

FLUID FILM BEARINGS FUNDAMENTALS

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ABSTRACT

Bearings constitute one of the most critical components in turbo and rotating machinery today. Their influence on the rotordynamic performance, life, and reliability of the machine cannot be ignored. Many of the problems we face with machinery today can be attributed to the design and application of the bearings. An understanding of how bearings work and some knowledge of the basic principles that underline their operation is therefore essential for making the proper choice for the particular design that best matches the service requirements of the machine in question. Even after the machine is designed and placed in operation, changes or modifications to the bearings constitute one of the most effective, direct, and economical means to alter and improve the machine's dynamic performance.

FLUID FILM BEARINGS

Fluid film bearings have been around for centuries, however the understanding of how they actually worked did not come about until 1882-1883. This is when an English inventor and railway engineer by the name of Beauchamp Tower [1] was appointed to study high-speed railway engine bearings. He demonstrated in his test rig that with a suitable supply of lubricating oil, the surfaces of the bearing and the shaft were separated by a continuous film of lubricant which prevented them from ever coming into contact. Tower was able to show that the load in the bearing was carried by the thin oil film through his measurements of the pressure profile in the bearing, but did not explain how and why this happens. Three years later (1886), Professor Osborne Reynolds at the University of Manchester in England published his 77 page paper reference [2] where he was able to predict and explain the experimental measurements conducted by Tower. Reynolds developed what is known as the classical Reynolds hydrodynamic lubrication equation through combining the simplified Navier Stokes equation (assumed Newtonian incompressible fluid, neglected inertia and variations of the viscosity across the film) and the continuity equation.

Fluid film bearings operating in the full hydrodynamic regime support the load on a very thin film and thus there is no contact between the shaft and the bearing. It is for this primary reason that fluid film bearings offer infinite life provided the lubricant is kept clean and the machine is operating in a safe dynamic

range away from stability and critical speed problems. This is why when long term operation (2, 3, 5, and even 10 years of continuous operation) and a high online factor (90% and higher) are required, the fluid film bearings are the bearings of choice.

The formation of an oil wedge to lift the shaft journal is similar to hydro planning (controlled hydro planning in the case of bearings) and is dependent on the speed (relative speed between the shaft and the bearing), load (weight of the rotor or any additional side loads from process fluid, or gear loads, or side loads due to misalignment), and the oil viscosity of the lubricant. These parameters are combined and presented by the ZN/P curve shown in Figure 1. This is also known as the Stribeck curve [3] the engineer who introduced it in 1902. The symbol "Z" represents viscosity, "N" is the speed in RPM, and "P" is the unit loading in "lb/in²". This curve describes the three regimes of operation a bearing passes through while the machine accelerates to operating speed or decelerates from the operating speed to stand still conditions. These regimes are "Dry Friction" where contact between the asperities of the shaft and the bearing exist, mixed lubrication regime or "Boundary Friction", and full hydrodynamic lubrication "Fluid Friction" where a thin film exists between the shaft and the bearing which supports the static and dynamic loads in the rotating shaft. These three regimes are also shown graphically with an exaggerated clearance in Figure 2 as the shaft accelerates from standstill to full operating speed. The discussion in this tutorial is limited to the full hydrodynamic regime. This regime has the most influence on the rotordynamic characteristics of the machine as it speeds up from stand still to possibly traverse one or more critical speeds on its way to reaching the design operation speed.

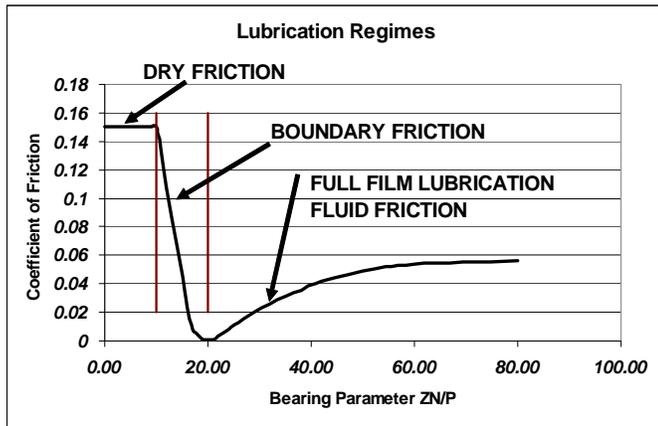


Figure 1. The ZN/P curve and the three lubrication regimes

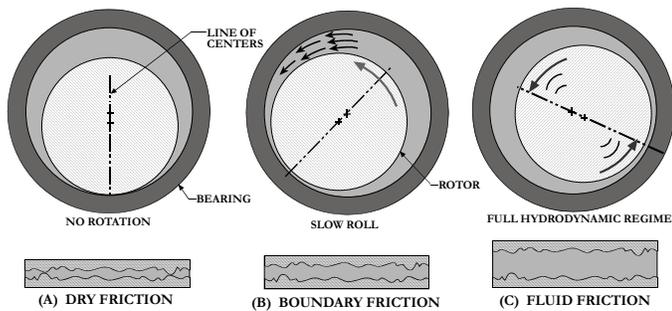


Figure 2. Three lubrication regimes in fluid film journal bearings

Fluid film bearings develop pressure in the converging wedge which supports the radial load on a thin oil film. Most fluid film bearings are of the “hydrodynamic” type, where the film pressure is produced by the shaft rotation dragging the oil into the converging wedge formed by the shaft and the bearing surface. In “hydrodynamic” bearings the oil pressure is just enough to keep the bearing supplied with oil and to remove the frictional heat generated by the viscous shear in the thin film.

From the standpoint of rotordynamics, the main advantage of fluid film bearings is their inherent damping characteristics. Fluid film bearings exist in a variety of configurations depending on the application, space availability, and the rotordynamic requirements. The trend over the last few decades has favored the variable geometry tilting pad type bearings over the fixed geometry sleeve type bearings particularly in high speed super critical machinery due to their inherent stability characteristics. However, these variable geometry tilt pad bearings are more complex, contain more parts, and generally have lower damping than fixed geometry bearings.

FIXED GEOMETRY SLEEVE BEARINGS

A fixed geometry or fixed profile sleeve bearing supports the weight of the rotating shaft by developing a hydrodynamic pressure in the converging wedge formed by the shaft and the bearing surface shown in Figure 3. The asymmetric pressure profile in the oil film is characteristic of fixed geometry journal

bearings. This gives rise to an attitude angle formed by the line of centers (line connecting the center of the shaft and the center of the bearing) and the load vector. This characteristic, present in all fixed geometry journal bearings, is indicative of the presence of cross-coupling in the bearing. The load, which in this case is due primarily to the weight of the rotor, is acting directly downwards on the journal bearing. Integrating the pressure profile results in a net force that balances the weight of the rotor. This force generated in the film is accompanied by a shaft displacement with a component that is along the load direction (direct) and a displacement that is orthogonal to the load direction and is shifted in the direction of rotation (cross). This cross coupling characteristic is present in all rotating machinery where a fluid or gas is rotating with the shaft in a small annulus.

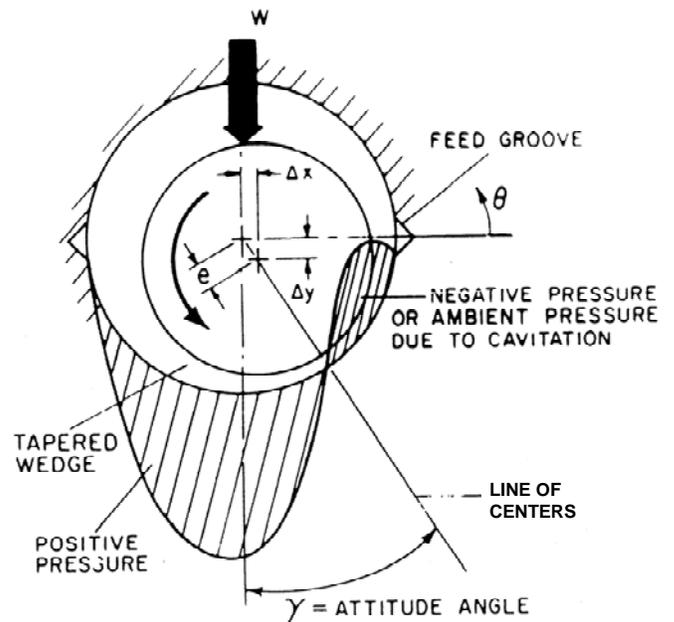


Figure 3. Pressure profile and attitude angle in a sleeve bearing

This phenomenon is unique to rotating machinery and is not present in other non-rotating structures or mechanisms. It is this cross-coupled stiffness force that promotes self-excited sub-synchronous vibrations and instability in rotating machinery. In fluid film bearings this is referred to as “oil whirl” where the shaft will whirl at a frequency that is equal to a fraction of the running speed frequency. This sub-synchronous component tracks the running speed until it latches on or locks on a natural frequency where its amplitude will increase rapidly and results in what is commonly referred to as “oil whip”. The frequency of the sub-synchronous vibrations in the “oil whirl” region is dependent on the L/D ratio of the bearing. The higher that ratio is, the closer the sub-synchronous oil whirl is to half running speed. This gives rise to the thought that the sub-synchronous frequency is directly related to the average circumferential flow of the lubricant between the shaft and the bearing. This circumferential flow is closer to half running speed as the length of the bearing increases and the effect of the side leakage is relatively smaller.

