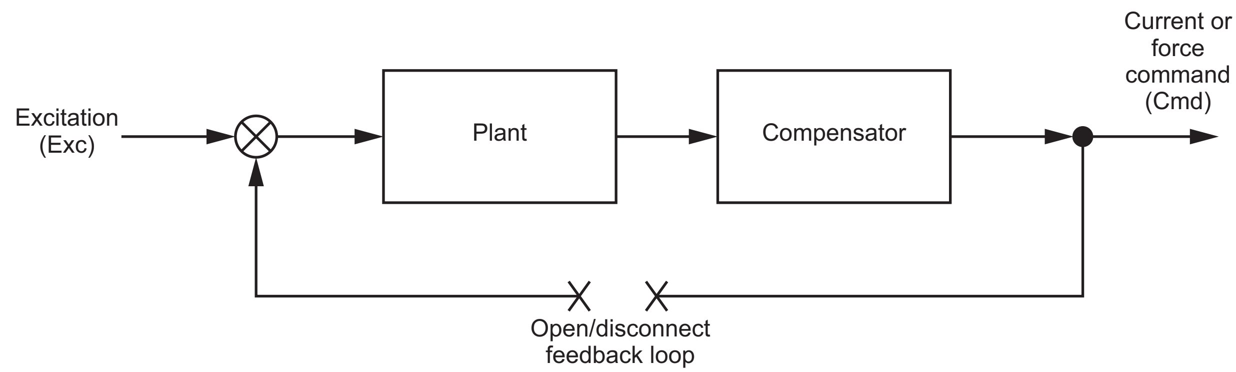

Figure 2-143—Typical Open Loop Transfer Function Measurement for Most Controlled Systems

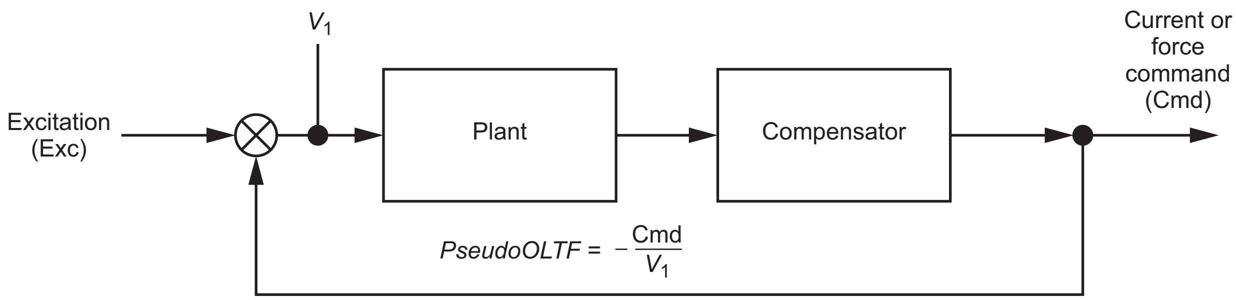

Figure 2-144—Practical "Pseudo Open Loop" Transfer Function Measurement for AMB System

An example of a pseudo OLTF is shown in Figure 2-145. This transfer function and the differences relative to a true open loop transfer function are discussed at length in references [2] and [3].

This pseudo open loop transfer function is not quite the same as the true open loop transfer function. There are two major issues that need to be considered:

— The coupling with the other control axes will generally increase the apparent damping of the free-free shaft modes. Thus, instead of almost no damping, as would be measured with a true OLTF or rap test, there modes could appear to have noticeable amounts of damping. The frequencies, however, are generally correct.

— The coupling with the other control axes will generally affect the rigid body modes. In a true open loop measurement, these modes would appear at 0 Hz. However, with the other control loops active and the coupling due to the shaft, these are likely to appear higher in frequency. This can lead to some confusion if only free-free modes were expected.

Many modern digital AMB control systems are capable of measuring the pseudo OLTF without any external instrumentation, since all of the signals are generally available in digital form.

2.10.7.4 References

[1] Nise, N.S., 2004, *Control System Engineering,* 4th Edition, John Wiley.

[2] ISO 14839-3, 2006, "Mechanical Vibration—Vibration of Rotating Machinery Equipped with Active Magnetic Bearings, Part 3: Evaluation of Stability Margin," No. 14939-3, International Organization for Standardization.

