APPLICATION OF HIGH SPEED AND HIGH PERFORMANCE FLUID FILM BEARINGS IN ROTATING MACHINERY

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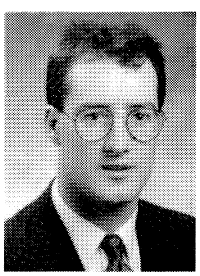

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ABSTRACT

Some of the critical parameters in the design and application of high performance fluid film bearings are emphasized. The limitations and problems associated with high speed and highly loaded bearings will be discussed. Examples of bearing failures, the symptoms associated with these failures, and their impact on the machine performance will be shown. Some of the common failure mechanisms will also be described with suggestions on how to eliminate the failures or reduce their consequences by changes to some of the bearing design features. New developments in bearing technology and testing specifically designed to address some of these limitations will be demonstrated. Case studies and analysis will be used in many common and newly developed turbomachinery equipment to help illustrate some of the key attributes in the design and application of high performance fluid film bearings and squeeze film dampers.

INTRODUCTION

The term high speed and high performance bearings is used in this tutorial to describe bearing applications that have a significant impact on the rotordynamic stability and performance characteristics of rotating equipment. Bearings whose performance characteristics are critical to the reliability and longevity of a rotating machine are included in this category. This will also encompass special applications of bearings outside what are usually considered normal operating parameters or practice.

Tilting pad bearings have become the standard with most rotordynamically sensitive and critical rotating equipment. Although these bearings are more complex and more expensive than the simple fixed geometry sleeve type bearings, their inherent stability characteristics make them the bearing of choice in most modern high speed and high performance rotating equipment. The low cross coupling provided with most variable geometry "tilting pad" bearings eliminated many of the bearing related stability problems. However, as is the case with most engineering problems, a solution that addresses a specific problem may generate some side effects and additional problems. These problems are generally more obscure and some may not become apparent until the bearings have been in service for some time. The degradation with time may be gradual leading the user to believe that the phenomenon is merely generated by normal wear of the bearing components. Quite often the bearings are replaced at scheduled maintenance intervals when in fact the problem is due to a design defect and should not be classified as a routine maintenance activity.

The focus herein is on these problems and provides the user with some of the common symptoms associated with such design flaws. The solutions commonly used to eliminate such failures or increase the life expectancy of the bearings and the machine are also addressed.

To facilitate the comprehension of some of the cases presented and the problems associated with them, a review of fluid film bearing fundamentals is due.

BEARING LOAD CAPACITY

The load on a journal bearing is commonly expressed in terms of unit loading. The unit loading is determined by dividing the rotor weight (portion of the rotor weight carried by the bearing) by the bearing projected area. The bearing projected area is simply the product of the length and diameter of the bearing. The unit loading is a rough estimate used only to gauge the load carried by the bearing in a particular application. It is important to remember that the pressure in the bearing is not uniform and that peak pressures can be two to three times higher than the unit load. Bearing loads in the lower end of the load spectrum fall in the 50 to 100 psi range, while the middle range of the load spectrum has unit loads in the 150 to 250 psi range. The upper end of the load spectrum, which is typical of some of the gear loaded applications and heavier rotor installations, is in the range of 300 to 350 psi. Although no mention is made of the lubricant for the above limits, it is assumed that lubricating oils in the common viscosity grades are used.

The classical ZN/P curve shown in Figure1 can provide better insight into the load capacity of a bearing and can account for variables such as the viscosity and speed in the expression for the load capacity. The ZN/P curve is divided into three operating regimes which are classified as follows:

- Full Film Hydrodynamic Lubrication
- Mixed Film Lubrication
- Boundary Lubrication

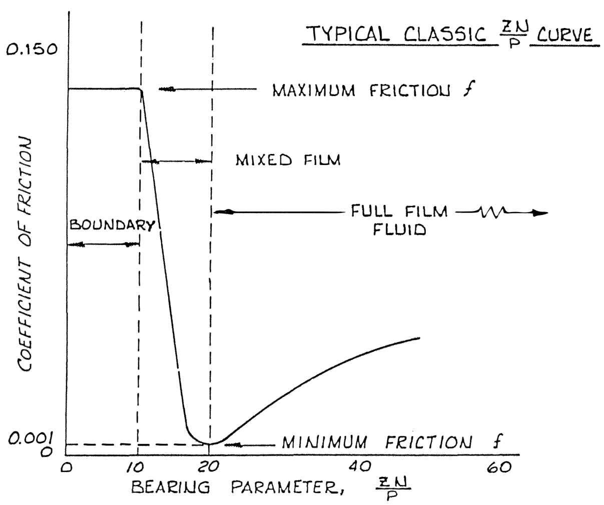

Figure 1. Classical ZN/P Curve.

