In Pursuit of Better Bearings . . .

by Donald E. Bently

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Throughout Bently Nevada's history, I've tried to ensure that we consistently exceed expectations when it comes to the solutions we offer our customers. I'm gratified that our business has grown over the years, in large part because we not only meet our customers' needs, we exceed them. So when I say that our new ServoFluid™ Control Bearing is the most important development in bearing technology in the last 100 years, I'm well aware that such a statement sets some pretty lofty expectations. However, as I watch the results of our testing with this bearing technology, my enthusiasm continues to build. I am convinced that the ServoFluid™ Control Bearing will revolutionize the way machinery is designed, built, operated, and maintained.

What is ServoFluid™ technology? Very simply, it is a fully lubricated, high-pressure fluid (liquid or gas) bearing. It exhibits the positive attributes of hydrostatic, hydrodynamic, rolling element, and magnetic bearings, but without their drawbacks. Its features and advantages are numerous, and I'll touch upon them in this article. However, to fully appreciate this bearing and the way it revolutionizes the future of machinery, a brief review of other bearing technologies is appropriate.

Fluid-Film Bearings

Fluid-film bearings have historically been the choice of machinery designers for high-speed turbomachinery – particularly where large load-carrying capacity is required. An early problem with such bearings, however, was fluid-induced instabilities from the bearing's lubricant. So-called "whirl" and "whip" phenomena in bearings have been known about for at least the last 80 years. Early researchers published some unfortunate papers that concluded that pressurization and full 360-degree lubrication led to instability in bearings. While the combination of full lubrication and *inadequate* pressurization can exacerbate fluid instability, these papers gave birth to "conventional wisdom" suggesting that bearings should never be fully lubricated or supplied with lubricant at pressures above a few dozen psi (7-175 kPa). The conclusions reached in these papers effectively stopped people from experimenting with the application of higher pressure bearings and full lubrication on turbomachinery. This belief has, unfortunately, spread around the world with very few exceptions. The result is that today virtually every fluid-film bearing on turbomachinery uses partial lubrication and very low lubricant supply pressures.

> *"... our new ServoFluid™ Control Bearing is the most important development in bearing technology in the last 100 years ..."*

Remarkably, our own recent research on full lubrication and pressurization has led us to a conclusion that is exactly opposite to "conventional" beliefs regarding fluid-film bearings: by properly pressurizing a fully lubricated bearing, it exhibits characteristics that are far superior to partially lubricated, essentially non-pressurized designs. It is vastly more stable, is adjustable in the field, has better stiffness, enjoys virtually no circumferential fluid flow, and can operate with rotor eccentricities and attitude angles near zero.

Returning to our historical discussion of fluid-film bearings, even with the intentional "starvation" of the bearing lubricant and minimized pressurization, bearing instabilities continued to be a significant problem for machine designers and users. Partial lubrication and very low pressures are merely attempts to keep the lubricant from being "dragged" into motion around the entire circumference of the shaft, and thus limit the circumferential lubricant flow. Because these methods (partial lubrication with little or no pressurization) did not always work, other approaches were devised. Hence, the rise of "pressure dams," "lobed bearings," and other variations on bearing geometries designed to alter the lubricant's circumferential flow path and attempt to prevent whirling and whipping from occurring. Tilting pad bearings also appeared – again, a form of bearing discontinuity. While these approaches helped, they still did not eliminate the problems entirely. Furthermore, the use of mechanical "obstacles" to disrupt the lubricant's flow path led to greater fluid drag and subsequent frictional/mechanical losses in the machine.

Today, instability is far from being a "solved" problem in machinery. Indeed, our Machinery Diagnostic Engineers rank instability and excessive amplification factors as some of the more frequent rotor dynamic problems they encounter in the field. As I'll discuss in a few moments, the ServoFluid™ Control Bearing readily addresses these problems much better than conventional sleeve or tilting pad bearings.

> *"Remarkably, our own recent research on full lubrication and pressurization has led us to a conclusion that is exactly opposite to 'conventional' beliefs ..."*

As an aside, I've sometimes heard people say, "Fluid instabilities are impossible in our machines because we have tilting pad bearings." Actually, fluid instabilities are not impossible in tilting pad bearings, and we've seen this in the field. What we have found is that the axial locking rings in some of these tilting pad bearings can themselves give rise to whirl and whip. So-called "pad flutter" is another problem that is thought to be a fluid-induced instability. Thus, tilting pad bearings alone don't guarantee the end of fluid-induced instability problems, as some of our customers can attest.

Rolling Element Bearings

With improvements in materials and manufacturing methods, rolling element bearings have come a long way in the last 100 years. There continues to be a place for rolling element bearings. Mist-type lubrication methods are appropriate here – it is widely understood that over-lubrication on these bearings leads to premature failure. However, rolling element bearings have their own limitations such as:

- Limited design life
- Very poor damping leading to high amplification factors
- Poorer load-carrying capacity than fluid-film bearings of similar size
- Maximum rotational speed limitations

As such, they simply will not displace other bearing technologies, particularly for high-speed turbomachinery with large loads and large rotors.