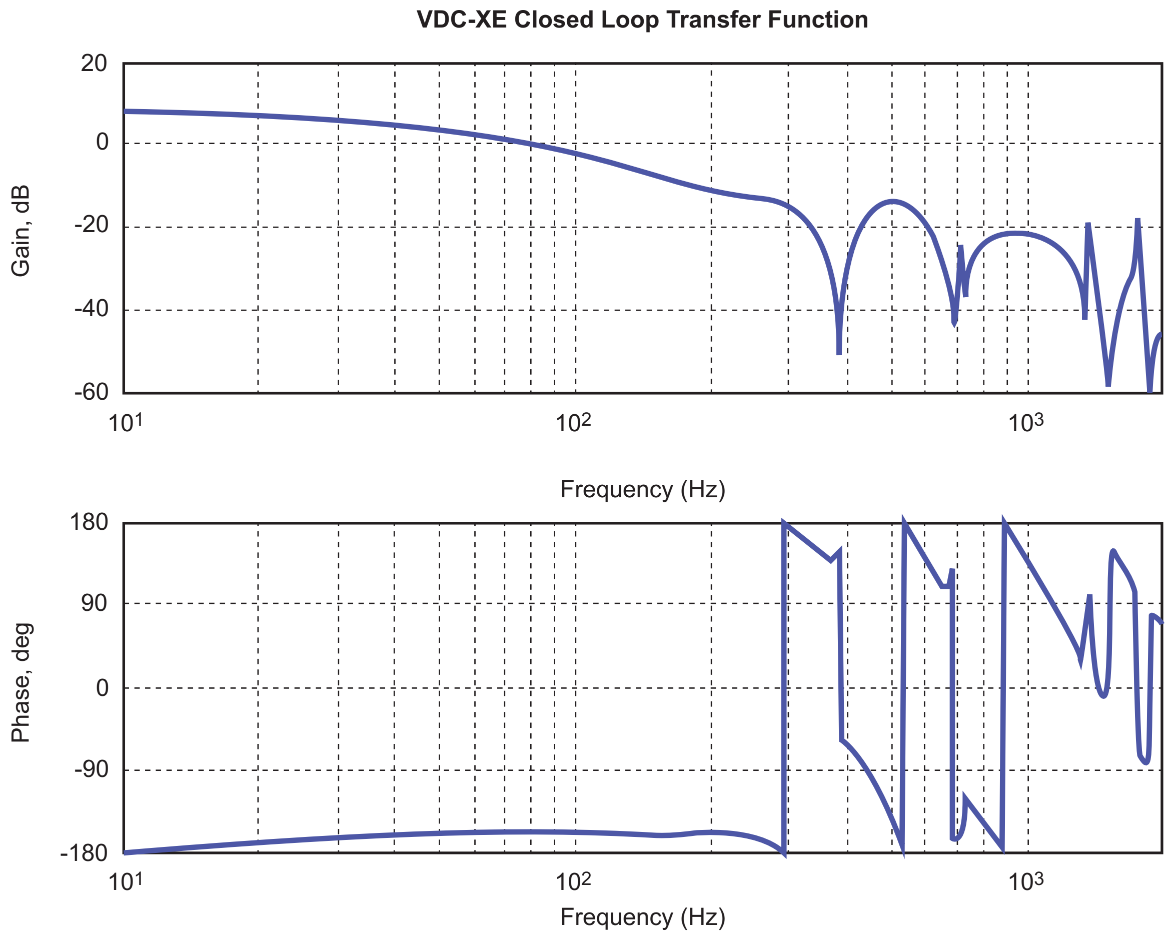

Figure 2-145—Example Pseudo Open Loop Transfer Function

[3] Schweitzer, G. and Maslen, E.H., editors, 2009, *Magnetic Bearings: Theory, Design and Application to Rotating Machinery*, Springer.

2.10.8 Model Verification Testing

2.10.8.1 Overview

AMB systems inherently have the ability to both measure shaft motion, as well as apply excitation forces to the rotor. Thus, many rotordynamic performance measurements are more easily facilitated by AMB equipped machines as compared to fluid film bearing equipped machines.

The AMB system allows measurements to be made with the machine running or with the machine not rotating. Calculation results for both the zero speed and running cases can be made available to compare with the measured results. A good correlation between the zero speed calculations and measurements will minimize the risks of any problems occurring when the actual running tests are performed. This is in contrast to the situation with a fluid film bearing machine, where the machine must be operating to make meaningful measurements since unbalance is typically the only practical source of excitation force.

The specifications include an option to replace the traditional unbalance response model verification test with a CLTF model verification test. This test compares the measured CLTF to the predicted CLTF. This approach is especially useful in the case of a sealed machine, where there is no convenient coupling hub on which to place a test weight. In these machines, the traditional unbalance test would require opening the machine casing.

2.10.8.2 Unbalance Response Tests

A traditional unbalance response test can still be used as a rotordynamic performance verification measurement. With an AMB machine, however, both shaft position and the AMB currents should be monitored. The AMB bearing current signals provide a measurement that is proportional to the actual dynamic loading of the AMB. In most systems with linearized actuator characteristics, the dynamic force will be directly proportional to the dynamic current over the useful range of bearing loading. Technically though, the dynamic load is dependent on both the bearing currents and the bearing air gap, but if the vibration level is a small percentage of the magnetic bearing gap (less than 10% of the air gap), then the dynamic component of the current measurement is a reasonable approximation of the dynamic bearing load.

The dynamic bearing currents (force) can be compared to the allowable bearing dynamic force envelope limits to ensure that adequate reserve margin is maintained.