Under full film lubrication conditions, no contact takes place between the metallic surfaces. In mixed film and boundary lubrication, the oil film is too thin to completely separate the journal and bearing surfaces. In this case, the oil is capable of carrying only part of the load, while the remainder is carried by contact between the two surfaces. As the load is increased, speed is reduced, or oil temperature is increased (reduced viscosity), a point is reached where the oil plays little or no part in carrying the load. This condition is referred to as boundary lubrication. Boundary lubrication and mixed film or thin film lubrication are two modes of operation in which friction and wear are affected by properties of the contacting surfaces, and by properties of the lubricant at the microscopic level.

While the ZN/P curve is not totally satisfactory as an absolute criterion, it illustrates the characteristic coefficient of friction curve plotted directly against speed and viscosity, and inversely against load. There are, however, other factors that influence the load capacity that are not reflected in the ZN/P curve. These factors include the journal diameter, surface finish, cleanliness of the lubricant, boundary lubrication properties of the bearing surface, relative hardness of the two surfaces, and the thermal conductivity of the bearing.

The speed can also be a limiting factor that is not completely reflected in the simple ZN/P curve. It appears that the load capacity can be increased as the speed is increased. Although this is true at low speeds, what is not reflected in the ZN/P curve is the fact that the higher speed will result in more churning and heat generation as well as hot oil carry over. This will in turn reduce the oil viscosity and subsequently the load capacity of the bearing. The higher temperature due to the higher speeds will also reduce the babbitt strength, further limiting the load capacity of the bearing. Therefore, the surface speed and the unit load should both be considered in evaluating the load capacity and limit of a certain bearing application. The surface speed is expressed in feet per second. The upper limiting value typical of some of the highly loaded bearing applications is usually set to 300 ft/sec. The authors have used optimized bearings with directed lubrication to further increase the speed and load limits of journal bearings. The design features that can help increase the load and speed limits are discussed in a later section.

A similar load capacity curve exits for thrust bearings. The ZN/P expression for journal bearings is replaced by ZU/PB, where Z is the viscosity, U is the surface velocity at the pad bolt or pivot circle, P is the unit loading, and B is the pad circumferential length. This last term is important in thrust bearings, because it makes the expression dimensionless and accounts for the circumferential length of the pad.

MEASUREMENT OF BEARING CLEARANCE AND CRUSH

Bearing clearance is one of the most important parameters in the operation of a bearing. Therefore, it is important to determine the installed clearance and the bore contour and concentricity to the outside fit diameter.

The final bearing clearance is influenced by the contour of the housing where the bearing is installed and also by the amount of interference or crush between the bearing shell and the housing. Proper crush is crucial in the operation of high speed and critical machinery. Improper crush can lead to either a hot bearing or a loose bearing fit. A loose bearing may contribute to synchronous or subsynchronous type vibrations. It is critical that a metal-to-metal fit to 0.002 in of crush on diameter be maintained for a proper bearing installation. This criterion applies to most thick shell bearings and linings. For thin shell bearings, and in situations where the rigidity of the housing is much greater than that of the bearing liner, the values for crush should be arrived at by consulting with the bearing and equipment manufacturer. What should be guarded against in these cases is excessive crush or a contact stress that could cause local yielding of the material. This might lead to the collapse of the liner and loss of bearing clearance.

In some special applications, and due to ease of assembly and other limitations dictated by the design, the cold bearing fit might be a loose fit. As the bearing heats up during operation, thermal expansion will eventually result in a tight fit. On the other hand, some applications might start with an interference fit at cold assembly conditions. However, at certain operating conditions, the bearing housing might expand more than the bearing shell, resulting in a loose fit.

In certain gear applications, the bearings might not have as tight a fit as the one recommended above. There are usually two reasons for a lighter fit in bearings designed for gears. One is to guard against distortion in the gear box top half and the oil leakage, which may ensue at the split line due to an excessive interference fit at the bearing. These problems may arise due to the fact that the bearing crush in the case of a gear box is sometimes accomplished by the top cover which is relatively flexible. The other reason that may justify a lighter fit is the fact that the static loading in gear applications is usually very high due to the gear separating forces. This tends to keep the bearing seated and does not allow the dynamic load to move it around.

The following is a recommended procedure to check bearing crush and is illustrated in Figure 2: Place shims of equal thickness (Ts) along both sides of the split lines. Lay a strip of plastigage or lead wire on top of the bearing shell along the shaft axis. Install the bearing cap or strap and tighten all split line bolts. The plastigage or lead wire should indicate a thickness (Tf) equal to or less than the shim thickness used at the split line. The amount of interference (crush) is equal to the difference between the indicated clearance (Tf) and the shim thickness (Ts). A negative value for the crush indicates a loose bearing and the problem has to be rectified by replacing the bearing or using shims around the circumference to maintain an interference fit. The use of shims is generally not recommended, since it may block the oil feed and therefore should be used with caution and only as a temporary fix.