It is also worth noting that the cross-coupling in the journal bearing tends to always move or push the shaft to the opposite side of the converging wedge. Therefore, this cross-coupling effect is always biased by the direction of rotation as shown in Figure 4. The term rotational bias is often used to describe this phenomenon, but what is also relevant is the fact that it is forward driving. Therefore, when diagnosing sub-synchronous vibrations it is important to note if the whirl of the sub-synchronous vibrations is forward or backward to help distinguish oil whirl and whip from rub and other anomalies that produce backward whirl.

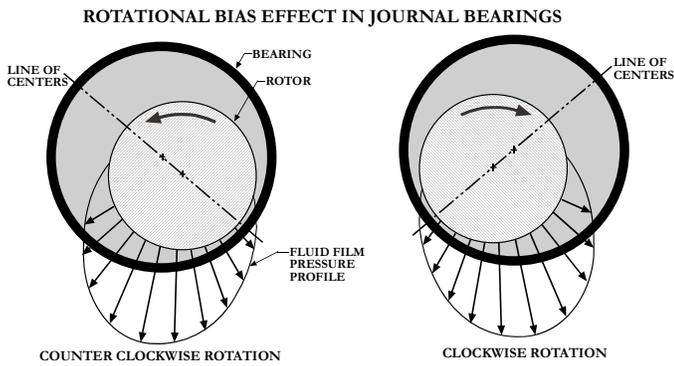


Figure 4. Rotational Bias caused by cross-coupled stiffness coefficients

The cross-coupling force vector shown in Figure 3-15 [Vance et al] provides an excellent graphic representation of the effects of the cross-coupled stiffness coefficients in the bearings and how they act on the rotor. The cross coupled stiffness forces $K_{xy}X$ and $K_{yx}Y$ combine vectorially to produce a force that is tangential to the circular whirl orbit and in the direction of rotation. This is why it is often referred to as a follower force (following the direction of rotation), forward driving force (pushing the rotor to whirl forward in reference to the direction of rotation), or destabilizing force (tends to add energy to the system which is destabilizing). All these terms are often used to describe this destabilizing force (generated from the cross-coupled stiffness) which acts in the same manner as direct damping (colinear with the instantaneous velocity) but in opposite direction to direct damping thus the term negative damping. Positive damping dissipates energy and in this manner it reduces vibrations and whirl amplitude, while the negative damping adds energy to the dynamic system and therefore increases vibrations and amplitude of motion. What may determine the final stability of the rotor-impeller-bearing-seal dynamic system is often dependent on the net effect of these two opposing forces; direct damping and the destabilizing cross-coupled stiffness (negative damping) present in the bearings, seals, and impellers.

The degree of the destabilizing cross-coupled stiffness in a fixed geometry bearing is mostly influenced by the circular geometry in the bearing and the fluid rotation. Although load and eccentricity also play a role in the stability of these bearings, the circular geometry and fluid rotations are directly related to the bearing design and configuration. These two parameters can be changed and/or modified to help an unstable

or marginally stable bearing become more stable. Therefore, fixed geometry bearings can achieve a higher stability threshold by adding axial grooves as shown in Figure 5 to reduce the fluid rotation. The more grooves that are added, the more the net fluid rotation in the bearing is reduced, and the higher the stability threshold. The increase in the stability threshold is demonstrated in Figure 6 which shows an increase in the log dec for the first forward mode (the mode most likely to go unstable) as a function of the number of axial grooves. There is a limit to the addition of grooves in the bearing where it can become counter productive. This is because as we add more grooves, we are effectively reducing the load area, increasing the bearing temperature, and eventually reducing the direct damping. There is also the possibility of the load vector being directly in line or very close to the groove which will affect the flow of oil into the bearing. Thus it is important to note that improving one aspect of the bearing performance may have deleterious effects on other operating parameters. Therefore, a complete bearing design must examine all aspects of the bearing performance and limits.

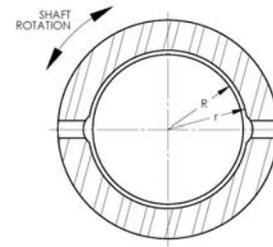


Figure 5. Sleeve bearing with two axial grooves

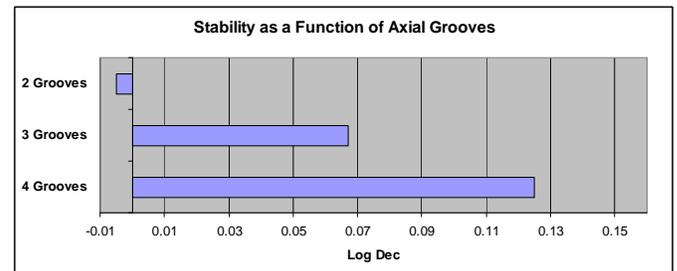


Figure 6. Stability as a Function of Axial Grooves

Additional and more practical gains in stability with fixed geometry bearings can be achieved by altering the purely circular geometry. This is achieved by using preloaded lobes; often referred to as elliptical or lemon bore in the case of two lobe bearings, and a preloaded three lobe or four lobe bearings as shown in Figures 7, 8, and 9. The elliptical two lobe bearing and the preloaded three and four lobed bearings are bi-directional and this feature makes them more commonly used. The elliptical bore essentially provides a geometric preload on the shaft journal. Typical preloads used in elliptical bearings can range from 0.25 to 0.5. A preload of 0.5 means that the machined bore (bore machined with shims at the split line) will have a clearance twice of the assembled bore (vertical bore once the shims are removed from the split line). Thus the horizontal clearance in this case is twice as large as the vertical

clearance and provides a significant asymmetry in the fluid film direct stiffness coefficients. The plot in Figure 10 shows the effect preload has on the stability of a rotor bearing system. The plot shows that there is an optimum value for preload and as we keep increasing the preload the stability will start to degrade. This is because the stiffness in the bearing increases with increasing preload and as a result of that, the damping becomes less effective. Another important characteristic of the preload optimization plot shown in Figure 10 is the slope of the curve below and above the optimum point. In the lower preload range (zero preload to 0.25), the curve is steeper and any wear will cause the stability to drop significantly. Above the optimum point (0.25 to 0.5), the slope appears to be more gradual and suggest that a proper design for long term operation should target a preload value slightly above the optimum point.

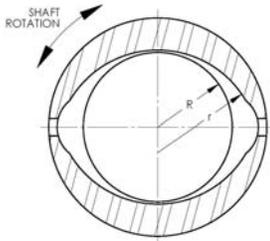


Figure 7. Elliptical or Lemon bore journal bearing

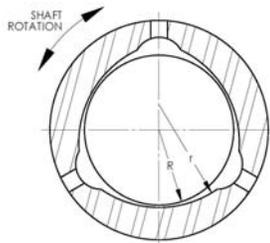


Figure 8. Three lobe preloaded journal bearing

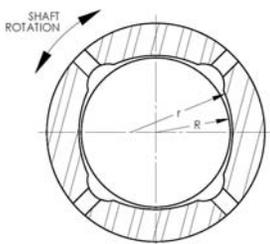


Figure 9. Four Lobes preloaded journal bearing

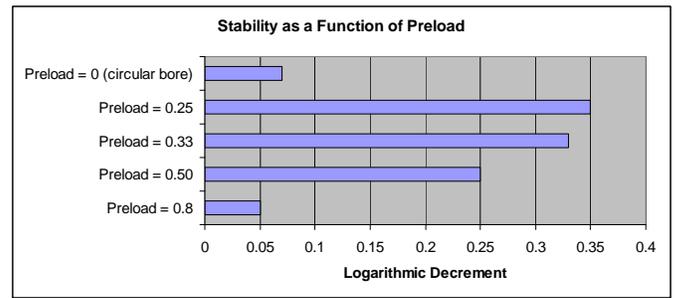


Figure 10. Influence of Preload on Stability

The offset half bearing shown in Figure 11 also produces a more stable bearing in comparison to a circular sleeve bearing. The offset half can be a circular bearing with the upper and lower halves offset or it can be an elliptical bearing with an offset between the upper and lower halves. Both the elliptical and offset half bearings produce a significant asymmetry in the fluid film direct stiffness coefficients. Tripp and Murphy [4] have shown how this asymmetry combines with the cross coupled stiffness coefficients to influence the total energy added to the dynamic system. Energy added is destabilizing, while energy dissipated due to direct damping is stabilizing. The net effect on stability can be physically explained by the schematic shown in Figure 12 and the following equation:

$$E_{cyc} = A(K_{xy} - K_{yx}) \quad (1)$$

The energy added to the system is calculated by integrating the force due to the bearing cross coupled coefficients over the displacement around the closed curve of the whirl orbit. Therefore, the energy added (destabilizing) to the dynamic system is equal to the product of the whirl orbit area (A) times the net value of $(K_{xy} - K_{yx})$. The more asymmetric the whirl orbit, the smaller the orbit area and the less destabilizing energy is added to the dynamic system. As is true with most engineering applications, there is a penalty associated with this approach. Asymmetry, which tends to reduce the destabilizing influence of cross-coupled stiffness, results in higher synchronous vibration amplitudes along the axis with the lower stiffness. The asymmetry may also cause a split critical speed as shown in Figure 13, which depending on its proximity to the operating speed may reduce the safe operating range of the machine. Operating between the peaks of the split critical speed can also make the machine very sensitive to rub since the rotor may be executing line to line or backward whirl (backward precession) under such conditions. This effect was measured and reported by Gruwell and Zeidan [5] and is also demonstrated with the backward whirl orbit when operating between the two peaks of the critical speed shown in Figure 13. The arrow in the upper left hand corner of the orbit plot indicates the direction of rotation, while the whirl motion is deduced from the blank-dark dot precession which is in the direction of rotation below the first peak and against between the peaks (split critical) and then in the direction above the second peak.

Although asymmetry in the support may increase the synchronous vibrations, it is an easier problem to deal with. When faced with an unstable system where the sub-synchronous component can grow rapidly and unpredictably, or with a system that is more stable but with a higher synchronous vibrations; the choice is simple. You can always control the synchronous response and can find means of reducing its magnitude through balancing, etc. The stability is always of a higher concern and requires immediate attention and priority.

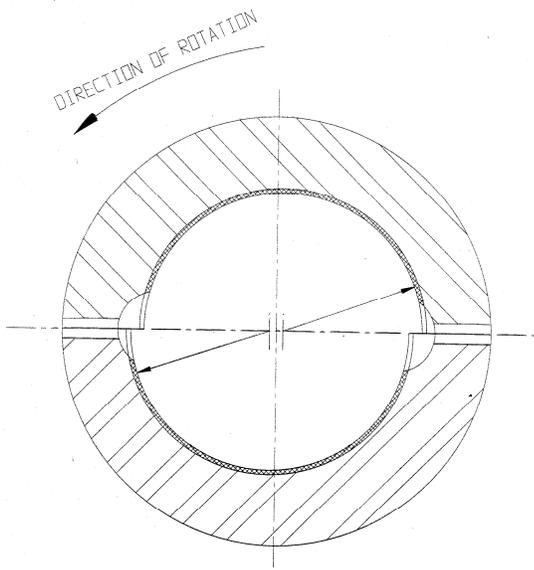


Figure 11. Offset half sleeve bearing

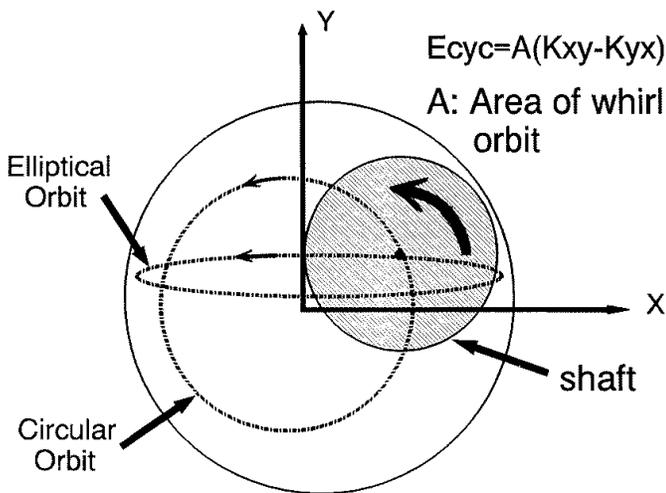


Figure 12. Schematic showing the stabilizing effects of asymmetry

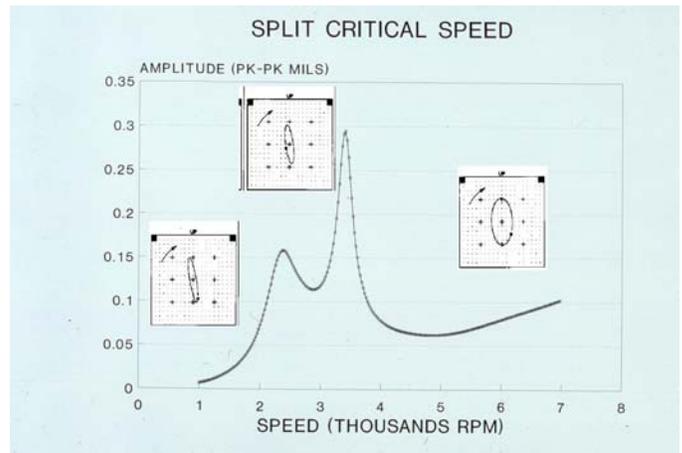


Figure 13. Split critical and backward precession or backward whirl

Another graphic representation from [6] and described in Chapter 3 on the effect of orbit ellipticity is worth mentioning here as well. The follower force or destabilizing force resulting from the cross-coupled stiffness coefficients tends to be in line with the instantaneous velocity of the shaft only when the orbit is circular. This is when the follower force is most effective and in fact most destructive as it transfers the destabilizing energy into the rotor dynamic system. When the orbit is elliptical, the follower force and the instantaneous velocity are no longer in line, and the effectiveness of the follower force in adding destabilizing energy to the rotor is diminished.

The canted bore bearing is a special type of the three lobe preloaded bearings with a taper or a fixed tilt. The lobes in this bearing configuration are continuously converging across each segment in the direction of rotation. A schematic of such a bearing configuration is shown in Figure 14 and it resembles a preloaded pad with a fixed pre-tilt about a large offset pivot location in the direction of rotation. This bearing is used in some high speed integrally geared compressors. They offer very high stiffness, which is not necessarily a good thing for most rotordynamic issues. Cavitation is a major problem with these bearings when used in high speed applications.

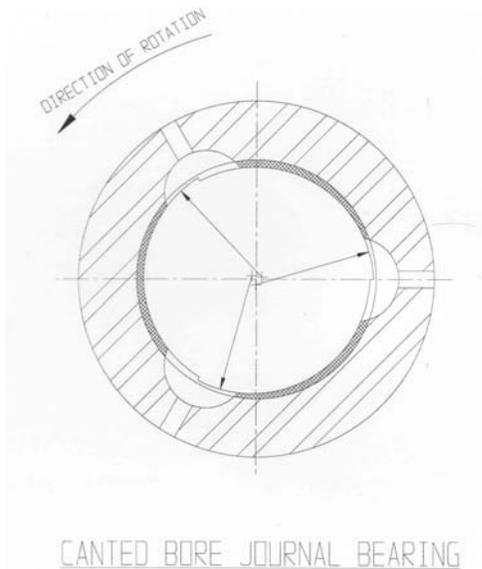


Figure 14. Canted Bore Journal Bearing

The pressure dam bearing, shown in Figure 15, is similar to a plain two-groove sleeve bearing with one relief track in the bottom half in the mid section of the bearing or two relief tracks at the ends. In the upper half it has a relief track in the mid section of the bearing and extends from the split line to around 120 to 135 degrees in the direction of rotation. The relief track in the upper half comes to an abrupt sharp edge or “dam”. The sharp edge was thought to be very critical to the proper operation of the pressure dam. Childs et al., [7] showed that not to be true and in fact a radius was determined to provide better stability than an abrupt or sharp edge at the bottom of the dam. There are two main features that make the pressure dam bearing more stable. The first and most important is due to the pressure from the fluid inertia effects hitting the dam and generating an additional downward load on the journal. This forces the shaft into a position of greater eccentricity and, consequently, greater stability. The other aspect is due to the relief in the bottom half of the bearing which essentially reduces the effective length of the bearing. This serves to increase the operating eccentricity and thus further enhances the stability of the bearing. The important parameters that influence the effectiveness of the pressure dam bearing is the optimum depth and width of the dam, the angular position of the dam, and the width of the relief track in the bottom half. These parameters must be optimized to achieve higher stability values while still maintaining good synchronous response characteristics. A good starting value for the optimization process for the pressure dam depth is a value five times the radial clearance based on Someya’s layout [8]. Nicholas and Allaire [9] performed extensive theoretical and experimental calculations and found this value to be between 2 to 3 times the radial clearance for stability. Often improving the stability through the use of a pressure dam bearing results in higher amplification factors when traversing the critical speed as shown by Carlson and Zeidan [10].