2.10.8.3 Model Verification Example

The results of one axis of a CLTF model verification test are shown in Figure 2-146. These measurements were made on one radial axis of a small, high-speed machine. As can be seen, the lower shaft modes are extremely heavily damped in both analysis and measurement. The frequency and amplification factor of the first shaft bending mode near 700 Hz are well predicted. There is some difference for the second mode near 1700 Hz. The machine has a maximum operating speed of 36,000 rpm. API 617's Annex E requires that "the frequency of radial resonance peaks from the closed loop transfer function up to $1.25 \times N_{mc}$ shall not deviate from the corresponding frequency predicted by the analysis by more than ±5 %, and the measured peak amplitudes must not be greater than 1.0 times, nor less than 0.5 times the predicted amplitudes." Thus, the match is acceptable. This measurement would be repeated on the remaining axes. If all are acceptable, then the model is validated.

2.10.9 Axial Rotordynamic Analysis

2.10.9.1 Introduction

In a fully AMB supported rotor, there is also an axial AMB system. Thus, it is necessary to perform an axial analysis. An axial analysis is not typically performed for most fluid-film or rotating element bearing machinery. The main concerns for the axial analysis are stability and stability robustness, which are discussed in Section 3. However, the specifications also require a damped natural frequency/modeshape analysis, which will be discussed in this section.

2.10.9.2 Axial Rotor Modeling Considerations

Generally, at least a partial train model is required to accurately model the axial dynamics. There have been cases where the dynamic behavior of a "flexible" disk-pack coupling were significant, so it is not a safe practice to assume that a flexible coupling decouples the various bodies in the train for the axial analysis.

An adequate axial model can generally be obtained by considering each body in a train to be a rigid mass, similar to a simplified torsional analysis. Some other considerations include:

— Coupling disk pack stiffnesses should be included in the model if flexible coupling are used.

— Rigid couplings may result in only one rigid body mode that needs to be considered.

— Axial stiffness and damping of dry gas seal assemblies may need to be included.

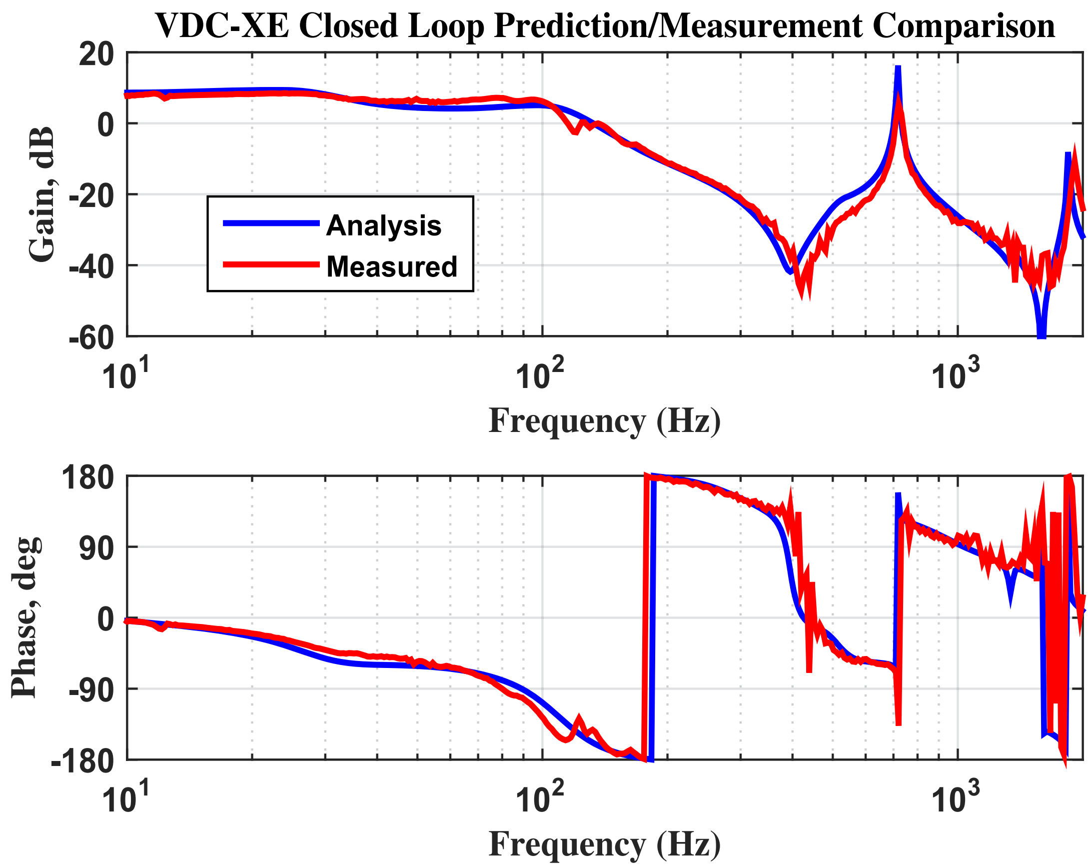

Figure 2-146—Closed Loop Transfer Function Model Validation

— Axial stiffnesses due to magnetic centering forces from motors or generators, as well as the radial AMB system may need to be included.

— Axial stiffnesses due to aerodynamic centering forces may need to be included.

