Pressurized Water Bearing Solves Environmental Problems

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The position of the bearings in the machine train often puts oil in close proximity to the path of water through the hydro turbine. This novel bearing design uses externally pressurized water lubrication to eliminate the source of this potential environmental liability and improve the rotodynamic performance of many types of high and low speed machinery.

Low pressure oil lubricated hydrodynamic bearings were specified 100 years ago as original equipment on Southern California Edison’s hydro turbines. While low pressure oil bearings have been adequate for this and many other installations, the advent of stricter environmental regulations and the cost of safety, health, and environmental compliance make the continued use of these bearings in any hydro turbine application a risky proposition.

Recent improvements have allowed water lubricated externally pressurized bearings to replace oil lubricated hydrodynamic (internally pressurized) bearings in many applications. For the water power industry, the primary advantage of the externally pressurized bearing is the option to use an environmentally benign fluid, such as water, for lubrication without loss of load carrying capacity, efficiency, or reliability. A properly engineered externally pressurized bearing also results in better stability and control of the vibration response to dynamic forces acting on the rotor.

Early History of Fluid Film Bearing Lubrication
A few years before the early hydro turbines were installed, the expansion of the railroads in the western United States drove the need for durable and efficient bearings. In 1893, the Institution by direct contact between the shaft and axle. This early theory, based on the work of Frenchman Charles Auguste Coulomb, predicted that once the surfaces were in motion, friction should be nearly independent of velocity.

![Figure 1](image)

Figure 1. A drawing of Tower's early experiment examining the role of lubrication on railway axle bearings.

Tower experimented by immersing the shaft in a bath of oil. He found that the coefficient of friction between the axle and bearing varied greatly with rotational velocity. Tower drilled a feed hole into the top of the bearing cap and was surprised to see that oil traveled up the feed hole and could not be contained even after he tried to plug the hole with a wooden stopper. In effect, Tower discovered that the load of the shaft was carried
In 1883, the Institution of Mechanical Engineers hired Englishman Bechamp Tower to find the best way to lubricate railcar wheel bearings. The illustration in Figure 1 shows the test rig Tower used to simulate the conditions found on railcar bearings. Previous researchers had erroneously believed that the load was carried by a pressure distribution in the oil film. Upon further study, he realized the summation of local hydrodynamic pressures multiplied by the projected area equaled the load supported by the bearing. Tower is generally credited with the discovery of hydrodynamic lubrication.

Further work by Sir Osborne Reynolds observed that hydrodynamic action was dependent on lubricant viscosity. Reynolds postulated that the lubricant adhered to the surfaces of the rotating shaft and stationary bearing and was dragged into the narrow clearance of the offset shaft forming a “pressure wedge” sufficient to carry the applied load. Reynolds published the famous “Reynolds Equation,” providing the mathematical model for oil pressure distribution in a hydrodynamic bearing. Hydrodynamic bearings are properly classified as “internally pressurized” because the pressure wedge is built up in the bearing clearance by the relative motion between the bearing and shaft.

During this same period, an exhibit was presented at the Paris Exhibition of 1878 called Le Chemin de Fer de Glace (Icem Railroad). This exhibit turned out to be one of the first examples of hydrostatic lubrication. The exhibit consisted of a large metal statue with four hollow legs on a flat steel plate. Oil was pumped down each leg with enough force to “float” the statue. The heavy statue could be moved with ease because the friction between the feet and the plate was greatly reduced by the thin oil film separating the two surfaces.

Hydrostatic lubrication applied to journal bearings can support load even with little or no relative motion between the rotating and stationary parts of the bearing because it uses external pressure to form the supporting layer of fluid. A modern example is high pressure “jacking oil” systems (usually Vickers pumps) used to reduce wear on start-up of large turbines. Hydrostatic journal bearings are fully lubricated around 360-deg of the bearing surface and tend to operate at relatively low eccentricity (near the center of the bearing clearance).

As technology advanced and rotating speeds increased in the early 20th century, many machines exhibited unstable behavior and high vibration. Experiments showed that flooding simple sleeve bearings with lubricant caused the bearings to exhibit instabilities. These fluid instabilities became known as whirl and whip phenomena. Whirl and whip instabilities are characterized by high sub-synchronous vibrations commonly occurring at a frequency just below 50 percent of running speed.

