Chapter 23

Externally Pressurized Bearings and Machinery Diagnostics

In previous sections of this book, we have discussed machinery diagnostics from the point of view of conventional bearing technology. The technology continues to evolve, and new bearing developments promise to change the behavior of machines and the way we look at machinery data. The externally pressurized bearing is undergoing rapid development, and, because of its external pressure source, it has the capability of producing variable spring stiffness under operator or automatic control. As the externally pressurized bearing enters general use, it promises to change rotor behavior in ways that will affect diagnostic methodology.

In this chapter, we will discuss the machinery diagnostic implications of variable stiffness. We will start by comparing the stiffness behavior of rolling element bearings, conventional hydrodynamic (internally pressurized) bearings, and externally pressurized bearings. Then, we will show how variable stiffness can affect rotor behavior. Finally, we will discuss the implications of variable stiffness for future machinery diagnostics.

Types of Bearings

The bearings used in large rotating machinery can be divided into two general types: rolling element bearings and fluid-film bearings. Fluid-film bearings can be further subdivided into internally and externally pressurized bearings.

Rolling element bearings most often find their application in small, balance-of-plant machinery, very low-speed machinery, or machines where the weight or complexity of a lubricant supply system for a fluid-film bearing cannot be justified. Rolling element bearings do not allow significant rotor motion near the bearing; they have very high spring stiffness and low damping, both of which are essentially independent of load. Because of their high stiffness, rotor nodal points tend to be located within or very close to the bearings.

To provide more damping to machines equipped with rolling element bearings, squeeze-film dampers are sometimes used. For the same bearing design, a squeeze-film damper will tend to reduce the overall bearing stiffness and allow more shaft motion in the bearing plane.

Fluid-film bearings are used in most large rotating machines, where a thin film of lubricant keeps the journal and bearing separated. The film thickness in these bearings is typically on the order of tens of micrometers (a few mils). The lubricant is most often oil, but it can also be the working fluid of the machine; for example, water or even a gas.

Fluid-film bearings can be *internally* or *externally* pressurized.

Internally Pressurized Fluid-Film Bearings

Internally pressurized bearings support the rotor with the dynamic pressure of the lubricating fluid created inside the bearing by the action of the rotating journal. In these bearings, a continuous flow of oil is supplied to the bearing at low pressure, typically around 1 to 2 bar (15 to 30 psi). The bearing normally supports the journal at a moderate to high average eccentricity ratio, and the journal acts like a pump that produces circumferential flow of the fluid. As the fluid is drawn into the reduced clearance by the journal, the local fluid dynamic pressure increases, some of the fluid escapes axially, and the remaining fluid provides the support for the rotor. When a rotor is normally loaded and operating at design eccentricity ratios, plain cylindrical journal bearings are partially lubricated; that is, the bearing cavity is often not completely flooded with lubricating fluid.

When plain cylindrical journal bearings are lightly loaded, perhaps due to misalignment or radial loads on the rotor that act against gravity, the journal can move to low eccentricity ratios, and the bearing can become fully flooded. When this happens, it can trigger fluid-induced instability. The root cause of this instability is the tangential force produced by the differential pressure associated with the converging and diverging circulating fluid (see Chapter 22).

In an attempt to counteract this effect by disrupting the fluid flow, various bearing geometries have evolved, such as axial groove, elliptical, offset half, and pressure dam bearings. Even with these designs, fluid-induced instability is occasionally a problem.

Tilting pad bearings were developed to provide an inherently more stable, internally pressurized, fluid-film bearing. They have several separate, pivoted pads that support the journal. Each pad rests on a single pivot or rocker that, ideally at least, cannot support a moment. The design ensures that the support force is directed through the pivot toward the center of the journal. When this condition is satisfied, no tangential forces can develop, and the bearing will be stable. However, fluid-induced instability has been known to occur in tilting pad bearing machines because of circumferential oil flow in the oil control (retainer) rings of these bearings or because of circumferential fluid flow elsewhere in the machine, such as in seals or around turbine or impeller disks. Tilting pad bearings are effectively fully flooded, with a hydrodynamic oil film existing at all times on all of the pads, and the journal normally operates at eccentricity ratios close to zero. Tilting pad bearings are in widespread use on many types of rotating machinery.

Externally Pressurized Fluid-Film Bearings

An externally pressurized fluid-film bearing operates in a fully flooded condition by design. Fluid at relatively high pressure is supplied to a set of inlet orifices that restrict the flow and create a pressure drop, and pockets distribute the fluid pressure over parts of the journal (Figure 23-1). The fluid escapes axially through the journal/bearing radial clearance.