Magnetic Bearings

Magnetic bearings have caught the attention of many people because they offer active rotor control and the elimination of lubrication systems. However, these bearings have suffered from very significant problems that outweigh their benefits and, consequently, have not enjoyed widespread success. I don't believe magnetic bearings will proliferate because of some very fundamental limitations:

- **The need for auxiliary bearings.** Rolling element bearings are used as a back-up measure ("catcher" bearings) if the primary magnetic support is lost. Loss of magnetic bearing control is catastrophic, resulting in damage to both the bearing windings and machine internals. For this reason, a "belt and suspenders" approach using redundant bearings has to be used. Thus, machines fitted with magnetic bearings also incorporate rolling element bearings which are "sacrificial" in the event of a magnetic bearing failure. Indeed, these auxiliary rolling element bearings can typically only sustain one or two magnetic bearing failures until they too must be replaced.
- **Inherently unstable.** The use of magnetic attraction force to hold the rotor in place is inherently unstable, and very sophisticated control systems must be used to overcome this inherent instability. This results in a very complex system that has poor default mechanisms and must instead rely on auxiliary bearings, as mentioned above.

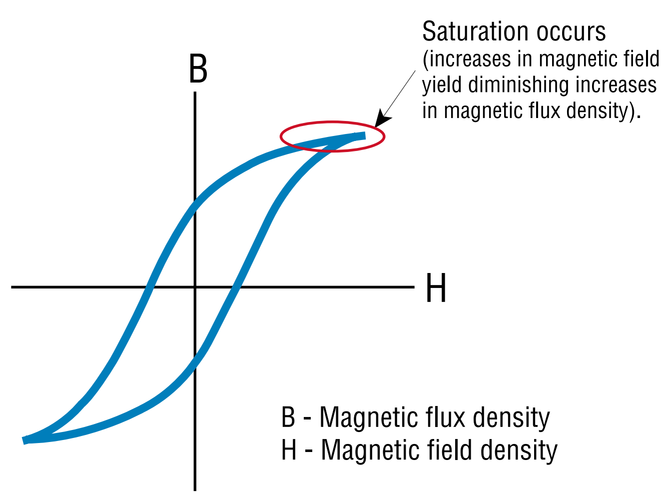


Figure 1. B-H curve showing magnetic saturation.

- **Heat.** Magnetic bearings can suffer from a sort of positive feedback mechanism where additional current is required to control the rotor, and the additional current flow in the windings creates more heat. As the windings heat up, they present more resistance to current flow, and it is necessary to push even more current through the windings. The heat continues to spiral upward and there is nothing inherent in the bearing's design that helps to counteract this effect or to remove heat from the windings. This is why some magnetic bearing installations have elaborate liquid cooling systems.

> *"Actually, fluid instabilities are not impossible in tilting pad bearings, and we've seen this in the field."*

- **Insufficient stiffness.** It takes a considerable amount of electric current and bearing area to generate strong restoring forces in a magnetic bearing. Magnetic bearing designers attempt to get better current-to-force conversion by using more core material in the bearing (to increase the magnetic permeability and subsequent magnetic field strength). However, this extra metal comes at a cost: increased system mass, which lowers the resonant frequency of the rotor dynamic system and is generally undesirable. Further, as the B-H curve in Figure 1 shows, some magnetic bearing designs allow saturation to occur. Very simply, there is a point at which additional current (proportional to magnetic field density H) does not translate to additional force (proportional to magnetic flux density B) and the bearing cannot "pull" on the rotor with sufficient force to keep it in place. Figure 2 shows this effect as a function of rotor eccentricity. We've superimposed the ServoFluid™ Control Bearing's performance for comparison. Notice that it provides higher restraining stiffness with increasing rotor eccentricities, unlike the magnetic bearing.

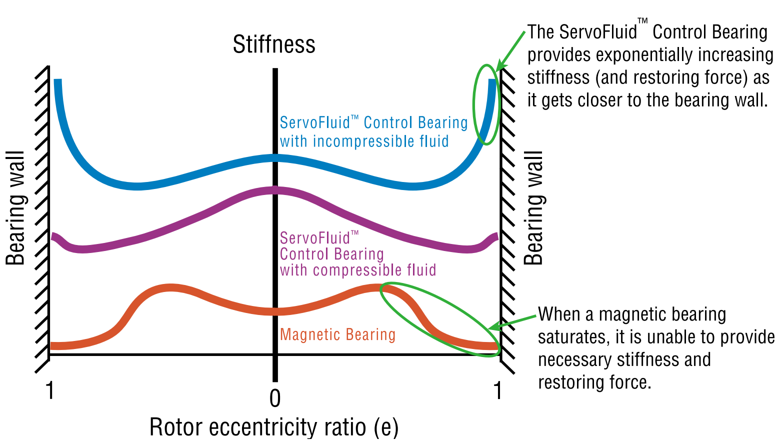


Figure 2. Relationship between stiffness and rotor eccentricity position for typical magnetic and ServoFluid™ Control Bearings.

Whether insufficient stiffness, limited lifespans, instability problems, the need for auxiliary bearings, or other problems, it is clear that none of the bearing technologies available in the past could claim to be ideal. It was the problems with conventional bearing technology – particularly instability – that propelled us in our search for a better bearing.

Foil Bearings

As the name implies, these bearings use foil and directed airflow to support a rotor. There are few to zero field applications of such bearings that I am aware of, and they seem to fall into the realm of "laboratory curiosity" only. If you know more about these bearing types and their application, I would be pleased to have you contact me so I can learn more.