Since fluid instability was associated with flooded bearings, it was erroneously assumed that a fully lubricated hydrostatic bearing would be inherently unstable. To improve stability, bearings were operated partially lubricated at low pressure. High pressure, hydrostatic lubrication was avoided for all but the slowest speed applications. Over time, several features such as the axial groove, elliptical, offset half, pressure dam, and tilt pad were added to the plain hydrodynamic journal bearing in order to raise the instability threshold. These features helped but the instability persisted up to 30 percent of running speed. A rotor can support load with relative motion because it uses external pressure to form the supporting layer of fluid. A modern example is high pressure “jacking oil” systems (usually Vickers pumps) used to reduce wear on start-up of large turbines. Hydrostatic journal bearings are fully lubricated around 360-deg of the bearing surface and tend to operate at relatively low eccentricity (near the center of the bearing clearance).

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The sum of the force vectors includes all of the forces acting on the rotor system. These forces can be static (unchanging in direction and time) or dynamic (exhibiting changes in magnitude or direction with time). Common examples of forces on rotor systems are gravity (static force) and unbalance (dynamic force).

The dynamic stiffness is derived from Newton’s Second Law: The sum of forces acting on a body equals the mass times acceleration. By making certain assumptions - the rotor system parameters are isotropic, gyroscopic and fluidic inertial effects can be ignored, and linearity – we can create a simplified equation of motion based on the free body diagram (see Figure 2).
The Rotor Model, Dynamic Stiffness, and Fluid Instability

Rotor systems can be modeled as spring systems using Hooke’s law. Hooke’s law states that the static displacement of a spring

$$\Sigma DS = K - M\omega^2 + jD\omega - jD\lambda\Omega$$

As shown in Figure 3, dynamic stiffness can be viewed as the sum of four vectors.

Figure 3. A representation of dynamic stiffness using vectors.

The four components of dynamic stiffness can be described in physical terms. The first term, \(K\), is the simple spring constant. The positive value of \(K\) indicates that it acts opposite to the direction of applied force. The second term, \(-M\omega^2\) is the mass stiffness that occurs because of the inertia of the rotor. The fact that this term is negative indicates it has a destabilizing effect on the rotor. The tangential stiffness term, \(-jD\lambda\Omega\) acts opposite to the damping term and is potentially destabilizing. The tangential stiffness term is proportional to the fluid circumferential angular velocity, \(\lambda\Omega\).

The tangential stiffness term introduces the term lambda (\(\lambda\)) to our rotor model. Lambda describes the fluid circulation around the circumference of the bearing journal. Whenever a viscous fluid is contained between two surfaces moving at different velocities, the fluid will be dragged into relative motion. Because of friction, the relative velocity of the fluid at the surfaces will be zero. Because the surfaces are moving at different rates, the fluid will develop a velocity profile similar to the one represented in Figure 4.

Figure 4. The average fluid angular velocity is represented by \(\lambda\Omega\).

If we return to our equation for radial position of our spring-mass-damper system,

$$r = \frac{\Sigma F}{\Sigma DS}$$

the maximum displacement will occur when the value of dynamic stiffness is small for any non-zero value of applied force. Instability, defined as the threshold where \(r\) becomes bounded only by the mechanical constraints of the system and non-linear effects, occurs when the value of the dynamic stiffness in our model equals zero.

Because the dynamic stiffness consists of both direct and quadrature terms, this condition is satisfied when \(K = M\omega^2\) and \(\omega = \lambda\Omega\). Combining these two conditions results in the Bently-
The Bently-Muszynska model predicts that the stiffer the bearing support and the lower the value of lambda, the higher the Threshold of Instability or stability margin.

