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Externally pressurized bearings may solve vibration problems

While rotating machinery design has made tremendous strides over the past 100 years, bearing design has essentially remained static. Even the mid-1990s excitement over magnetic bearings seems to have passed, leaving behind little commercial success. Now, one old powerplant pro plans to change the way we think about rotor dynamics, machinery control, and bearing design.

By Dr. Robert Peltier, PE, Senior Editor

It's the middle of the summer on-peak period, and the vibration level on your steam-turbine No. 2 bearing is slowly pushing up toward the alarm mark. The tech rep was here for the second time last week and couldn't find anything wrong, and the millwrights just finished realigning the turbine last night.

You've been to the training classes and understand that rotor-vibration amplitude is related to the fluid-film bearing stiffness, which is a function of the eccentricity ratio. You think there is something wrong with the rotor support stiffness that lowered the resonant frequency and increased the vibration level.

But the manufacturer's tech rep says this model turbine is notorious for having a natural frequency close to the operating speed, and the slightest change in alignment can have a profound impact on vibration level. His only recommendation: Keep the turbine aligned. Brilliant.

Unfortunately, you don't want to spend your time aligning turbines; you want to operate them—at least until the vibration level hits the alarm setpoint, and then what do you do?

Until now, the answer to this common plant conundrum was to call the maintenance team and start the alignment process all over again, missing significant revenues, paying for replacement power, and perhaps denominator among many stability problems seems to be that they tend to grow with time as the affected components begin to wear or fatigue.

But what if there were *active* rotor dynamic controls available that could change the bearing stiffness in response to a change in the shaft vibration level? What if, instead of picking up the phone to call the alignment crew, you could make a minor adjustment to the bearing lubrication supply pressure on your control station, and the vibration levels would drop toward zero?

Sound like science fiction? So did proximity transducers in the late 1950s, when Donald Bently began selling them out of his garage. But Bently's transistorized eddy-current proximity transducers were not only very real, they also gave birth to the entire machinery condition-monitoring industry.

Last year, the company that bore his name, Bently Nevada (Minden, Nev.), was sold to GE Power Systems (Schenectady, N.Y.), and the turbomachinery authority formed Bently Pressurized Bearing Co (Minden, Nev.) to focus on his new bearing designs (see box). According to Bently, "the externally pressurized designs enable machinery engineers to adjust the bearing's pressure, which in turn influences rotor dynamic response. Operating situations that previously mandated a shutdown to correct instability problems can be resolved by adjusting the bearing." In other words, high vibration in rotating equipment would

A rotor dynamics primer

The most commonly recurring problems in rotor dynamics are excessive steady-state synchronous vibration levels and subsynchronous rotor instabilities. Steady-state synchronous vibration levels may be reduced by:

- Improving machine balance/alignment.
- Modifying rotor-bearing systems—tune system-critical speeds out of the rpm operating range.
- Introducing damping to limit peak amplitudes at critical speeds that must be traversed.

paying for replacement power, and perhaps even damaging equipment.

Rotor-dynamic instabilities have become more and more common as the speed and horsepower of turbomachinery have increased. These instabilities can be erratic, seemingly increasing vibration amplitudes for no apparent reason. The common

Subsynchronous rotor instabilities may be avoided by:

- Raising the natural frequency of the rotor system as much as possible.
- Eliminating the instability mechanism (that is, change the bearing design if oil whip is present).
- Introducing damping to raise onset speed above the operating speed range.

become one less thing for the plant operator to worry about.

Bently boldly asserts, "This is the bearing for the 21st century."

A little bearing background

Most fluid-film bearings operate with low lubrication pressures, just enough to push some oil through the bearings and keep the metal temperatures within specification. This design approach is a direct holdover from early theories of fluid bearing behavior that assumed the load from the rotating shaft was directly applied to the stationary part of the bearing, and that the only role for lubrication was to decrease friction.