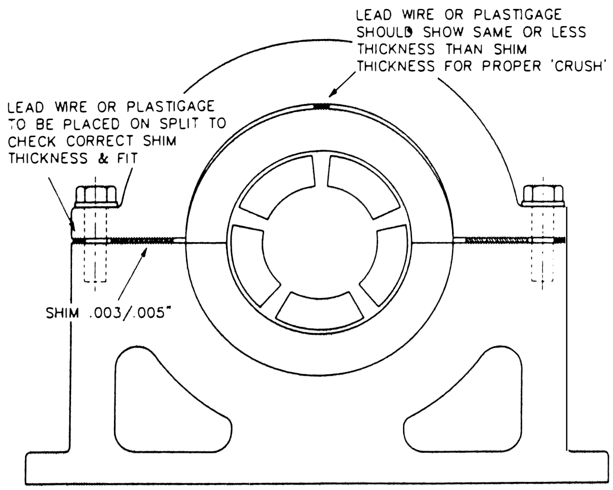

Figure 2. Procedure to Check Bearing Crush.

Once the desired crush has been obtained, the bearing clearance should be checked again to ensure that the crush is not excessive to the point of significantly reducing the bearing clearance. A lift check is commonly utilized to accomplish this task. It is important to recognize that the field lift check is fraught with problems that hinder it from providing an accurate measure of bearing clearance. Therefore, it should be used with caution and mainly to check gross problems and not as a qualifying check for acceptance or rejection of the bearing. Some of the problems encountered in a field lift check are, sticky or uncalibrated dial indicators, a soft foot on the machine, or a flexible housing support. Placement of the indicators along with the means by which the rotor is lifted can significantly influence the outcome of this measurement.

The lift check or bump check method shown in Figure 3 will result in values larger than the specified clearance on certain bearing configurations. This is due to the fact that under static conditions the shaft will sink between the two bottom pads on bearing configurations having a load between pad design. Similarly, the shaft will rise above the clearance bore (between the two top pads) at the upper end of the lift for certain bearings such as a four pad bearing with a load between pad configuration. Nicholas [1] presented a graphic description of the shaft sink. This is illustrated in Figure 4. A quick reference guide is

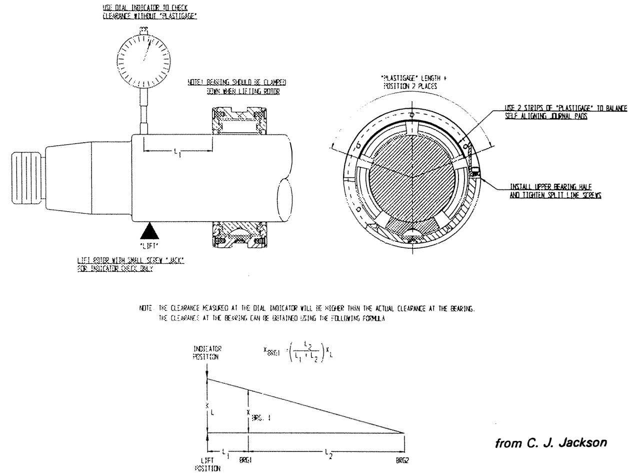

Figure 3. Lift Check Method for Checking Bearing Clearance.

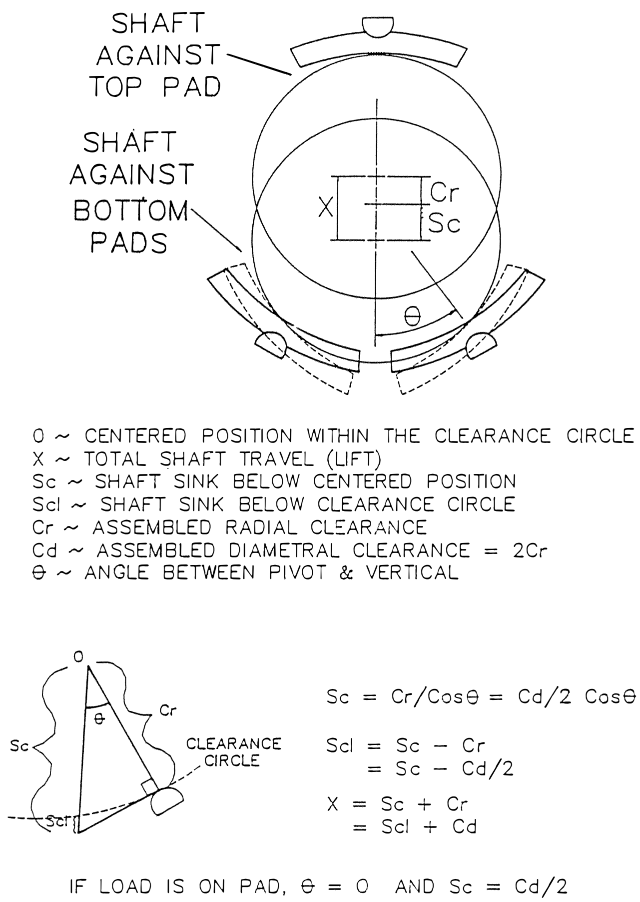

Figure 4. Graphic Description of Shaft Sink.

provided in Table 1 to determine the lift and shaft sink for the majority of tilt pad bearing configurations in use.