The Relationship Between Stability and Dynamic Stiffness

The general expression for Dynamic Stiffness (excluding gyroscopic effects) is shown in Equation 1 below:

$$K_{DS} = \underbrace{K - \overbrace{M\omega^2}^{\text{mass inertia effect}} - \overbrace{M_f(\omega - \lambda\Omega)^2}^{\text{fluidic inertia effect (approximate)}}}_{\text{Direct Dynamic Stiffness (DDS)}} + j\underbrace{(D\omega - D\lambda\Omega)}_{\text{Quadrature Dynamic Stiffness (QDS)}} \qquad [1]$$

where:

K = modal stiffness (including radial rotor and fluid-film stiffness)

M = rotor modal mass

ω = perturbation frequency

M_f = fluid inertia coefficient (approximate)

λ = fluid circumferential average velocity ratio

Ω = shaft rotative speed

j = $\sqrt{-1}$

D = modal radial damping (primarily from the fluid film)

Very simply, instability occurs in a bearing when both Direct and Quadrature Stiffness terms become

Lambda (λ) – A Basic Understanding

When analyzing typical radial hydrodynamic bearings, the circumferential flow of the fluid (almost always oil in conventional hydrodynamic bearings) inside the bearing (between the journal and the bearing) and other important parameters of bearing operation are represented by these terms:

λ : Lambda, the fluid circumferential average velocity ratio

K_B : Fluid bearing radial stiffness term (lb/in)

D : Fluid radial damping term (lb•s/in)

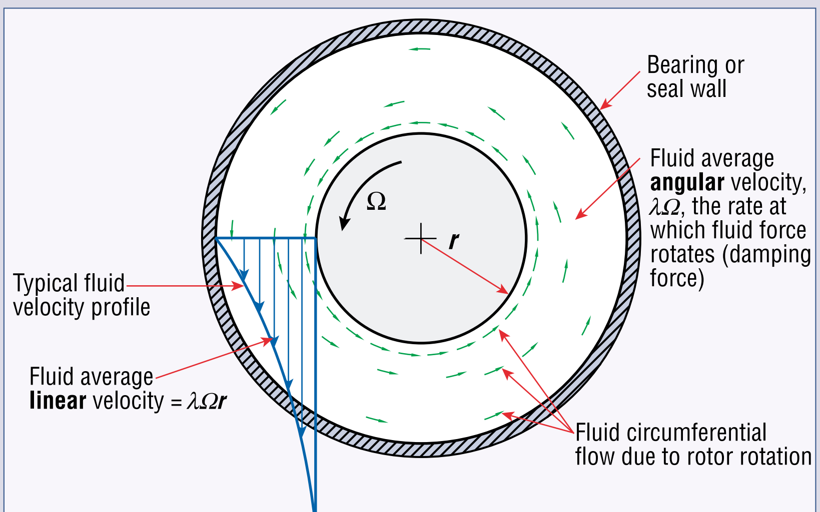
These terms are functions of many different parameters, but the most significant parameter is the eccentricity ratio, e. This parameter is observable in any shaft centerline plot which contains an accurate clearance circle.

The fluid bearing radial stiffness, K_B, and radial damping, D, increase with shaft eccentricity (as the rotor approaches the bearing wall), while λ decreases. This is intuitive since as eccentricity increases, the rotor moves closer to the bearing wall, reducing the clearance between rotor and bearing wall, and thus the circumferential fluid flow in the bearing. In effect, the fluid flow is "pinched off" and its average velocity decreases.

The figure below illustrates the concept of λ. At the rotating shaft, fluid is dragged into motion, rotating with the shaft. The fluid immediately adjacent to the shaft has its highest angular (rotational) velocity: namely, the shaft rotative speed (Ω). Because the bearing wall is stationary, the fluid immediately adjacent to the bearing wall has its lowest angular velocity (zero). This is noted in the fluid velocity profile in the figure. The **average** fluid angular velocity is therefore obviously between its two extremes: 0 (at the bearing wall) and Ω (at the rotating shaft). λ is simply a ratio: the fluid average angular velocity divided by the shaft rotative speed. This ratio (λ) is typically less than one-half, and plain, sleeve-type bearings usually exhibit values for lambda of around 0.42 to 0.48. Finally, it is important to note that the threshold of stability is inversely proportional to λ. This is why lower values of lambda are desirable in a bearing – they denote greater stability.

Most rotating equipment engineers have observed the relationship between eccentricity ratio and λ when fluid-induced instabilities occur and "quick fixes" are used. One of the most common fluid-induced instabilities is oil whirl. One method used to suppress oil whirl is to intentionally misalign the machine, sometimes called "friendly misalignment." Why does this often stop whirl? Simply because the rotor has been "aligned" closer to the bearing wall, which equates to an increased eccentricity ratio, increased stiffness, and a reduction in λ (see Figure 5a on page 38). This increase in stiffness stops the fluid whirling.

It is important to note that while these "fixes" may provide a temporary method of dealing with an instability, they do not address the root cause: namely, poor bearing designs, which are prone to instabilities. As always, Bently Nevada advocates treating the cause, not the symptom. Our ServoFluid™ Control Bearing is an excellent way to address the fundamental causes of instability in bearings. Its design creates exceptionally low values of lambda and results in bearings with unparalleled stability.