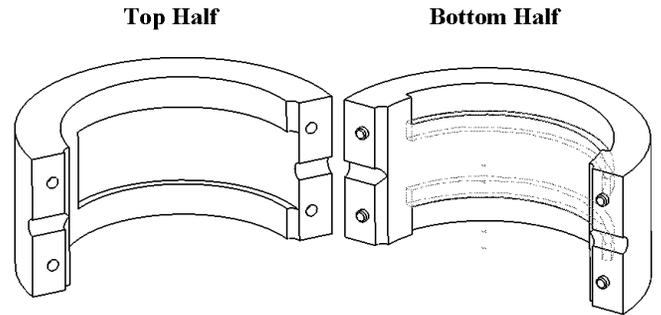


Figure 15. Pressure Dam Bearing

All the bearings discussed thus far fall in the category of what is commonly referred to as fixed geometry or fixed profile journal bearings. The geometry of the bearing is fixed (no moving parts) and cannot be changed or adjusted to different loading conditions. These bearings are simple and therefore very economical to manufacture. They are very limited in stability when used in machinery operating super critical (above their first critical speed) and are very susceptible to variations in operating parameters such as load, alignment, and increase in wear and clearance. A general guide on the relative stability of the different fixed geometry bearings can be assessed from the plot shown by Zeidan and Paquette, [11].

One of the major limitations of the fixed geometry bearings in general is the fact that they are designed for a certain load magnitude and direction. In speed increasing and speed reducing gears, as shown in Figure 16, the load magnitude and direction can cover a wide angle making it very difficult for these bearings to perform well under such conditions. This is also true in integrally geared compressors and expanders where the radial load is the net effect of the weight of the rotor which must be added vectorially to the gear reaction forces that are a function of the torque or power applied. In pumps, the side load from the pump volute must also be taken into account. In power generation and process steam turbines, the partial steam admission can also change the magnitude and direction of the load as shown in Figure 17. A bearing that is designed for a load in the bottom lobe could end up with a load in line with the oil feed groove. These are some of the drawbacks of fixed geometry journal bearing and the reasons why in such equipment they (fixed geometry bearings) have a distinct limitation.

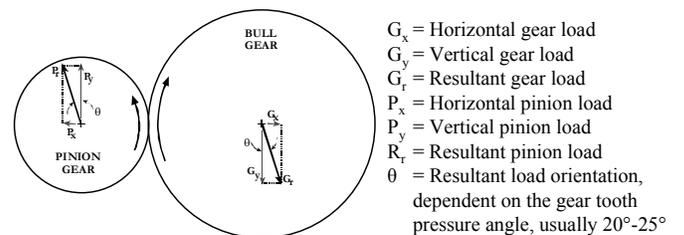


Figure 16. Gear Reaction Forces in a speed increasing gear box

NOZZLE FORCES ON POLAR PLOT

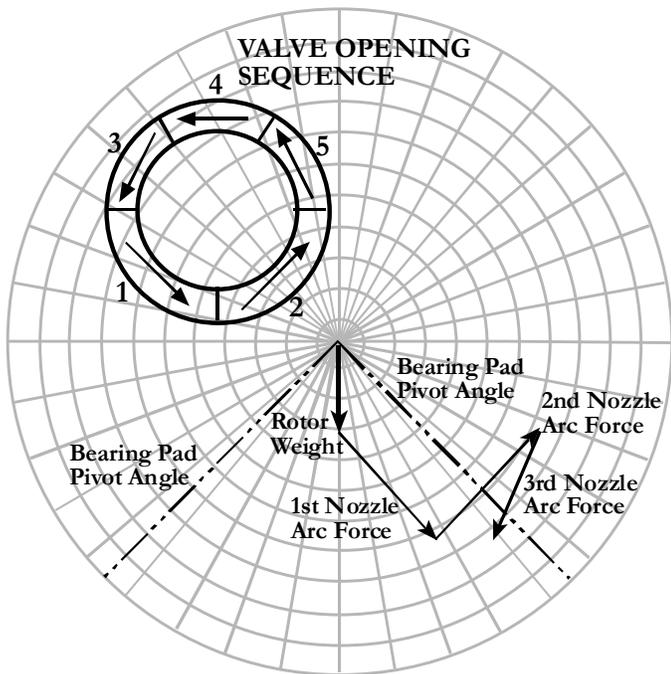
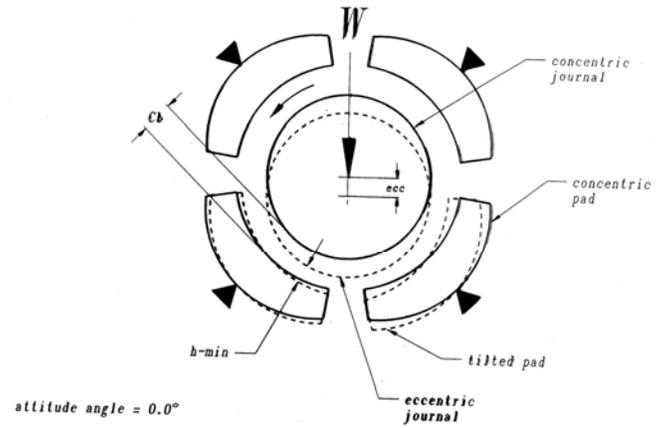


Figure 17. Partial steam admission effects on bearing load magnitude and direction

VARIABLE GEOMETRY TILTING PAD BEARINGS

Variable geometry tilt pad bearings are characterized by the inherent stability that arises from the low or negligible cross-coupling present in these bearings. The pads tilt or rotate about their pivot in response to the radial load applied by the shaft journal, and will always produce a reaction force that is in line with the journal center as shown in Figure 18. There is no attitude angle (attitude angle is zero) and therefore no corresponding cross-coupled stiffness. The attitude angle is almost zero provided the pad inertia and the friction in the pivot are low or negligible.

Conventional tilt pad bearings use a point or a line contact for the pivot and achieve the tilt motion through rocking motion. The contact stresses can be very high on these bearings particularly when used in integrally geared compressors or when the radial load or weight is high. Under these circumstances, the pivot wear and brinnelling will accelerate and cause a loss in the preload factor and a drop in stiffness and damping. Depending on the proximity of the critical speed to the operating speed, this may result in a loss of the separation margin and a further degradation in the bearing pivot leading to further increase in synchronous vibrations. The loss in the preload can, and in many instances does, reduce the preload and cause the rotor-bearing dynamic system to become unstable.



Geometry of a Tilting Pad

Figure 18. Schematic of a tilt pad Bearing

The spherical pivot tilt pad bearing shown in Figure 19 and is often referred to as ball-in-socket tilt pad bearing. This bearing tends to reduce the contact stresses in the pivot and allow the pad to tilt in the circumferential direction as well as in the axial direction. The latter (axial tilt) allows the bearing to avoid edge loading caused by the shaft sag or misalignment between the rotor and the bearing. These bearings are advantageous for retrofitting older equipment since they provide a higher tolerance to deteriorating field conditions. They work very well when the radial loads applied are moderate. However, when used in high speed integrally geared compressors, the highly loaded pads can lock in a fixed position under load. This can cause the bearing to behave like a fixed geometry bearing resulting in sub-synchronous vibrations.



Figure 19. Ball-in-socket Tilt Pad Bearing

The conventional variable geometry tilt pad bearings offer greater stability than the fixed geometry sleeve bearings, but this benefit comes at a cost and brings some additional drawbacks that we need to be aware of. By virtue of having multiple parts, the tilt pad bearings are more costly to make and require longer lead time. The unloaded pads are inherently susceptible to pad instability often referred to as pad flutter. The phenomenon is limited to the unloaded pads and often

leads to Babbitt fatigue at the leading edge of the pad as demonstrated by Adams [12]. In some applications, the pad instability can lead to significant wear in the pivots of the unloaded pads, increasing the clearance and significantly influencing the performance of the bearing.