An alternate method to a lift check is a mandrel check. The lift check can only provide a measure of the vertical clearance in the bearing. The mandrel check can provide the clearance in all directions and therefore can also be used to check concentricity. This is shown in the schematic of Figure 5.

Table 1. Static Shaft Sink and Total Shaft Lift.

# of Pads	Load Config	Theta	Static Shaft sink below centered pos. Sc=Cd/2cosθ	Static Shaft sink below clear. circle Scl=Sc-(Cd/2)	Total Shaft Travel (Lift) X=Scl+Cd
3	LBP	60	Cd	0.5 Cd	1.5 Cd
3	LOP	0	0.5 Cd	0	1.5 Cd
4	LBP	45	0.7071 Cd	0.2071 Cd	1.4142 Cd
4	LOP	0	0.5 Cd	0	Cd
5	LBP	36	0.6180 Cd	0.1180 Cd	1.1180 Cd
5	LOP	0	0.5 Cd	0	1.1180 Cd
6	LBP	30	0.5774 Cd	0.0774 Cd	1.1547 Cd
6	LOP	0	0.5 Cd	0	Cd
7	LBP	25.71	0.5550 Cd	0.0550 Cd	1.0550 Cd
7	LOP	0	0.5 Cd	0	1.0550 Cd
8	LBP	22.50	0.5412 Cd	0.0412 Cd	1.0824 Cd
8	LOP	0	0.5 Cd	0	Cd

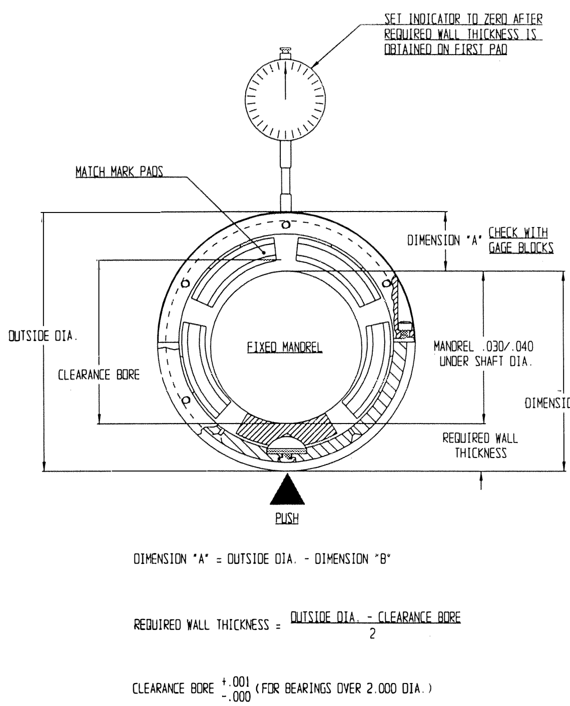

Figure 5. Mandrel Check Method for Checking Bearing Clearance.

STABILITY CHARACTERISTICS OF FIXED GEOMETRY SLEEVE BEARINGS

Journal bearings support the weight of the rotating shaft by developing a hydrodynamic pressure in the converging wedge, as shown in Figure 6. The unsymmetric pressure profile in the oil film is characteristic of fixed geometry journal bearings. This gives rise to an attitude angle between the line of centers and the

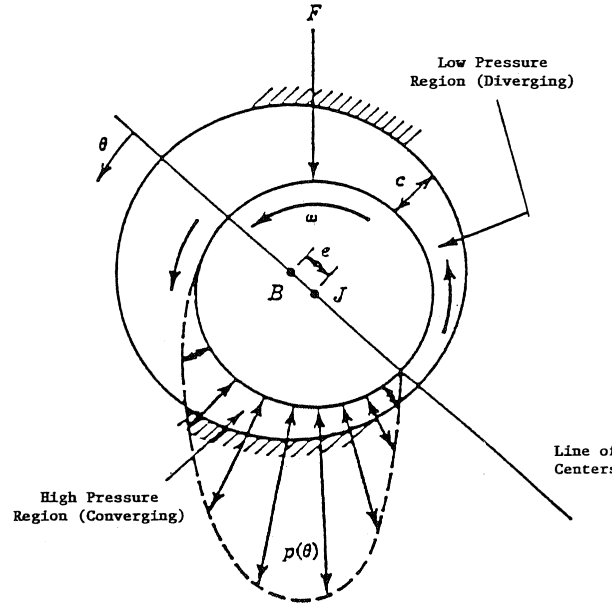

Figure 6. Journal Bearing Pressure Profile.

load vector. This characteristic, present in all fixed geometry journal bearings, is indicative of the presence of cross coupling in the bearing. The load acting directly downward on the journal bearing results in a displacement along the load direction (direct), and a displacement orthogonal to the load direction in the direction of rotation (cross).