λ, fluid circumferential average velocity ratio

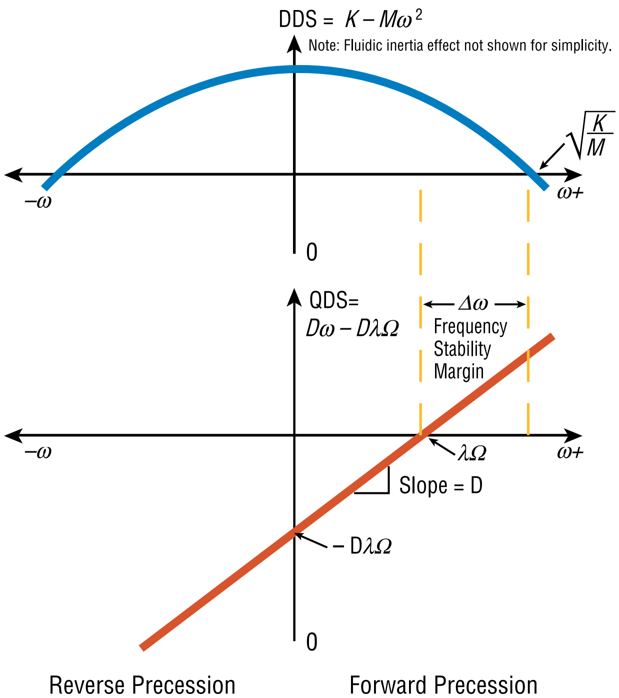


Figure 3. Frequency stability margin.

zero simultaneously. As shown in Figure 3, the separation $(\Delta\omega)$ between where Direct Stiffness becomes zero and where Quadrature Stiffness becomes zero is one measure of stability margin. Anything that can change in the machine and cause these two zero-crossings to coincide will result in instability. One might think that this never happens and can always be avoided by appropriate designs and safety margins with conventional bearings. However, it *does* occur – and more frequently than might be assumed. The reasons why are related to λ (lambda) and its affect on both the Direct and Quadrature parts of Dynamic Stiffness. More about this in a moment.

Returning to our Dynamic Stiffness equation, our Direct Dynamic Stiffness term represents the bearing's ability to supply a restoring force that acts counter to the applied force (load). This is desirable and represents the bearing's ability to carry load. Notice that our expression for Direct Dynamic Stiffness is not just K. It also contains a "mass effect" and a "fluidic inertia effect." Remember, we indicated that Direct Dynamic Stiffness was a measure of how well the bearing can "push back" in the same direction as the applied load. Under certain conditions, Direct Dynamic Stiffness can actually become dominated by the fluidic inertia effect. In this case, something called a "negative spring effect" occurs which means that the response pushes in the same direction as the load, rather than against the load. This is obviously undesirable. Thus, we think of the K term in Direct Dynamic Stiffness as desirable, but the fluidic inertia effect, $M_f\,(\omega - \lambda\Omega)^2$, as undesirable since it acts in the opposite direction from K. Notice that the fluidic inertia effect is a function of lambda and rotational speed (Ω) *squared*. Thus, larger values of lambda and higher speeds have a dramatic effect on the fluidic inertia effect, which in turn reduces the actual Direct Stiffness. This fluidic inertia effect has been called the "Coriolis Effect" or the "Bernoulli Effect" by some. I often refer to it as a "ghost" since it disappears the instant the fluid film is broken and becomes instead a partial or half Sommerfeld film, with mixed turbulent flow of gas and liquid. Since a gas bearing does not exhibit any fluidic inertia effect, I have come to the conclusion that fluidic inertia is simply a display of negative stiffness. It is notable that, unlike conventional fluid-film bearings, the ServoFluid™ Control Bearing does not appear to exhibit any fluidic inertia effect (even when a liquid, rather than gas, is used) because full 360-degree lubrication is maintained.

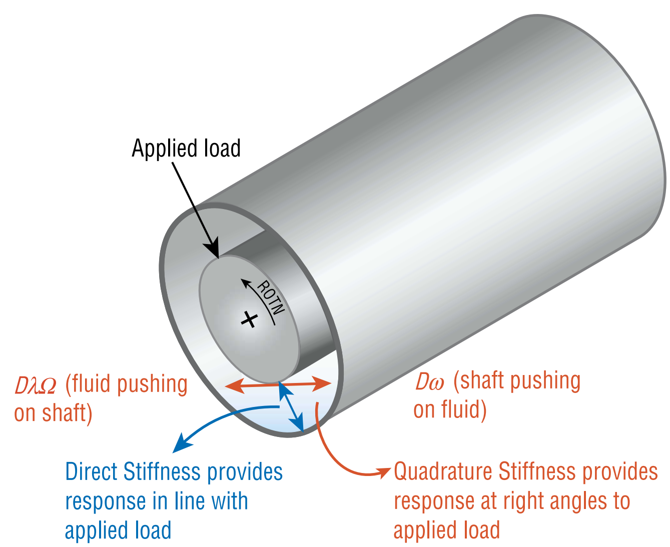


Figure 4. Directions in which response to an applied load occurs.

