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Damping & stiffness, a primer

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What is rotordynamics?

Most simply, it is the study of the vibration characteristics of a rotating shaft supported on bearings. In almost all cases, the object of rotordynamic analysis is to predict and then reduce or eliminate actual rotor vibrations. A rotor spinning frictionlessly on a perfect axis of rotation with zero lateral and axial vibrations would be the geometric ideal. This condition is also preferred for reducing tip clearances and increasing efficiencies.

More philosophically, rotordynamics can also be considered an effort in determinism. Determinism is the idea that everything happens for reason, and that these reasons can be understood. It is recognized that rotor bearing systems can be very complicated, to the point where their response appears random. But nothing is random and rotordynamics is our tool to simplify the complexity and deepen our understanding.

The question explored in this work is how fluid film lubrication in bearings affects our ability to make rotordynamic predictions.

Let's start with the property of stiffness, sometimes referred to as a spring rate. It is

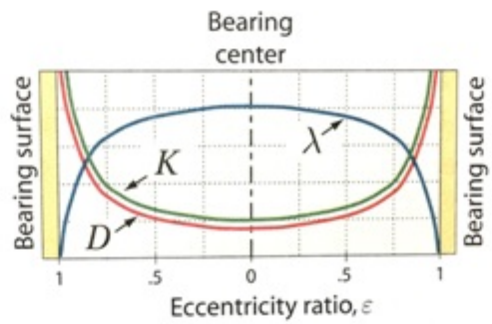

Figure 1. Qualitative plots of fluid-film bearing parameters versus journal eccentricity ratio. Stiffness, K, and damping, D, are minimum when the journal is at the center of the bearing, and they are approximately constant for low eccentricity ratios. As the journal nears the bearing surface, stiffness and damping increase dramatically. λ behaves the opposite way. When the journal is at the center of the bearing, λ is maximum. As the journal nears the wall, the fluid flow is increasingly restricted, until λ nears zero at the wall.

Source: *Bently, D. E. (2002). Fundamentals of rotating machinery diagnostics. Minden, Nv: Bently pressurized bearing Press.*

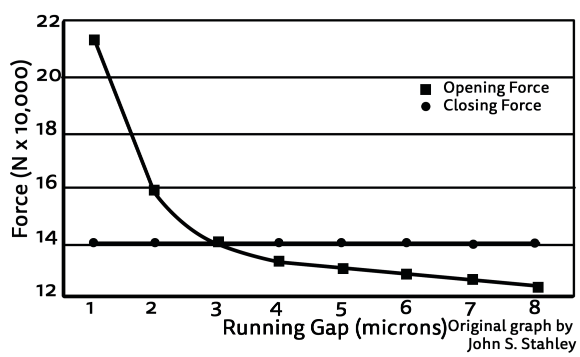

Figure 2. The aerodynamic bearing opening force of a dry gas seal face. The slope of the line is representative of the stiffness at a gap. Note that the gap is in microns.

Source: Stahley, J. S. (2005). *Dry gas seals handbook*. Tulsa, OK: PennWell Corp.

the force required to compress a spring though a unit of distance (force/distance) here in lbs./in.

The stiffness of rotor support is very important, as is the stiffness of the rotor itself. It is recognized that taken together with the base, they comprise a structural loop that needs to be considered as a system. In this case though, we will focus on the bearing film properties, considering both oil and gases as fluids used in both dynamic and static bearing technologies.

The spring rate of a fluid film is primarily dependent on its film thickness- the thinner the gap the higher the stiffness. This is well known and documented in eccentricity ratios of hydrodynamic oil journal bearings, figure 1. Stiffness curves of dry gas seals, figure 2, and commercially available externally pressurized gas bearings, figure 3. Notice in every case that the stiffness is increasing exponentially at the smaller gaps.

Hydrodynamic oil bearings common in large turbines and aerodynamic bearings used as sealing elements in dry gas seals have no static stiffness, as dynamic bearings require motion to generate the lubricating film. Once spinning, the speed of rotation will be the primary determinant of film thickness. Temperature change will affect the viscosity of the oil and so the stiffness of the film significantly. Shear heating of the film is based on a cube of the film thickness, so heat generation can increase rapidly, again having an unstable feedback effect on viscosity.

When we look to the eccentricity ratio of oil hydrodynamic bearings, figure 1, it can be seen that both the stiffness and the damping of the film increase exponentially as the rotor approaches the bearing surface. The viscosity of oil also changes in a non-linear away with the temperature increase from the shearing oil. One of the main challenges in rotordynamics is that the conventionally used bearings (hydrodynamic oil bearings) have multiple properties changing in non-linear fashion with respect to the speed and temperature.