Cross coupling characteristics are present in all rotating machinery operating in fluid film bearings or seals. Cross coupling has no equivalent in other structural nonrotating equipment. The difference between rotating and nonrotating structural vibration problems is illustrated in Figure 7. In a nonrotating structure, one expects the structure to deflect in the direction of the applied load as shown on the left side of the figure. On the right side of the figure, the structure, or rotating beam in this case, is allowed to rotate while a load is applied to the rotating shaft. A deflection in the direction of the applied load occurs as expected, but there is also deflection perpendicular to the applied load in the direction of rotation. This component is due to the cross coupling present between the vertical and horizontal directions. The cross coupling is generated by the fluid rotation in the annulus between the rotating shaft and the housing. It is the force generated

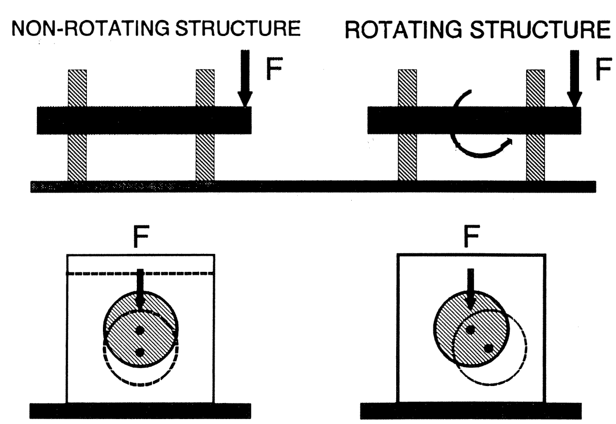

Figure 7. Fundamental Difference Between Rotating and Non-Rotating Structural Vibrations.

by the cross coupled stiffness that promotes self-excited vibrations and instabilities in rotating machinery. Vance [2] provides an excellent graphic presentation of the cross coupled stiffness force. This presentation is reproduced in Figure 8. The destabilizing cross coupled force is represented as a resultant of the force components due to the (Kxy) and (Kyx) stiffness terms. A positive (Kxy) and a negative (Kyx) will result in a net force that is tangential to the whirl orbit and is in the direction of rotation. The term "forward whirl driving force" is used to describe such a destabilizing motion. Follower force and circulatory force are other expressions also used to describe the destabilizing cross coupling force. The degree of cross coupling is greatly influenced by the fluid rotation and circular geometry present in fixed geometry bearings.

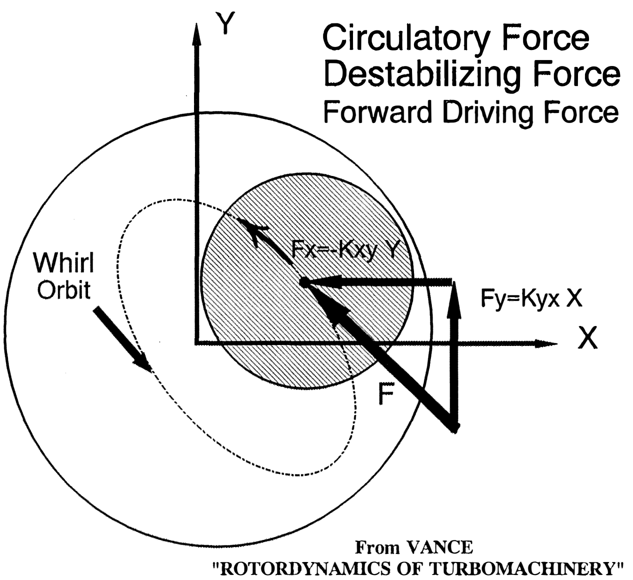

Figure 8. Cross Coupling in Fluid Film Bearings.

Fixed geometry journal bearings can achieve a higher stability threshold by incorporating modifications to reduce fluid rotation and alter the circular geometry. The addition of grooves in journal bearings tends to reduce the destabilizing effects due to fluid rotation. Additional gains in stability for fixed geometry bearings are realized through altering the circular geometry. This is achieved by using preloaded lobes, canted lobes, lemon shaped or elliptical bearings, and offset half bearings. A schematic of each of these bearings is shown in Figures 9, 10, 11, 12, and 13. As will be seen in one of the case studies on cavitation, these bearings, when used in high speed and high load applications, can experience severe cavitation, which limits their life in service. Some of these bearing geometries, such as elliptical and offset half bearings, introduce significant asymmetry which generally tends to enhance stability. Tripp and Murphy [3] have shown how asymmetry combines with the cross coupled stiffness coefficients to influence the total energy added to the dynamic system. Energy added is destabilizing, while the energy dissipated due to direct damping is stabilizing. The net effect on stability can be physically explained by the schematic shown in Figure 14, and the following equation:

$$E_{cyc} = A\,(Kxy - Kyx) \tag{1}$$

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 (Kxy-Kyx). 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 cou-

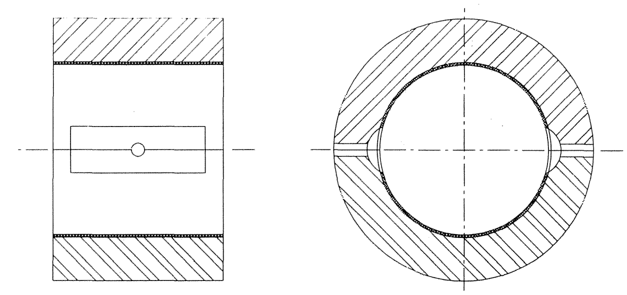

Figure 9. Plain (Circular Bore) Journal Bearing.