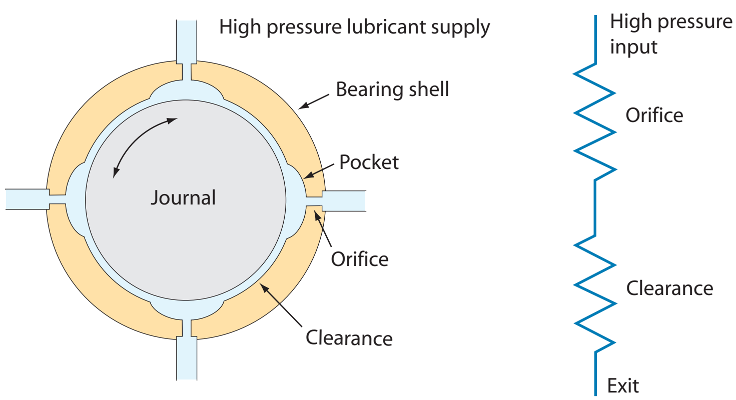

Figure 23-1. Externally pressurized bearing and equivalent resistance network. The high pressure supply feeds multiple inlet ports. The pressure drops across an orifice to an intermediate level. The clearance between the journal and the bearing creates another flow resistance that changes when the journal position changes in the bearing.

The pressures and fluid flow in an externally pressurized bearing are similar to the voltages and electric current in a series resistor circuit. The orifice is equivalent to one resistor; the flow resistance created by the journal/bearing clearance is the second. Thus, the fluid in the pocket is at a lower pressure than the supply pressure, but is higher than the pressure at the exit of the bearing.

When a nonrotating journal is centered in an externally pressurized bearing, the fluid flow is equally divided between the bearing pockets. In this condition, the pressure in each bearing pocket is equal, and no net force is produced on the rotor.

When a nonrotating journal is displaced from the center, the flow becomes more restricted in the narrow clearance and less restricted in the wider clearance. The lower restriction in the larger clearance lowers flow resistance, reducing the pressure in that pocket. Meanwhile, on the opposite side of the bearing, the narrower clearance increases flow resistance, increasing the pocket pressure on that side. The net result is a pressure differential and a restoring force that attempts to push the rotor back toward the center of the bearing. The differential pressure is approximately proportional to the displacement from the center and produces the equivalent of a spring stiffness.

If the journal is rotating, then displacement from the center also creates a hydrodynamic pressure wedge similar to that in an internally pressurized bearing. Thus, the stiffness of an externally pressurized bearing is a due to a combination of static and dynamic pressure effects (Figure 23-2). If the supply pres-

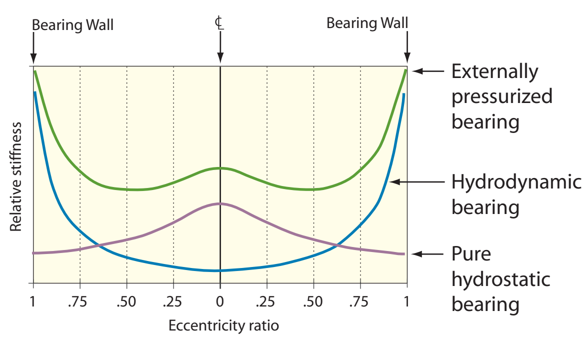

Figure 23-2. Stiffness versus eccentricity ratio for different bearing types. Hydrodynamic bearing stiffness (blue) is minimum when the journal operates at the center of the bearing (eccentricity ratio zero); stiffness rises rapidly near the bearing wall (eccentricity ratio one). Pure hydrostatic bearing stiffness (violet) is maximum when the journal is near the center of the bearing (low eccentricity ratio). Typical externally pressurized bearing stiffness (green) has characteristics of both types: high stiffness at low eccentricity ratios and high stiffness at high eccentricity ratios.

sure is sufficiently high, static pressure effects dominate, hence the name *hydrostatic* bearing.

For the same size bearing, externally pressurized bearings have higher stiffness than internally pressurized bearings. Also, an externally pressurized bearing can be pressurized before startup to lift the rotor, eliminating metal to metal contact and bearing wear, and allowing very low starting torque. The high stiffness of this bearing forces the journal to operate near the center of the bearing clearance most of the time. At such low eccentricity ratios, the attitude and position angles can become unimportant.

The hydrodynamic behavior of the externally pressurized bearing can cause fluid-induced instability problems. The hydrodynamic pressure wedge produces a tangential force component similar to that in an ordinary, plain cylindrical, internally pressurized bearing. In relatively low-speed turbomachinery applications, it is possible to defeat this circumferential flow problem by antiswirl injection. At the time of this writing, considerable research is taking place on how to eliminate circumferential flow and ensure stability in high speed turbomachinery applications.