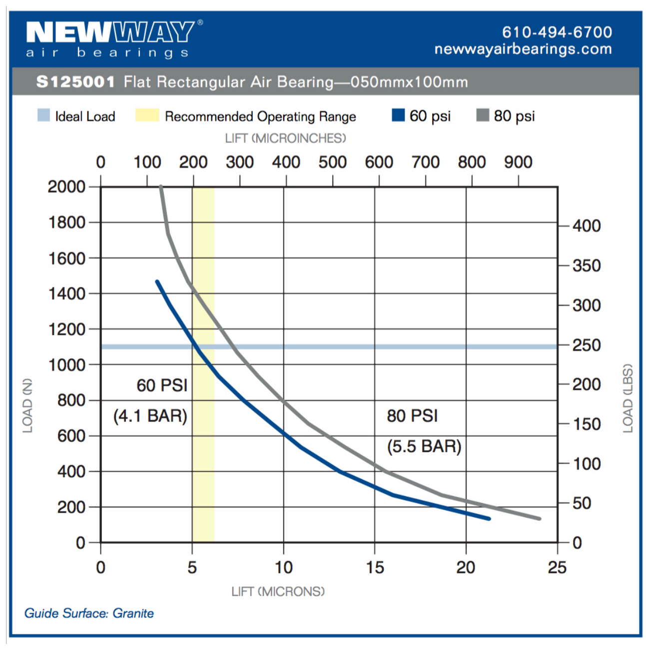

Figure 3. Lift-Load Chart for commercially available Externally Pressurized Porous (EPP) gas bearing.

Source: Frictionless Motion™. Retrieved November 20, 2017, from http://www.newwayairbearings.com/

This can make them a challenge to model.

Hydrostatic oil bearings common in large machine tools and aerostatic air bearings in measuring machines, have an external supply of pressurized fluid and so can support a load statically, that is without relative motion of the bearing faces.

Oil hydrostatic bearings will suffer in some degree from heat generated by shear in the film at higher speed as was case with the oil hydrodynamic bearing, but because of the external pressure, the film thickness may be made to remain the same, near an eccentricity of zero but with enough stiffness to damp radial motions of the rotor. As compared to the oil hydrodynamic bearing where lower viscosity oil will result in lower film thickness. The lower film thickness results in higher stiffness and damping as shown in the eccentricity ratio plots, but the increases may not be useful.

Aerostatic bearings have very low shear relative to oil, so they will not generate significant heat, and most gases do not change their properties much with a change in temperature. So, externally pressurized gas bearings have similar stiffness properties regardless of the rotation speed or temperature. It should be pointed out that this makes them easier to model and predict then oil hydrodynamic bearings. Because the stiffness may be adjusted by input pressure, they have flexibility and adjustability that can be used as feedback to check the model. And because they have an easily measurable static stiffness, functional quality rather than just dimensional checks may be performed before assembly.

It is hard for some to imagine how an air film could be too stiff in a structural loop made of metal but this can easily be the case. Stiffness of externally pressurized gas bearings, as we have seen, is a function of film thickness, which is a result of the input pressure and load conditions. A gas bearing with 0.5 square foot (6 x 12 inches of face area), fed 60psi compressed air and loaded with 2,500lbs will have a film thickness of .0002 inches and almost 10 million lb./in stiffness with flow less than 1 SCFM. The same bearing with the same input pressure but with just 10% the load (250lbs), would have higher flow and stiffness less than 1000 lb./in. This is a change of four orders of magnitude in the stiffness for a one order change in the load. This effect can also be seen in the lift load plot in Figure 4. At large gaps, above 0.0005in, the curve is near horizontal, which would be zero stiffness; at small gaps the curve is near vertical, which would be infinite stiffness.