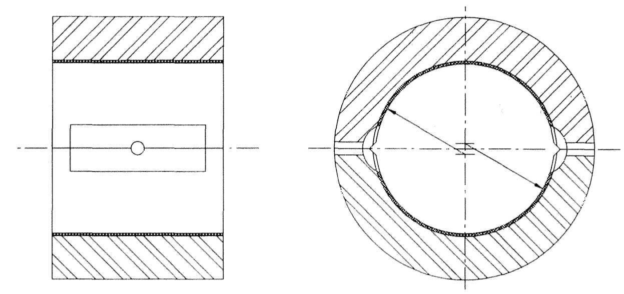

Figure 10. Lemon Bore (Elliptical) Bearing.

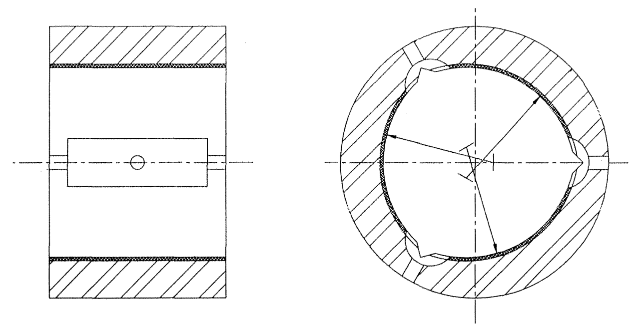

Figure 11. Three Lobe Bearing.

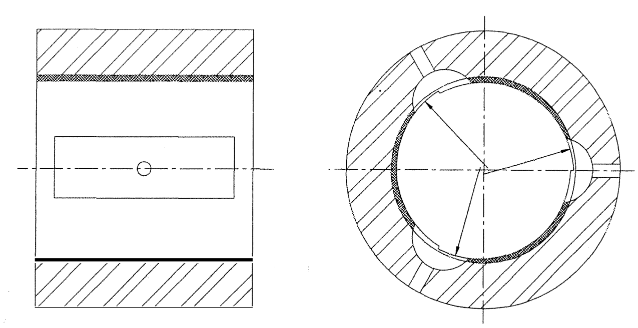

Figure 12. Canted Bore Bearing.

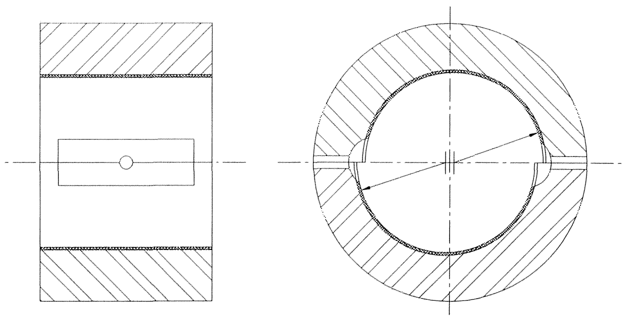

Figure 13. Offset-Half Bearing.

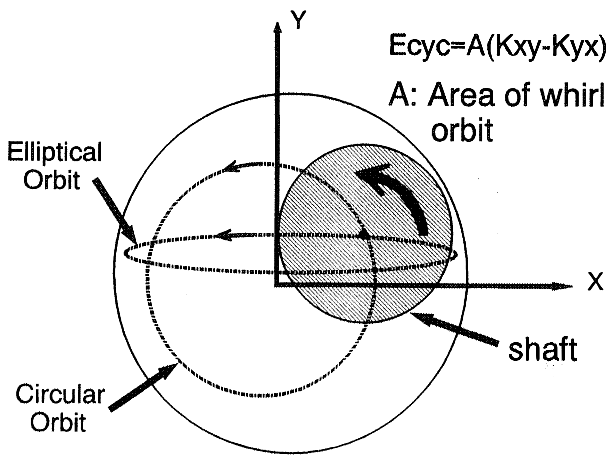

Figure 14. Effect of Asymmetry on the Destabilizing Energy.

pling, results in higher vibration amplitudes along the axis with the lower stiffness. This approach might also cause a split critical, as shown in Figure 15, reducing the safe operating speed range of the machine. Furthermore, a fixed geometry bearing design that enhances stability for a certain load condition may be more destabilizing when subjected to a change in the load direction or magnitude. This is particularly important in gear loaded applications where the machine is expected to run at part load and full load conditions.