A Frenchman named Gerrard in the days of Napoleon invented pressurized bearings. Perhaps the earliest example of a pressurized bearing was the "ice railroad" shown at the 1878 Paris Industrial Exhibition. The exhibit consisted of a large four-legged iron structure sitting on a flat steel plate. Pressurized oil was pumped down all four legs until the structure could be moved almost without friction, like a train might move on rails of ice, hence "ice railroad." A more recent example is the popular arcade game of "air hockey," in which the table has a large number of equally spaced holes whose air jets make the puck virtually weightless.

Beauchamp Tower demonstrated in 1883 that the lubricant actually supplied the supporting force in a bearing. Osborne Reynolds' theoretical results followed three years later, explaining what Tower observed. Reynolds' milestone publication included the now-famous "Reynolds equation," which has challenged engineering students ever since, as well as the familiar mathematical model for oil-pressure distribution in a bearing.

In 1919, W.J. Harrison theorized that a fully lubricated hydrodynamic bearing would be unstable, and later researchers agreed, concluding that even moderate pressures in a fully lubricated bearing would cause instability. For more than 80 years Harrison's conclusions were unchallenged, and few researchers rigorously investigated the effects of higher pressures in fully lubricated bearings.

The standard industry design practice remains to this day as a partially lubricated, low-pressure (typically less than 25 psig per American Petroleum Institute standards) fluid-film bearing for high-speed turbomachinery. Commercial bearings rely on the relative motion between static and rotating surfaces to generate the fluid film. Turbomachinery instabilities remain, but now with pressure dams, offset designs, and tilting pads adding to bearing design and operational complexities. Some low length-to-diameter tilting pad designs actually reduce stiffness and the load-carrying capacity of the system.

Today we find pressurized bearings typically only in very slow-speed applications—such as automotive engines, positioning mechanisms for large telescopes, and some space-flight equipment. One unique high-speed application of pressurized bearings is dental drills.

Engineering fundamentals

Bearings that do not rely on the rotation of the journal to develop a fluid film are generally known as *hydrostatic* bearings—the film is present even with no shaft rotation. These bearings are capable of supporting large weights with the external oil pressure forming a supporting layer of lubrication in the bearing.

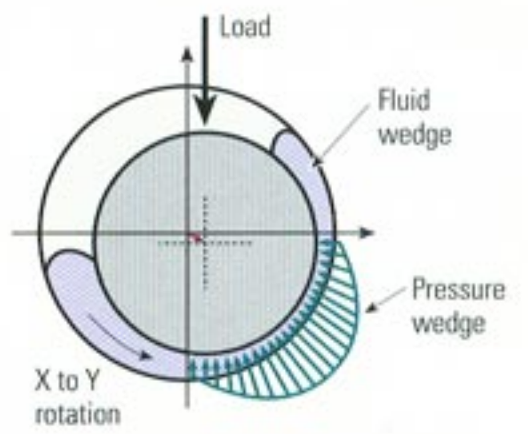
In contrast, *hydrodynamic* bearings rely on relative velocity between the rotating and stationary parts of the bearing to develop a self-sustaining oil wedge that supports the machine's shaft. For this reason, purely hydrodynamic bearings cannot be used for slowly rotating or stationary applications. With hydrodynamic bearings, there is no oil wedge formed unless the bearing's rotating and stationary parts are above some minimum rotational speed.

In reality, nearly all bearings exhibit both hydrostatic and hydrodynamic characteristics. The rotational speed and eccentricity define the degree of each characteristic that is present in any design.

A bearing-shaft system can be represented as a simple spring-mass system and modeled by the well-known equation:

