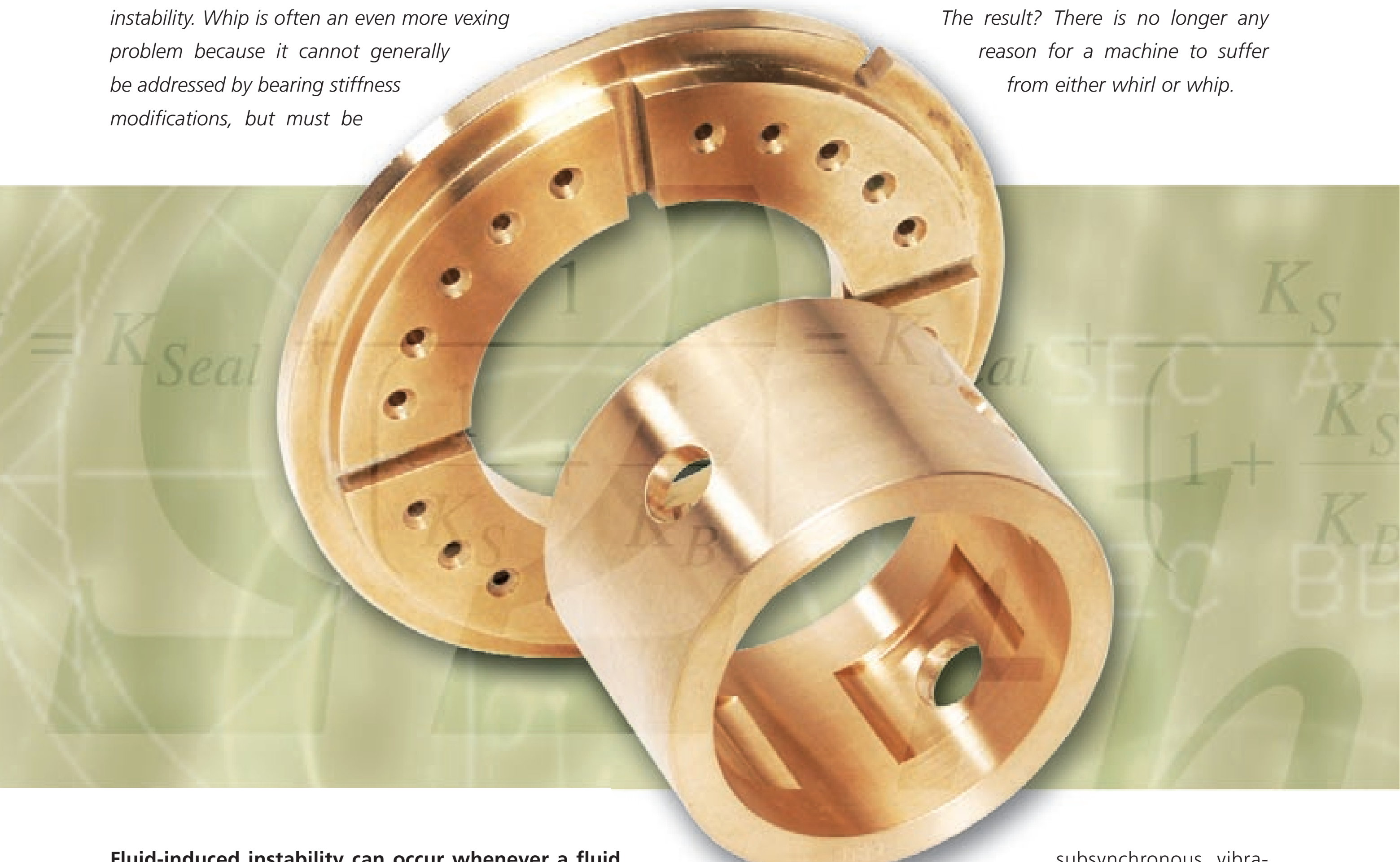
The Death of Whirl AND Whip

Use of Externally Pressurized Bearings and Seals for Control of Whirl and Whip Instability

Editor's Note: In the First Quarter 2001 issue of ORBIT, we featured an article titled "The Death of Whirl." While important, whirl is not the only form of fluid-induced instability. Whip is often an even more vexing problem because it cannot generally be addressed by bearing stiffness modifications, but must be addressed by rotor stiffness changes instead. In this article, we show how an externally pressurized bearing can be used as a mid-span seal to effectively eliminate whip. The result? There is no longer any reason for a machine to suffer from either whirl or whip.