Damping in externally pressurized bearings is more independent of stiffness than in hydrodynamic bearings. When oil is used as a lubricant, damping can be largely controlled by changing lubricant temperature (viscosity). Spring stiffness is largely independent of damping in these bearings.

Stiffness and Modal Damping in Fluid-Film Bearings

Compared to rolling element bearings, all fluid-film bearings have inherently high damping. The damping force produced by a fluid-film bearing is proportional to the amplitude and frequency of vibration in the bearing. However, even if a bearing has a high damping coefficient, if the amplitude of vibration in the bearing is low, little damping force will be generated. To generate a significant amount of damping force, the shaft must laterally move in the bearing; the amount of motion will depend on the mode shape.

Modal damping is the damping that is actually available for a particular mode, and it is inversely related to the observed Synchronous Amplification Factor *for that mode*: a low SAF produces high modal damping. High modal damping will occur when nodal points are located well away from bearings; conversely, when nodal points are located inside a bearing, little or no shaft motion can take place, resulting in low modal damping.

Rigid body modes (high shaft stiffness relative to bearing stiffness) tend to have nodal points located well away from the bearings, relatively large vibration amplitudes inside the bearings, and high modal damping. For this reason, rigid body modes are often overdamped (supercritically damped); machines will not exhibit a visible balance resonance associated with overdamped modes during startup or shutdown. Rigid body modes that are underdamped still tend to have relatively high modal damping with low Synchronous Amplification Factors.

Flexible rotor modes (low shaft stiffness relative to bearing stiffness) tend to have nodal points closer to or inside the bearings and lower vibration amplitudes inside the bearings. High bearing stiffness, by constraining rotor motion, tends to move nodal points closer to the bearing. (This is easy to understand if you imagine tightly squeezing the rotor at the bearing. The tighter you squeeze, the smaller the allowable motion at that point.) Nodal points near the bearing reduce shaft vibration in the bearing, decrease interaction with the lubricating fluid, and result in low modal damping. Thus, bearing stiffness, because of its effect on the location of nodal points relative to bearings, has a powerful effect on modal damping. Small changes in nodal point location can have a dramatic effect on modal damping. It is not sufficient to have bearings with high damping; the bearing stiffness must position nodal points in a way that allows enough movement in the bearing to provide adequate *modal* damping. Often the best compromise design occurs when bearing stiffness is approximately equal to the shaft stiffness.

Variable Stiffness in Internally Pressurized Bearings

While internally pressurized bearings are passive devices, they have spring stiffness that is a strong function of eccentricity ratio (Figure 23-2). Spring stiffness is minimum near the center of the bearing (eccentricity ratio of zero) and increases dramatically near the bearing wall (eccentricity ratio of one). In horizontal machines with plain cylindrical bearings, operation near the center of the bearing can only occur when a bearing is very lightly loaded, either as a result of misalignment or a radial load that cancels the gravity load of the rotor; for example, partial arc steam admission in a steam turbine.

Rotor speed and radial load both affect the average shaft centerline position of the journal within the bearing. The dependence of stiffness on eccentricity ratio means that it also depends on speed and load. As rotor speed increases, the dynamic pressure in the fluid film also increases, causing an increase in stiffness and a change in rotor position. This is easily visible on an average shaft centerline plot as a rotor moves off the bearing during startup. At any given rotor speed, changes in the magnitude of the radial load will also produce changes in the rotor eccentricity ratio and changes in stiffness.

Stiffness changes will change the rotor dynamic behavior. The undamped natural frequency, ω_n, of a rotor system can be expressed in its simplest form as

$$\omega_n = \sqrt{\frac{K}{M}} \tag{23-1}$$

where K is the modal stiffness of the rotor system, and M is the modal mass. In this equation, K includes the modal effects of the series and parallel combination of all the stiffnesses in the rotor system: the shaft stiffness, bearing stiffness, bearing support stiffness, and foundation stiffness. If we assume that the support and foundation stiffnesses are essentially infinite (rigid), K can be expressed as the series combination of the bearing stiffness, K_B, and the modal shaft stiffness, K_S:

$$K = \frac{1}{\left(\dfrac{1}{K_S} + \dfrac{1}{K_B}\right)} = \frac{K_B}{\left(1 + \dfrac{K_B}{K_S}\right)} = \frac{K_S}{\left(1 + \dfrac{K_S}{K_B}\right)} \tag{23-2}$$

When the bearing stiffness is relatively low, as is usually the case for the lowest rigid body modes, it will control the stiffness of the combination, and K can be expressed by the middle equation in Equations 23-2. In this case, because K is a function of eccentricity ratio in the bearing, ω_n will also be a function of eccentricity ratio.