Now, consider the Quadrature Dynamic Stiffness term. As shown in Figure 4, Quadrature Dynamic Stiffness acts at right angles to the applied load. It effectively creates a moment on the shaft, attempting to push it circumferentially around the bearing clearance. One part of Quadrature Dynamic Stiffness – the $D\omega$ term – is a measure of the shaft's ability to push tangentially on the fluid wedge in a direction *against* rotation. This has a stabilizing effect. The other part – the $D\lambda\Omega$ term – is a measure of the fluid pushing tangentially on the shaft and acts in the *same* direction as rotation. It is the de-stabilizing part of Quadrature Dynamic

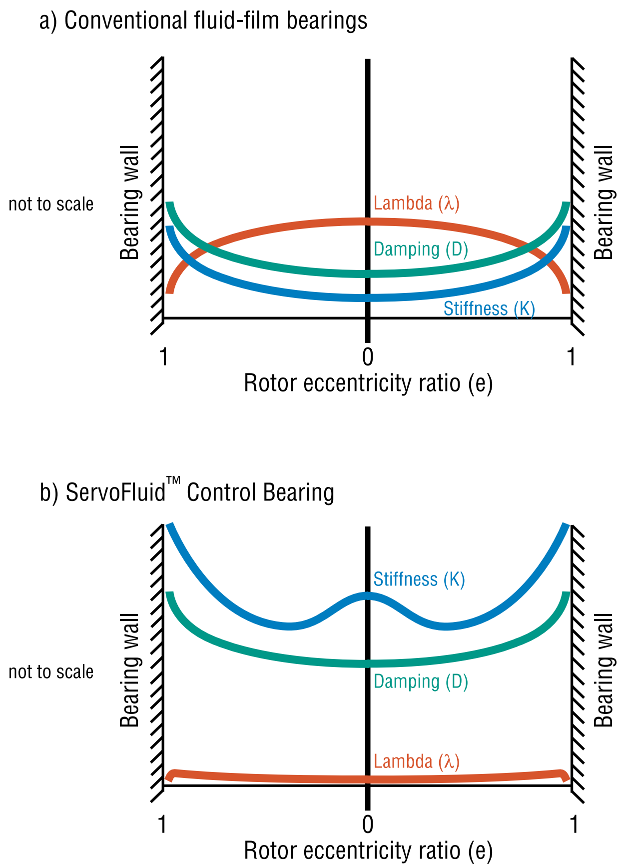


Figure 5. Various bearing characteristics as a function of rotor eccentricity position for: a) conventional fluid-film bearings; b) ServoFluid™ Control Bearing.

Stiffness. In simple terms, if the tangential force of the fluid pushing on the shaft becomes larger than the counter force of the shaft pushing back on the fluid, the fluid and shaft will begin a whirling motion around the inside of the bearing clearance, moving in the same direction as shaft rotation. While some Quadrature Dynamic Stiffness is generally desirable because it provides system damping, there is really only one part of Quadrature Dynamic Stiffness that we would like to keep – the $D\omega$ part. The $D\lambda\Omega$ part we would just as soon minimize or eliminate. How do we do this? By making λ (lambda) as small as possible.

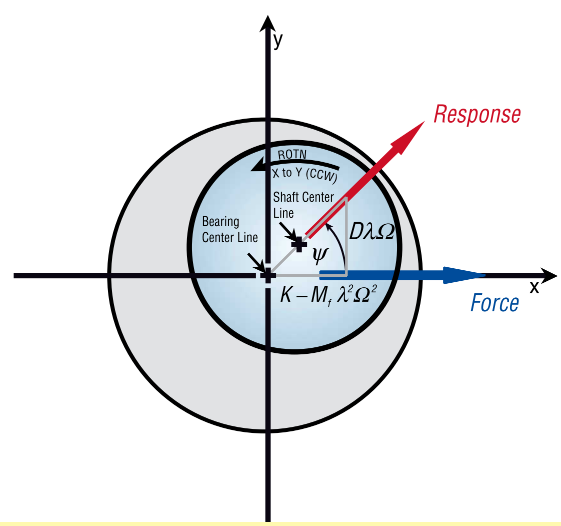
The Relationship Between Rotor Eccentricity and Stability

Figure 5a shows various bearing characteristics as a function of rotor eccentricity for conventional fluid-film bearings. Notice that desirable bearing attributes (high stiffness, low values of lambda, high damping) all require relatively large rotor eccentricities. However, as eccentricity approaches zero in a conventional bearing, these desirable attributes decrease, and as a result the bearing becomes more unstable. Indeed, as we have talked to people about our new ServoFluid™ Control Bearing and its ability to operate at very small rotor eccentricity positions, some have been almost horrified and remarked, "Mr. Bently, you *can't* run a bearing like that – it will be unstable." That's because they are thinking in terms of conventional bearings, not our ServoFluid™ Control Bearing.

> *"This relates directly to fewer losses and higher efficiencies – very important considerations as new machinery designs push for the highest efficiencies possible."*

Referring to Figure 5b, we have shown the same characteristics as in Figure 5a, but for the ServoFluid™ Control Bearing. You will immediately notice that excellent values of stiffness, lambda, and damping are all achieved at low rotor eccentricities. In fact, we have designed the ServoFluid™ Control Bearing to operate at typical rotor eccentricities of about 0.05 (depending on static loads), and under worst-case loading (both static and dynamic) conditions not to exceed 0.25. Even if loads exceed worst-case assumptions, the ServoFluid™ Control Bearing has a wonderful default mechanism … the closer it gets to the bearing wall (higher eccentricity), the more the bearing "pushes" back.