**Root Locus Stability Analysis**

The best way to evaluate the stability of rotor systems and bearing designs is to use a graphical technique known as the Root Locus Method. This method graphs the roots of the characteristic equation of the rotor system:

\[ M\ddot{r} + Dr + (K - jD\lambda\Omega)r = 0 \]

The solution of the characteristic equation takes the following exponential form:

\[ r = Re^{st} \]

Solving for s in the case when R, \( \lambda \), and \( \Omega \) are non-zero we obtain two complex roots:

\[ S_1 = Y_1 + j\omega_d \quad \text{and} \quad S_2 = Y_2 - j\omega_d \]

Because the solution for r (displacement) has an exponential form, values will grow unbounded with time when the roots are positive and decay when the roots are negative. Negative roots indicate that the rotor system is stable and will return to an equilibrium position when disturbed. Positive roots indicate instability. No real machine can operate in the region of instability. When the roots equal zero, r will remain constant. This represents behavior at the Threshold of Instability. Figure 5 shows a typical root locus plot over a range of operating speeds from 0- to 1000-rad/s. Based on this plot, the predicted threshold of instability is expected to be 870-rad/s.

Root Locus plots can be used to evaluate the effect on stability for a wide range of machine parameters, making it an extremely useful tool for rotodynamic analysis.

**The Externally Pressurized Bearing (EPB)**

In contrast to hydrodynamic bearings that rely on the relative motion between the rotating journal and stationary bearing to create a pressure wedge to support the load of the shaft, externally pressurized bearings (EPB) rely on an externally generated source of pressure. Therefore, they have several unique features and advantages that make them superior to the low pressure hydrodynamic journal bearings and even the latest modified geometry bearings such as the lobe and tilt-pad bearings.

The distance to the bearing wall is measured by the eccentricity ratio (0 represents a centered journal, 1 a journal touching the bearing wall). As seen in Figure 6, a hydrodynamic sleeve bearing must operate off center to create the pressure wedge that supports the bearing load. The stiffness of the hydrodynamic bearing is affected by the eccentricity ratio, which is lowest when the eccentricity ratio is near zero. Misalignment during installation or changing radial loads can result in a hydrodynamic bearing that is forced to operate at low eccentricity ratio and low stiffness. Since we showed earlier that the threshold of instability and the vibration response is related to stiffness, a normally stable hydrodynamic bearing can have problems when operating at low eccentricity ratio.
in Figure 7. While both the EPB and the hydrodynamic bearings have good relative stiffness at high eccentricity, only the EPB retains its stiffness at zero eccentricity. A pure hydrostatic bearing operating at very slow speed has good stiffness at zero eccentricity but does not have the advantage of hydrodynamic lubrication at high eccentricity.

![Figure 7. Relative stiffness as a function of eccentricity ratio.](image)

The superior stiffness value of the EPB makes it less prone to instability even when subject to misalignment, changing radial loads, or operation at low eccentricity. This feature of the EPB makes it ideal for installation on vertical machines found in many large hydro installations. In vertical hydro turbines, the lack of gravity side load makes it possible for the guide bearing journals to be forced to operate at low eccentricity. This is often the case during transient loading conditions that occur during start-up or shut-down.

Figure 8 show the typical pressure profile of a four pocket EPB design. The pressure profile around the EPB differs from the low pressure bearing because it acts around 360-deg of the circumference. The unique feature of the EPB is the pressure profile that acts along the longitudinal axis of the bearing. This axial pressure gradient drives flow out the end boundaries of the bearing geometry, effectively reducing the value of lambda (average circumferential velocity ratio).

External pressurization controls stiffness and lambda, the two parameters that directly contribute to the stability of the bearing. Not only are these two parameters controllable during the bearing design process but they are also controllable by the operator “online.” The properties of the bearing can be changed after installation by changing the external supply pressure (stiffness and lambda) or the supply temperature (damping).

Almost all types of hydrodynamic bearings in use today rely on petroleum based lubricating oils. Physical properties of lubricating oils facilitate the formation of the pressure wedge that supports the load in the internally pressurized bearing. Water has poorer lubricant properties and does not work well in hydrodynamic bearings. The EPB can operate with conventional petroleum lubricants but can also operate with other incompressible fluids such as water. Water can be used to support load in the EPB because the internal pressure profile is dependent on an external pressure source and not primarily the relative motion between bearing and journal. The EPB opens the possibility of choosing a bearing fluid not solely on its lubricant properties but also for its compatibility with the process and the environment.