— The effect of eddy currents on the dynamic characteristics of the axial actuator may need to be considered. If significant, these effects should be included in the displacement to force transfer function provided by the AMB vendor.

— In some cases, accurate models have required additional degrees of freedom at thin, large diameter disks (thrust disks, impellers, etc.) to account for disk flexibility effects from one or two nodal diameter modes of the disk.

— It may be necessary to consider the axial dynamics of the casing if there are casing modes in the control system bandwidth. The specification requires that the basis for the structural model be provided. In particular, if the structure is assumed to be rigid, the basis for this assumption should be documented.

2.10.9.3 Axial AMB Control System Considerations

Most of the comments related to radial control system models also apply for the axial dynamic model. In particular, the full dynamics of the AMB plant (sensor, controller, power amplifiers and actuator) must be included in the analysis model.

2.10.9.4 Damped Natural Frequency/Modeshape Analysis

The specifications require that the axial damped natural frequencies (eigenvalues) for all modes with an amplification factor greater than 2.5 be calculated. Given that a simplified, lumped mass model is being used for the axial system, this would be expected to be a fairly small list of modes, so no bandwidth limits are specified. From a practical perspective, all of the axial modes would be calculated, since it is generally not possible to know in advance which ones would have an amplification factor less than 2.5.

The reporting requirements indicate that the modeshapes also need to be plotted or at least described in the report. A typical report table for a two body system with a flexible coupling is shown in Table 2-6.

There are no explicit acceptance criteria for the axial modes in the standard. However, the axial sensitivity function must fall within zone B or better. Implicitly, this indicates that the modes must be stable, since the sensitivity function analysis is meaningless for an unstable system. It is also noted that modes which are dominated by motion within a flexible coupling are excluded, since the AMB system generally has little or no control authority over these modes.

Table 2-6—Example Axial Damped Natural Frequency Analysis Results (Amplification Factors > 2.5)

Mode	Frequency (Hz)	Amplification Factor	Description
1	80	2.5	Bodies Out of Phase
2	310	50	Spacer Mode

2.10.10 As Installed Analysis

One very unusual feature of the AMB specifications is the requirement to perform a final set of analyses using the as-tuned parameters following initial field commissioning. The final rotordynamics report is not considered complete until this step is performed. At the time this requirement was developed, it was recognized that it will cause some administrative complications. However, given that many AMB systems currently require at least some field tuning, it was felt that it is crucial that the final rotordynamic report include consideration of the implications of this tuning.

As currently written, there are no acceptance requirements related to the results of this analysis. It is intended to be purely informative for all parties. Obviously though, it is certainly desirable that the as-tuned system meet all of the standard acceptance criteria.

2.10.11 Auxiliary Bearings

2.10.11.1 Introduction/Overview

It is expected that all API machinery using AMB's will have an auxiliary bearing (also called "touchdown bearings," "backup bearings," or "catcher bearings"). These bearings are typically located adjacent to the radial actuators as shown previously in Figure 2-130. As described in the standard, these bearings are purely machinery protection devices. In many cases, they will have a very limited operational life. In addition, the rotordynamic performance of the machine when operating on the auxiliary bearings may be quite unacceptable by the criteria used to evaluate normal operation. However, it must still be evaluated in so far as possible, to ensure that the auxiliary bearings can fulfill their required function.

The goal of this section is to provide some background on the unique requirements for auxiliary bearings, evaluation of the rotordynamic performance through analysis and modeling, as well verifying the performance through drop testing.

It should be noted that there was no proven, industry standard approach for modeling and analysis of auxiliary bearings when the AMB specific standards were developed in 2009–2010. Therefore, the standards require that the auxiliary bearing performance be evaluated, but do not provide specific guidance for how the evaluation is to be performed. The intent was to allow OEMs a great deal of latitude with regards to how to meet machinery operational goals, while hopefully encouraging the development of a consensus approach. The results of the evaluation are included as part of the overall rotordynamics report to allow comparison with future experience.

2.10.11.2 Auxiliary bearing function

The function of the auxiliary bearing system is to provide additional shaft support for three conditions:

— rotor not levitated by AMB system;

— AMB system failure;

— AMB system overload.

For each condition, the auxiliary bearing system must ensure that there is no rotor-stator contact at any close clearance locations other than the auxiliary bearings (with the possible exception of abradable/compliant seals). The following paragraphs provide more detail for each of the scenarios in which the auxiliary bearing system is necessary.

The standard requires that the auxiliary bearings be designed for a minimum of two drops with a complete coastdown, and an agreed upon number of momentary contacts due to transient overload during operation, without requiring replacement. Although infinite life would clearly be desired, this will probably be beyond the state of the art for quite some time. The intent of the standards is to encourage a realistic discussion among all of the parties to ensure that the machine is capable of performing its intended function.

2.10.11.2.1 Rotor support when not levitated

Any time the AMB system is powered down for maintenance, inspection, or simply because it is not in use, the auxiliary bearing system must support the weight of the rotor. Thus, even a fault tolerant AMB system capable of supporting all possible operating loads will still have some sort of “auxiliary bearing” surface to support the rotor when the AMB system is inactive. For some machines, the auxiliary bearing must also support the rotor during transportation or installation. In this case, consideration must also be given to the relevant loading, which may be larger, be oriented differently, and/or have significant components due to vibration. Likewise, seismic events may have to be considered for some installations.

