard rotordynamics audit, and will not be discussed. However, a couple of important points that should be noted if a transient response analysis needs to be performed for an AMB supported system are:

- The analysis needs to consider the nonlinear effects of the gap between the actuator pole face and the rotor.
- The analysis may need to consider the nonlinear saturation effects in the magnetic material.
- The analysis may need to consider the maximum voltage limit for the amplifiers.
- The analysis may need to consider all of the effects related to the auxiliary bearings.

**Auxiliary Bearings**

All practical AMB systems must include a second bearing system. This auxiliary (or backup) bearing system is required to support the rotor and any rotor loads (aerodynamic, magnetic, gearing, fluid, etc.) under one or more of the following conditions:

- AMB system powered down for storage, transport and/or machine not operating

![Figure 15. Quadrant Force Versus Current for the Electromagnets Depicted in Figure 4 Using Finite Element Analysis.](image-url)
• Short-term AMB overload from forces that exceed the design capacity of the bearings. These could include surge, liquid slugs, seismic forces, rotating stall, blade loss, etc.

• Complete AMB system or subsystem failure or damage

In discussing auxiliary bearings, it is important to note that an AMB can generally be designed to support the rotor during any transient events that can be predicted or bounded during design (surge is a common example). It is also important to note that there is a substantial amount of field experience that suggests that total AMB failures are extremely rare. Most customers, however, do require an auxiliary bearing system capable of supporting the rotor during normal operation.

The presence of this second, auxiliary bearing support system means that a complete rotordynamic evaluation of an AMB supported machine must include an additional evaluation that considers the auxiliary bearing system. Normally, such an evaluation assumes that the AMB generates no forces and the rotor is supported entirely by the auxiliary bearing. Operation in a load-sharing mode during short-term overload has also been proposed, and does occur, but is beyond the scope of this tutorial.

As shown in Figure 16, a typical industrial auxiliary bearing consists of a mechanical bearing with shaft-bearing air gap and a mount system. The mount system includes a carefully selected compliant element with some means to dissipate vibrational energy (i.e., damping). The allowable range of motion of the bearing is generally restricted by an outer hard stop. This general arrangement is also used for a dual purpose radial/thrust thrust bearing as shown in Figure 17. Arrangements with separate thrust bearings are also used. There are also arrangements with an inside-out arrangement, where a hollow shaft segment makes contact with the outer surface of the auxiliary bearing, as well as arrangements where the bearing is attached to the rotor rather than the stator.

If there is a drop event, the basic sequence for a horizontal machine is as shown in Figure 18. At the start of the drop, the spinning rotor is initially in a free-free condition, falling in a radial direction under the influence of gravity, operating loads, any rotor unbalance, and the rapidly decaying AMB magnetic field. Once the rotor falls through the auxiliary bearing clearance gap, it makes contact with the auxiliary bearing. The force of this impact is taken up primarily by the compliance and damping of the auxiliary bearing mount system. For heavy, highly loaded, or out-of-balance rotors, the compliant mount may move far enough to hit the hard stop. Even for a well-designed auxiliary bearing system, the rotor will probably bounce around for a brief period before settling down to a fairly stable, fairly small orbit in the bottom of the auxiliary bearing. The rotor motion then remains in this small orbit during an emergency, rapid machine shutdown.

The sequence for a vertical machine would be similar, with the major difference that the gravity induced motion is axial, and a small orbit may not be obtained. Therefore, to limit the dynamic loads on the backup bearings, it is important that the whirl frequency is much lower than synchronous. In the case of a momentary overload, the AMB system remains active, which gives rise to additional forces from the AMB system, which put this case outside the scope of this tutorial.

The rotor-bearing gap is generally about 50 percent of the air gap between the AMB stator and the rotor. This gap balances the need for noncontact operation when the rotor is levitated, and the need to prevent contact between the rotor and stator at the AMB, seals, impellers, etc., when operating on the auxiliary bearing.

![Figure 16. Typical Radial Auxiliary Bearing Arrangement.](image)

![Figure 17. Typical Dual Radial/Thrust Bearing Arrangement.](image)

![Figure 18. Typical Stable Drop Sequence.](image)
speeds and loads that are generally well in excess of standard rotating machine design practice for these bearing types. The compliant mounting systems have also received attention. Each bearing developer has one or more proven mount configurations. The stiffness and damping of the mount system are tuned for the specific application or class of applications.

ANALYSIS TOOLS

Model Structure Issues

Modeling of the linear behavior of the AMB system is most readily accomplished using a state-space formulation of the equations of motion:

\[
\frac{d}{dt} \begin{bmatrix} x_B \\ y \end{bmatrix} = \begin{bmatrix} A_B(\Omega) & B_B(\Omega) \\ C_B(\Omega) \end{bmatrix} \begin{bmatrix} x_B \\ y \end{bmatrix} + \begin{bmatrix} f_{x_{\text{new}}} \\ f_{y_{\text{new}}} \end{bmatrix}
\]

(15)

Note that some of the terms may be operating speed dependent, thus allowing for gain scheduling and possibly synchronous disturbance rejection.

This first order form model is relatively straightforward to combine with a standard rotordynamics model when the rotordynamics model is also expressed in first order form. Detailed discussions of how to obtain a rotordynamic model in first order form are provided in most of the standard rotordynamics texts (for example, Childs, 1993; Chen and Gunter, 2005). Briefly though, the rotordynamic model can be expressed in first order form as:

\[
\frac{d}{dt} \begin{bmatrix} x_R \\ y \end{bmatrix} = \begin{bmatrix} A_R & B_R C_R \\ B_C \end{bmatrix} \begin{bmatrix} x_R \\ y \end{bmatrix} + \begin{bmatrix} B_R \\ 0 \end{bmatrix} f_s
\]

(16)

in which:

\[
A_R(\Omega) = \begin{bmatrix} 0 & 0 & I & 0 \\ 0 & 0 & 0 & I \\ -M^{-1}K & 0 & -M^{-1}C - \Omega M^{-1}G \\ 0 & -M^{-1}K & \Omega M^{-1}G & -M^{-1}C \end{bmatrix}
\]

(17)

\[
B_R(\Omega) = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ M^{-1} & 0 \\ 0 & M^{-1} \end{bmatrix}
\]

\[
C_R(\Omega) = \begin{bmatrix} I & 0 & 0 & 0 \\ 0 & I & 0 & 0 \end{bmatrix}
\]

This is combined with the control system using rotor displacements as control system inputs, and forces to be applied to the rotor as outputs as shown in Equation (18). Significant care is required to ensure that the proper degrees of freedom are used in each case. Note that this model fully incorporates the control system, sensor, amplifier, and actuator dynamics. Not all rotordynamics analysis packages are capable of accepting a state space model. An equivalent formulation using a matrix of transfer functions is also possible. As noted above, modeling the AMB system as a set of speed dependent stiffness and damping coefficients is not adequate for stability calculations or the other AMB specific calculations presented below.

\[
\frac{d}{dt} \begin{bmatrix} x_R \\ y \end{bmatrix} = \begin{bmatrix} A_R & B_R C_R \\ B_C \end{bmatrix} \begin{bmatrix} x_R \\ y \end{bmatrix} + \begin{bmatrix} B_R \\ 0 \end{bmatrix} f_s
\]

(18)

Analysis

With the combined system in first-order form, the eigenvalue/eigenvector (whirlspeed/stability) analysis procedure is essentially the same as with a conventional bearing system. Some care is required in the interpretation of the eigenvectors (mode shapes) to separate rotor degrees of freedom from control system degrees of freedom.

The AMB specific analyses, including sensitivity functions, open loop transfer functions, and closed loop transfer functions are also conveniently calculated with the equations in first order form. Procedures for doing these calculations can be found in standard multivariable control system texts (Nise, 2008, for example).

Many rotordynamics codes calculate unbalance (forced) response using a second order system based on the mass, damping, gyroscopic, and stiffness matrices. Unfortunately, the first-order form cannot in general be reduced to a simple set of second order matrices. Thus, for AMB system analysis, it is desirable to use the first order form, or an equivalent transfer matrix form, directly. The input is generally the set of unbalances forces. Some care must be taken to properly apply these forces to the correct degrees of freedom with the correct phasing.

As noted previously, a scaled maximum singular value analysis provides a powerful method to ensure that the response to the worst possible unbalance condition at each speed has been considered (Cloud, et al., 2002).