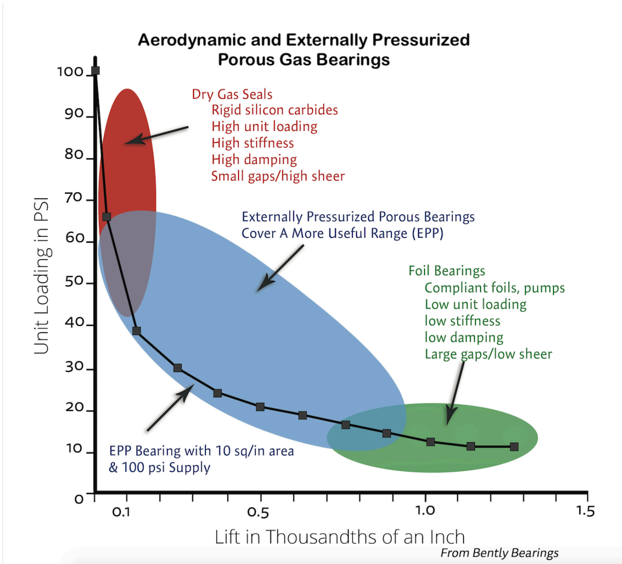

Figure 4. This is a "big picture" chart to conceptualize gas-bearing technology in turbo equipment. It is a lift vs. load chart for an externally pressurized porous (EPP) gas bearing that has 10 square inches of bearing face; 100 psi is fed to the bearing as the load on the bearing is increased. Resulting air gaps are shown on the bottom axis. The slope of the curve is representative of the film stiffness. The typical operating regions of dry gas seals and foil bearings are super imposed, as is the operating area for EPP Bearings. It can be seen that the EPP Bearings operate with a larger gap than dry gas seals, reducing heating and contact issues but still having good stiffness, damping and speed capabilities.

Don Bently focused on this adjustable stiffness property as a panacea for the many rotordynamic problems he had spent his life identifying. After selling Bently Nevada to GE in 2002, he established the Bently Pressurized Bearing Company to commercialize externally pressurized bearings, both hydrostatic and aerostatic, more about this later.

Our next topic, damping is certainly affected by stiffness. Damping, the property of dissipating energy is force/velocity, here in units of lb-s/in. We see in the eccentricity ratios that stiffness and damping are both increasing together as the rotor moves towards the bearing wall. As a way of conceptualizing this, it can help to think of damping in extremes of stiffness as noted above; if there is zero stiffness, there can be no damping because the rotor has nothing to exert force on. A large film thickness is like ringing a rotor hanging on surgical tubing to get its undamped natural frequency. If the stiffness is infinite, then there is no motion, and again zero damping (damping being force over velocity). So, for a given system and frequency, somewhere in between zero and infinite stiffness, an optimum point will be reached to maximize the damping, after which point further increases in stiffness will begin to lock up the motion, reducing effective damping.

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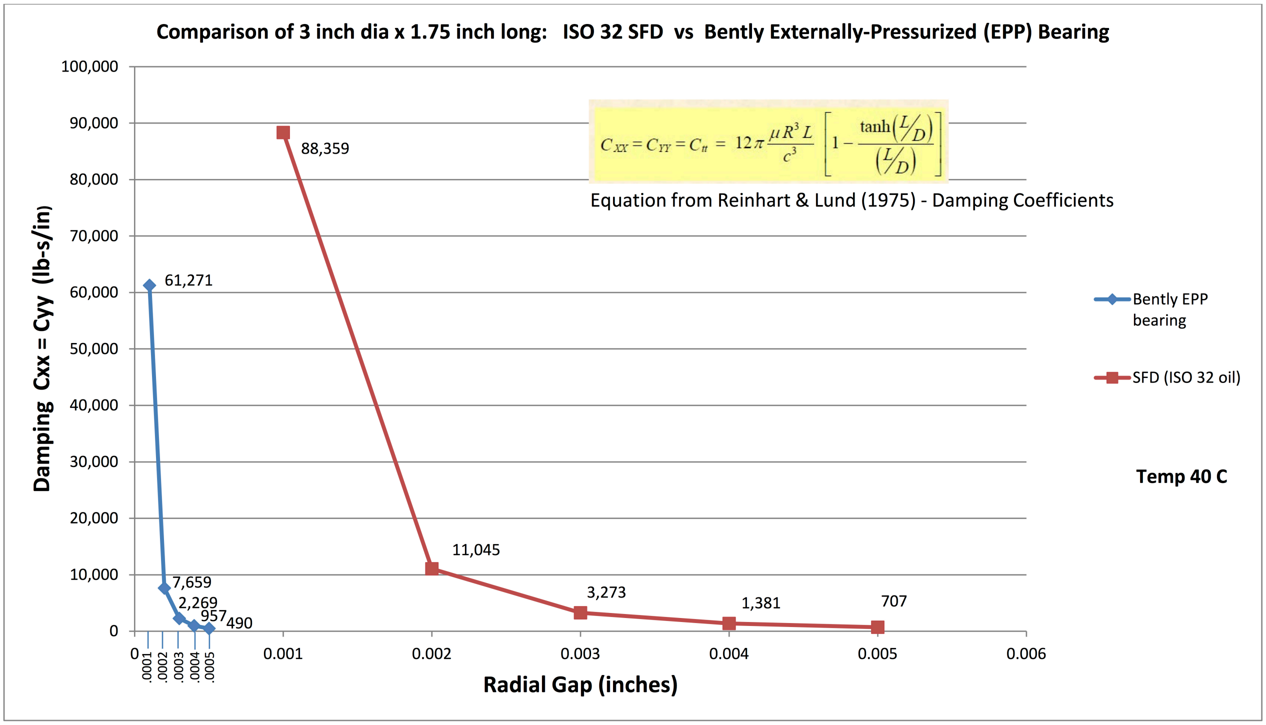