The multi-piece assembly in conventional tilt pad bearings, where each major component is made with a set of manufacturing tolerances, can lead to a significant tolerance stack-up during assembly. The stack-up can be a significant percentage of the bearing clearance particularly for the smaller size bearings (5 inches and lower). This has a significant effect on the range of preload that can be achieved and subsequently on the dynamic bearing coefficients

The drawbacks with the conventional tilt pad bearing have given rise to what is known as Flexure Pivot™ tilt pad bearing shown in Figure 20. This bearing achieves the tilt necessary for bearing stability through flexure of the web supporting the pad as shown in the schematic of Figure 21. The construction of the Flexure Pivot™ tilt pad bearing eliminates the multiple piece features inherent with conventional tilt pad bearings, thus eliminating the relative movement between parts which produces wear. The fact that the pad, pivot, and bearing shell are all made from the same piece of material in the Flexure Pivot™ bearing also reduces the tolerance stack up as illustrated in the schematic of Figure 22. The fact that the unloaded pads cannot drop from their pivots, and the higher rotational stiffness in these pivots, are two factors that are necessary for pad flutter to take place. These features in the Flexure Pivot™ prevent pad flutter from taking place or move the threshold speed at which this takes place to a much higher speed than is attainable with conventional tilt pad bearings.

The possibility of achieving the optimum preload is greatly enhanced with the Flexure Pivot™ bearing due to the narrow range for preload variation. Furthermore, the manufacturing process that utilizes Computer Numerical Controlled (CNC) combined with state of the art wire Electric Discharge Machining (EDM) ensures a very high repeatability. Every part performs in the same manner and there is no variation in the dynamic coefficients between different batches of manufactured bearings.

Tilt pad bearings, like fixed geometry sleeve bearings, come in many variations and configurations. These bearings have many parts and therefore a potential for more variations and configurations. These variations allow greater flexibility for the designer and rotordynamicist to tailor the bearing for each specific application. These variables include the orientation of the bearing for a load on pivot (LOP) versus a load between pivots (LBP), the number of pads used, the pad pivot offset angle, axial length of the pad, clearance, lubricant viscosity, and preload are all factors that can be modified to achieve the desired thermal and rotordynamic characteristics. These factors will be examined further to provide a better understanding of their use in the bearing design and influence on the rotordynamics of modern turbomachinery.

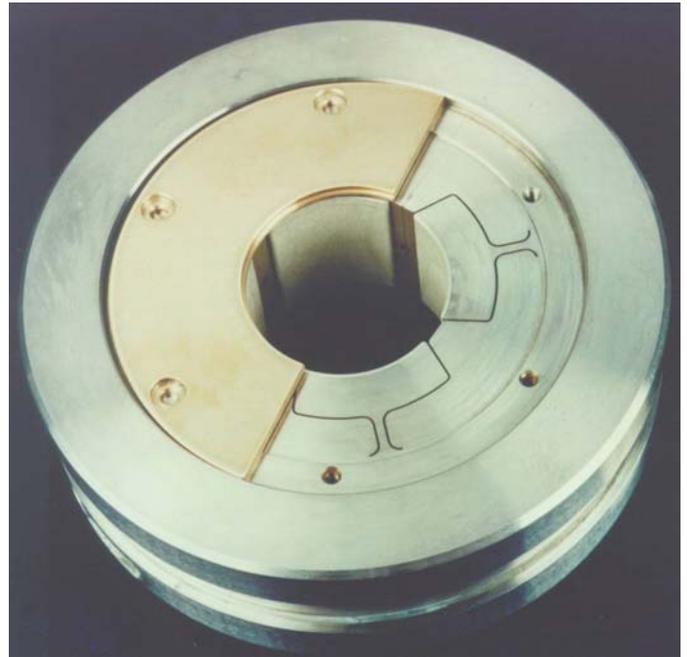
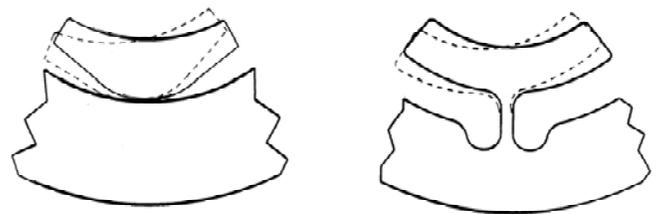


Figure 20. Flexure Pivot™ Tilt Pad Bearing (Courtesy of KMC, Inc.)



Rocker Back Tilt Pad

Flexure Pivot™ Tilt Pad

Figure 21. Schematic of Rocker back and Flexure Pivot™ tilt pad bearings

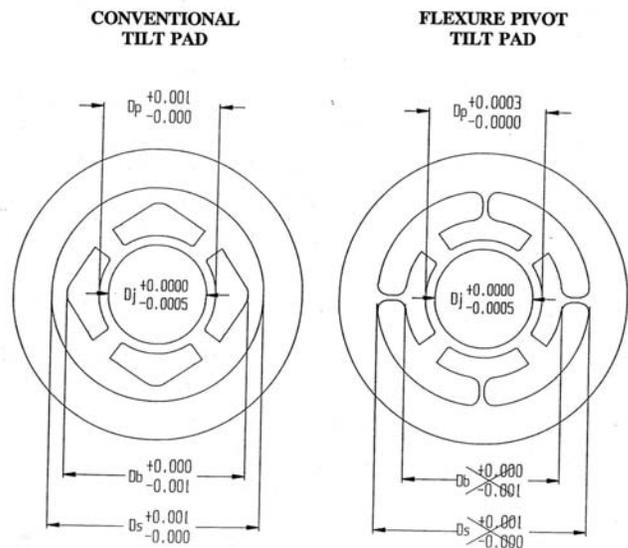


Figure 22. Tolerance stack up as a result of manufacturing tolerances in tilt pad bearings

LOAD BETWEEN PIVOT (LBP) VS. LOAD ON PIVOT (LOP)

The orientation of a tilt pad bearing with respect to the radial load plays an important role as this directly relates to the stiffness and damping characteristics of the bearing. The schematic shown in Figure 23 illustrates the orientation with respect to the load vector for a LOP and a LBP tilt pad bearing. The vertical and horizontal stiffness shows a large stiffness asymmetry for the LOP orientation as shown in the plot in Figure 24. Whereas the LBP shows very symmetric characteristics and the vertical and horizontal stiffness curves are virtually identical and cannot be discerned in the plot. Furthermore, the LBP configuration tends to allow both bottom pads to share the load and thus this configuration offers higher load capacity than the LOP configuration.

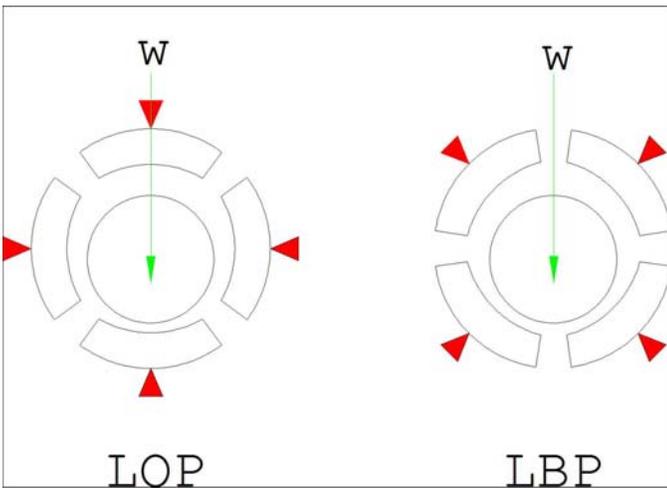


Figure 23. Schematic representation of LOP vs LBP in a four pad bearing

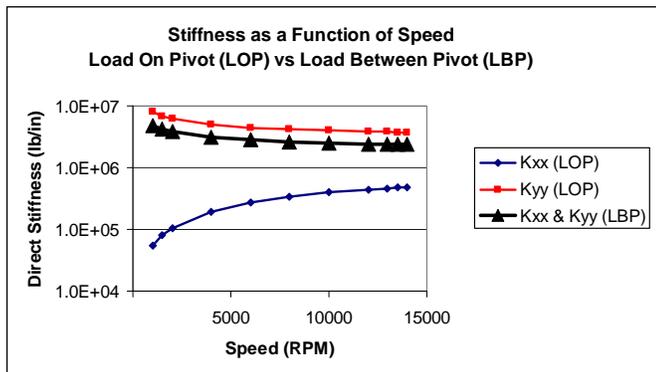


Figure 24. Direct stiffness coefficients for a 4 pad bearing in LOP vs LBP configuration

While it is clear that the LBP offers higher load capacity and most likely better synchronous response characteristics, there are applications in which the LOP offers a better solution. The benefits of the LOP configuration become more apparent when coupled with a high degree of aero cross coupling. The stability analysis for an industrial high pressure centrifugal compressor showed that the basic log dec (the log dec evaluated with no aero cross-coupling or any destabilizing seal

effects) was higher for the LBP configuration as expected. This is shown in the eigen analysis and mode shape plots for the first forward mode in Figures 25 and 26 for the LBP and LOP respectively. However, when the aero cross coupling effects were also included in the model, the log dec for the LOP dropped as expected but not at the same rate as for the LBP case. The stability for the first forward mode with the LOP bearing configuration resulted in a more stable system than with the LBP. This is mainly due to the beneficial asymmetry effects introduced by the bearings which more than compensate for the lower damping. The mode shape for the first forward mode with the LBP and LOP with aero cross coupling included is shown in Figures 27 and 28. One can clearly see the difference in the orbit shape for each of the two bearing configurations and the high degree of ellipticity introduced by the LOP configuration.