Why is it significant that we can achieve these characteristics at low rotor eccentricities? Because a rotor that can be



NOTE: Smaller attitude angles are indicative of bearings with greater stability. Please refer also to our ServoFluid™ Control Bearing brochure, included with this issue of ORBIT and available on our website at www.bently.com, for a graphical comparison of attitude angles in conventional bearings versus ServoFluid™ Control Bearings.

Figure 6. Attitude angle, ψ.

centered (and remain centered) within its bearing clearance – and whose rotor dynamic properties can be precisely controlled – translates to seals and other mechanical clearances within the machine that can be smaller and more precise. This relates directly to fewer losses and higher efficiencies – very important considerations as new machinery designs push for the highest efficiencies possible.

Attitude Angle – An Indicator of Stability

Earlier, we discussed the basic equation for Dynamic Stiffness and indicated that Direct Dynamic Stiffness was a measure of the bearing's ability to "push back" on the shaft in the same direction as the load. We also discussed a part of Quadrature Dynamic Stiffness – the $D\lambda\Omega$ component – that was shown to be destabilizing and therefore undesirable. The measurement known as the shaft attitude angle is, in effect, a measure of how much Direct Stiffness exists in a system relative to $D\lambda\Omega$.

Attitude angle (ψ) is defined as:

> The included angle between the direction of the vector sum of all the unidirectional, steady state, radial loads on a rotor and a line connecting the bearing and shaft centers.

Although I will not do so here, it can be shown that this angle is also equivalent to the relationship illustrated in Figure 6 and expressed by Equation 2 below:

$$\tan(\psi) = \frac{D\lambda\Omega}{K - M_f\,\lambda^2\Omega^2} \qquad [2]$$

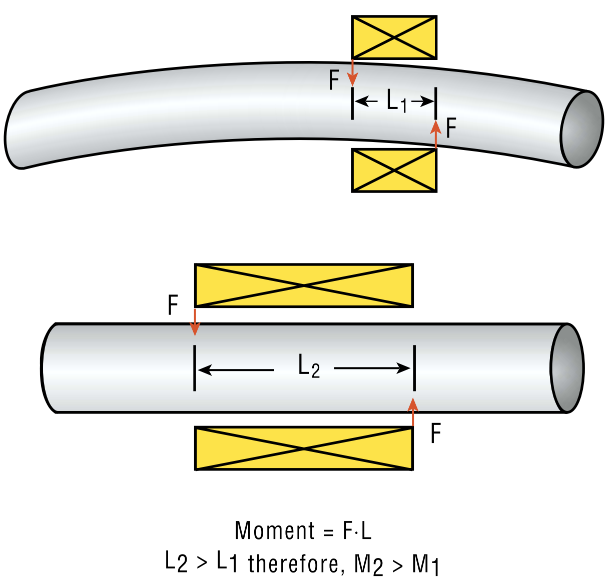


Figure 7. Longer bearings can create a stronger effective moment, thus stiffening a shaft considerably.

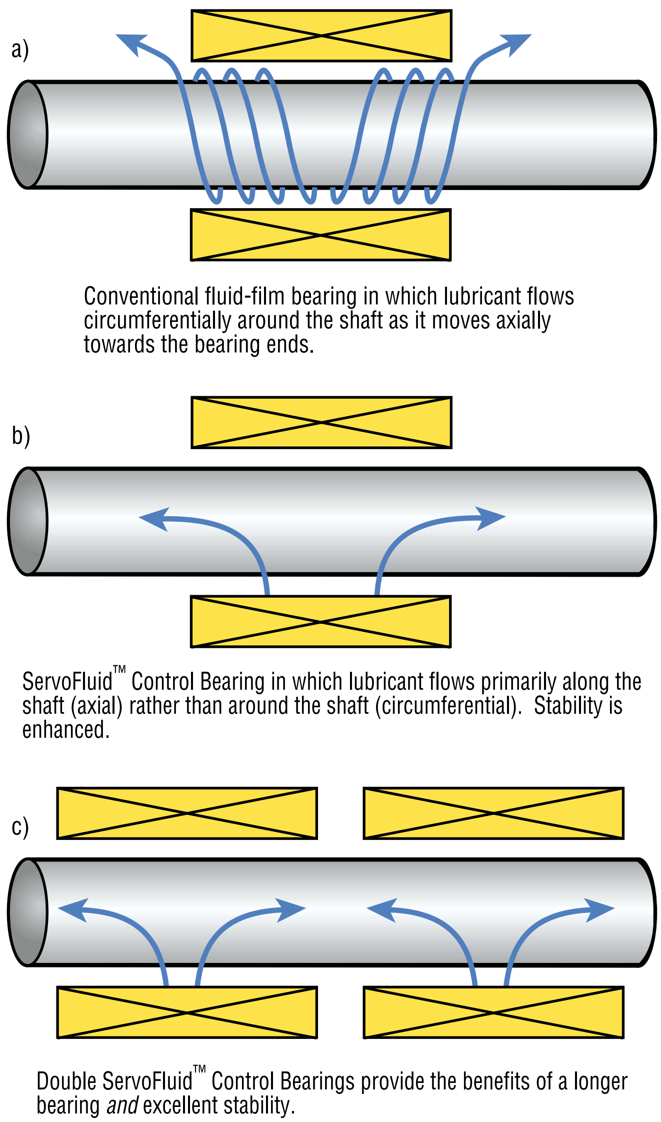


Figure 8. Comparison of bearing lubricant flow paths for: a) Conventional fluid-film bearing; b) Single ServoFluid™ Control Bearing; c) Back-to-back ServoFluid™ Control Bearings.