Figure 5. Comparison of oil and air based squeeze film dampers. Air damping is similar to oil at 10 times smaller gaps.

Hydrodynamic oil bearings are considered to have good damping properties. We can see in the eccentricity plot that damping increases with the stiffness but please remember that damping will be variable in a non-linear way, based on speed and temperature for the same reasons that the stiffness was.

Hydrostatic oil bearings offer a method to adjust the damping of oil bearings, again by having control over the film thickness and stiffness via the external pressure.

Regarding aerodynamic bearings, it is frequently noted in turbo-related articles and papers that foil type gas bearings have little stiffness and damping. This is in large part due to the low stiffness of the bumps, leaves and foils supporting the gas films and or the relatively large clearance left for the films to develop and so typically do not have enough stiffness to give much damping.

Yet there seems to be universal acknowledgement that aerodynamic bearings in dry gas seal applications have very small gaps < 0.0002in. and very high stiffness, as seen in figure 2. Considering that DGS are “balanced”, that is sealed pressure is allowed behind the stationary face, forcing it against the rotating face with unit loading of the pressure being

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sealed, often that may be thousands of psi. This is much higher than oil bearings are capable of, but only because the DGS dynamic bearing is charged by the sealed pressure trying to escape though the gap. The dynamic pumping of the groves only provides the small pressure differential that keeps the gap. So, if there is 20,000 lbs of closing force, (excluding spring bias for now), say 2000psi across 10 square inches and the pumping groves generate 4psi there will be 20,040lbs of force separating the faces. Again though, there is no ability to adjust the film thickness; you just take what you get for a film in dynamic bearings.

Aerostatic bearings do allow for the adjustability of stiffness in order to maximize the damping as noted above. There are equations for calculating the damping in fluid films. Using one that is often cited in the industry, we calculated the damping from an oil squeeze film damper at 5 different film thicknesses from 0.001 to 0.005in. Again, we see the same effect where damping increases exponentially at the smaller film thicknesses. Using the same equation, changing only the oil viscosity for that of air, and calculating the damping at 5 film thickness exactly 10 times smaller than with the oil (0.0001 to 0.0005in.) we notice that the damping values are only marginal less than the oil, figure 5.

Cross coupled stiffness coefficients in rotordynamic calculations are a way to account for the destabilizing effects of friction in the fluid film.

With oil hydrodynamic journal bearings, and to a lesser degree in oil hydrodynamic tilt pads, as the rotation of the rotor drags oil underneath it, there is an equal force on the rotor causing it to climb up the side of the bearing wall. At first thought it may be hard to see how this could have much effect, but consider it this way, a gas turbine with 6-inch diameter journals may consume some 1000hp for cooling the heat added to the oil from shear. This shear heating is the measure of the “traction” the rotor gets in the bearing. Imagine a 1000hp dragster burning rubber and accelerating down the track. When considering that this is the same power and traction the rotor has in the journal bearings, it is easier to conceptualize the destabilizing effects of friction. This friction may be seen graphically in figure 6.