This effect can often be observed in the lowest, rigid body modes on a critical speed map, a plot of rotor system natural frequencies versus bearing stiffness (Figure 23-3). On these plots, the natural frequencies of these modes often show a strong dependence on bearing stiffness, while the natural frequencies of the higher, bending modes, where stiffness is controlled by the shaft stiffness, are

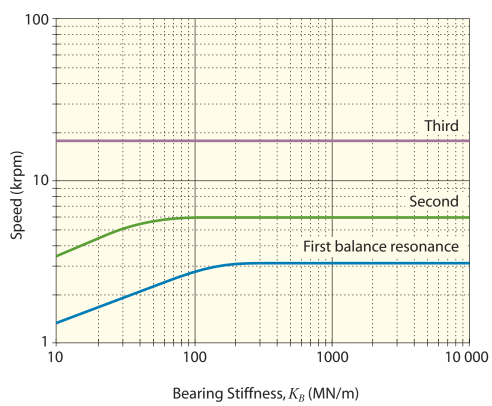

Figure 23-3. Typical critical speed map. The map shows how rotor system balance resonance speeds (vertical axis) change with different values of bearing stiffness (horizontal axis). Several modes are shown. The lowest modes tend to be more sensitive to changes in bearing stiffness than higher modes.

relatively insensitive to changes in bearing stiffness. This last situation can be represented by the rightmost equation in Equations 23-2.

These stiffness effects are responsible for the observed behavior of whirl and whip during fluid-induced instability (Figure 23-4). During whirl, the frequency of the instability vibration tracks rotor speed because, as the subsynchronous orbit diameter increases, the dynamic eccentricity ratio increases, which increases the bearing stiffness and the natural frequency. In the speed range where whirl is taking place, the bearing stiffness is significantly less than the shaft stiffness, the middle equation of Equations 23-2 applies, and changes in eccentricity ratio and bearing stiffness cause a change in rotor system natural frequency. When the eccentricity ratio becomes high enough, the bearing stiffness becomes significantly larger than the shaft stiffness, and the system transitions to whip. In whip, the natural frequency is constant because the shaft stiffness is significantly less than the bearing stiffness, and the shaft stiffness controls the combination stiffness, the right equation of Equations 23-2, and cannot be changed.

At very high eccentricity ratios, where the bearing stiffness is higher than the shaft stiffness, the natural frequency of that mode is relatively high compared to the natural frequency at lower eccentricity ratios; this natural frequency is called the *high-eccentricity natural frequency*. The high-eccentricity natural frequency is approximately the same as the balance resonance frequency documented by the manufacturer, the *nameplate critical*. Most often, in large horizontal machines, the high-eccentricity natural frequency is observed in conjunction with shaft bending modes.

If, for some reason, the lightly loaded rotor operates at an abnormally low eccentricity ratio, the bearing stiffness and natural frequency will also be relatively low, the *low eccentricity natural frequency*. It is only likely to be encountered during abnormal conditions, where the bearing operates below design load.

Thus, a particular balance resonance frequency may exist in a *frequency band* that depends on the operating conditions of the machine. When the machine is operating normally, the journals operate, by design, at moderate to high eccentricity ratios. (This discussion pertains primarily to plain cylindrical journal bearings. Journals supported by tilting pad bearings tend to operate at lower eccentricity ratios.) When the machine is significantly misaligned or subjected to unexpected radial loads, the journals operate at very low or very high eccentricity ratios. If the machine has primarily rigid body modes, where the bearing stiffness is the weakest spring in the system during normal operating conditions, then these rigid body natural frequencies may actually occur at speeds from significantly below to above their nameplate criticals. Systems with