API/ISO SPECIFICATIONS

Although there are a significant number of successful AMB supported machines in commercial service, industry standards are just starting to consider AMB systems. The familiar API specifications (610, 612, 617, 673, etc.) were developed using operational experience from rolling element and fluid-film bearing supported machinery. As discussed previously, AMB systems have some characteristics that are quite different from those conventional bearing systems. Thus, it is reasonable to expect that some of the requirements should be different. Recently emerging AMB-specific standards from ISO and API, which do take into account some of the unique characteristics of these bearings, are discussed below.

In general, the overall goal of machinery standards/specifications is to provide design requirements that result in machines with a relatively low risk of machine damage related to rotorbearing system dynamics. In this assessment, there are generally two concerns:

1. Motion due to forced response excitation (unbalance, misalignment, torsional, etc.).

2. Motion due to dynamic instability (bearing whirl, seal forces, etc.).

This section will discuss how API and ISO attempt to achieve these goals and what additional assessment should be applied to AMB machinery.
API Specifications—Introduction

The API approach to rotordynamics is to include a very specific set of analysis and test stand requirements that API compliant machinery must meet in each of several machinery specific standards. These standards for machines where magnetic bearings are most likely to be employed are shown in Table 2. The rotordynamic content of these standards, in turn, is generally based on a set of standard paragraphs. These standard paragraphs and a great deal of background information are presented in API 684 (API 684, 2005), which is a rotordynamics tutorial document.

Table 2. API Specifications.

<table>
<thead>
<tr>
<th>Standard</th>
<th>Machine Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>610</td>
<td>Centrifugal Pumps</td>
</tr>
<tr>
<td>612 (ISO 10437)</td>
<td>Steam Turbines</td>
</tr>
<tr>
<td>617</td>
<td>Axial and Centrifugal Compressors</td>
</tr>
<tr>
<td>673</td>
<td>Centrifugal Fans</td>
</tr>
</tbody>
</table>

The sections below will discuss selected API standard paragraphs for lateral rotordynamics as presented in API 684, Second Edition (API 684, 2005), and the AMB specific material in Appendix 4F of API 617, Seventh Edition (API 617, 2002).

For machines that must meet API specifications, it should be noted that there are some cases where an AMB machine design will inherently require that the purchaser agree to an exception to the API standards. For example, the API 612 steam turbine specifications specifically state that the bearings shall be hydrodynamic bearings (API 612, 2005). Likewise, API 673 (API 673, 2002), centrifugal fans, seems to suggest that the bearings must be either antifriction (for smaller machines only) or hydrodynamic bearings.

For more detail than is provided below, the reader is encouraged to consult the rotordynamics tutorial, API 684 (2005).

API—Damped Lateral Unbalance Response Analysis

In the API standards, forced response (generally due to unbalance) is addressed through a requirement that a damped unbalance response analysis must be performed. The major issues covered include (references are to standard paragraph numbers in the Second Edition of API 684, 2005):

- Discussion of modeling issues that might be important (SP6.8.1)
- Unbalance level for use in analysis and typical distributions to use (SP6.8.2.7-SP6.8.2.7.2)
- Separation margin requirements between operating speeds and critical speed peaks in the unbalance response (SP6.8.2.10)
- Maximum allowable vibration probe amplitudes (SP6.8.2.11)
- Maximum allowable motion at close clearance locations (SP6.8.212)
- An approach to use for experimental validation of the analysis (SP6.8.3)

The approach presented in these paragraphs requires that the analyses demonstrate that any critical speeds are adequately separated from the operating speed range, and that the predicted vibration response for a standard level of unbalance is acceptable. The vibration acceptance requirements are based on both predicted vibration probe measurements and checks at close clearance locations such as seals.

An important difference between common AMB practice and the API requirements is the probe vibration limit. The standard API probe vibration limit (peak to peak) is given as:

\[ \text{Amplitude(mils)} \leq \sqrt{\frac{12000}{N}} \]  

or 0.001 inch (25.4 microns), whichever is smaller. This results in amplitudes of 0.001 inch or less. These levels are required for long service life with rolling element and hydrodynamic bearings. For AMBs, many would argue that a higher limit is appropriate due to the larger clearances and control system features that can allow the rotor to spin about the mass center without large transmitted vibration forces.

Another issue that needs some consideration for the application of the API vibration level criteria with AMB supported machinery is the requirement to consider “minimum” and “maximum” bearing clearances. For fluid-film bearings, the actual bearing clearance is a key uncertainty with regard to system rotordynamic characteristics. Thus, an API unbalance response analysis is performed at both extremes. For an AMB system, there is no clearly analogous parameter to clearance that exerts such a strong effect. Thus, the example analysis presented below will be performed for the nominal system only. Robustness with regard to system parameter variations will be addressed in a separate set of calculations.

API Lateral Stability

The latest revisions of API 617 (2002) and API 684 (2005) have introduced an approach for evaluation of stability robustness with respect to destabilizing aerodynamic effects. This evaluation is primarily directed at higher pressure compressors, although the basic method can be used in a variety of machines. The approach is divided into an initial conservative screening analysis (“Level I”), and a more detailed analysis for machines that do not meet the Level I screening criteria.

As presented, the Level I screening analysis is intended to be a relatively quick analysis that separates machines requiring a detailed analysis to determine stability, from those that do not. The Level I analysis is a parametric study of the effect of increasing levels of assumed cross-coupled stiffness (the standard model for destabilizing aerodynamic effects). The assumed cross-coupled stiffness is based on a semiempirical estimate of the expected level based on machine parameters. The criteria include a minimum stability level and a minimum stability margin with the predicted level of cross-coupling applied.

For machines that do not meet any of the criteria (many, if not all, high pressure compressors), a Level II analysis is to be performed. This analysis essentially boils down to the use of the vendor/analyst’s state-of-the-art approach. Again, a minimum stability level is specified. However, a stability margin is not specified.

For details of the API Level I procedure, the reader should consult API 684 (2005) or API 617 (2002). A recent paper by Nicholas and Kocur (2005) may also be helpful.

Three issues that need to be addressed are the min/max clearance requirement, the focus on the stability of the first forward mode, and the minimum logarithmic decrement (log dec) of 0.1. The min/max clearance issue was discussed previously. As with unbalance response, the example analysis will consider the nominal system only.

The focus on the first forward mode needs to be reconsidered. For compressors supported on conventional fluid-film bearings, the first forward mode is the mode that will be driven unstable by the added aerodynamic cross-coupled stiffness (or one of the first two modes in the case of double overhung compressors). This is generally a subsynchronous mode. However, as pointed out by Li, et al. (2006), there is no reason for this to be true for an AMB system. Indeed, there are good reasons to expect that the unstable mode will not be the first forward mode for many AMB machines.

Thus, for an AMB system, it is advisable to consider the stability of all modes when determining which mode is driven unstable by
the added cross-coupled stiffness. This, however, complicates the generation of the sensitivity function plot as envisioned by the Level I analysis.

**API 617, Seventh Edition, Appendix 4E: Auxiliary Bearings**

This appendix makes a number of recommendations specifically for AMB supported machinery. These include a looser vibration tolerance and a discussion of some issues related to the auxiliary bearing.

The looser vibration limit is based on the idea that the traditional amplitude limit is appropriate for relatively stiff, small clearance rolling element and fluid-film bearings, but that it generally does not make as much sense for AMB systems with their much larger clearances. In these machines, it is common practice to allow larger responses.

API 617 (2002), Appendix 4E, explicitly acknowledges this fact when discussing the test stand vibration acceptance criterion: “The maximum allowable rotor movement relative to the center of the auxiliary bearing for any given axis of levitation is 0.3 times the minimum radial clearance in the auxiliary bearing in that axis.”

The standard indicates that the motion is the total motion due to all forces, including unbalance, aerodynamic loading, mechanical distortions, etc. The standard indicates that this criterion replaces all of the other criteria specified for fluid-film bearing supported machinery. For field acceptance tests, the standard relaxes this limit to 0.4 times the clearance as an acceptance criterion. This specification is very similar to the ISO specification.

The comments related to auxiliary bearings include (API 617, 2002):

- “The auxiliary bearing system shall be provided with a damping mechanism, if required, to prevent destructive whirl during coastdown. The damping system shall be provided by the vendor and shall be a proven design.”
- “The radial and auxiliary stiffness of the bearing assembly must be sufficient to withstand a sudden shock load equal to the full capacity of the magnetic bearing (plus kinetic energy), without allowing contact between any portion of the rotor and stator.”
- “The auxiliary bearing system shall be designed to survive at least two delevitations from maximum continuous speed to zero speed with the normal aerodynamic braking and nominal process induced thrust load (which should not be larger than 75 percent of the thrust bearing rated capacity).”
- In several places, the standard states that the auxiliary bearing system is considered to be a consumable backup system only. Thus, damage to the auxiliary bearing system is acceptable, so long as it prevents rotor-stator contact for at least a specified number of delevitation and overload events.