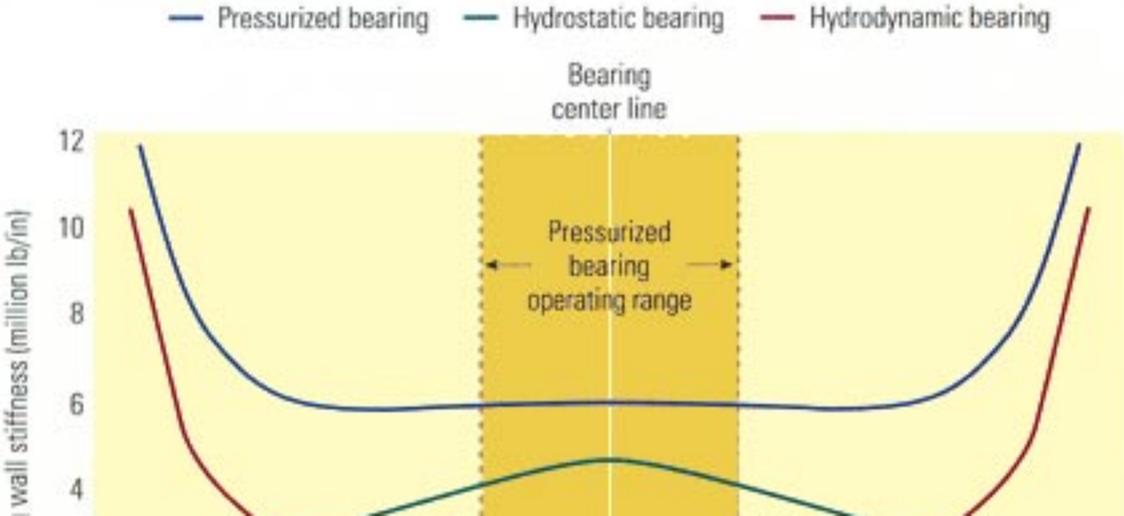
$$\text{Force} = \text{Stiffness} \times \text{Displacement}$$

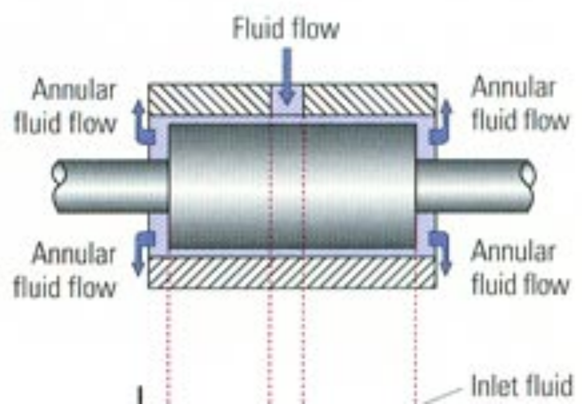
For a spring, there is only a single direction of motion, and the displacement is in the direction of the force, so each quantity

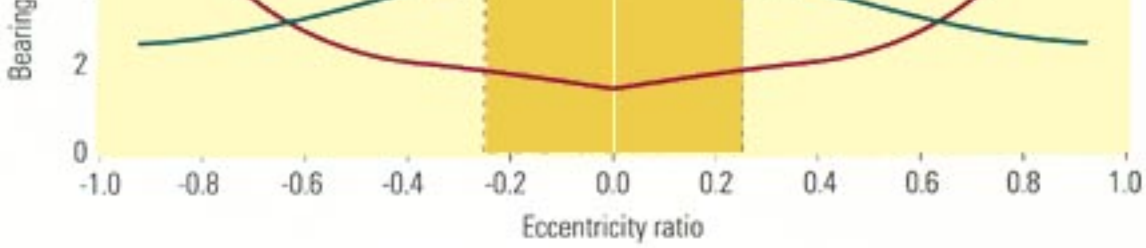


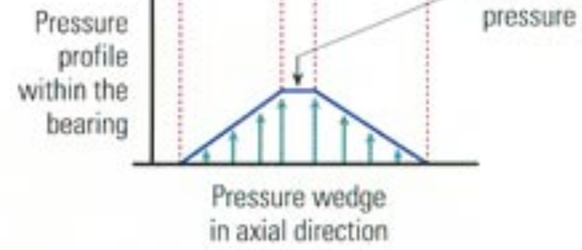
2. Hydrodynamic bearings

Typical hydrodynamic bearings form a fluid and pressure wedge in the circumferential direction.









1. Pressurized bearings

Pressurized bearings have a unique characteristic: the closer the shaft comes to the bearing wall, the more the bearing "pushes" back from the wall. Stiffness of the Bently pressurized bearing is a combination of the hydrodynamic and hydrostatic effects.