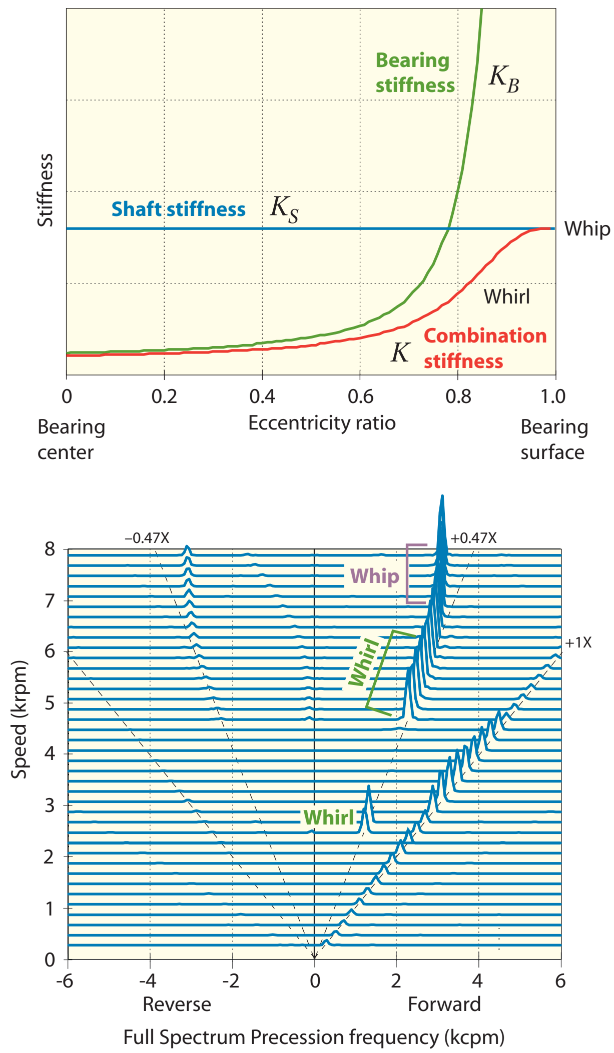

Figure 23-4. Bearing stiffness, shaft stiffness, and whirl and whip. The upper plot shows stiffness as a function of eccentricity ratio. The combination stiffness (red) is always less than its weakest component, bearing stiffness (green) or shaft stiffness (blue). As journal eccentricity ratio increases during whirl, the overall stiffness increases until it becomes approximately equal to the shaft stiffness. At this point, the whirl transitions to whip and locks to a single natural frequency controlled by shaft stiffness (lower plot).

flexible rotors, where shaft stiffnesses control resonance frequencies, will show a much smaller change.

Variable Stiffness in Externally Pressurized Bearings

The stiffness of externally pressurized bearings includes a combination of static and hydrodynamic effects. The static component of stiffness depends on the supply pressure; the diameter of the orifices; the number, area, and shape of pockets; the bearing radial clearance; and, to a lesser extent, the eccentricity ratio. The hydrodynamic component of stiffness depends on eccentricity ratio. At low eccentricity ratios, the combination of these effects produces an overall bearing stiffness that is higher than that of an internally pressurized bearing and is also a function of eccentricity ratio.

The externally pressurized bearing has an additional, important advantage. For any given physical bearing geometry, the static component of stiffness can be controlled by varying the supply pressure; higher pressure produces higher stiffness. The externally pressurized bearing also provides the opportunity to put bearing stiffness under either operator or automatic control. Control is accomplished by statically or dynamically adjusting pocket pressures in groups or individually.

In its most basic form, the externally pressurized bearing operates in a completely passive mode, with a preset operating pressure applied to all the inlet orifices. In this case, each pocket will produce a uniform, constant hydrostatic stiffness component around the bearing.

The bearing can also be operated in a semiactive mode to provide variable stiffness under either automatic or operator control. In this operating mode, a single valve can be used to change all the pocket pressures simultaneously when desired. When bearing stiffness is comparable to or less than shaft stiffness, variable stiffness control gives the operator the opportunity to shift the balance resonance frequencies to different speeds. The amount of the frequency shift will depend on how much influence the bearing stiffness has over the system natural frequency for that mode. Under some conditions, the resonance can be moved rapidly through running speed, and some or most of the vibration amplification associated with passage through a resonance can be avoided.

If pocket pressures are adjusted individually by separate control valves, it is possible to produce a change in average shaft centerline position. For example, given a radial load which moves the journal away from the center of the bearing, if the pressure is increased in pockets opposite the load and decreased in pockets on the same side of the load, the net force will push the average shaft centerline of the journal back to the bearing center. This can be done by sensing the position of the journal, comparing it to a predefined set point, and using an automatic control system to make the adjustments. Because the average of the individual pocket pressures does not change, the overall bearing stiffness remains approximately the same.

Finally, the bearing can be operated in a fully active mode, where rotor position information is provided to an automatic feedback control system that continuously adjusts the individual pocket pressures and the forces applied to the journal. The type of control used determines the effects of these pressure changes. The use of proportional control can provide the equivalent of a rotating stiffness, derivative control can provide synthetic damping, and integral control can be used to adjust the dc offset to move the rotor to the desired average set point.

The fully active bearing will suppress vibration (for example, in the event of sudden blade loss), and the control signal can be used to provide input to a diagnostic system.

Rotor Dynamic Implications of Variable Stiffness Bearings

Changing bearing stiffness changes how the rotor system responds to the static and dynamic forces that exist in the machine. The effects of variable bearing stiffness depend on the ratio of bearing stiffness to shaft bending stiffness. In the following discussion, we assume a shaft with stiffness K_S is supported by two variable stiffness bearings with total stiffness K_B. The modal stiffness depends on the combination of the shaft stiffness and the deflection mode shape of a particular mode. Modal damping depends on both the amount of damping in the bearings and the locations of nodal points relative to the bearings. These effects can be divided into three general cases. The following discussion is qualitative in nature; particular machines may behave differently.