This standard proposes a variety of other useful requirements that could be included in a review or design specification. These include various electrical considerations, required load capacity, and various mechanical design details. It also gives an approach to dealing with the need to validate the performance of the auxiliary bearing system with reduced risk to the machine relative to a full drop and coastdown sequence.

**ISO 14839/10814**

Recognizing the need to develop a standard that addresses the unique characteristics of AMBs, an ISO working group was established in 1996. This working group has published a standard titled “Mechanical vibration—Vibration of rotating machinery equipped with active magnetic bearings” (ISO 14839, 2004, 2006). This standard currently includes three parts:

- ISO 14839-1, Vocabulary
- ISO 14839-2, Evaluation of Vibration
- ISO 14839-3, Evaluation of Stability Margin

This standard addresses unbalance response (amplification factors and separation margins) by incorporating a reference to ISO 10814 (1996) “Mechanical vibration—Susceptibility and Sensitivity of Machines to Unbalance.”

These standards cover many of the same issues addressed by the API standards. The primary difference is higher vibration limits and presentation of a testing method to evaluate stability of the control system that is suitable for field measurements. The ISO standards also differ in that they are recommendations rather than acceptance criteria. The discussion below will focus on ISO 14839 Part 2 (ISO 14839, 2004), 14839 Part 3 (ISO 14839, 2006), and ISO 10814 (ISO 10814, 1996).

It should be noted that ISO 14839 (2004) applies only to machinery larger than 15 kW.

**ISO Unbalance Response Analysis**

Although the standard is structured a bit differently, and more oriented toward a discussion of measured machine performance requirements, the combination of ISO 10814 (1996) and 14839 Part 2 (2004) provides a set of evaluation guidelines that can be applied to analysis results.

ISO 14839 Part 2 (2004) provides an extensive discussion of the issues related to measured vibration in an AMB machine (it is very similar to API 684 [2005] in this regard). Unlike the API standards, however, the results are stated as recommendations for performance at “agreed conditions,” rather than the more explicit requirements of the API specs. The standard also explicitly states that the evaluation is “not intended to serve as an acceptance specification on either a test stand or the commissioning installation. The acceptance specifications shall be subject to agreement between the machine manufacturer and customer.”

The standard indicates that the appropriate metric of shaft motion is total shaft displacement relative to the nominal machine centerline at the displacement sensor location. This displacement is the vector sum of both the steady-state vibration and any static motion due to static sag, thermal effects, etc., as shown in Figure 19. Both radial and axial displacements need to be considered (separately).

Figure 19. ISO 14839 Displacement.

The standard indicates that the appropriate evaluation criterion for total shaft displacement is a specified fraction of the minimum radial or axial clearance at locations such as the auxiliary bearings, seals, impellers, etc. It is suggested that the typical design practice is for the auxiliary bearing clearance to be the controlling clearance.
The evaluation of the vibration is given in terms of four zones related to the minimum clearance (C_{min}) at each calculation point as shown in Table 3.

<table>
<thead>
<tr>
<th>Zone</th>
<th>Boundaries</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.3 x C_{min}</td>
<td>New machines</td>
</tr>
<tr>
<td>B</td>
<td>0.4 x C_{min}</td>
<td>Acceptable for long term operation</td>
</tr>
<tr>
<td>C</td>
<td>0.5 x C_{min}</td>
<td>Excessively high vibration</td>
</tr>
<tr>
<td>D</td>
<td>Amplitude</td>
<td>Machine damage expected</td>
</tr>
</tbody>
</table>

ISO 10814 (1996) discusses separation margin and amplification factors. This specification develops a set of plots and formulas that relate a vibration amplification factor to the damping of a resonance (critical speed), and how far it is from an operating speed. Like API, the net result is that lightly damped responses are required to be farther away from the operating speed range than heavily damped responses. As with ISO 14839 (2004, 2006), this specification again states that, “The proposed sensitivity values are not intended to serve as acceptance specifications for any machine group but rather to give indications of how to avoid gross deficiencies as well as exaggerated or unattainable requirements.”

ISO 10814 (1996) has two significant differences relative to the API specifications. The first is an explicit acknowledgement that the evaluation is based on a single mass model: “The sensitivity values should be used on simple machine systems, preferably with rotors having only one resonant speed over the entire service speed range. They may also be used for machines that have more resonant speeds in the service speed range if the resonant speeds are widely separated (e.g. more than 20 percent spaced)” (ISO 10814, 1996).

The second is that the evaluation is stated in terms of three machine types and five sensitivity ranges. The three machine types range from machines in a very clean service (Type I) that are unlikely to experience degradation of the balance state, to machines in a fouling or abrasive/corrosive service that would be expected to have significant balance shifts (type III). The five sensitivity ranges are described as ranging from very low sensitivity to unbalance (A) to overly sensitive for reliable machinery (F).

ISO 10814 (1996) region “A” for a “type II” machine, shown in Figure 20, results in recommendations that are similar to the API requirements.

The method ensures that the AMB system is robust against changes in the gain and phase of the sensor and amplifiers/actuators. Unlike the unbalance standard, the stability margin standard indicates that an AMB machine must pass the proposed testing criteria prior to shipment, with field tests optional. While the method proposed is intended for testing purposes, it is also suitable for analytical modeling.

The standard provides a detailed description of the approach. Essentially, the method requires an axis-by-axis injection of an excitation signal into the control loop with the rotor levitated, then the measurement of the ratio between the excitation and the response at a specified location within the control loop. This measurement is made over a range of frequencies. From the audit perspective, a straightforward calculation provides the equivalent information.

A subtle but important point with regard to this standard is the measurement point. The standard requires the use of transfer functions measured at the four lateral sensors and the axial sensor (Figure 21). Measurements made after sensor outputs have been summed to obtain tilt/translate coordinates, for example, are not correct. Two of the examples provided with the standard do not show the measurements being made in the correct location. The zones specified in the standard are only valid for these five specific measurements. Measurements of combined sensor outputs amount to a change in the coordinate basis, which can significantly change the magnitudes of the measured sensitivity functions.

Appendix D of the specification discusses several issues related to analytical calculations. One key point is that the calculation is not valid if the AMB control system transfer function is reduced to equivalent synchronous stiffness and damping. The model used must also account for noncollocation of the sensors and actuators.

The ratio of the specified signals as a function of frequency is the “sensitivity function,” S. The specification summarizes the appropriate results from multivariable control theory to show how the magnitude of this ratio is a reasonable estimate of the stability margin of the AMB/rotor system. A low value of the sensitivity function implies a robust system. A high value implies a system that has very little stability margin. The standard indicates that the sensitivity function should be measured or evaluated with excitation frequencies up to three times the rated speed or 2 kHz, whichever is larger, but no more than half of the sampling frequency for a digitally controlled system.

The maximum value of the sensitivity function (S_{max}) over all axes and all frequencies is evaluated relative to four zones as shown in Table 4.

<table>
<thead>
<tr>
<th>Zone</th>
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<th>Description</th>
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<tr>
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</tr>
<tr>
<td>C</td>
<td>4 ≤ S_{max} &lt; 5</td>
<td>Excessively high vibration</td>
</tr>
<tr>
<td>D</td>
<td>S_{max} ≥ 5</td>
<td>Machine damage expected</td>
</tr>
</tbody>
</table>
ISO 14839 Stability Margin Issues

The ISO 14839 (2004, 2006) stability margin evaluation using sensitivity functions provides a very good measure of the margin of the AMB/rotor system relative to a number of uncertainties within the AMB system. A machine with a large peak value of the sensitivity function (zone C or D) will almost certainly have problems. However, the presence of a low peak value does not imply that the machine will be robust with respect to other changes in the system.

A paper by Li, et al. (2006), discusses the issues in detail. The basic problem is that a low peak value of the sensitivity function does not imply system robustness with respect to other common issues such as aerodynamic cross coupling or seemingly minor changes in rotor natural frequencies as might result from an overhaul. Certain controllers might also have a very high sensitivity to strong gyroscopic effects, without being flagged by the sensitivity function approach.

Thus, the ISO 14839 (2004, 2006) specification provides a necessary stability margin requirement, but it is not sufficient in many machines of practical industrial interest. Li and his coauthors (Li, et al., 2006) do recommend using the ISO spec, but with an awareness that other issues should also be examined.

API/ISO Discussion

From this very brief review of the two sets of standards, it seems that neither quite provide a complete basis for the evaluation of an AMB supported machine in the way that the API specification does for traditional machine designs with fluid-film or rolling element bearings.