Fluid-induced instability can occur whenever a fluid, either liquid or gas, is trapped in a gap between two concentric cylinders, and one is rotating relative to the other. This situation exists when any part of a rotor is completely surrounded by fluid trapped between the rotor and the stator, for example in fully lubricated (360° lubricated) fluid-film bearings, around impellers in pumps, or in seals. Fluid-induced instability typically manifests itself as a large amplitude, usually subsynchronous vibration of a rotor, and it can cause rotor-to-stator rubs on seals, bearings, impellers, or other rotor and stator parts. The vibration can also produce large-amplitude alternating stresses in the rotor, creating a fatigue environment that can result in a shaft crack. Fluid-induced instability is a potentially damaging operating condition that must be avoided.

During the 1980s, Don Bently and Dr. Agnes Muszynska showed that whirl and whip fluid-induced instabilities were actually manifestations of the same phenomenon, not separate malfunctions as previously believed. This groundbreaking work, modeling the two malfunctions in a single, harmonized, modern algorithm, is summarized in the April 1989 issue of *ORBIT* in the article "Fluid-Generated Instabilities of Rotors" [1].

"Both whirl and whip can be eliminated without introducing the disadvantages inherent in other commonly applied technologies."

Fluid-induced instability can originate in bearings or seals, but most often in fluid-film bearings; it appears suddenly and without warning, as the rotor speed reaches a particular threshold speed, which we call the Bently-Muszynska Threshold of Instability. Through rotor stability analysis, we can obtain a very useful expression for the Threshold of Instability, Ω_{th} :

$$\Omega_{th} = \frac{1}{\lambda}\sqrt{\frac{K}{M}} \tag{1}$$

where λ is the Fluid Circumferential Velocity Ratio, a measure of fluid circulation around the rotor, K is the rotor system spring stiffness, and M is the rotor system mass.

There is an important point regarding this equation: If the rotor speed is less than Ω_{th}, then the rotor system will be stable. Or to look at it another way, if Ω_{th} is above *the operating speed*, then the rotor system will be stable. Thus, to ensure rotor stability, all we have to do is keep the Threshold of Instability above our highest anticipated operating speed.

Rotor dynamic stability analysis itself is a separate topic, best performed using a technique called Root Locus Analysis, and covered in a previous *ORBIT* article [2]. Root Locus was pioneered by Walter Evans, a brilliant control systems engineer, and represents one of the most important contributions ever introduced to the field of control. It is covered in Evans' own text [3] as well as numerous other modern texts, two of which are noted at the end of this article [4,5].

Controlling Lambda

One way the Threshold of Instability is commonly raised is to reduce λ. You can see in Equation 1 that if we reduce λ, we will increase Ω_{th}. λ is a measure of the amount of fluid circulation in the bearing or seal, and it can be influenced by the geometry of the bearing or seal, the rate of end leakage out of the bearing or seal, the eccentricity ratio of the rotor in the bearing or seal, and the presence of any pre- or antiswirling that may exist in the fluid. Fluid-induced instability originating in fluid-film bearings is commonly controlled by bearing designs that break up circumferential flow. Examples of such bearings include tilting pad, lemon bore, elliptical, and pressure dam bearings. λ can also be reduced by antiswirl injection of fluid into the offending bearing or seal.

Controlling Stiffness

Fluid-induced instability can also be eliminated by increasing the rotor system spring stiffness, K. This article shows how an externally pressurized bearing or seal can be used to control fluid-induced instability by increasing K. First, however, we need to discuss how the various sources of spring stiffness combine to produce the symptoms of whirl and whip.

"To eliminate whirl, add stiffness at the bearing; to eliminate whip, add stiffness at or near the center of the shaft."

Spring Model

A flexible rotor can be thought of as a mass that is supported by a shaft spring, which is in turn supported by a bearing spring (Figure 1). Thus K actually consists of two springs in series, the shaft spring, K_S, and the bearing spring, K_B. For these two springs connected in series, the stiffness of the combination is given by these equivalent expressions:

SERIES SPRING COMBINATION WITH MASS

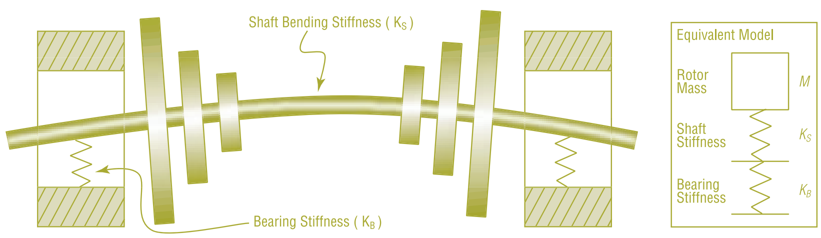


FIGURE 1

$$K = \frac{1}{\left(\frac{1}{K_S} + \frac{1}{K_B}\right)} = \frac{K_B}{\left(1 + \frac{K_B}{K_S}\right)} = \frac{K_S}{\left(1 + \frac{K_S}{K_B}\right)} \quad (2)$$