$K_B << K_S$. The bearing stiffness is significantly less than the shaft stiffness. In this situation, the rotor is effectively rigid, and bearing stiffness controls the rotor system natural frequency (the middle equation of Equations 23-2). Changing bearing stiffness will have a relatively strong influence on the rotor system natural frequency. Nodal point locations will remain far from bearings, producing relatively small changes in modal damping.

$K_B \approx K_S$. Bearing stiffness is comparable to shaft stiffness. Changes in bearing stiffness will have some effect on the rotor system natural frequency. Stiffness changes will move nodal points and affect modal damping.

$K_B >> K_S$. The bearing stiffness is significantly greater than the shaft stiffness. In this situation, the rotor is flexible, and shaft stiffness controls the rotor system natural frequency, which will be relatively insensitive to changes in bearing stiffness. With stiff bearings, rotor nodal points will be close to the bearings, and stiffness changes will have a large influence on nodal point locations relative to the bearing. As the bearing stiffens, nodal points will approach and enter the bearings, shaft vibration in the bearings will decrease, modal damping will drop, and amplification factors at resonance will increase. Similarly, reducing bearing stiffness will increase shaft vibration in the bearings, increase modal damping, and reduce amplification factors at resonance. Thus, for this case, variable stiffness will have more of an effect on modal damping than on natural frequency. Note that the American Petroleum Institute recommends that $K_B \leq 4K_S$ to avoid high amplification factors at resonance [1].

Figure 23-5 shows, when $K_B \approx K_S$, how the balance resonance frequency can be changed by changing bearing stiffness. The left Bode plot displays the vibration response of the simple rotor model of Chapter 10 for three ratios of K_B/K_S. This model is not sophisticated enough to show the effects of changes in nodal point location. The value of K was calculated using Equations 23-2, and the values of mass and damping were unchanged. The resonance frequencies increase with increasing stiffness ratio. The Synchronous Amplification Factor is affected somewhat by the increase in stiffness, even though the modal damping was not changed.

The right Bode plot displays the vibration behavior for the same three stiffness ratios, using a finite element rotor model. This model is sophisticated enough to show nodal point location effects on modal damping. The change in amplification factor is much more dramatic because the stiffness changes produced changes in the nodal point location and, hence, in the modal damping. The change in modal damping is also revealed by the changes in the slope of the phase data.

For most machines, variable stiffness bearings will allow some adjustment of natural frequencies and mode shapes while the machine is running. The stiffness adjustment will also affect modal damping. Since spring stiffness and modal damping are a part of Dynamic Stiffness, changes in bearing stiffness will cause changes in vibration amplitude and phase (see Figure 23-5). Shifting the resonance away from running speed may cause dramatic changes in amplitude and phase.

At any particular speed, a change in bearing stiffness will change the influence vectors of the machine, which are closely related to the Dynamic Stiffness (Chapter 16). This has important implications, discussed below, for the repeatability needed for balancing.

Changes in bearing stiffness will also have an effect on the unbalance response of the rotor system. As we discussed in Chapters 16 and 18, 1X vibration depends on the amount, angular location, and axial distribution of unbalance along the rotor and *how well that distribution fits the rotor vibration mode* *shape*. If that mode shape is changed by adjustments in bearing stiffness, the unchanged unbalance distribution may fit the new mode shape better or worse than the original mode shape, increasing or decreasing the unbalance response of the rotor. Thus, when bearing stiffness is changed, 1X vibration will change for two reasons: the unbalance force distribution fit to new the mode shape and the new Dynamic Stiffness.