Based on API 617-4F (2002) and the ISO spec, one reasonable approach to auditing many AMB supported systems might be to eliminate the API probe amplitude limits, but to keep the close clearance requirements (SP6.8.2.12). Thus, the maximum peak-to-peak response at any close clearance location, over the range of zero to trip speed, should not exceed 75 percent of the minimum design diametric clearance at all locations. This requirement falls between zone A and B of the ISO vibration limits, which also match the specifications of API 617-4F (2002).

The ISO-14839 (2004, 2006) sensitivity function analysis should be performed to ensure that the basic AMB/rotor system stability has a reasonable margin. Likewise, for machines with significant aerodynamic excitation sources, the API Level I screening analysis is a useful approach. For most machines, additional analyses are recommended to ensure that the machine will be robust to potential changes in natural frequencies, gyroscopic effects, etc. These evaluations might include a Monte Carlo analysis to consider a wide range of possible uncertainties.

In machines that use some form of synchronous unbalance force cancellation (autobalance), the analysis procedure needs to be modified to reflect the fact that the rotor is allowed to rotate about the mass center, rather than the geometric center. The details of the implementation vary by vendor, but it should be possible to at least approximate the algorithm in the analysis. The inclusion of an autobalance algorithm in the analysis should reflect the actual machine operation. If an autobalance algorithm can be activated, then an additional analysis with the autobalance algorithm included needs to be performed. If it is a normally controller function over some or all speed ranges, then the analyses should account for this fact.

It should be noted that most of the API standards address lateral and torsional dynamics only. For an AMB system, axial dynamics also need to be considered.

AUXILIARY BEARINGS

From an overall design audit perspective, there are three basic issues related to the auxiliary bearing system. These are:

- Does the bearing prevent other rotor-stator contact during all phases of operation?
- Will the design have adequate standby life (corrosion, vibration, heat, etc.)?
- Will the design have adequate operational life?

From the perspective of a rotodynamics audit, the main issue of concern is the first—would the bearing be expected to prevent other rotor-stator contact during all phases of operation? Note that the rotodynamic characteristics do have implications for the operational life question. In particular, if the rotodynamic characteristics are such that large orbits are expected, then auxiliary bearing life will probably be reduced because of the large forces that tend to accompany these large orbits. Ways to connect the rotodynamic performance with life prediction are a current area of active research.

Auxiliary Bearing Dynamics

A machine operating on auxiliary bearings has exactly the same set of rotodynamic concerns as a machine operating on any bearing system. The stability and unbalance response will be controlled by the machine’s combination of rotor mass, rotor stiffness, unbalance distribution, operating speed, bearing characteristics, seal forces, etc. The rotor can be readily driven to very large response amplitudes by excessive unbalance. The relatively large clearance between the rotor and auxiliary bearing means that the response amplitudes can be quite large. These large response amplitudes, could in turn, lead to bearing damage, and rubs at other close clearance regions such as seals.

One major difference between auxiliary bearings and conventional bearings is the potential for bearing excited backward whirl. Due to the clearance space, both rolling element and solid lubricated bearings provide a tangential force at the rotor surface that tries to force the rotor to roll around the clearance space in a direction opposite to rotation as shown in Figure 22. If excited, a full clearance space backward whirl can lead to enormous radial forces that are likely to damage the backup bearing, and have a high potential for damaging rotor-stator contact in other rotor locations. Interestingly though, most of the anecdotal evidence is that large amplitude auxiliary bearing whirl events have generally exhibited forward whirl at frequencies considerably below the rotor spin speed.

![Figure 22. Backward Whirl Due to Friction Interface.](attachment:image)

The other major difference is that the auxiliary bearing system is very nonlinear. This means, for example, that the response does not scale linearly with unbalance magnitude. Experiments have shown that there can be thresholds where a small increase in unbalance magnitude leads to a large change in the response amplitudes. It also means that the largest response frequency may not be synchronous.

Rotor operation on the auxiliary bearing results in a new and different set of rotodynamic characteristics than for the AMB system. Not only do the auxiliary bearings have quite different dynamic characteristics, but they are also generally located in...
slightly different axial locations than the AMB actuator centerlines. Thus, most of the analyses performed to evaluate the AMB system do not apply.

However, simply performing a second, relatively standard rotordynamics analysis for the new bearing system is not as practical as it might at first seem. As noted above, the auxiliary bearing characteristics are very nonlinear. When the rotor is out of contact (falling, for example), the auxiliary bearing has no effect. When the rotor is in contact with small orbits, the mount system controls the motion. For larger orbits, most mount systems hit a hard stop, even if the rotor remains in contact with the auxiliary bearing. This stop leads to a sudden increase in stiffness and decrease in damping. For intermediate size orbits, where the rotor is bouncing, there is no single “stiffness,” since the rotor is alternatively in contact, and out of contact. Even during the desired small amplitude orbits, the overall motion also includes a “pendulum” like motion where the rotor rocks back and forth in the bottom of the auxiliary bearing. The typical linear rotordynamics analysis does not consider this motion.

Most designers currently appear to use experience, insight from basic linear analyses, and simplified time transient dynamic simulations for the auxiliary bearing design process.

As will be discussed below, although time transient simulations seem very appealing, they cannot prove system stability in the general case (unlike the more familiar linear rotordynamic stability analysis). Each analytical run only indicates what happens for a particular combination of parameters, balance state, operating speed/decal speed, etc. There is always the possibility that there is another combination that will perform differently. For the end-user who is familiar with linear system analysis, which provide reliable assessments of general machine behavior, this is likely to be disturbing. Time transient analysis run times also tend to be quite long (order of hours rather than minutes as seen for critical speed, unbalance, and stability analyses).

There are several techniques that have been proposed that do provide a relatively general answer, subject to certain conditions. For example, under a reasonable set of conditions, Maslen and Barrett (1995) have shown that it is possible to evaluate a machine’s global propensity for backward whirl. However, these techniques do not appear to be in wide use. They also are not possible in a standard, off-the-shelf rotordynamics analysis package. As more and more AMB systems are applied, it is hoped that these and similar techniques will become available to the general community. However, since they are not presently in widespread use, the evaluation procedures below do not discuss these advanced tools.

**Auxiliary Bearing Rotordynamic Evaluation**

Despite these challenges, the auxiliary bearing rotordynamics audit analysis needs to address the same two fundamental questions that any rotordynamic analysis attempts to address:

- Will the machine operate with small amplitude motions about some operating point (stability)?
- How will the machine respond to realistic levels of mass unbalance (unbalance response)?

As noted above, the standard tools of rotordynamic analysis (eigenvalue analysis, unbalance analysis), are of limited value. This considerably complicates the situation, and forces end-users to rely heavily on the AMB supplier’s experience base.

From the audit perspective, however, there is a fair amount of published information that does provide some guidance. For the purposes of this tutorial, the authors have grouped this guidance into two sets of evaluations—an initial checklist, and suggestions for time transient analysis if budget and application suggest that a larger research effort is appropriate. Note that this approach could change in the future as new techniques for analysis of auxiliary bearing dynamics are developed.

**Evaluation for a Typical Audit**

This section presents some general guidance and preliminary checks that can be performed using most commercial rotordynamics software. Much of this is taken from a paper by Kirk, et al. (1997), with some additional ideas from Reitsma (2002), and Hawkins, et al. (2006, 2007). Note that this evaluation is not go/no-go. It is only intended to provide some guidance about the level of application risk. With careful design and operation limits, successful auxiliary bearing systems that do not meet one or more of these criteria have been developed.

Note also that the analysis should take account of seal forces in high-pressure machines. Converging and diverging seals, for example, can have significant direct stiffness terms. As with almost all aspects of an audit for a high-pressure machine, the analyst needs to have enough familiarity with these machines to be able to exercise some engineering judgment. Likewise, some bearings and motors give rise to a large negative direct stiffness which would need to be considered.

The guidelines and checks are as follows:

- Heavier machines, faster machines, and long coastdowns would all be expected to be a more severe challenge for the auxiliary bearing system. For rolling element auxiliary bearings, it would be useful to confirm that the auxiliary bearing DN (inner diameter in mm times speed in rpm) is within the AMB vendor’s experience base. For solid lubricated bearings, a similar question could be asked with regard to PV (unit pressure times sliding velocity). Likewise with coastdown time, or coastdowns with very high axial load. Applications that exceed the vendor’s experience base in any of these regards have higher risk.

- For most industrial scale machinery, it is essential that the auxiliary bearing be mounted in a compliant, damped mount system. Pumps are a possible exception, due to damping arising from seals. If this mount is not included, or the total travel allowed is less than the deflection under static load, then the application has a very high level of risk.