2.10.11.2.2 Rotor protection in the event of magnetic bearing failure

Although AMB system reliability is historically high, it is still possible that a component of the system can fail, be damaged by electrical transients (lightning strike, for example), or physically damaged by some outside source (interconnecting cabling accidentally cut, for example). In these situations, the auxiliary bearing system must provide the necessary shaft support at the operating condition of the machine. This includes supporting the weight of the rotor, the dynamic loads due to unbalance, and the radial and thrust loads associated with the process (i.e. side loads and thrust loads). The auxiliary bearing system must provide this support instantaneously upon the event of a failure. It is typical to initiate an emergency stop in the event of AMB failure, so the auxiliary bearing system will only be required to provide support long enough to bring the rotor safely to rest. This failure scenario is often referred to as a “landing” event, since the rotor essentially lands on the auxiliary bearings.

2.10.11.2.3 Rotor protection in the event of magnetic bearing overload

Because AMB systems have a limited load capacity, there can be overload conditions in which the AMB system cannot provide the necessary support and additional bearing capacity is required. Typical examples are unexpected process upsets that generate forces not considered during the design process, or bearings that cannot be practically sized large enough to handle all possible operating conditions. In this scenario, the auxiliary bearing system must provide the extra load capacity required to prevent contact between the stator and rotor. Unlike the landing event described above, in this scenario the AMB system is still active, so the auxiliary bearing system only has to provide partial load support. Operation in a load sharing mode may continue until normal operation is restored, or brief load sharing immediately followed by a trip and delevitation event where auxiliary bearings carry the entire load.

2.10.11.3 Auxiliary Bearing Types

Almost all API relevant AMB supported machinery use either rolling element bearings or solid lubricated bushings for the auxiliary bearings. A possible exception is pumps, which might use process lubricated bearings or seal(s) that double as auxiliary bearing(s).

Figure 2-147 provides sketches of typical arrangements. As shown in the sketches, it is most common for the auxiliary bearings to be located outboard of the AMB components. The choice of bearing type is a function of application requirements and AMB vendor experience base. In each case, there are radial and axial clearances between the bearing surfaces and the shaft during normal operation.

Typically, rolling element auxiliary bearings are either deep groove (radial only) or a duplex pair of angular contact (radial and thrust) bearings. Frequently, ceramic balls are used. Considerable care is required in the bearing geometry and lubricant selection because of the combination of very high operating speeds and relatively large bearing diameter that is typically required. The operating DN values are often well beyond what would be acceptable for continuous operation. The bearing race/ball/cage acceleration rates when the shaft contacts the bearing race are also quite high. Special cages or cageless (full ball complement) designs may be required. The choice of materials for the race and shaft, along with any lubrication or coating is also important.

A variety of solid lubricated bushing designs have been used. The bearings may or may not have segmented pads. The radial bearing may or may not be separate from the thrust bearing. Considerable care is required in selection of bushing geometry, materials (bearing and shaft), lubrication, coatings and mounting due to the very high surface speeds involved.

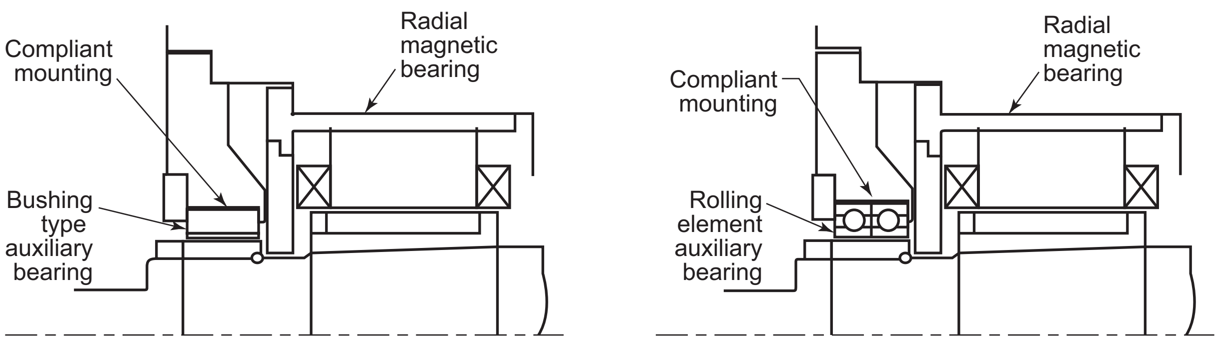

Figure 2-147—Common Auxiliary Bearing Configurations

2.10.11.4 Bearing Mounting Stiffness and Damping

The auxiliary bearings are almost always mounted in a support system that provides stiffness and damping (API 617 requires this for compressors). The dynamic characteristics of this mount largely control the overall rotordynamic behavior of the system during a delevitation or overload event. Figure 2-148 shows a typical arrangement.