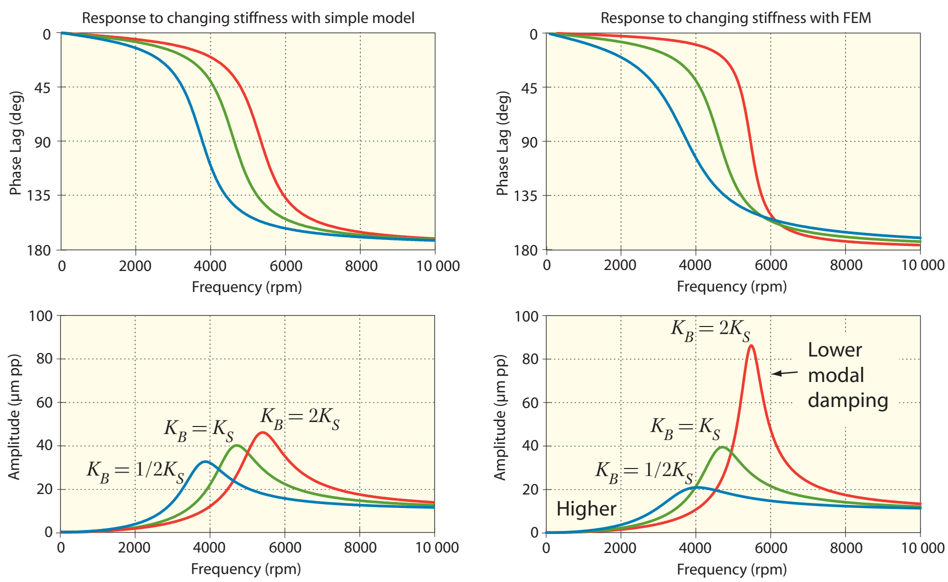

Figure 23-5. Variable stiffness effects on natural frequency and modal damping. The left Bode plot displays the vibration response of the simple rotor model of Chapter 10 for three ratios of K_B/K_S. The value of K is calculated using Equations 29-2, and the values of mass and damping are held constant. The resonance frequencies increase with increasing stiffness ratio. The Synchronous Amplification Factor is affected by the stiffness increase, even though the modal damping is not changed (the phase slope remains constant). The right Bode plot displays the vibration behavior of a more sophisticated finite element rotor model for the same three stiffness ratios. The model includes the effects of nodal point location changes. The change in amplification factor is much more dramatic because the stiffness changes produce changes in the nodal point location and hence in the modal damping.

Diagnostic Implications of Variable Stiffness Bearings

Changes in static spring stiffness are often the primary indicator of various malfunctions: rub increases stiffness, a shaft crack decreases stiffness, and misalignment can do either. Spring stiffness is a component of Dynamic Stiffness. We use the rotor behavior resulting from forces interacting with Dynamic Stiffness to interpret the state of a machine; for example, we deduce the heavy spot location for a particular mode by examining the Bode or polar plot data from a startup or shutdown. This vector data depends on the Dynamic Stiffness of the machine, which, in healthy machines with conventional bearings, remains approximately constant over speed and time.

However, variable stiffness bearings allow changes in spring stiffness that affect Dynamic Stiffness. Thus, changing bearing stiffness during a startup or after a machine reaches running speed alters the Dynamic Stiffness of the machine and the machine response. This has important implications for the diagnosis of machine malfunctions.

For example, the balance resonance is usually thought of as occurring at a fixed operating speed, where running speed coincides with a rotor system natural frequency. With an externally pressurized bearing, the natural frequency and balance resonance speed now become variables under the machine operator's control. By changing the bearing spring stiffness, the natural frequency can be quickly moved to another speed, enabling the operator or machine control system to jump the resonance rapidly through the machine during startup or shutdown. This will greatly alter, and possibly even eliminate, the usual balance resonance signature that is used to identify the heavy spot location in a polar or Bode plot (Figure 23-6). If a resonance is shifted to a different speed, then heavy spot/high spot relationships may have a different appearance. For example, what was above a resonance might now be below, or vice versa. Response that was out of phase might now be in phase. Influence vectors will depend on bearing settings, and bearing settings will have to be similar to provide repeatability of data.

Changes in bearing stiffness can also change the rotor mode shape. A mode associated with low bearing stiffness, for example, a rigid body mode, could be modified by higher bearing stiffness to a bending mode. This change in mode shape could change the match to the unbalance distribution, producing a change in unbalance response. It is possible that the existing unbalance distribution would become a better match to the new mode shape and that the rotor would have to be balanced specifically at particular bearing settings.

Some malfunctions manifest themselves as a self-excited vibration at a system natural frequency. Because of the new, variable nature of the balance resonance, this natural frequency will exist somewhere in a frequency band, which will depend on the range of bearing settings and their effect on rotor modal stiffness.

Some malfunctions produce vibration at a system natural frequency. Subsynchronous rub vibration and fluid-induced instability whip are examples. It is a common diagnostic practice to look at a cascade plot to see if a malfunction vibration frequency is near a balance resonance frequency. With rub, the vibration frequency is often shifted above the original balance resonance speed because of the stiffening of the rotor due to the rub contact. With variable stiffness bearings, these relationships may be shifted farther than expected, and in a different direction. For example, the frequency of a resonance encountered with a stiff bearing setting may be higher than the frequency of a subsynchronous rub vibration with a softer bearing setting.