- The rotor balancing/assembly procedure should be reviewed. Some have suggested that component/stack balancing to minimize distributed unbalances is highly desirable. It is advisable to be careful that any field trim balance weights will not excite the free-free bending mode during a drop event.

- Machines in fouling service that do not maintain a consistent balance state have a higher level of risk. Questions regarding the allowable unbalance, and how the balance state will be monitored would be advisable.

- If load sharing between the AMB system and the auxiliary bearing is proposed, questions regarding AMB dynamic force capability versus expected auxiliary bearing dynamics loads are appropriate. The issue here is whether the AMB has enough force capacity to capture a whirling rotor. There have been reports of test stand cases where recapture was not possible.

- The lowest rotor free-free mode can be readily calculated. Operation above the free-free mode has higher risk.

- The undamped critical speed map for the machine for bearings located at the auxiliary bearing centerline can readily be calculated. The intersections between the expected range of effective auxiliary bearing stiffness and the critical speeds can be examined. If the intersections below operating speed occur in the “pinned” region, then the support damping is not likely to be effective.

- A standard, linear eigenvalue (damped whirlspeed) analysis can be performed under the assumption that the rotor remains in contact with the auxiliary bearing system and has very small orbits. These assumptions make it reasonable to replace the nonlinear auxiliary bearing system with a linear bearing based on the estimated mount stiffness and damping (and possibly mass if the
combined bearing and mount moving mass are heavy). The results of this analysis can be examined for very lightly damped subsynchronous modes and modeshapes that indicate that the effective mount damping is not adequate. Note, however, that this analysis cannot in general address whether the rotor orbits will remain small. For example, the “pendulum” rocking mode is not generally modeled, and the effects of the various stiffness nonlinearities, which are not modeled, are likely to be important.

- An auxiliary bearing and compliant mount system that weighs a significant fraction of the rotor weight can lead to a situation where the bearing does not track rotor motion, or substantially reduces the normal rotor critical speeds if it does track. The linear analysis suggested above should give insight into this case.
- A static analysis using the worst-case combination of gravity and normal operating loads using the design rotor-auxiliary bearing gap and the estimated mount stiffness can be performed. The results can be evaluated to confirm that the rotor still has reasonable clearances at seals or other close clearance components, and that the mount travel does not exceed the allowable travel. Some margin would be desirable to account for dynamic shaft motion. Note that for an overhung rotor, the worst case may correspond to one bearing at the top of the clearance and one at the bottom.
- A static analysis using the worst-case combination of gravity and normal operating loads, plus enough additional force (or impeller load for an overhung machine) to force the auxiliary bearing into the stops can be performed. The results can be evaluated to confirm that the rotor still has reasonable clearances at any seals or other close clearance components under this condition. Some margin would be desirable to account for dynamic shaft motion.

More Nearly “Research” Options

For higher risk, or new classes of applications, a more detailed research program may be justified. This section will describe several options that have been reported to be useful. The approaches proposed involve time transient simulations. At the present time, these are not recommended for a typical design audit due to the poor tradeoff between predictive power and time involved in running the analysis. Given the current state-of-the-art with regard to auxiliary bearings and bearing mount designs, the authors propose to treat time transient analysis as more of a research tool than a part of a standard design audit for four primary reasons:

1. Many mount configurations have complex features such as friction interfaces. It is difficult to predict the behavior of these features without experimental data.
2. Auxiliary bearings are a highly nonlinear system. Thus, a stable time transient analysis solution does not rule out the possibility of unstable behavior for a different set of initial conditions.
3. A complete evaluation of a mount system using time transient analysis really requires running many cases and parametric variations to build an understanding of how the system works and what are bounds on the parameters. This level of effort seems beyond the scope of a “design audit.”
4. The severe operating conditions for a backup bearing lead to significant thermal issues that may impact the actual behavior in the machine. A full accounting of these effects is a current topic of ongoing research.

As a research tool, a carefully setup transient analysis that accurately accounts for the various stiffness nonlinearities, the mount characteristics (including the hard stop), and makes a reasonable estimate of the tangential friction at the rotor-auxiliary bearing surface can be quite accurate with regard to the general character of the motion. Such a model will require a fair amount of detailed auxiliary bearing design information from the AMB vendor. Some improvement for rolling element bearing systems is also obtained by including the rolling element bearing dynamics. The complexity of the analysis and the level of detail of the input data required, however, increases substantially.

The cautions with regard to seals and other sources of stiffness and damping presented above should be considered for time transient response calculations as well.

Given these cautions, there are several time transient research studies that can give useful insight into the performance of an auxiliary bearing system.

- A time transient drop analysis for a balanced rotor, but with the worst-case combination of operating loads and gravity. This analysis would help confirm that there will not be rotor-stator contact at other close clearance regions. The relative magnitude of the initial bounce for such an analysis has also been reported to be useful in judging the adequacy of the damping system.
- A time transient drop analysis parametric study for the initial response with varying levels of unbalance and friction coefficients has been reported to be useful in understanding the overall system behavior.
- A series of time transient analyses for a drop and complete coastdown with nominal operating conditions and friction coefficient at the rotorbearing contact (or a rolling element bearing model) can be performed with varying levels and distributions of unbalance. The aggregate results can be useful to bound the unbalance required for full clearance forward whirl. Full clearance whirl for low levels of unbalance indicate a higher level of risk. This analysis could also be used to examine the mechanical strength requirements for the auxiliary bearing supports, as well as the possibility of contact at other close clearance locations.
- If there is considerable uncertainty in one or more of the auxiliary bearing or mount system parameters (stiffness, damping, allowable travel, rotor gap, etc.), then a series of analyses to evaluate the changes in overall behavior for changes in these parameters might also be appropriate.

Auxiliary Bearing Rotordynamic Evaluation—Summary

This section has presented some background on the auxiliary bearing system and suggestions for ways to evaluate this important system during a design audit. Evaluation of auxiliary bearing system is considerably complicated by the fact that the dynamics are highly nonlinear with large uncertainty. AMB vendors have had considerable success in developing reliable, effective systems that allow machinery to be installed with confidence. There are active research projects to develop alternative backup bearing technologies, and hybrid bearing systems that combine a process lubricated bearing, for example, with an AMB system. Some of this research is also aimed at allowing conventional auxiliary bearings to act in a load-sharing mode for occasional high load conditions such as surge. As these technologies develop, they may change the picture considerably.

From an audit perspective, the authors have tried to present some general guidelines to use in evaluating an auxiliary bearing system. However, end-users ultimately have to rely heavily on AMB supplier experience and validation from test stand drop test results where appropriate.

EXAMPLE AUDIT

To tie all of this information together, and help highlight some of the special concerns related to AMB supported rotors, selected results for an example rotordynamic audit are presented in this section.

The machine selected for this audit is a 33,000 hp (24.8 MW), single-stage pipeline compressor. This is an older unit, originally delivered in 1990. It had some early issues, then received some upgrades in the mid to late 1990s that have made this a very reliable unit. A summary of the machine is presented in Table 5 below.
The general rotor cross section is shown in Figure 23. This compressor is representative of a number of between-bearing machines placed in natural gas service around this time.

Using a standard finite-element based rotordynamic analysis tool, the rotor was discretized into sections, and various components such as the laminate stacks, seal runners, impeller, balance piston, etc., were reduced to lumped masses (API 684, 2005). One important consideration was to ensure that the final model included stations (degrees of freedom) at not only the magnetic bearing centerlines, but also the noncollocated sensor centerlines, as well as the auxiliary bearing centerlines for use during the analyses. As is standard practice for the construction of this type, the laminate stacks and sleeves were assumed to add no additional stiffness to the rotor.

The lateral controller is an analog system with a “tilt/translate” approach. For tilt/translate, the goal is to control symmetric modes (rigid cylindrical, first bend, etc.) separately from the antisymmetric modes (rigid conical, second bend, etc.). This was achieved by adding the sensor signals from each end to produce a signal that corresponds to the symmetric modes (cylindrical, first bend). This combined signal is routed through the translate controller, then applied in-phase to the bearing actuators. Tilt control is based on the difference between the two sensors. The output of the tilt controller is applied out-of-phase to the bearing actuators. Thus, this system cannot be modeled as two separate bearings. A multi-input, multi-output approach must be taken.

The characteristics of this multi-input, multi-output control system were provided as a set of transfer functions that were implemented in the rotordynamic model.

Magnetics Audit

The objective of the magnetics component of the audit is to determine the static and dynamic capacities of the two radial AMBs as well as their linearized properties. The results of this component of the audit are summarized in Table 6.