Tradeoffs between competing requirements are generally required during auxiliary bearing mount design. Adequate damping is required to reduce the dynamic loads on the auxiliary bearings during rotor impacts and to suppress rotor whirl motion within the auxiliary bearing clearances. However, as with squeeze film dampers, too much damping would be undesirable, since it could prevent the mount from moving. This would again lead to very large whirl amplitudes and forces. Stiffness has two opposite effects: high stiffness will reduce rotor vibration amplitudes. However, high stiffness also tends to increase the rotor natural frequencies on auxiliary bearings, triggering higher frequency rotor whirl and larger auxiliary bearings loads. Very high stiffness can also prevent the damping from being effective.

The mount stiffness characteristics are typically nonlinear for large loads due to the presence of a hard stop which limits maximum mount motion Additional nonlinearities due to friction damping, nonlinear springs, etc. are also frequently present.

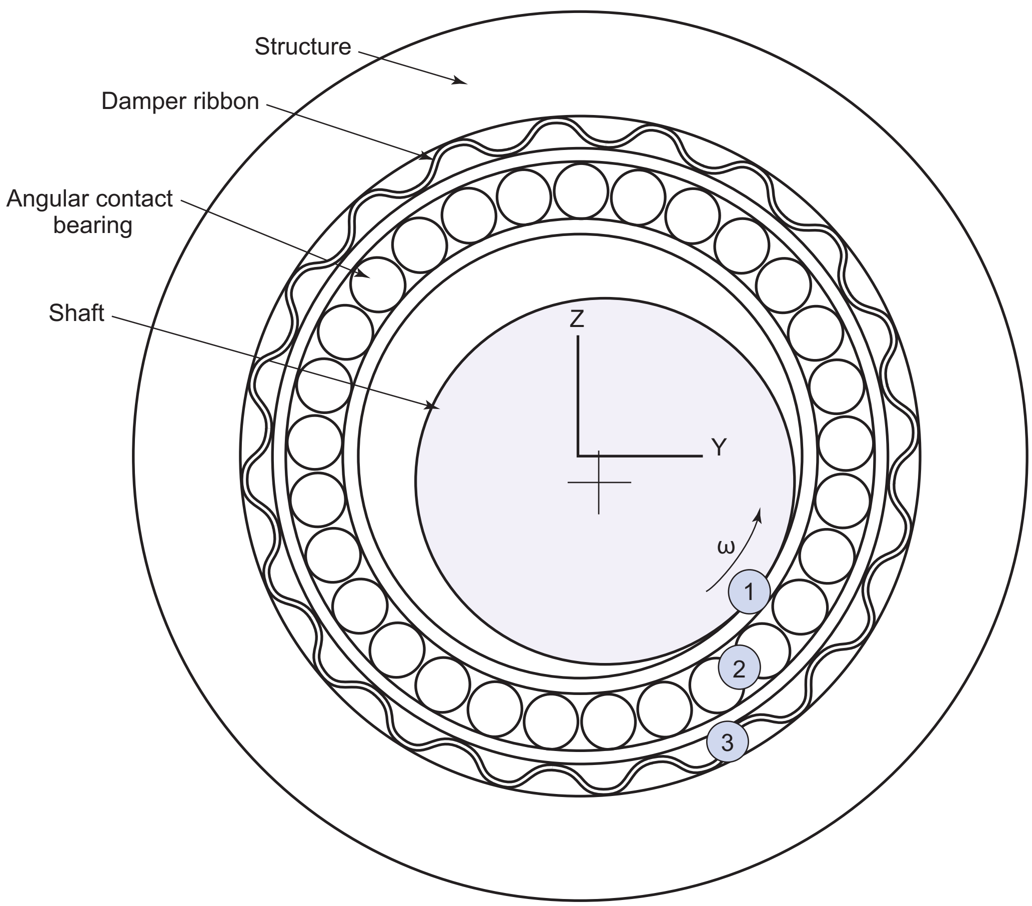

Figure 2-148—Typical Auxiliary Bearing Mount System [4]

2.10.11.5 Vertical Versus Horizontal Machinery

By their nature, AMB systems allow for operation in any orientation. The choice of the orientation has some influence on the design of the auxiliary bearing system and the system rotordynamic characteristics.

For the horizontal application, there is almost always adequate radial load so that the rotor tends to sit at the bottom of the auxiliary bearing. This radial load is primarily the weight of the rotor, although additional side loads may have to be accounted for as well. Two common examples are suction and discharge nozzles for compressors and pumps. These can apply significant radial loads that increase or decrease the bearing load. Converging seals can also have very large centering forces. In the case where the auxiliary bearing can be unloaded by side forces, there is typically an increased likelihood of large amplitude whirl.

The axial or thrust load will generally be related only to the fluid forces for horizontal machines. Depending on the design of the machine and the secondary flow management scheme, these fluid forces can be quite large, but they will generally decay quickly with shaft speed. So, for the design of horizontal machines, gravity and fluid forces are the main drivers in auxiliary bearing selection with regard to load capacity. Schmied and Pradetto [1] present typical whirl results for a horizontal machine which experiences a few circular orbits prior to settling down into the bottom of the auxiliary bearing race for the remainder of the landing event (Figure 2-149).