Other problems that can arise and result in exceeding the limits of fixed geometry bearings have to do with the not-so-

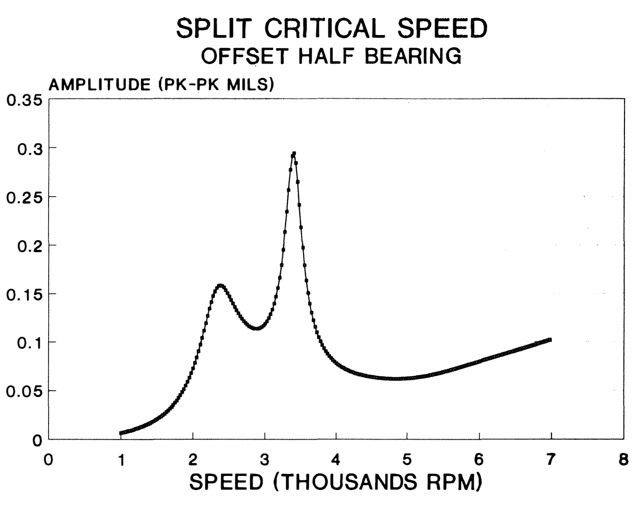

Figure 15. Split Critical Due to Asymmetry.

ideal conditions in the field. Misalignment, gear coupling lock-up, and partial steam admission in a steam turbine, are some of the common field conditions that can unload a bearing or slightly change the net load direction. This sensitivity to changes in load direction and magnitude can result in high synchronous or subsynchronous vibrations. Fixed geometry sleeve bearings are also more sensitive to changes in the running clearance.

Plain Sleeve Bearings

The plain sleeve bearing still continues to be used extensively because of its simplicity and low cost. The most common plain journal bearing is shown in Figure 9. It is horizontally split with axial oil distribution grooves along each horizontal joint. Oil is fed to these grooves at pressures anywhere from a few inches of Hg to pressures in excess of 30 psig. This bearing has a very limited stability threshold. The grooving that tends to break the oil film and reduce fluid rotation will increase the stability threshold. The more grooves present in the bearing, the higher the stability threshold and the logarithmic decrement, as shown in Figure 16. The logarithmic decrement (log dec) has been widely accepted as a measure of the stability of the rotor bearing system. The more positive the log dec value is, the more stable the rotor-bearing system. However, as the number of grooves increases, the effective bearing load area is reduced. This will result in a lower load capacity and higher operating temperatures. Thus, it is important to note that improving one aspect of the rotor 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.

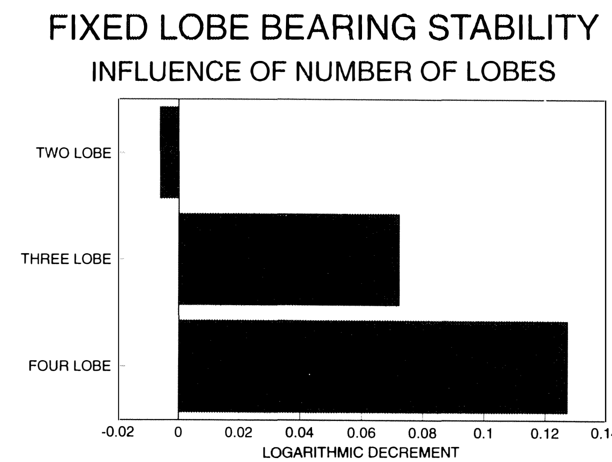

Figure 16. Stability Vs *Number of Bearing Grooves.*

Lemon-Bore Bearings

A lemon bore or elliptical bearing is shown in Figure 10. This bearing suppresses whirl by vertically preloading the two halves of the bearing. In marginally unstable bearing designs, these can provide satisfactory stability. The tight clearances in the vertical direction generate higher film temperatures. This bearing configuration is easy to manufacture by machining to the major bore diameter with shim stock placed at the horizontal splits.

Multilobe Preloaded Bearings

These can be made with a number of fixed bearing segments (lobes) bored to a larger radius than the bearing set clearance, thus creating a built in preload. These lobes can be three, four, five, to over ten in number. Three lobes and four lobes are the most common configurations. The bearing is conventionally manufactured by cam-boring each of the lobes. This manufacturing method limits the accuracy and precision of the geometric preload that can be obtained particularly in smaller size bearings (1.0 to 5.0 in). A three lobe bearing with load on pad configuration is shown in Figure 11. In order to illustrate the influence of preloading the lobes, the stability analysis was performed using these bearings on a multistage centrifugal process compressor. The results showing the logarithmic decrements for the first forward mode using different values for the preload are shown in Figure 17. Note that preload will improve the stability of the compressor compared to the zero preload case (circular bore). However, there is an optimum preload value after which stability will start to drop.