<table>
<thead>
<tr>
<th>Property</th>
<th>NDE</th>
<th>DE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Static Capacity (lbf)</td>
<td>3979</td>
<td>3979</td>
</tr>
<tr>
<td>Low Frequency Dynamic Capacity</td>
<td>3147</td>
<td>2675</td>
</tr>
<tr>
<td>(lbf)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dynamics Capacity Frequency Limit (Hz)</td>
<td>89.5</td>
<td>105.3</td>
</tr>
<tr>
<td>Actuator Gain (lbf/A)</td>
<td>250</td>
<td>236</td>
</tr>
<tr>
<td>Magnetic Stiffness (lbf/in)</td>
<td>-2.24e5</td>
<td>-2.37e5</td>
</tr>
</tbody>
</table>

**Load Capacity—Method**

Following the strategy outlined previously, the quadrant force was computed as a function of quadrant coil current with all of the other quadrant coils turned off and with the journal centered. Based on the power amplifier’s capacity of 50 A, the static load capacity of each quadrant is 3979 lbf. Note that, at this force level, the bearing force is only 50 percent of what an unsaturated magnet would produce, indicating a significant level of saturation. The consequences for operation are high heat at large forces and potentially significant nonlinearity in the actuator. Note that some AMB engineers carry out the static capacity analysis with the journal moved as far away from the working quadrant as clearances will permit. This yields a lower quadrant capacity. If a digital controller were implemented on this machine, it may make sense to introduce a linearization process, such as a lookup table, to manage this nonlinearity. When operating with static forces above the linear range of the force/current characteristic, linearization by lookup table or curve fit is necessary to make up for the diminished local actuator gain.

**Linearized Properties—Actuator Gain**

To obtain the actuator gain, FEA analyses were run for the full bearing model with coil currents in quadrants 2 and 4 set to 20 A, coil currents in quadrants 1 and 3 set to max (0, 20 + ic) and max (0, 20 − ic), respectively, with ic ranging from −30 to +30 A. The nominal operating point was identified as ic, 0 = 3.265 A for the nondrive-end (NDE) bearing and ic, 0 = 5.18 A for the drive-end (DE) bearing. The slope of the force versus current curve was found at this operating point and this number is reported as Ki. For the NDE bearing, the value is Ki, = 5.18 A, while for the DE bearing, K = 236 lbf/A.

**Linearized Properties—Magnetic Stiffness**

With the operating point identified, the FEA was rerun with quadrant currents of 20 ± ic, 0 as appropriate to produce the static bearing load for DE or NDE. In this case, the journal was moved along the quadrant 2-4 line and the forces computed for each position. The slope of this force versus position curve was then determined at the journal centered condition. The resulting slope is the magnetic stiffness. For the NDE bearing, this was computed as K = −2.24e5 lbf/in, while for the DE bearing, it was computed to be K = −2.37e5 lbf/in.

**Lateral Undamped Results**

As is common with most machines, once the model has been assembled, the audit will begin with an examination of the undamped characteristics of the machine. The first two results to be considered will be the rotor free-free characteristics. The audit starts with the free-free characteristics to provide some insight into the rotor’s innate behavior.

Figure 24 presents a plot of the first six free-free natural frequencies as a function of speed. Figure 25 presents the corresponding 0 rpm mode shapes with the sensor (S) and actuator (A) locations...
indicated. The first thing to note from these figures is that the first free-free mode occurs very nearly at the normal 5200 rpm running speed. This is a characteristic of this particular set of machines. This indicates that the control system will be required to shift this mode and add damping to ensure that it is not a problem. This also suggests that conventional autobalance algorithms cannot be applied to this rotor at running speed, since the conventional algorithms would not allow a rotor to pass through a free-free critical speed.

Figure 24. First Six Free-Free Modes Versus Operating Speed.

Figure 25. Zero RPM Free-Free Modeshapes.

The mode shapes are fairly typical for this between-bearings machine. One important thing to note is that there are no free-free mode nodes between the sensor and actuator (region shown shaded). This is desirable, since a node between the sensor and actuator would indicate that the applied force would be out-of-phase with the measured displacement. A node in this location, especially at or below the running speed range, requires careful consideration. However, both the second and third modes have a node that is right at the edge of the undesirable region. Thus, it is desirable to consider whether gyroscopic effects over the speed range might shift the node into the undesirable region. Figure 26 presents the node locations as a function of operating speed. The undesirable regions between the sensors and actuators are again shaded. This plot indicates that the gyroscopic induced changes to the free-free mode shapes are relatively minor for this machine over this speed range. Thus, with regard to free-free modes, the same control strategy that works at 0 rpm should remain effective over the full speed range. If this were not the case, changes to the control system design, and/or gain scheduling would be expected.

Figure 26. Free-Free Undamped Node Locations as a Function of Speed.

The final undamped results are shown in Figure 27, which presents a conventional undamped critical speed map for a bearing stiffness applied at the actuator centerline. This figure suggests that the effective bearing stiffness needs to be at least $2 \times 10^5$ lb/in, but not more than $6 \times 10^5$ lb/in to avoid modes in the operating speed range.

Figure 27. Conventional Undamped Critical Speed Map.

Lateral Damped Stability/Whirl Speeds

With the undamped, undamped characteristics examined, the multi-input/multi-output control system was added to the model for the remaining analyses. The first set of system results to be considered is the eigenvalue/eigenvector analysis (also sometimes referred to as whirlspeed/stability). Figure 28 presents the result in Campbell diagram form, showing the variation in damped natural frequencies as a function of operating speed. The 5200 rpm normal operating speed is indicated with a vertical line; the 6000 rpm overspeed trip is indicated with a vertical dashed line. A $1 \times$ synchronous line is shown dashed. The intersections between this line and the modes are indicated and summarized in a table adjacent to the plot. Table 7 summarizes the modes at 0 rpm and running speed.

Figure 28. Lateral Campbell Diagram (Damped Whirlspeed Map/Eigenvalues).
Table 7. System Damped Natural Frequencies.

<table>
<thead>
<tr>
<th>Mode</th>
<th>0 RPM</th>
<th>5200 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Freq (CPM)</td>
<td>Log Dec (-)</td>
</tr>
<tr>
<td>1 (Cyl. Rigid)</td>
<td>820</td>
<td>6.65</td>
</tr>
<tr>
<td>2 (Cyl. Rigid)</td>
<td>820</td>
<td>6.64</td>
</tr>
<tr>
<td>3 (Con. Rigid)</td>
<td>845</td>
<td>6.61</td>
</tr>
<tr>
<td>4 (Con. Rigid)</td>
<td>845</td>
<td>6.61</td>
</tr>
<tr>
<td>5 (1st Flex)</td>
<td>2843</td>
<td>0.61</td>
</tr>
<tr>
<td>6 (1st Flex)</td>
<td>2843</td>
<td>0.61</td>
</tr>
<tr>
<td>7 (Con./Flex)</td>
<td>4342</td>
<td>2.10</td>
</tr>
<tr>
<td>8 (Con./Flex)</td>
<td>4342</td>
<td>2.10</td>
</tr>
<tr>
<td>9 (2nd Flex)</td>
<td>8534</td>
<td>0.39</td>
</tr>
<tr>
<td>10 (2nd Flex)</td>
<td>8542</td>
<td>0.38</td>
</tr>
</tbody>
</table>

The interpretation of these results is essentially the same as for a conventional bearing system. There are two pairs of very well damped rigid modes well below running speed. There is a pair of shaft bending modes similar to the first free-free bending mode that are below running speed. These modes are followed by a pair of modes that are conical with some shaft bending. Finally, there is a pair of modes above the maximum continuous operating speed. These results suggest that there will be adequate separation margin. From the AMB perspective, this plot again indicates that gyroscopic effects in the modal frequencies are relatively small, thus a gain scheduled controller would not seem to be required.

As with a conventional bearing system, a key finding from this analysis is that all of the modes are predicted to have a log dec greater than 0, thus implying that the modes are all stable (note that aerodynamic cross coupling was not present, and will be considered below). Unlike a rotor with fluid-film bearings, however, all modes must be considered over the full speed range. Any of them could be unstable. This differs substantially from fluid-film bearings, which typically do not require supersynchronous modes to be considered. An improperly tuned AMB system can have unstable motion even when the shaft is not rotating. This system is predicted to remain stable.