Thus, it can be seen that smaller values of K and/or larger values of $D\lambda\Omega$ are reflected in larger attitude angles. Typical fluid-film bearings have attitude angles of 40 to 60 degrees. In contrast, because the ServoFluid™ Control Bearing has large values of K (stiffness) and virtually no lambda (which results in very small values for the destabilizing terms $M_f\,\lambda^2\Omega^2$ and $D\lambda\Omega$) it also exhibits attitude angles very close to zero.

Length-to-Diameter (L-to-D) Ratios

It is well known that longer bearings (larger L-to-D ratios) are desirable because they can apply a moment to the shaft, keeping it straighter (see Figure 7). This moment has a dramatic effect by increasing the stiffness of the shaft. This reduces mid-span deflections and permits tighter clearances in seals and other machine elements. Large stiffness also results in raising the first balance resonance of the machine, theoretically allowing many machines to be designed that operate *below* their first balance resonance. However, there is a catch. It is also well known that longer bearings that use conventional fluid-film technology are more prone to instability. In fact, this is so prevalent that bearings with L-to-D ratios of greater than 0.5 are almost unheard of – they are simply too unstable.

One way to understand this relationship between bearing length and stability is to consider Figure 5a once again and recall that smaller rotor eccentricity positions result in lower stiffness, less damping, and higher values of lambda for conventional bearings. By making a longer bearing, we effectively increase the area over which it can support the rotor. This additional "support" tends to push the rotor nearer the bearing's centerline, which means it operates with a smaller eccentricity ratio. In a conventional bearing, we know that smaller eccentricity ratios are inconsistent with the bearing properties that enhance stability. Also, as shown in Figure 8, longer bearings simply mean that the lubricant (which is moving primarily in the circumferential direction around the shaft) has farther to travel axially before it can exit the bearing. This contributes to the bearing's instability since the fluid has more opportunity to start "swirling" around the shaft before exiting the bearing. As a result, to reduce the likelihood of instabilities, most bearings today observe L-to-D ratios of around 0.5, as previously mentioned.

> *"... stiffer shafts, reduced mid-span deflections, tighter clearances, and higher first balance resonances – we can see that they are all practical with the ServoFluid™ Control Bearing because longer bearings are no longer synonymous with instabilities."*

In contrast, the ServoFluid™ Control Bearing enjoys excellent characteristics and stability at very small eccentricities. As discussed earlier, this is because the ServoFluid™ Control Bearing has virtually no lambda (very little circumferential fluid velocity). Instead, it relies on axial lubricant flow, which forms a very different pressure wedge profile. [Editor's Note: Refer to ORBIT Vol. 21 No. 1, 2000, pp. 18-24 for a more extensive discussion of axial versus circumferential pressure wedges in fluid bearings.] This virtual absence of lambda in the ServoFluid™ Control Bearing permits much longer bearings, with L-to-D ratios up to 2.0 possible.

> "Perhaps the most exciting benefit of ServoFluid™ technology is that it makes possible a whole new generation of machines with adjustable rotor dynamic characteristics."

ServoFluid™ technology also permits the use of back-to-back radial bearings to provide benefits similar to that of a long bearing. Thus, returning to our original discussion of the benefits of longer bearings – stiffer shafts, reduced mid-span deflections, tighter clearances, and higher first balance resonances – we can see that they are all practical with the ServoFluid™ Control Bearing because longer bearings are no longer synonymous with instabilities.

Lubricant Considerations

Almost without exception, the fluid-film bearings in turbomachinery use some form of petroleum-based lubricating oil. While the ServoFluid™ Control Bearing can certainly operate with conventional lubricants, it can also operate with water and other incompressible fluids, as well as compressible fluids such as air, carbon dioxide, or nitrogen. This opens up numerous possibilities to choose a lubricant not just for its lubrication properties, but also for its compatibility with the process and any hazardous (explosion-prone) environments that might be present. In some cases, the process media itself can be used as the lubricant.

As we have presented the ServoFluid™ concept to customers, a frequent question that arises is the pressure ranges we can accommodate. Often, they want to know what we mean by "high pressure." We have found that most machines can be addressed with ServoFluid™ Control Bearings using pressures less than 1000 psi. In cases where more pressure is necessary to achieve adequate bearing characteristics, we can go as high as 2000 psi, but we don't believe such pressures would be required for typical machines. Our primary concern with extremely high pressures is that fluid velocities through the bearing ports could potentially become large enough to begin cutting or eroding the shaft. Thus, we have kept our pressure limitations extremely conservative.

Lubrication Services – Our Capabilities Go Further

The key to trouble-free lubrication can be summarized as "clean and dry." In other words, once the correct lubricant is selected, keep it free of contaminants and particles, and keep it free of water. As simple as this sounds, considerable expertise is required for a successful lubrication program.

While many companies, including Bently Nevada, provide services surrounding conventional petroleum-based lubricants, our capabilities go further. We can custom design applications of our ServoFluid™ Control Bearing in conjunction with alternative lubricants that are more compatible with your particular process or hazardous area requirements. Or, we can help you use conventional lubricants – whether for your existing bearings or our new ServoFluid™ Control Bearing.