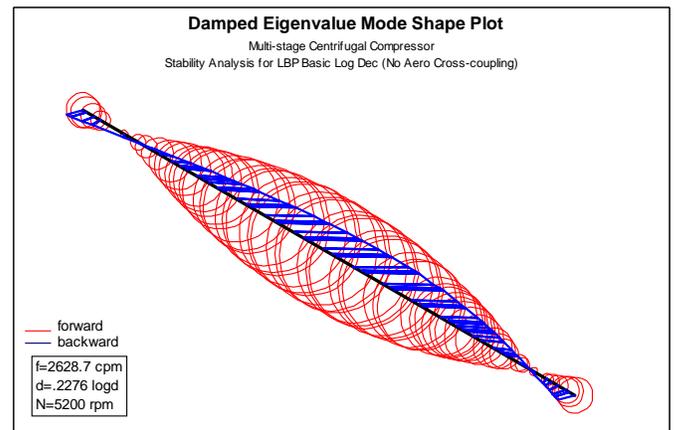


Figure 25. Natural frequency for first forward mode with basic log dec for LBP bearings

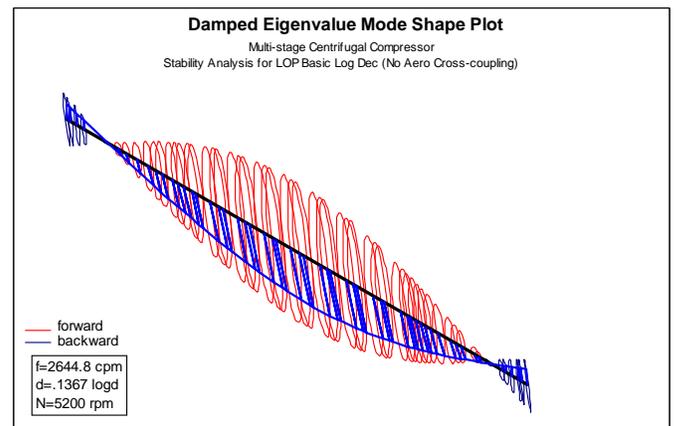


Figure 26. Natural frequency for first forward mode with basic log dec for LOP bearings

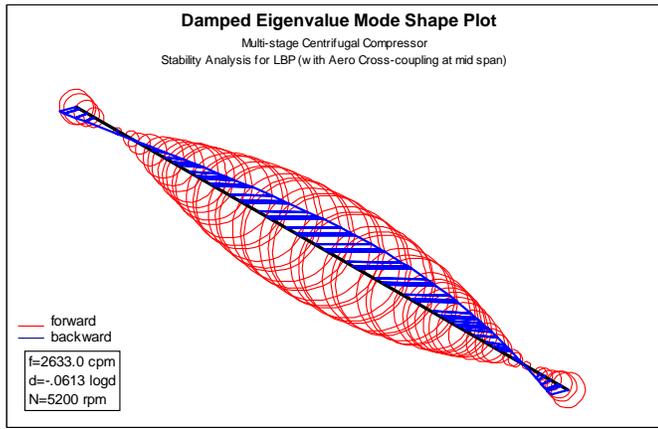


Figure 27. Natural frequency for first forward mode with aero cross coupling for LBP bearings

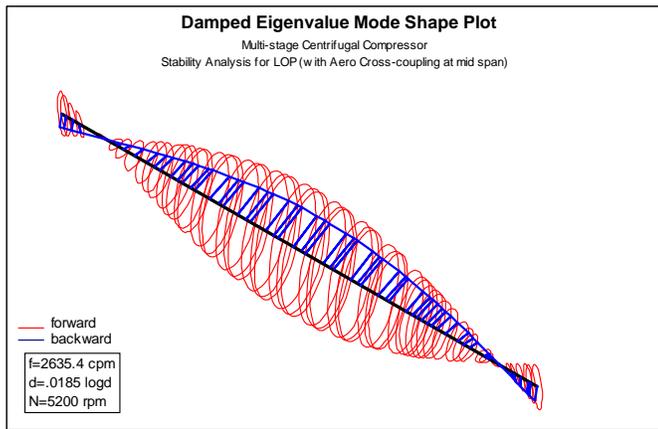


Figure 28. Natural frequency for first forward mode with aero cross coupling for LOP bearings

However, when we examine the response to imbalance, it is clear there is a distinct advantage for the LBP configuration. The response plot shown in Figure 29 clearly shows a more damped and symmetric response exhibited by the identical amplitudes predicted for the vertical and horizontal response. The amplification factor is also lower for the LBP with 11.08 as opposed to the 17.58 for the LOP configuration shown in Figure 30. The LOP response also shows a split critical which is introduced by the asymmetric stiffness characteristics for this bearing configuration. A comparison for the two bearings configuration is summarized in Table 1 and shows that the

synchronous response is better with the LBP. The LOP is only advantageous when relatively high cross-coupled stiffness values are present.

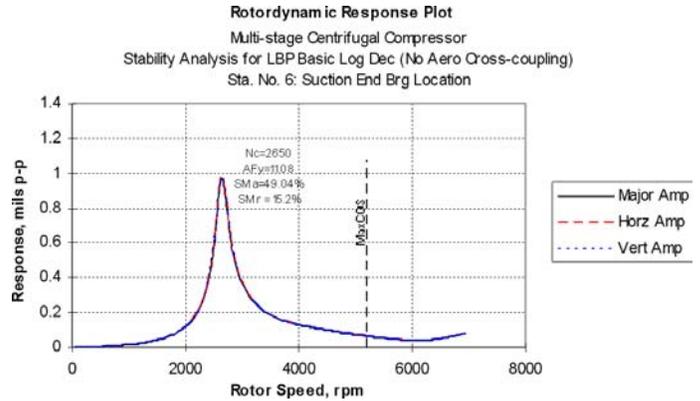


Figure 29. Unbalance Response with LBP bearings

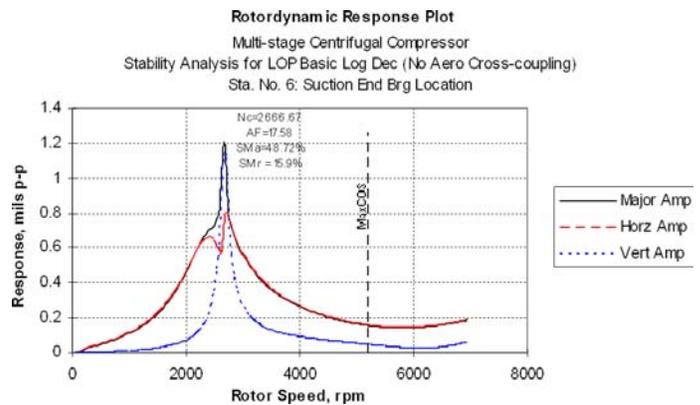


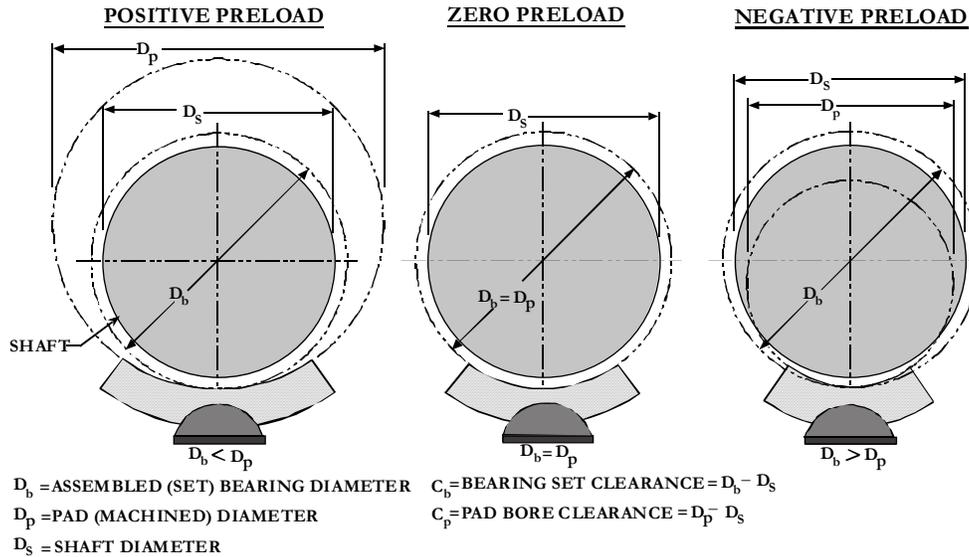
Figure 30. Unbalance Response with LOP bearings

INFLUENCE OF PRELOAD ON THE DYNAMIC COEFFICIENTS IN TILT PAD BEARINGS

The geometric preload in tilt pad bearings is a very critical and effective parameter to alter the magnitude of the force coefficients. Preload can be explained in the graphic shown in Figure 31. A positive preload is essential also from the stand point of providing a larger inlet for the oil at the leading edge of the bearing. In contrast, zero preload due to manufacturing tolerance can quickly lead to a negative preload. This will reduce the effectiveness of the oil entering the leading edge of the bearing and will cause pad flutter or pad instability.