In vertical applications, the radial bearings are not loaded by the shaft weight. Therefore, the design load for the radial bearings is more difficult to assess. Because there is no gravity load to bias the rotor to any one lateral direction, the rotor is more likely to whirl within the auxiliary bearing clearance. There are now two important terms which determine the loads sustained by the auxiliary bearings; the radius of the whirl (*i.e.* the orbit of the shaft) and the frequency of the whirl. Clearly, the rotor begins the landing event spinning at the operating speed, but it will establish a second frequency of rotation, in which the unit whirls about the center of the bearing clearance, while still spinning. An excellent example of this phenomenon is provided by McMullen et al. [2] and shown in Figure 2-150. In this case, there is clearly an orbit of the rotor around the circumference of the auxiliary bearing. There is also a synchronous component of whirl which is due to rotor unbalance.

Depending on the whirl frequency (and the whirl radius), the radial load due to whirl can be large. Therefore, as a design philosophy, it is important to minimize the whirl frequency of the vertical rotor while supported on auxiliary bearings. Prediction of the whirl frequency continues to be an active research subject, but is generally closely related to the support stiffness of the auxiliary bearing system. Research in flywheel applications has demonstrated that the support stiffness is the determining factor for these machines, whether that support stiffness is dominated by bearing components or by the structure itself [3]. For oil and gas applications, the support stiffness may not be the only determining factor. Ransom et al. [4] found that typical aerodynamic forces related to labyrinth seals and impeller shrouds can drive the whirl frequency up. This is due to the presence of cross-coupled stiffness, which converts radial motion into a forward whirl force.

For a vertical machine, the thrust bearing will experience the direct gravity load of the rotor in the event of delevitation. As mentioned above, fluid forces developed within the machine can be significant as compared to the rotor weight. Eventually though, the auxiliary bearing will carry the entire rotor weight as the fluid forces decay with shaft speed.

2.10.11.5.1 References

[1] Schmied, J. and Pradetto, J.C., 1992, “Behavior of a One Ton Rotor Being Dropped into Auxiliary Bearings,” *Proceedings of the Third International Symposium on Magnetic Bearings*.

[2] McMullen, P., Vuong, V., and Hawkins, L., 2006, “Flywheel Energy Storage System with AMB’s and Hybrid Backup Bearings,” *Proceedings of the Tenth International Symposium on Magnetic Bearings*.

[3] Caprio, M.T. et al., 2004, “Spin Commissioning and Drop Test of a 130 kW-hr Composite Flywheel,” *Proceedings of the Ninth International Symposium on Magnetic Bearings*.

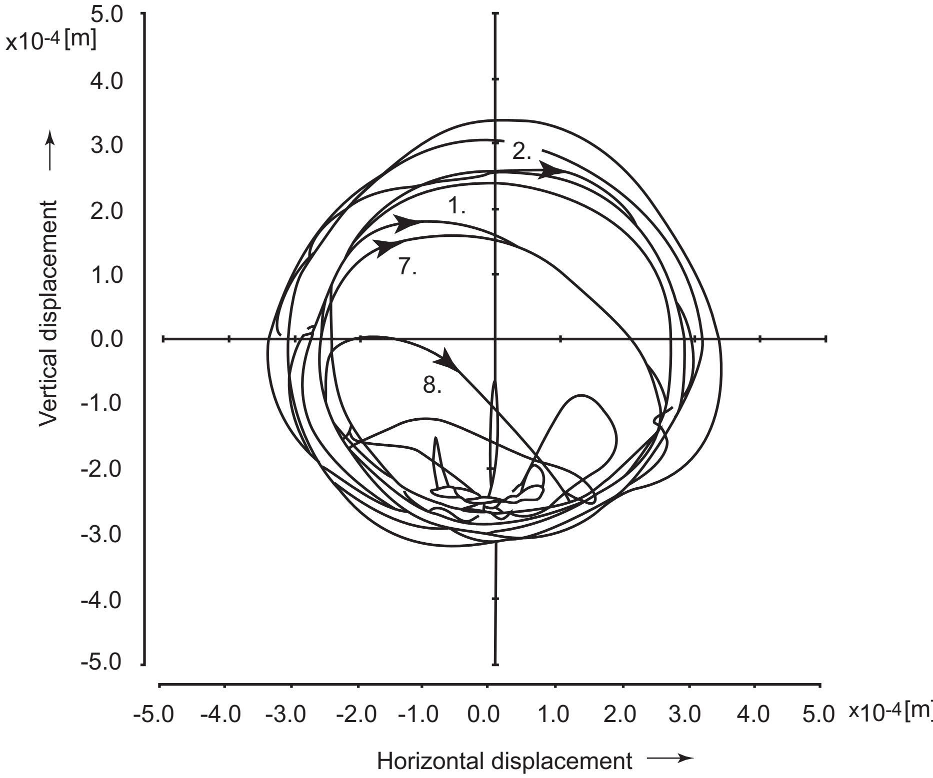

Figure 2-149—Typical Horizontal Rotor Landing [1]