Below, we've provided an example of a lubricant specification for a typical application of our ServoFluid™ Control Bearing where a conventional lubricant is used.

Example Oil:	ISO[1] 32 grade turbine oil:
TAN[2]:	Typical new oil range: 0.5-0.6
Alert limit:	new oil value + 0.2 (in this example, alert at 0.7-0.8)
Condemn limit:	new oil value +1 (in this example, condemn at 1.5-1.6)
Water:	100 ppm, maximum
Viscosity:	
Nominal:	32 cSt[3] @ 40°C
Alert Limits:	30-34 cSt @ 40°C (Alert limit is ±5% of nominal value)
Condemn Limits:	28-36 cSt @ 40°C (Condemn limit is ±10% of nominal value)
Cleanliness:	17/14/12 ISO4406 (1999, MTD[4])
Wear Metals[5]:	An increase of 5-10 ppm, or 100 % increase from baseline, whichever is larger, for each element.
Contaminant Elements[6]:	An increase of 10-20 ppm, or 100% increase from baseline, whichever is larger, for each element.
TBN[7]:	A decrease of 50% from new oil value. For example, if the new oil has a TBN of 12, condemn at TBN of 6.
Particle Count:	An increase of one or two ISO ranges, in any size category. In this case, an ISO result of 18/14/12 would be a signal to start cleanup, and a result of 19/14/12 should cause immediate cleanup action.
Trend Analysis:	Because of differences in oils, equipment types, environment, and service, no simple universal guideline can be established for determining the limits for metals in most oils. In most cases, trending specific, consistently measured data is more important than absolute numbers. Best results will be obtained using software appropriate to monitor the data for periodic changes.

1 International Standards Organization (ISO)

2 Total Acid Number – the quantity of base, expressed in milligrams of potassium hydroxide, required to neutralize all acidic constituents present in 1 gram of sample. (ASTM Designation D 974.)

3 The stoke is a unit of kinematic viscosity. The centistoke (1/100 stoke) (cSt) is commonly used to quantify typical lubricant viscosities.

4 Medium Test Dust – the new material standard for calibration of light-extinction-based particle counters. This material replaces the outdated ACFTD (AC Fine Test Dust) used by the previous standard.

5 Typical of this category are elements such as Fe, Cr, Pb, Sn, Al, Cu, Ni, etc., depending on the materials used in the construction of the machine's internal wear surfaces.

6 Typical of this category are elements such as Si, B, K, and Na, depending on possible contamination sources.

7 Total Base Number – the quantity of acid, expressed in terms of the equivalent number of milligrams of potassium hydroxide, required to neutralize all basic constituents present in 1 gram of sample. (ASTM Designation D 974.)

Another advantage of our bearing technology is that its fully lubricated design reduces foaming. Unlike partially lubricated bearings, it is more difficult for air to become entrained in the lubricant and create foaming. Consequently, the need for de-foaming agents or other additives to the lubricant may be greatly reduced.

Rotor Management

Until now, the rotor dynamic properties of a machine were something you simply "lived with." Making deliberate changes to the machine's rotor dynamic characteristics meant at least two things:

1. Removing the machine from service.
2. Physical modifications to the geometry of bearings, seals, blades, rotor(s), or support structures.

> *"While the ServoFluid™ Control Bearing can certainly operate with conventional lubricants, it can also operate with water and other incompressible fluids, as well as compressible fluids ..."*

Perhaps the most exciting benefit of ServoFluid technology is that it makes possible a whole new generation of machines with *adjustable* rotor dynamic characteristics. In other words, you can make adjustments to the machine without removing it from service and physically modifying the rotor and associated items.

Because it is very hard to manage an asset over which only limited control exists, past efforts at managing machinery condition could often identify and isolate problems, but could not do anything to alleviate the problems without stopping the machine and performing bearing and/or rotor modifications. The ServoFluid™ Control Bearing permits changes to rotor dynamic properties through relatively simple means, such as adjusting the lubricant supply pressure or viscosity (via temperature control). This bearing also allows independently adjustable damping and stiffness characteristics, providing more flexibility for machinery designers.

There are numerous reasons why adjusting rotor dynamic characteristics is useful. I'll list just a few here:

- Complex machines that can't be fully modeled on the drawing board. Often, this results in machines whose true rotor dynamic characteristics aren't known until they are installed in the field. Unfortunately, it is then too late to make modifications easily.
- Machines whose rotor dynamic characteristics change over time.
- Malfunctions for which short-term corrective action can be taken by adjusting the machine's rotor dynamic characteristics while waiting for a more opportune time to open the machine and perform maintenance.
- Compressors that are de-staged will change the rotor dynamics of the machine. The ability to compensate for these changes with adjustable bearings, rather than total redesigns, is highly desirable.

Conclusions

What started as a search for a more stable fluid-film bearing has resulted in a design that has exceeded even my expectations. In this article I've described our ServoFluid™ Control Bearing and its numerous benefits, comparing it to conventional technologies and elaborating on how it achieves its excellent performance. I'd be delighted to hear from you with your questions or with applications that you think might benefit from this new bearing. While the bearing lends itself very nicely to retrofit applications, I think its true potential will be realized through incorporation in new machine designs – machines that can be more efficient, more stable, more controllable, and enjoy better power-to-size ratios than their predecessors. As such, I would very much like to hear from machinery manufacturers and packagers about how this bearing could be used in your next generation of machinery.