3. Pressurized bearings

The lubricating oil-pressure wedge in a pressurized bearing is maximum at the entry point and flows axially along the shaft.

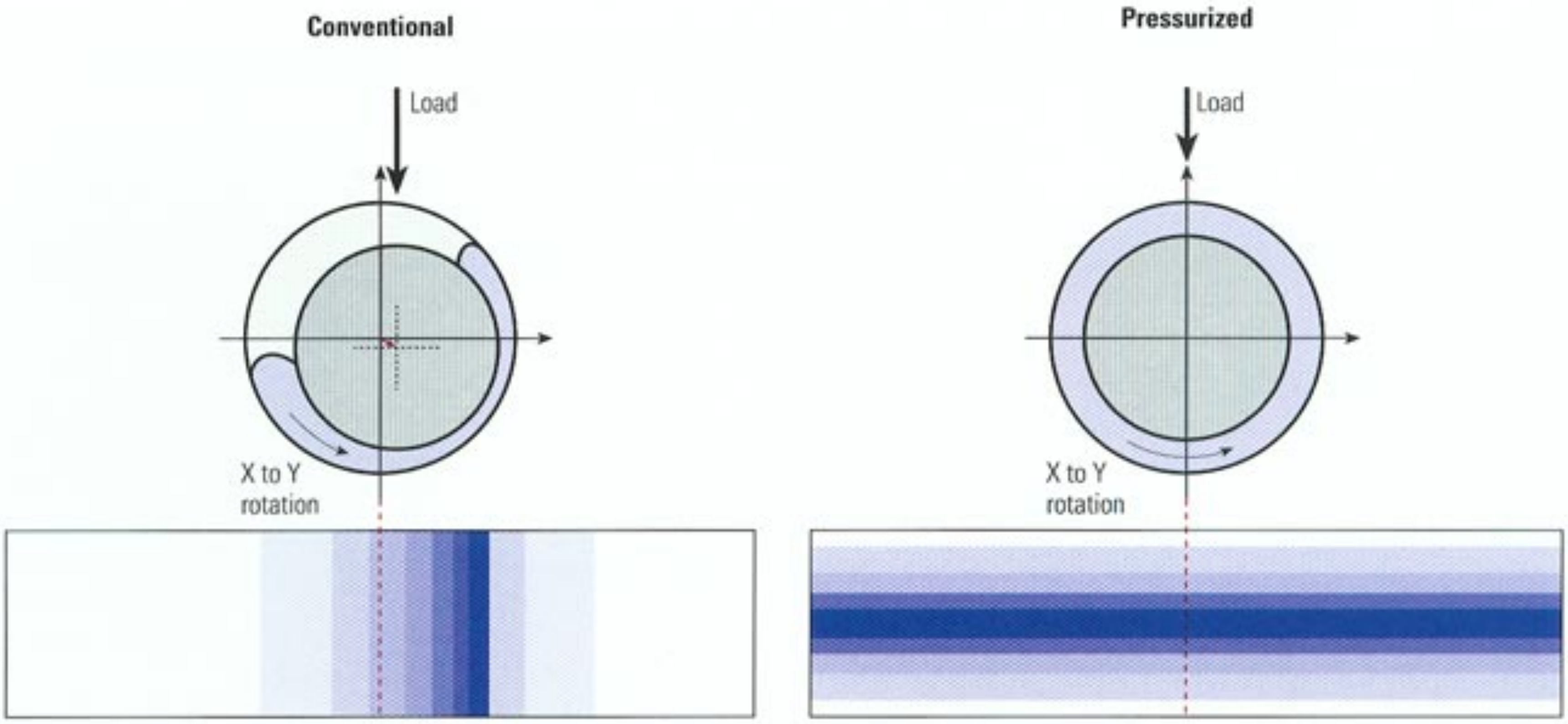


Figure 4. Conventional vs. pressurized bearings

A conventional oil-film bearing is partially lubricated, and oil flows around the journal (left). Pressurized bearings, in contrast, are fully lubricated, and oil flows primarily along the journal (right). Darker areas denote higher pressure.

is a scalar (it has only a magnitude). However, for a fluid-film bearing, the stiffness is a vector (it has both a magnitude and a direction), with components in the radial and tangential directions. The vector sum of these two components is called the "dynamic stiffness."