For any series combination of springs, the stiffness of the combination is always less than the stiffness of the weakest spring. The weak spring controls the combination stiffness. For example, assume that K_B is significantly smaller than K_S. Thus, K_S is much larger than K_B, and the middle equation can be used. As K_S becomes relatively large, K becomes approximately equal to K_B. For this case, the system stiffness, K, can never be higher than K_B; in practice it will always be less. A similar argument can be used with the rightmost equation when K_B is relatively large compared to K_S; the system stiffness will always be lower than K_B.

Eccentricity and Stiffness

Let's assume that the source of the fluid-induced instability is a plain, cylindrical, hydrodynamic bearing, an example of an *internally pressurized bearing*. Typically, when the journal is close to the center of the bearing (the eccentricity ratio is small), the bearing stiffness is much lower than the shaft stiffness. In that case, the ratio K_B/K_S is small, and the middle of Equation 2 tells us that the combination stiffness is a little less than K_B. In other words, *at low eccentricity ratios, the bearing stiffness is the weak stiffness* and it controls the combination stiffness.

On the other hand, when the journal is located relatively close to the bearing wall (the eccentricity ratio is near 1) the bearing stiffness is typically much higher than the rotor shaft stiffness. Because of this, the ratio K_S/K_B is small. Then, the rightmost of Equation 2 tells us that the combination stiffness is a little less than K_S. Thus, at *high eccentricity ratios, the shaft stiffness is the weak stiffness*, and it controls the combination stiffness.

Fluid-induced instability *whirl* begins with the rotor operating relatively close to the center of the bearing. The whirl vibration is usually associated with a rigid body mode of the rotor system (Figure 2, top). During whirl, the rotor system precesses at a natural frequency that is controlled by the softer bearing spring stiffness.

Whip is an instability vibration that locks to a more or less constant frequency. The whip vibration is usually associated with a bending mode of the rotor system (Figure 2, bottom). In this situation, the journal operates at a high eccentricity ratio, and K_B is much higher than K_S. K_S is the weakest spring

DIAGRAM SHOWING CONICAL WHIRL MODE SHAPE AT TOP AND BENDING WHIP MODE SHAPE AT BOTTOM

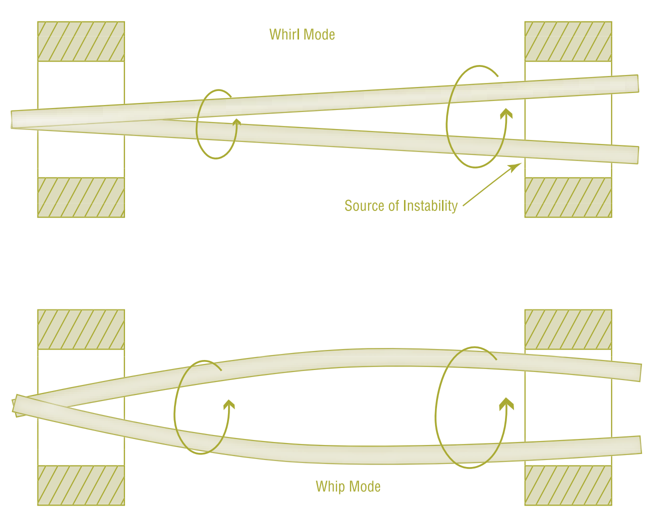


FIGURE 2

in the system, and it controls the natural frequency of the instability vibration.

To summarize, at low eccentricity ratios, the bearing stiffness controls the rotor system stiffness. Therefore, any changes in bearing stiffness will show up immediately as changes in the overall rotor system spring stiffness, K. On the other hand, at very high eccentricity ratios, the constant shaft stiffness is in control, and the overall rotor system spring stiffness will be approximately independent of changes in bearing stiffness.

Externally Pressurized Bearings and Seals

Conventional hydrodynamic bearings generate the rotor support force through the dynamic action of fluid drawn around by the rotation of the shaft journal. For that reason, they are called *internally pressurized bearings* (Figure 3). These bearings are normally designed to operate in a partially lubricated condition. They are vulnerable to fluid-induced instability problems because they can become fully lubricated if the shaft journal operates at a low eccentricity ratio, as can happen due to misalignment or an unanticipated high radial load.