In hydro power applications, two convenient sources of pressurized water can be used to supply the EPB. If a clean water source with sufficient head is available from the penstock, it may be used to supply the EPB. Typically, filtration in the 3 to 5 micrometer range is necessary to protect the close tolerances of the EPB. The pressure necessary for the operation of the EPB must be calculated by the bearing designer on a case by case basis. Typically, the pressure needed is in the range of 150-psi to 1000-psi, or between 350-ft and 2300-ft of static head.

If a suitable source of water is not available from the environment, pressurized water can be produced by a closed loop fluid delivery system. The fluid delivery system is composed of several components, typically a high pressure pump, a high pressure filtration unit, and a low pressure pump and heat exchanger on a kidney loop to control the fluid temperature. Sensors are employed to monitor the fluid temperature, pressure, and flow. Controls and interlocks can be employed to insure that the bearing is operating at the desired design conditions. A pressurized accumulator tank may be used to provide a smooth flow.

![Externally Pressurized Bearing](image)
The Pelton wheel (left) and generator (right) instrumented with temporary proximity probes for diagnostic analysis.

Creek generating facility. A typical machine train is composed of two major components: a Pelton water wheel and a General Electric three-phase generator. The generator and Pelton wheel are mounted on a common horizontal shaft supported by three bearings. Bearing 1 is located outboard of the Pelton wheel, bearing 2 is located between the Pelton wheel and the generator, and bearing 3 is located outboard of the generator.

The bearings appear to be those supplied with the original equipment. They utilize soft metal babbitt faces with diagonal grooves to help distribute the oil. Soft metal is used to avoid damaging the shaft upon start up and shut-down. A leather

The center pedestal (Bearing 2) in close proximity to the Pelton wheel (right).

slinger is employed on the shaft to contain the oil in the bearing pedestal. Based on nameplate information, it appears that the Pelton wheel and generator were manufactured in 1909.

A common occurrence on these types of machines is oil leakage from the bearing pedestals. This problem is prevalent at bearing 2 which is in close proximity to the Pelton wheel. The housing of the Pelton wheel operates under slight vacuum, causing oil to be sucked into the water discharge. Bearing 2 also carries most of the load of the machine. For these reasons, it was decided to replace the hydrodynamic oil bearing in the middle of the machine with an externally pressurized water bearing.

Because of the historical nature of these machines, Southern California Edison requested that no external modifications be made to the bearing pedestals. The EPB retrofit which consisted of new bearings and backers was designed to fit inside the original pedestal with small penetrations for pressurized water delivery and return.

<table>
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<tr>
<th>Properties of Original Oil Bearing</th>
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<tr>
<td>Length</td>
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<td>Diameter</td>
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<td>Bearing construction</td>
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The field data collected from this machine revealed no surprises. The machine appeared to be well behaved and exhibited normal bearing operation. As expected, the water bearing performed as expected.

Test Stand Data

To validate the pressurized water bearing design, a near full-scale model of the turbine generator machine train was constructed in Minden, NV. The original number 2 bearing pedestal from Southern California Edison's Mill Creek site was used to support a pressurized water bearing and replacement backer. The simulated machine train uses two rotating wheels with a weight of 3,300-lb each to model the turbine and generator. The total rotating weight with shaft is close to 8,000-lb. The shaft is driven by a variable speed electric motor at 450-rpm. Two roller bearings are used on either side of the wheels to simulate bearings 1 and 3.

The test stand pressurized water bearing was designed to carry slightly more than half the total load. Pressurized water is supplied by a closed loop fluid delivery system that supplies 100 psi at the bearing face.
Conclusions
The externally pressurized bearing solves the age old problem in the many vertical configuration Francis turbines worldwide. Many potential applications of pressurized bearing technology have been identified in addition to the initial success with this particular installation. Designs can be modified to accommodate the specific needs of each application and the potential is extensive.
of fluid bearing instability. It takes the century-old principles of hydrostatic bearing lubrication, and applies them in such a way that enhances bearing stiffness and reduces lambda. In doing so, the externally pressurized bearing can positively influence the two bearing factors that have the greatest impact on the stability of rotating machinery.

While this report details the application of the externally pressurized water bearing in a radial load application, the technology works equally well on vertical machines and can be successfully applied to thrust bearings, resulting in similar benefits and advantages. This makes application of the technology ideal.