The stiffness component in the direction of the applied load is called the direct dynamic stiffness (DDS), and the stiffness component in the orthogonal (at 90 degrees to the direct dynamic stiffness) is called the quadrature dynamic stiffness (QDS). According to equations jointly developed by Donald Bently and Dr. Agnes Muszynski, our simple spring equation, applied to bearings, becomes:

Force = [DDS + j(QDS)] x Displacement

where "j" just means the component is perpendicular to the first component, and force and displacement are vector quantities.

The DDS also plays a large role in determining the overall system stability: Stabili-

Rotor eccentricity

Desirable bearing attributes (high stiffness, high damping) all require relatively large rotor eccentricities in hydrodynamic bearings. However, as eccentricity approaches zero in a conventional hydrodynamic bearing, these desirable attributes decrease, and as a result the bearing becomes unstable. For a pressurized hydrostatic bearing, stiffness and damping are achieved at low rotor eccentricities. Typical design rotor eccentricities are around 0.05, not to exceed 0.25. Even if load exceeds the worst-case assumptions, the closer the bearing gets to the wall (the higher the eccentricity), the more the bearing "pushes" back (Figure 1).

Why are low rotor eccentricities significant? Because a rotor that can be centered and can remain centered within its bearing clearance and whose rotor dynamic properties can be precisely controlled will allow seals and other mechanical clearances within the machine to be smaller and more precise. This translates into fewer energy losses

The key concept in Bently's new design is that the DDS originates from the axial pressure wedge, rather than the circumferential wedge, and that it acts opposite to the QDS. The supply pressure determines flow in the axial direction, rotating speed, lubricant viscosity, and the pressure drop along the bearing.

Therefore, the best mechanism for increasing the direct stiffness, Bently reasons, is to employ a fully lubricated bearing and to increase the axial pressure wedge (Figure 3). Bently accomplishes this by increasing the inlet pressure at which the bearing is supplied with lubricating fluid.

Ironically, this is the polar opposite of what is generally taught today—that externally pressurized bearings should be avoided because they are less stable. This is the essence of Bently's pressurized bearing design (Figure 4).

In the externally pressurized hydrostatic bearing system, lubricating oil is supplied to the bearing using a pressurized fluid

ty requires that the DDS be of the same order of magnitude as the QDS. The DDS represents the bearing's ability to supply a restoring force that acts counter to the applied force (the load). The DDS can be improved by increasing the pressure drop along the bearing.

The QDS acts at right angles to the applied load and effectively creates a moment on the shaft, attempting to push it circumferentially around the bearing clearance. In the worst case, if the tangential force of the fluid pushing on the shaft becomes larger than the counterforce of the shaft pushing back on the fluid, then the fluid and shaft will begin a "whirling" motion around the inside of the bearing clearance and in the same direction as shaft rotation.

and higher efficiencies.

Driving a wedge

The ability of a hydrodynamic bearing to carry load is based on a converging-diverging fluid wedge in the circumferential direction (Figure 2). For many years this wedge was considered to be the sole contributor to a bearing's ability to carry load. What the researchers didn't consider was fluid flow in the *axial* direction.

The fluid wedge contributes primarily to the tangential forces in the bearing, which produce QDS. But consider also the pressure wedge in the axial direction. Bently has shown that the DDS is proportional to the pressure at which the bearing is supplied with lubricating oil.

5. Bently's pressurized bearings

Donald Bently modified a compressor with his new pressurized bearing to demonstrate the technology. Testing showed stable operation under all operating conditions.

source (generally in the hundreds of psig, up to around 1,000 psig). The pressurized oil flows through orifices to specially designed bearing ports and pockets. Rotor motion and the bearing geometric design create a pressure differential between opposing pockets (180 degrees apart) to provide a restoring force to the rotor while it is operating, thus centering the rotor and creating a stiff and stable bearing.