Lateral Forced Response (Unbalance Response)

As with an analysis for a rotor supported with conventional bearings, the next analysis to be considered is the system response to unbalance excitation. As above, this model should include the full control system dynamics. Much of this analysis can proceed as per a standard API analysis. Some agreement is required with the end-user regarding whether API sensor amplitude limits are appropriate, the unbalance magnitude to use, and whether to use the API or the ISO clearance check limits. The presence of the AMB system suggests that several additional plots related to actuator force capability limits should also be considered as part of this analysis.

Since the modes of primary concern based on the eigen (whirl-speed) analysis are cylindrical and conical modes, two unbalance cases were considered. For case 1, a total of 8W/N was applied at midspan. For case 2, the unbalance was applied out-of-phase at the two ends of the rotor per the API spec. In the interests of space, only the first unbalance case will be presented in this tutorial.

**Forced Response**

Figure 29 presents the major axis response at the actuators and at the impeller for twice the allowable unbalance (8W/N) applied at midspan. As expected, the only critical speed response peak occurs at the bending mode (mode 6 in Table 7). The displaced rotor shape is shown in Figure 30. These two plots confirm that the expected mode is being excited, that the response is relatively symmetric, and that the amplitudes are generally small. One of the API requirements if this machine were on conventional bearings, is that the sensor responses for twice the allowable unbalance do not exceed 1.52 mils (sqrt(12,000/Nmcos)). Figure 31 presents one of the required sensor Bode plots (NDE). The drive-end plot (not shown) is similar. The sensor responses indicate that this machine would meet this API requirement. Note that as has been discussed, there is a compelling argument that it may be appropriate to relax this criterion for many AMB machines.
test stand and field commissioning conditions. For the purposes
of this tutorial, the ISO 14839 (2004, 2006) criterion will be
considered. This standard does not define a specific balance
condition, so the same twice the maximum API residual unbalance
condition will be used. Note that a simple ratio can be used to
achieve the level corresponding to the API (or AMB vendor
specified) sensor vibration limit if desired.

Figure 32 presents a plot of the maximum rotor displacement for
speeds up 5200 rpm, versus axial location. Estimated clearances at
key locations are identified, along with the corresponding ISO 14839
(2004, 2006) acceptance zones. This plot indicates that the response
for twice the allowable residual unbalance is well within Zone A.
Indeed, it is so small that it barely is visible at the bottom of the plot.

Figure 33. Actuator Force, Midspan Unbalance.

The other consideration is the dynamic load capacity of the
bearings. As discussed previously, the inductance of the radial
actuator and the power amplifier output voltage results in a second,
decreasing, frequency dependent load capacity that is less than the
rated static capacity for higher operating frequencies. Figure 34
presents the required dynamic force versus speed as a percentage
of the smaller of the static force limit, or the maximum dynamic
force based on the inductance and voltage calculation. This plot
indicates that the radial AMB system has very good margin with
regard to dynamic load required to control unbalance forces.

Figure 34. Slew Rate Limit Check, Midspan Unbalance.

A similar analysis was performed for a couple mode unbalance
(two out-of-phase unbalances, one at each end). The results
indicate that this unbalance will excite the eighth mode at around
4800 rpm. The amplification factor is less than 2.5, and would not
be considered to be a critical speed. In the interests of space, these
results will not be presented.

Lateral AMB System

As discussed above, and in ISO 14839 (2004, 2006), the
AMB/rotor system should also be examined using a few tools from
control theory. In this section, the plots required by ISO 14839-3
(2006) will be presented. These results will be augmented by
several additional plots that help to ensure an adequate evaluation.
The first plots to be presented are the sensitivity functions

The maximum value of the four specified sensitivity functions
(four sensors, four actuators) at each frequency between 0 and 2
kHz (range per ISO 14839 [2004, 2006]) at 0 rpm is presented in
Figure 35. The corresponding data for 5200 rpm are presented in
Figure 36. These figures also show the four ISO evaluation ranges.

Figure 35. Zero RPM, Maximum of Sensor/Actuator Sensitivity Functions.

Figure 36. 5200 RPM, Maximum of Sensor/Actuator Sensitivity Functions.
According to ISO 14839 (2004, 2006), the maximum values should all be in zone A for the test stand configuration and zone B for the initial field installation. Ideally, the predicated sensitivity functions would remain in zone A (9.5 dB). However, both plots show a high sensitivity peak at 210 Hz. A second high peak at 0 rpm is shown at 364 Hz. These peaks were probably reduced through field tuning during commissioning. The actual operational history of this machine is quite good.

The other two results that could be generated are the open and closed loop transfer functions. If plots for eventual field model verification are required, the open and closed-loop transfer functions would be a good choice. To create plots for this purpose, it may also be necessary to generate an additional set of plots in an alternative coordinate system (such as tilt/translate) to accommodate physically accessible field measurements.

The results presented below will be for the DE X output as a function of all four sensor inputs. The plots for the other four outputs are similar.

Figures 37 and 38 present the open loop transfer function as a function of frequency from the four sensor inputs to the DE X displacement (this plot is the series combination of the shaft and the control system, without feedback). This plot shows sharp peaks in the vicinities of the free-free natural frequencies as would be expected. The absence of additional peaks in the response confirms that the sensors, control system, and amplifiers are not adding any new high amplification factor modes.

These plots also show the effect of the gyroscopic coupling due to rotation. At 0 rpm (Figure 37), the only two lines that appear are the two X axis inputs, since the Y axis inputs have no influence on the behavior. At 5200 rpm (Figure 38), all four inputs are shown, since they all have an influence on the output due to gyroscopic cross coupling.

The corresponding closed loop transfer functions are shown in Figures 39 and 40. These plots show that the AMB control system has eliminated any high amplification factor modes near operating speed.

Lateral Stability Robustness

The previous sections covered the evaluation of the nominal system, for the rotor and AMB system alone. The audit also needs to address the issues of destabilizing aerodynamic cross-coupling, as well as robustness against other uncertainties. These will be covered in this section.

The first issue to be considered is the effect of destabilizing aerodynamic cross-coupling. As presented in the API 617 (2002) and API 684 (2005) documents, the stability analysis begins with a Level I screening analysis that is based on the application of increasing amounts of cross-coupled stiffness at the rotor midspan. The amount to be applied ranges from none, to the smaller of 10 times a conservative estimate of the anticipated cross-coupled stiffness based on a semiempirical formula, and the amount required to drive the system unstable.

The resulting sensitivity plot is shown in Figure 41. This plot augments the standard API plot that shows just the first forward mode. At 0 rpm (Figure 37), the only two lines that appear are the two X axis inputs, since the Y axis inputs have no influence on the behavior. At 5200 rpm (Figure 38), all four inputs are shown, since they all have an influence on the output due to gyroscopic cross coupling.
Using the parameters selected for this tutorial, the machine would pass the Level I screening test. The machine is compressing a relatively light gas, and is operating at less than 2.5 times the rigid support first critical speed. Thus it is required to have a margin of at least 2.0 between the anticipated level of cross coupling (14,213), and the amount required to drive the system unstable (71,897).

The effect of the added cross-coupled stiffness also should be considered from the AMB perspective. Figure 42 plots the peak value of all of the sensitivity functions described in ISO 14839 (2004, 2006), over the full frequency range, at an operating speed of 5200 rpm, as the cross-coupled stiffness is increased (Figure 36 is one of these plots summarized). This figure shows that the added cross coupling in fact reduces the sensitivity function until a threshold value of approximately 5.8e4 lb/in, at which point it dramatically increases the maximum value of the sensitivity function value. In keeping with the spirit of the API Level I screening test, this amount of cross coupling is much greater than twice the expected cross-coupling, thus the margin is probably acceptable.

Note that similar sensitivity analyses for impeller mass, mode frequencies, etc., could also be performed. The effect of uncertainty in the location of the 210 Hz mode, for example, would be useful to consider. Combination effects could also be examined via a Monte Carlo analysis. In the interest of space, these additional analyses possibilities will not be considered in this tutorial.

Axial Results

The next subsystem that would be considered is the axial (thrust) control system. Since the axial bearing is an actively controlled system, the performance of this system also needs to be considered. For this evaluation, the rotor is generally modeled as a single lumped mass. This significantly simplifies the analysis. However, the eddy-current effects for a nonlaminated stator severely limit the bandwidth of the overall system as described previously.

The basic analysis is essentially the same as for the lateral system, and will not be presented. A forced response analysis similar to the unbalance response analysis may or may not be performed depending on the machine.

Lateral Auxiliary Bearings

The other major subsystem that needs to be considered for a rotodynamics audit of an AMB supported rotor is the auxiliary bearing system. This section presents an evaluation of this system.