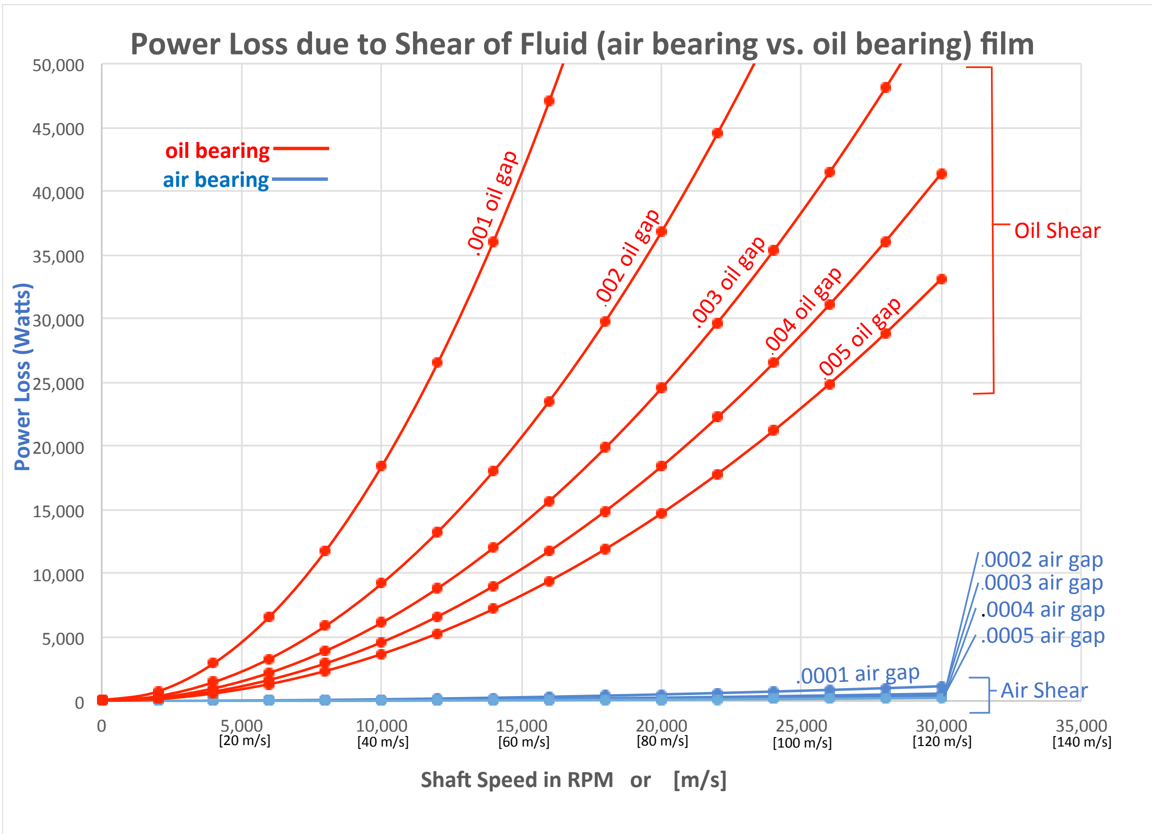

Figure 6. Comparison of shear power losses between oil and air within their operating range. These losses are from the traction the rotor gets in the oil and responsible for the destabilizing cross-coupled stiffness terms.

Considering a journal bearing of the same dimensions used in the damping example figure 5 (3-inch dia. 1.75binch long), but in figure 6, we solve for shear (power loss) at speed. It is a representation of the "traction" we were referring to previously, and it becomes clear again that the thickness of the film is key. We have seen that stiffness, damping and now the destabilizing effects of shear are a cubed function of the film thickness. At smaller gaps where dynamic bearings operate, multiple rotordynamic properties are near asymptotic, with both heat generation and the destabilizing cross coupling changing dramatically with small changes in other properties.

Oil hydrostatic bearings would be more deterministic, again in that it is possible to control the film thickness and so the other rotordynamic properties.

Aerodynamic journal bearings like leaf, bump or metal mesh type generate lift from the same principle as the oil hydrodynamic bearings. The viscosity of air is so low though, that it is difficult for the rotor to create a destabilizing cross coupled stiffness. This is also the fundamental reason the unit load capability of aerodynamic bearings is low.

Externally pressurized gas bearings, or aerostatic bearings, are also represented in figure 6. It is evident from the nearly zero power loss that there must be near zero cross couple stiffness. Dr. San Andres from Texas A&M Turbo Lab investigated this in a 2015 STLE (Society of Tribology Lubrication Engineers) paper on aerostatic bearings. He noted "External gas pressure has a large effect on the system damping ratio, increasing as supply pressure decreases, yet there is near zero cross couple stiffness". Source: Andres, L. S. (2015). Experimental Assessment of Drag and Rotordynamic Response for a Porous Type Gas Bearing. *STLE*.

Notice that Dr. San Andres's findings regarding the damping increasing with decreasing supply pressure seems counterintuitive. The key is, by reducing the supply pressure the fluid film thickness is reduced and so its stiffness increases to allow more damping, yet still there was not 100 W of shear.

This was Don Bently's dream after selling Bently Nevada to GE; by avoiding the destabilizing effects of oil lubrication and providing adjustable stiffness and damping, Don was going to revolutionize bearings in rotating equipment. He wanted to simplify rotordynamic analysis by making the bearing properties more deterministic. If he could not control it, he did not want it.

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