Table 1. Stability and Imbalance Response Summary for LBP and LOP 4-pad Bearing
Summary for Stability and Imbalance Response for LBP vs LOP

Bearing Configuration	Logarithmic Decrement		Amplification Factor
	Basic Log. Dec. (No Aero Cross Coupling)	Log. Dec. (with Aero Cross Coupling)	
4 pad LBP	0.2276	-0.0613	11.08
4 pad LOP	0.1367	0.0185	17.58



$$\text{PRELOAD} = 1 - \frac{C_b}{C_p} = 1 - \frac{D_b - D_s}{D_p - D_s} = \frac{D_p - D_b}{D_p - D_s}$$

* D_b CAN BE CHANGED IN CERTAIN DESIGNS TO ALTER THE PRELOAD ONCE THE BEARING HAS BEEN MANUFACTURED. OTHER FACTORS THAT MAY INFLUENCE THE PRELOAD IN OPERATION ARE THERMAL DISTORTION OF THE PAD, AND MECHANICAL DEFORMATION OF THE PAD AND PIVOT SUPPORT.

Figure 31. Schematic representation of positive, zero, and negative preload in tilt pad bearings

Increasing the preload in the bearing as shown in Figure 32 will result in higher stiffness particularly at the higher Sommerfeld numbers. Increasing the stiffness in the bearing may be desirable from the stand point of shifting a critical speed located just above operating speed further up in frequency to provide a better separation margin. However, it is also important to note that while increasing the preload will increase the stiffness, it will result in a lower effective damping. The lower stiffness is essential for the damping to be effective. There have been several incidents when one would attempt to provide a higher separation margin from synchronous vibrations by increasing the preload and stiffness, and the end result is a sub-synchronous instability due to the loss of effective damping. Therefore, the total rotordynamic consequences of a change should address both; stability (Eigen analysis), as well as synchronous response (Forced analysis). The latter determines the critical speed location and the safe separation margin between the critical speed and the operating speed.

Direct Stiffness as a Function of Sommerfeld Number

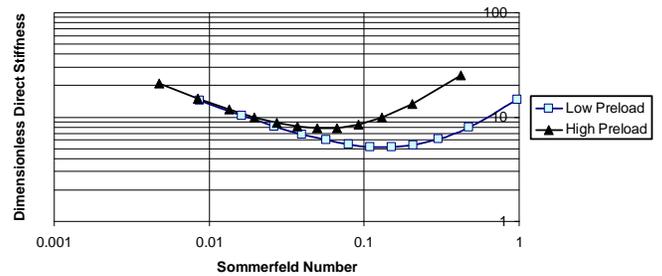


Figure 32. Effect on preload on the direct stiffness

INFLUENCE OF THE BEARING LENGTH OR PAD LENGTH

Although manufacturers tend to standardize on a certain L/D ratio for their bearings, field conditions and varying process requirements may push the standard bearing design close to its limit. One of the parameters that can be changed to impart a desirable change in the stiffness and damping coefficients in a bearing is the pad length. This approach has been utilized in addressing stability problems as well as critical speed problems. One such field problem that the third author experienced and left a lasting impression in his mind was the redesign of a bearing to help shift the critical speed on a high speed compressor. A meeting was held to address this critical speed that was just above operating speed. This meeting was attended by the chief compressor engineer and the chief turbine engineer representing the two product lines offered by this manufacturer. While everyone agreed that the problem should be addressed by

changing the length of the bearing, the two chief engineers offered opposing views on the effects the length has on the bearing and its stiffness characteristics. The compressor chief engineer, who is experienced with relatively light rotors and running at high speeds and thus high Sommerfeld numbers, suggested that increasing the length will increase the stiffness. On the other hand, the turbine engineer whose experience focused on heavier rotors running at lower speeds, and therefore generally lay in the lower range for the Sommerfeld numbers, stated that reducing the length would result in higher eccentricity and therefore higher stiffness. What was unique about this dilemma was the fact that both chief engineers were correct in their assessment based on the range of equipment they were experienced with. In the case of the turbine engineer who was more used to dealing with driven equipment. This equipment tends to be relatively heavy and runs at lower speeds at least in process and conventional power generation equipment. A heavier rotor and lower speed means that the range of Sommerfeld number covers the left side of the dimensionless stiffness curve shown in Figure 33. When operating in that region, decreasing the length in the expression for the Sommerfeld number will result in a lower number for the Sommerfeld number and moves further to the left increasing the stiffness. On the other hand, driven equipment such as compressors in this case tend to be relatively light and run at higher speeds, and therefore in the higher range of the Sommerfeld number. In this case, increasing the length in the Sommerfeld expression will increase the Sommerfeld number and shift the value to the right in the right half of the curve shown in Figure 33, thus resulting in a higher stiffness.

Direct Stiffness as a Function of Sommerfeld Number

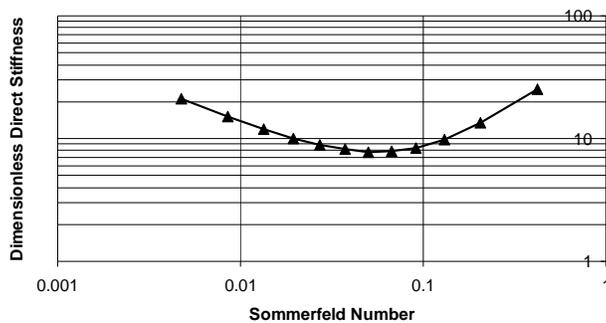


Figure 33. Dynamic stiffness as a function of Sommerfeld number

INFLUENCE OF THE PIVOT OFFSET

The pivot offset is generally expressed in dimensionless form as the ratio of the angle from the leading edge to the pivot divided by the pad arc. The influence of the pivot location or offset position has been established in the use of tilt pad thrust bearing to increase load capacity and also reduce operating temperatures. The schematic shown in Figure 34 shows that with the offset pivot, the pad rotation will result in a larger gap at the leading edge and thus more cool oil or lubricant will enter the pad. Although both of these effects; load capacity and

cooler temperatures, are also true when applied to journal (radial) bearings, the offset can also be beneficial from the distinct rotordynamic characteristics such a parameter has on the dynamic coefficients.