Auxiliary Bearing Screening Criteria

This evaluation begins with an assessment of the auxiliary bearing system in light of the proposed screening criteria. Table 8 presents this summary. There are three items that are flagged as high risk. An audit report would suggest that these issues be thoroughly discussed by the vendor and the end-user. Note that the flagging of these items as higher risk would not result in an evaluation of the machine as “unacceptable.” Field experience indicates that this machine is acceptable. It has been running quite successfully for many years.

Table 8. Auxiliary Bearing Screening Criteria.

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Evaluation</th>
<th>Risk Level</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>Moderate speed (5200 RPM)</td>
<td>Moderate</td>
</tr>
<tr>
<td>Countdown time,</td>
<td>&lt; 30 sec countdown</td>
<td></td>
</tr>
<tr>
<td>Machine weight</td>
<td>Moderate weight</td>
<td></td>
</tr>
<tr>
<td>Vendor experience</td>
<td>Near upper limit (at time built)</td>
<td>High</td>
</tr>
<tr>
<td>Compliant mount system</td>
<td>Present (although not when originally built)</td>
<td>Acceptable</td>
</tr>
<tr>
<td>Rotor Balance Procedure</td>
<td>Unknown procedure, possible to balance a free-free mode from coupling hub</td>
<td>High</td>
</tr>
<tr>
<td>Fouling service</td>
<td>No</td>
<td>Low</td>
</tr>
<tr>
<td>Lead Sharing</td>
<td>Not proposed</td>
<td>Low</td>
</tr>
<tr>
<td>Free-Free Mode versus</td>
<td>Between M/COS and trip</td>
<td>High</td>
</tr>
<tr>
<td>Operating Speed</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Static Loading Results

The first analysis to be considered for the auxiliary bearing system is the shaft static deflection for gravity load only. This deflection is the sum of the rotor 1 G deflection, the auxiliary bearing clearance, and the compliant mount deflection due to shaft loading. The resulting shaft deflection is presented in Figure 43. The various close clearance regions are also shown on this plot. These results indicate that reasonable clearance exists at all close clearance locations under gravity load alone. The worst case location is the NDE inner labyrinth seal, which sees a static rotor displacement of 72 percent of the available clearance.

Note that similar sensitivity analyses for impeller mass, mode frequencies, etc., could also be performed. The effect of uncertainty in the location of the 210 Hz mode, for example, would be useful to consider. Combination effects could also be examined via a Monte Carlo analysis. In the interest of space, these additional analyses possibilities will not be considered in this tutorial.
The second result to be considered is a static analysis where the gravity loading is ratioed by the amount required to bottom-out both mounts. The resulting shaft deflection (shaft static loading at 1.5 times normal gravity load combined with the auxiliary bearing deflection, and the mount stop clearance) is shown in Figure 44. Under this much higher loading, the rotor still does not contact at any of the close clearance regions, although several locations do use more than 75 percent of the available clearance. Given the application, this is deemed “acceptable.”

Figure 44. Bottomed Mounts Static Analysis for Auxiliary Bearings.

Undamped Critical Speed Map

With the static behavior deemed acceptable, the next evaluation will be a standard undamped critical speed map for the auxiliary bearing locations. For this calculation, it is assumed that the rotor remains in contact with the auxiliary bearing, that the mount system is linear, and that it does not hit the stop. These assumptions translate to a linear system that is amiable to analysis. Figure 45 presents the resulting map. The estimated series combination of the auxiliary bearing stiffness with the mount stiffness is shown as a vertical green line. This plot suggests that the machine will pass through only one critical speed during the coast down. The general location of the intersections between the assumed stiffness and the critical speeds is to the left of the flat pinned-pinned region of the curves, thus the mount damping would be expected to be effective.

Figure 45. Undamped Critical Speed Map for Auxiliary Bearing System.

Eigenvalue Analysis for Assumed in Contact

A more detailed examination of the character of the system rotordynamics while operating on the auxiliary bearings is provided by a damped eigenvalue analysis with mount damping included. The resulting lateral Campbell diagram is presented in Figure 46. This plot confirms that the system will pass through one critical speed during coastdown, assuming that the orbit remains small enough that the assumption of continuous small amplitude contact between the rotor and auxiliary bearing is reasonable. The mode logarithmic decrements at the critical speed intersections suggest that the mount damping also has some effectiveness for the mode just above running speed mode.

Figure 46. Eigenvalue (Whirlspeed) Analysis for Auxiliary Bearings and Mount System.

The mode shapes for the modes corresponding to the forward damped critical speeds are shown in Figures 47 and 48. The first important thing to note from these plots is that the mount motion tracks the shaft motion (although this is difficult to see in the reduced size plots). The second is confirmation that the lightly damped critical speed at about 2200 rpm is a cylindrical mode. Given the low damping of this mode, it seems likely that the rotor orbits will become large when passing through this critical speed, and loss of rotor-auxiliary bearing contact over some speed range is likely for a high enough level of unbalance.

Figure 47. Mode Shape at First Forward Damped Critical Speed on Auxiliary Bearings.

Figure 48. Mode Shape at Second Forward Damped Critical Speed on Auxiliary Bearings.

WRAP-UP AND SUMMARY

The requirements and process for performing a rotordynamics audit of an AMB supported rotor system are not well defined in the literature or standards. This is especially true in the case of a machine that must be API compliant. This tutorial attempts to gather a variety of information from different sources to assemble an example of what might be included in an audit and some of the relevant AMB background information to help understand why the specific analyses were chosen and how to interpret some of the unfamiliar results.
Some of the key points that should be emphasized include:

- For an AMB supported rotor, the entire system must be modeled as a whole. It can be extremely misleading to think of the AMB system and the rotor as separate systems. This is particularly true for modern multi-input/multi-output control designs such as tilt/translate.
- The basic concerns of unbalance response amplitudes and machine stability are unchanged for an AMB supported rotor.
- Stability concerns for AMB systems are not limited to subsynchronous modes as with fluid film bearings. The stability of all modes within the control system bandwidth must be considered over the full speed range.
- ISO 14839 (2004, 2006), the new ISO AMB standard, introduces a “sensitivity function” analysis procedure that should be performed as part of an audit. However, this analysis alone is not a sufficient guarantee of acceptable performance. Additional evaluations are required. Typical examples include the effect of aerodynamic cross-coupled stiffness and stiffness/mass uncertainty.
- The auxiliary bearing system must be examined separately. The current state-of-the-art for auxiliary bearing analysis does not include an adequate method for evaluating the stability of the dynamic behavior of a rotor supported by the auxiliary bearing system through a complete coastdown. However, experience based guidelines do provide some guidance.
- Existing standards provide some guidance for evaluation of AMB system rotordynamics. Application of the standards, however, will require some engineering judgment. As currently written, the standards do not provide for a complete evaluation of the rotordynamic characteristics of an AMB supported rotor.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Pole area</td>
</tr>
<tr>
<td>A,B,C</td>
<td>First order system matrices</td>
</tr>
<tr>
<td>A,B,C,D</td>
<td>ISO evaluation zones</td>
</tr>
<tr>
<td>A/D</td>
<td>Analog to digital converter</td>
</tr>
<tr>
<td>AAF</td>
<td>Anti-aliasing filter</td>
</tr>
<tr>
<td>AMB</td>
<td>Active magnetic bearing</td>
</tr>
<tr>
<td>B</td>
<td>Magnetic flux density</td>
</tr>
<tr>
<td>Bsat</td>
<td>Magnetic saturation flux density</td>
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<tr>
<td>C</td>
<td>Damping, damping matrix, clearance</td>
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<td>D</td>
<td>Rotor displacement</td>
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<tr>
<td>D/A</td>
<td>Digital to analog converter</td>
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<td>DE</td>
<td>Drive-end</td>
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<td>E, F</td>
<td>Force</td>
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<tr>
<td>G</td>
<td>Transfer function/subsystem dynamics, gyroscopic matrix</td>
</tr>
<tr>
<td>g</td>
<td>Pole face to rotor gap</td>
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<td>K</td>
<td>Stiffness, stiffness matrix</td>
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<td>Kj</td>
<td>Actuator gain</td>
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<tr>
<td>Ks</td>
<td>Magnetic stiffness</td>
</tr>
<tr>
<td>L</td>
<td>Inductance</td>
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<tr>
<td>M</td>
<td>Mass, mass matrix</td>
</tr>
<tr>
<td>N</td>
<td>Number of turns on pole, rotor speed (rpm)</td>
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<tr>
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<td>Nondrive end</td>
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<td>PID</td>
<td>Proportion-integral-derivative</td>
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<td>Effective magnetic pressure</td>
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<td>Resistance</td>
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<td>r_p</td>
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<td>( \mu_0 )</td>
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REFERENCES


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