CROSS SECTIONS OF HYDRODYNAMIC BEARING AND EXTERNALLY PRESSURIZED BEARING

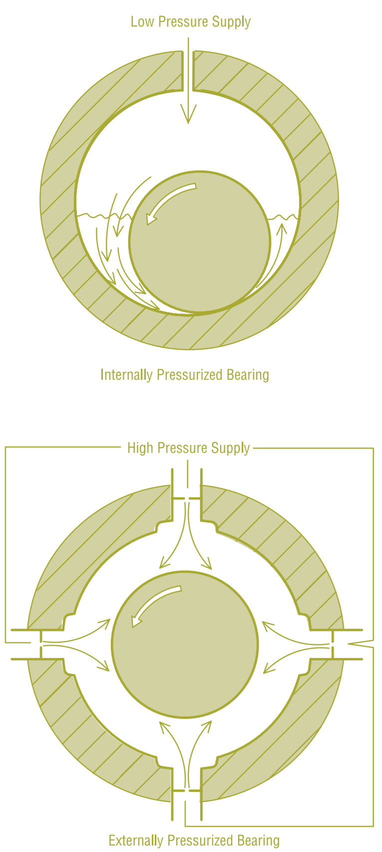


FIGURE 3

Externally pressurized bearings operate in a fully lubricated condition by design. The spring stiffness of these bearings strongly depends on the pressure of the lubricating fluid that is supplied to the bearing. This pressure is generated by an external pump; hence the name. By varying the pressure supplied to the bearing, it is possible to control the spring stiffness of the bearing, providing the possibility of *variable stiffness* control of a machine.

Seals can also act like bearings and have been responsible for triggering fluid-induced instabilities, even in machines supported by instability-resistant bearing designs, such as tilting pad bearings. Seals can also be externally pressurized with either gas or liquid, and they present the same possibilities for variable stiffness operation.

Ironically, the use of external pressurization in bearings occurs worldwide, especially in Europe, as many large machines use "jacking oil" to lift the rotor during startup, before rotative speed can develop a self-sustaining oil wedge. Unfortunately, it is widely believed that external pressurization at operating speeds degrades, rather than enhances, rotor dynamic stability. For this reason, the jacking oil pressure is removed once the rotor reaches an appropriate rotative speed. As this article will show, external pressurization of a properly designed bearing *enhances* stability and can *eliminate* whirl and whip. Thus, externally pressurized bearings and seals offer a new approach to the control of fluid-induced instability.

In whirl, the bearing stiffness is the weak stiffness of the system. We can increase the pressure of an externally pressurized bearing, increasing the bearing spring stiffness, K_B, and the system spring stiffness, K. The result is an increase in the Threshold of Instability, Ω_{th}. Thus, it is possible to design a rotor system using externally pressurized bearings that prevent fluid-induced instability whirl.

In whip, the bearing stiffness is very high, and the shaft stiffness, K_S, is the weak spring in the system. In this situation,

SIDE VIEW OF ROTOR SYSTEM WITH ADDITIONAL SPRING AT MIDSPAN SEAL LOCATION

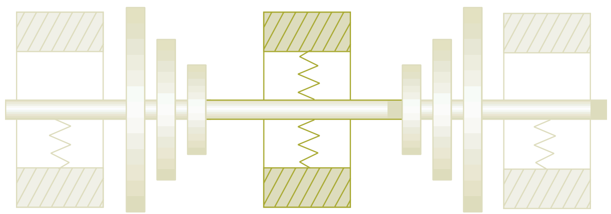


FIGURE 4

increasing the bearing pressure and stiffness *will have no effect* on the overall system spring stiffness, K. It is controlled by the shaft stiffness, which cannot be changed. However, we *can* add an additional spring to the system that acts *in parallel with* the shaft spring. This can be done by pressurizing a seal at or near the midspan of the rotor shaft (Figure 4). The total stiffness of two springs in parallel is simply the sum of the two stiffnesses. With an externally pressurized center seal with stiffness K_{Seal}, Equation 2 becomes

$$K = K_{Seal} + \frac{1}{\left(\frac{1}{K_S} + \frac{1}{K_B}\right)} = K_{Seal} + \frac{K_S}{\left(1 + \frac{K_S}{K_B}\right)} \qquad (3)$$

Thus, increasing K_{Seal} has the effect of directly increasing K and increasing the Threshold of Instability, Ω_{th}. Pressurizing the seal is equivalent to increasing the stiffness of the rotor shaft.