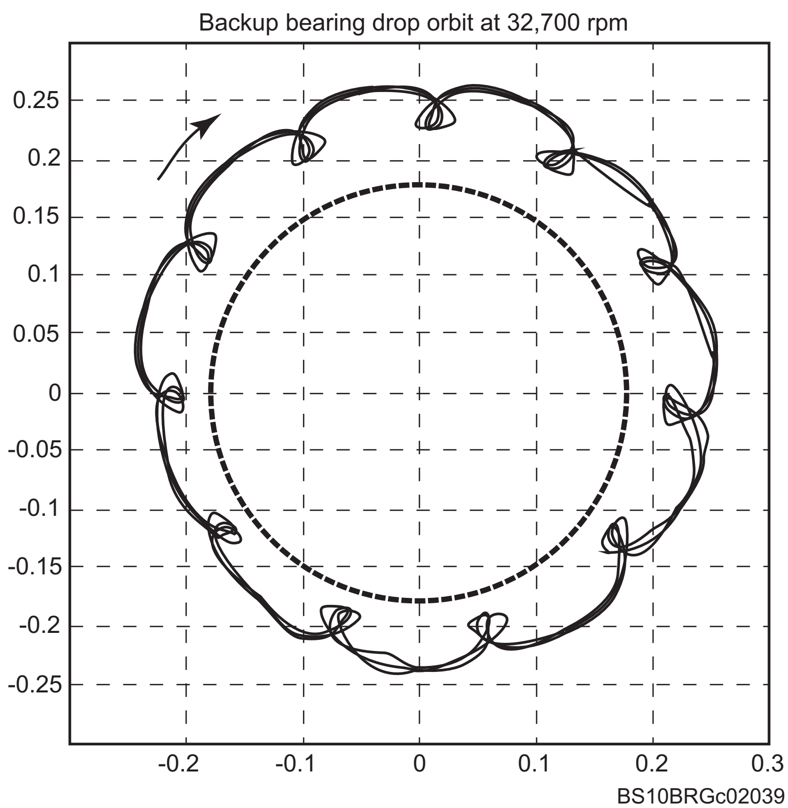

Figure 2-150—Vertical Rotor Landing Orbit [2]

[4] Ransom, D., Masala, A., Moore, J.J., Vannini, G., Camatti, M., 2008, "Numerical and Experimental Simulation of a Vertical High Speed Motorcompressor Rotor Drop onto Catcher Bearings," *Proceedings of the 11th International Symposium on Magnetic Bearings*, Nara, Japan, August 26-29.

2.10.11.6 Auxiliary Bearing Life Considerations

Auxiliary bearing systems are almost always very life limited systems. Thus, the key question we would really like to answer from the rotordynamics/bearing analysis is, "how many drops will be possible?" The answer to this question is a complex combination of:

- rotor weight and operating speed;
- auxiliary bearing design;
- rotordynamic characteristics;
- Whether process fluids are present at the auxiliary bearing location during normal operation, machine upset, or storage, and the composition of these fluids and thermodynamic conditions;
- potential for contaminates in the bearing compartment (rust, unexpected process fluids, etc.);
- thermodynamic conditions (temperature, pressure, cooling air) at the auxiliary bearing location.

Unfortunately, the current state of the art is not adequate to allow a reliable prediction of bearing life in the general case using readily available tools. As the statistics and experience become more widely known, statistical analysis, such as Weibull analysis, to quantify life data and predict component reliability may become practical.

From the rotordynamics perspective, a key to maximizing auxiliary bearing life is to avoid large amplitude dynamic forces due to unbalance and rotordynamic effects. Analytical predictions of dynamic loads of one or even two orders of magnitude larger than the static load have been presented in the literature [1]. Clearly, these large loads can have a very detrimental effect on bearing life, if not leading directly to failure. Adequate control of unbalance state, and appropriate rotordynamic characteristics of both the rotor and the bearing/mount system are clearly required. There is also interaction with the length of the contact or coastdown time.

The capability to meet service life, reliability, and observability requirements may involve tradeoffs in the design of auxiliary bearing systems. The space envelope may grow, resulting in compromises in the rotordynamics of the machine, both when supported on magnetic bearings and when supported on auxiliary bearings. Compromises in rotordynamics may become a vicious circle where the rotor vibration amplitude and loads imparted to the auxiliary bearings increase, accompanied by increases in the bearing temperatures as well. Rotor-bearing system stability could even be compromised in the extreme case. Thus, the design of an adequate auxiliary bearing is challenging.

2.10.11.6.1 References

[1] Kirk, R.G. 1999, "Evaluation of AMB Turbomachinery Auxiliary Bearings." *Journal of Vibrations and Acoustics*, 121, 2, pp. 156–162.

2.10.11.7 Rotordynamic Analysis and Modeling

There are two main concerns with regard to the rotordynamics of a rotor running on auxiliary bearings:

- excessive loading during the landing and stable whirl events;
- rubbing of shaft seals during landing and whirl events.