According to Bently, the concept offers significant advantages in a wide range of applications, because the pressurized fluid would not be limited to oil—it also can be other liquids or gases.

Full-scale testing

Theory is all well and good, but engineers need to see actual operating hardware before they embrace a new technology. Bently understands that, so he had his team construct several test cells to demonstrate the externally pressurized bearing. In one test cell is a modified Clark 1M6 compressor retrofitted with two pressurized bearings and a pressurized thrust bearing, driven by a 130-hp electric motor (Figure 5).

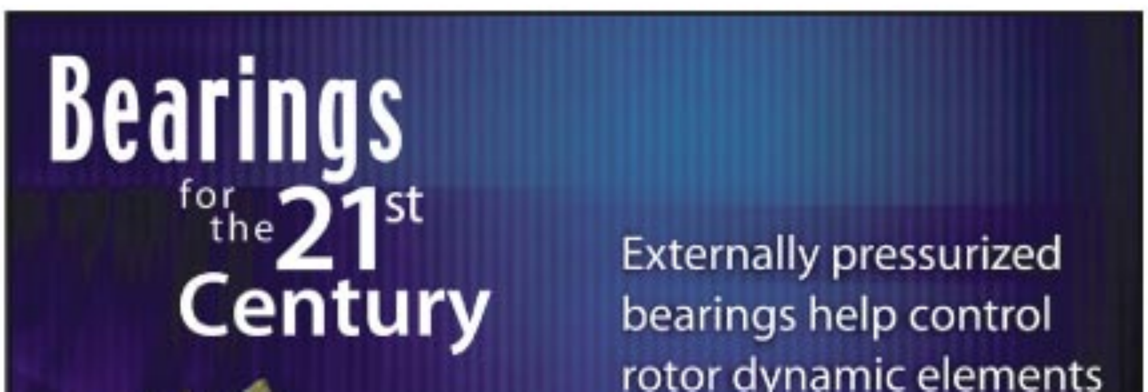
Operating conditions for the pressurized lubricant were 1,000 psig for the supply and 700 psig at the bearing "ports." The compressor was modified for an extensive instrumentation suite, the number of compression stages were reduced from six to two, a mid-span radial balance plane was added, and the compressor was fitted with a special perturbation wheel attached to the rotor drive end. The perturbation wheel can rotate on the shaft independent of the main rotor speed and direction, and can accommodate weight placement much like a balance ring, allowing an external force input to the drive train.

Testing of the compressor with the perturbation wheel was conducted with the compressor stopped, and with it at 7,000 rpm. At these compressor rotation speeds, the perturbation wheel itself was used both with and without weight at perturbator rotational speeds between 0 and 10,000 rpm to investigate effects of nonsynchronous perturbation.

The experimental testing demonstrated virtually no change in shaft centerline between 0 and 7,000 rpm—meaning that QDS was negligible, and eccentricity ratio remained small (less than 0.2), so the shaft virtually self-centers in the bearing from 0 to 7,000 rpm.

Radial stiffness increased much faster with increasing shaft eccentricity, demonstrating that, as the rotor moves away from the center of the bearing, the stiffness—and therefore the restoring forces—are much larger and increase much faster. This confirmed that the pressurized bearing has significantly better stiffness and load-carrying capacity than a conventional hydrodynamic bearing, particularly at low eccentricities. There was virtually no difference in stiffness characteristics between the compressor when stopped and when running at 7,000 rpm, indicating that the system is extremely stable. ■

Donald Bently recently published a book that covers all aspects of rotating equipment stability and dynamics, machinery diagnostics, operation, and maintenance. *Fundamentals of Rotating Machinery Diagnostics* presents these complex topics from an intuitive and practical—rather than a theoretical—point of view. Included are case histories illustrating how the knowledge developed over 40 years by Bently and his staff has been successfully applied on real-world equipment. Details on the 700-page reference are available at bpb.press@bpb-co.com.



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