Both of these approaches to instability control follow the same basic principle: the stiffness of the weakest component is increased using externally pressurized bearing or seal technology. In whirl, the relatively weak bearing is stiffened; in whip, the relatively weak shaft is stiffened.

Externally pressurized bearings and seals offer one additional technical advantage. The working fluid can be injected radially or tangentially. If fluid is injected tangentially against rotation (*antiswirl injection*), λ is reduced. This has a direct and positive effect on rotor system stability. Antiswirl injection is easy to incorporate into an externally pressurized application, by design.

By applying all of these approaches in the design of a machine, it is relatively easy to produce a machine that is immune to fluid-induced instability problems. This has been demonstrated, both in the lab and in public venues (See page 47), by Bently Rotor Dynamics Research Corporation (BRDRC). Through the use of externally pressurized bearings and seals, both whirl and whip can be eliminated *without* introducing the disadvantages inherent in other commonly applied technologies or approaches such as tilting pad bearings, intentional misalignment (including gravity), and sleeve bearing variations such as tapered land, lemon bore, pressure dam, and others. These approaches introduce other problems (moving parts to wear out, greater mechanical losses, increased machine stress), and none are able to address whip instabilities. In addition, externally pressurized bearing/seal technology provides advantages far beyond the elimination of whirl and whip and can replace other bearing types such as magnetic and rolling element, overcoming their numerous disadvantages. It can be used to provide a variable stiffness machine, easily adjusted in the field manually or using automatic controls; it can be used with a variety of working fluids (gas or liquid), allowing "oil-free" operation, sometimes with the process gas itself; it allows greater use of vertical machine designs, rather than primary reliance on horizontal designs which use gravity as a stabilizing preload; and, it can be used at very slow rotational speeds since it does not depend on rotation to develop a supporting fluid wedge.

For more information on externally pressurized bearing technology from Bently Nevada, contact your local sales professional, or visit our Web site devoted to this technology at www.servofluid.com. ORBIT

References:

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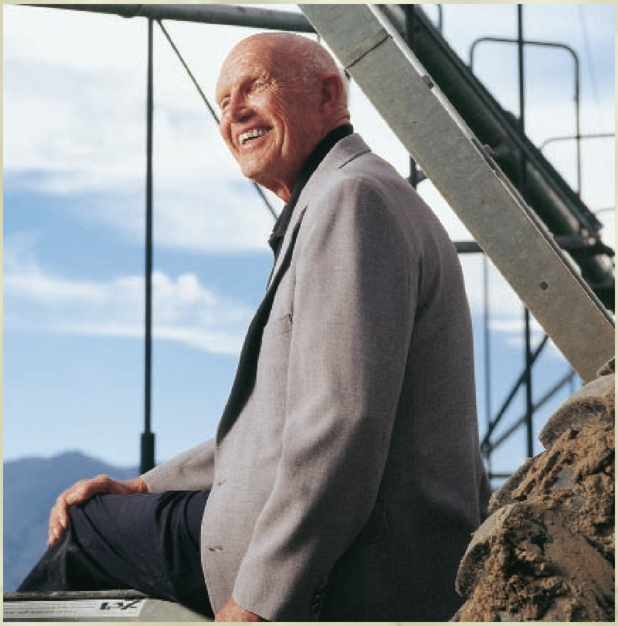
BENTLY PUBLICLY DEMONSTRATES WHIP CURE



Don Bently – Externally Pressurized Bearing Entrepreneur.

The use of our externally pressurized bearing technology to eliminate whirl and whip is an historic event, certainly among the most important rotor dynamic breakthroughs in the last 100 years. While external pressurization in bearings is not new – it dates to the year 1878 when an exhibit called the "Ice Railroad" was shown at the Paris Industrial Exhibition – its use to cure whirl and whip *is* revolutionary, and attributable principally to one man: Donald E. Bently.

At this year's ISCORMA stability conference, Bently Nevada took the opportunity to demonstrate the principles discussed in this article and publicly demonstrate the elimination of not just whirl instabilities, but whip as well. Externally pressurized bearing entrepreneur and nominee for the National Medal of Technology, Don Bently, was on-hand to answer questions and discuss the technology's implications for all aspects of rotor dynamic and machinery improvements – not just stability control. Read more about this historic event, and the biennial ISCORMA stability conference, at www.iscorma.com, as well as in our article beginning on page 40.