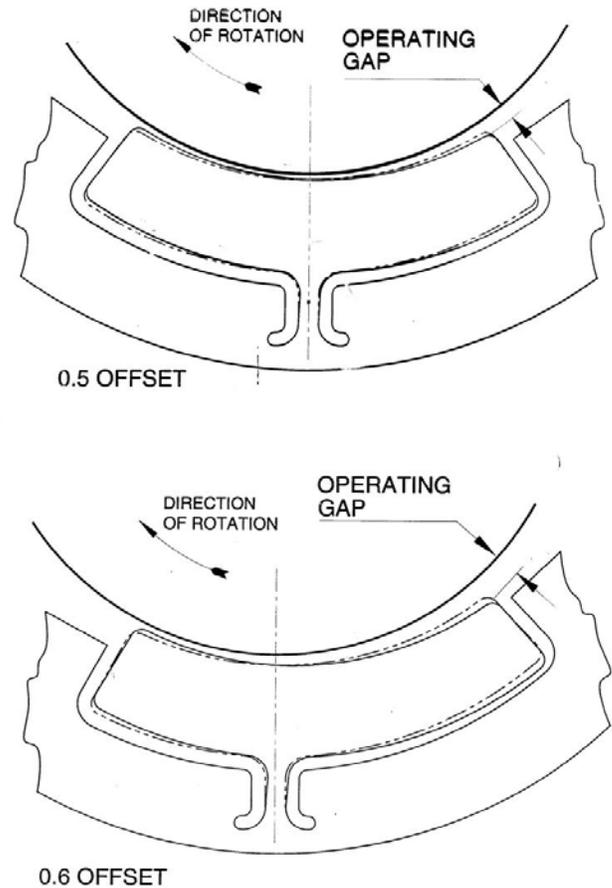


Figure 34. Schematic of center pivot (0.5 offset) and 0.6 offset pivot bearing

Integrally geared compressors such as the one shown in Figure 35 tend to have a relatively large overhung mass (the impellers) and run at very high speeds and very high loads imparted by the gear reaction forces. The bearings designed from these compressors often utilize the offset pivot feature to provide lower temperatures at these speeds and loads as well as higher stiffness. The compressor in reference [13] operated above two rigid modes and above a flexible mode with a conventional center pivot bearing. The use of the Flexure Pivot™ offset pivot bearings allowed operation above the two rigid modes but below the flexible mode as shown in the Undamped Critical Speed map presented in reference [13]. This is significant in terms of allowing the unit to tolerate more imbalance and eliminating the necessity for trim balancing. The other important advantage offered with the offset pivot is the reduced sensitivity to bearing clearance when evaluating stability. The unit had a larger range of stability with increased clearance while it had a much narrower range with the center pivot pads.

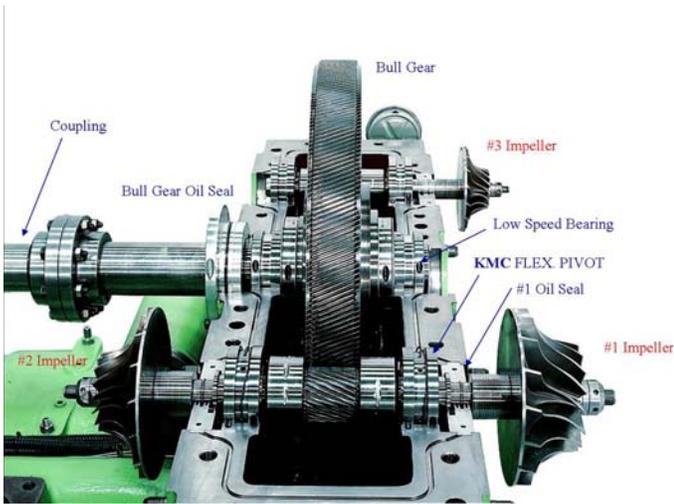


Figure 35. Typical Integrally Geared Compressor

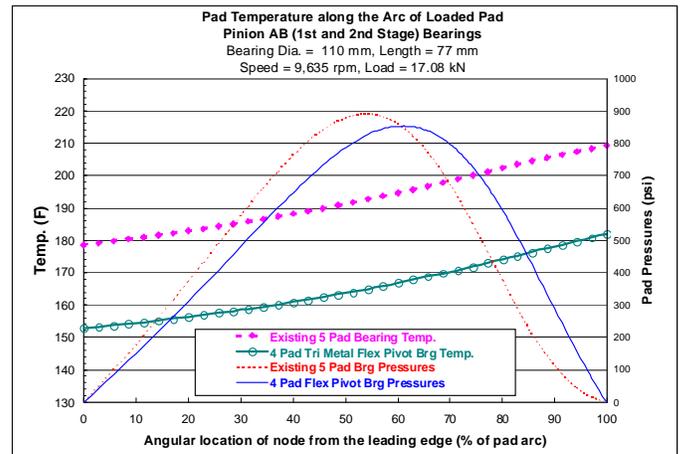


Figure 36. Temperature profile from leading to trailing edge of 5-pad and 4-pad bearing

INFLUENCE OF THE NUMBER OF PADS

The optimum number of pads in the case of thrust bearings has been established and centers on having an optimum aspect ratio. This more or less determines the optimum number of pads for thrust bearings. This is not the case for journal (radial) bearings as there are other aspects that need to be considered, in particular, their influence on the rotordynamics of the system. The predominant number of pads used in tilting pad journal bearing has been 5 pads in the past, but this trend is migrating towards the use of four pad bearings today.

The 4-pad bearings tend to provide a larger unit load area than a 5-pad bearing and thus can carry a higher load. They also tend to provide better synchronous response characteristics, and these two advantages are the main reason why more of the new equipment uses 4-pad bearings. The plot shown in Figure 36 shows the advantages offered with the 4-pad in a high load and high speed integrally geared compressor. The x-axis has been normalized to provide the temperature profile for both the pad in a 4-pad and in the 5-pad since they have different arc length. The four pad bearing runs at roughly 30° F lower temperature although this is partly due to the offset pivot configuration as well.

There are also applications particularly in gears or integrally geared compressors where operation dictates different loading direction and magnitude. The bearing dynamic stiffness coefficients will see a significant variation in the direct stiffness as the load changes directions from LOP to LBP. This variation in the direct stiffness can be minimized as the number of pads is increased as illustrated in Figure 37. Some gears use 7-pad tilt pad bearings for this same reason. In some instances the limited radial space dictates the use of more pads in order to reduce the pad deflections. The narrower radial space may result in very thin pads with long arcs and these may experience excessive thermal and mechanical deformations. Another reason for using more pads or in effect a shorter pad arc is the potential in the unloaded pads to experience pad flutter. The larger the number of pads and the shorter the pad arc, the higher the threshold for pad flutter or pad instability. Another aspect that is not commonly considered when selecting the number of pads, but should be accounted for in some applications is the increase in startup torque as the number of pads is reduced.

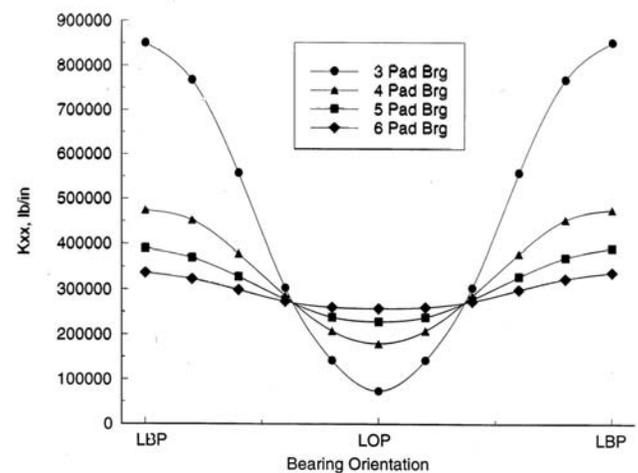


Figure 37. Variation in stiffness for different # of pads as the load swings from LBP to LOP

The swing in load direction with tilt pad bearings does produce cross-coupled stiffness coefficients even when the pivot inertial

and friction are neglected. This is shown in Figure 38 and is attributed to the coordinate transformation. The cross-coupling in this case is not destabilizing since the K_{xy} and K_{yx} are of the same magnitude and sign. These cross coupled coefficients do not produce any destabilizing force since they will always sum to zero.

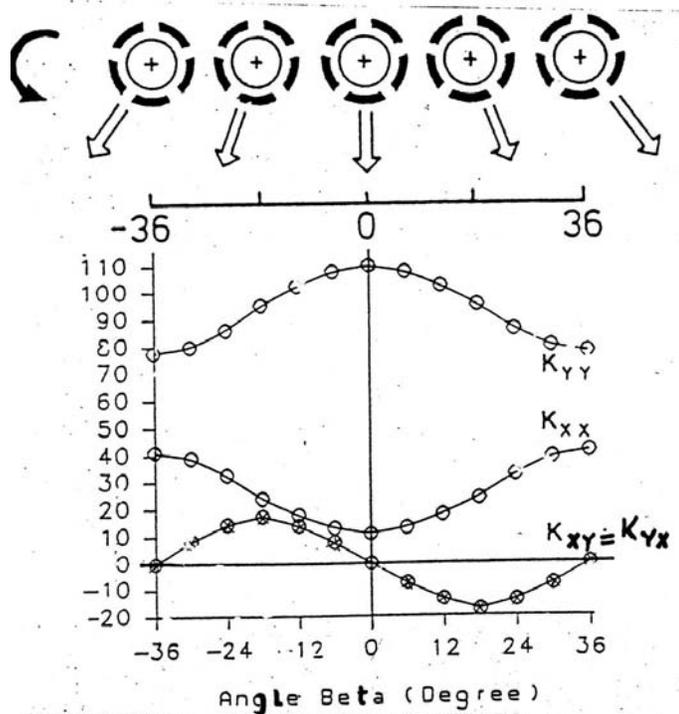


Figure 38. Cross Coupled coefficients in a tilt pad bearing when load is between LOP and LBP

ACKNOWLEDGMENTS

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