Under some circumstances, the bearing will allow the operator to move the natural frequency to a place where the malfunction vibration cannot occur. The diagnostician will need to understand how this kind of variable-parameter bearing operation will affect his or her interpretation of the data and how it can be

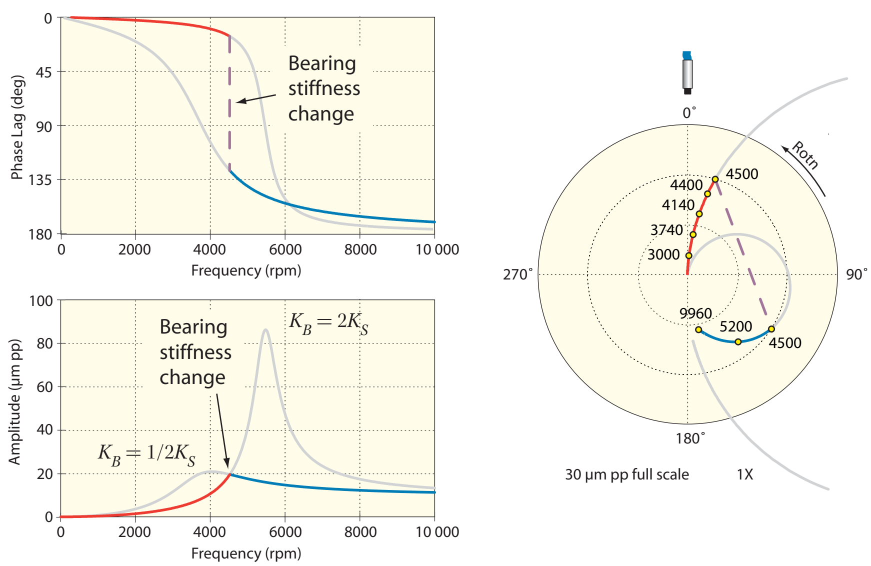

Figure 23-6. Effect of changing bearing stiffness during startup or shutdown. Finite element rotor model data shows the effect of suddenly changing bearing stiffness. The machine is started up with relatively high bearing stiffness (red). The bearing stiffness is suddenly lowered when the machine reaches 4500 rpm, changing the machine response (blue). A sudden phase change (dashed) is associated with the resonance moving through the machine, and the polar plot looks significantly different from the circular shape that would be expected with a normal bearing.

used to suppress unwanted vibration. For example, for ½X rub to occur, a machine must operate at a speed that is more than twice the natural frequency for the no rub condition; an externally pressurized bearing can be used to move the natural frequency associated with the ½X vibration to a higher frequency where this condition is no longer satisfied, eliminating the subsynchronous vibration.

Externally pressurized bearings, because of their high stiffness, will force the journal to operate near the center of the bearing clearance by default. Thus, average shaft centerline plots will show smaller changes during a normal start-up or shutdown. Variable stiffness, when used, will change the average shaft centerline position somewhat. Externally pressurized bearings can also be used to control the average shaft centerline position without significantly changing bearing stiffness. In either case, the appearance of average shaft centerline plots will be different from what would be expected with a conventional bearing.

All of the possible changes caused by variable stiffness will need to be kept in mind by the machinery diagnostician when examining data: Shifts in natural frequencies and resonances, including changes in amplification factors; misalignments between subsynchronous vibration frequencies and balance resonance frequencies; changes in mode shape with associated unbalance changes; and shaft centerline position changes.

Summary

The bearings used in large rotating machinery can be divided into two general types: rolling element bearings and fluid-film bearings. Fluid-film bearings can be further subdivided into internally and externally pressurized bearings.

Internally pressurized bearings support the rotor with the dynamic pressure of the lubricating fluid created inside the bearing by the action of the rotating journal. Internally pressurized bearings are normally partially lubricated. They are passive devices, and they have spring stiffness that is a strong function of eccentricity ratio.

An externally pressurized fluid-film bearing operates in a fully flooded condition by design. Fluid at relatively high pressure is supplied to a set of inlet orifices that restrict the flow and create a pressure drop, and pockets distribute the fluid pressure over parts of the journal. The stiffness of externally pressurized bearings includes a combination of static and hydrodynamic effects. The static component of stiffness can be controlled by varying the supply pressure; higher pressure produces higher stiffness.

The externally pressurized bearing also provides the opportunity to put bearing stiffness under either operator or automatic control. Control is accomplished by statically or dynamically adjusting pocket pressures in groups or individually. Changing bearing stiffness changes how the rotor system responds to the static and dynamic forces that exist in the machine. The effects of variable bearing stiffness depend on the ratio of bearing stiffness to shaft bending stiffness.

Variable stiffness can produce shifts in natural frequencies and resonances; changes in amplification factors; misalignments between subsynchronous vibration frequencies and balance resonance frequencies; changes in mode shape with associated unbalance changes; and minor shaft centerline position changes.

References

1. American Petroleum Institute, *Tutorial on the API Standard Paragraphs Covering Rotor Dynamics and Balancing: An Introduction to Lateral Critical and Train Torsional Analysis and Rotor Balancing*, API Publication 684 (Washington, D.C.: American Petroleum Institute, 1996).