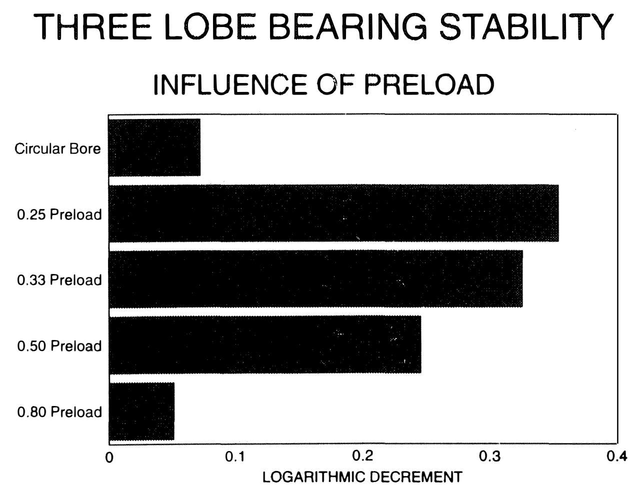

Figure 17. Stability Vs *Bearing Preload.*

There are also some hybrid configurations for the multilobe bearings. These will sometimes have a plain segment in the loaded quadrant and lobed or tapered segments in the remainder of the bearing.

Canted Bore Bearings

This is a special type of the multilobe preloaded bearings. 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 12. This bearing is used in some high speed integrally geared compressors. Cavitation is a major problem with these bearings when used in high speed applications as shown in a later section.

Offset Half Bearings

This is a unique type of the plain journal bearing. The offset half bearing has the upper and lower halves displaced transverse to the shaft axis a slight amount at the horizontal split line (usually about one half the radial clearance). A schematic of this bearing is shown in Figure 13. While more stable than the plain journal bearing, there is still a tendency for instability. They are usually more effective if clocked 45 degrees counter the direction of rotation from the standard horizontal installation, as shown in Figure 18.

Pressure Dam Bearings

The pressure dam bearing, shown in Figure 19, is similar to a plain two groove bearing except it has a relief track machined into the unloaded (usually upper) half. This relief comes to an abrupt sharp edge or "dam." Pressure due to the fluid inertia

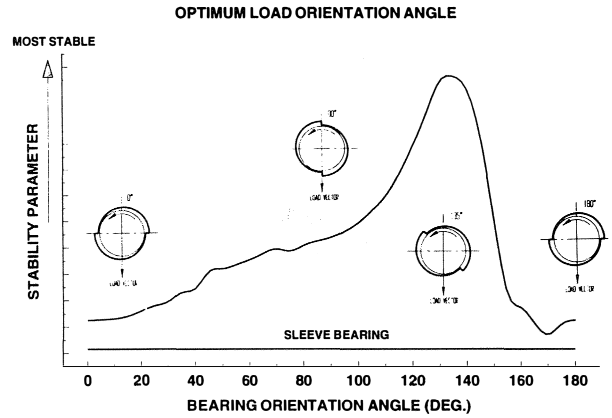

Figure 18. Optimum Orientation for Maximum Stability of an Offset-Half Bearing.

effects is generated at the dam imposing an artificial downward load on the journal. This forces the shaft into a position of greater eccentricity and, consequently, greater stability. The bottom half of the bearing also has relief tracks on both sides of the liner or a single relief track through the middle of the bearing. This relief essentially reduces the effective length of the bearing, which also serves to increase the operating eccentricity and thus the stability of the bearing.

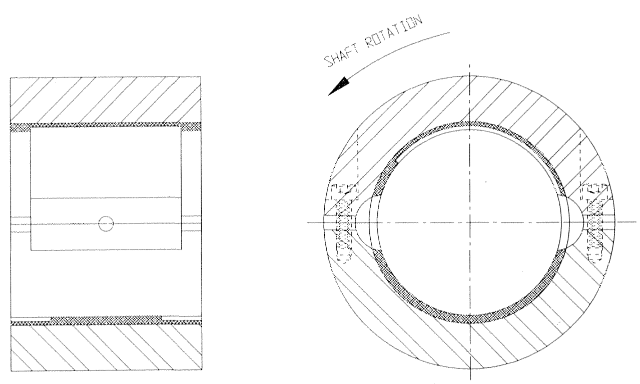

Figure 19. Pressure Dam Bearing.

Pressure dam bearings have greater power consumption than plain bearings and are more expensive to manufacture because of the precise machining required to produce the correct dam geometry. Important factors in the effectiveness of this design are the angular position of the dam and the precise depth and width of the relief track. These parameters must be optimized to achieve higher stability values while still maintaining good synchronous response characteristics.

In general, fixed geometry sleeve bearings are very economical to manufacture, very simple, and relatively cheap. They are very limited in stability and very susceptible to variations in operating parameters common in the not-so-ideal field conditions. A general guide of the relative stability of the different fixed geometry bearings can be assessed from the comparison shown in Figure 20.

CHARACTERISTICS OF VARIABLE GEOMETRY TILTING PAD BEARINGS

Variable geometry tilting pad bearings are characterized by the inherent stability that arises from the low or negligible cross coupling present in these bearings. The pads pivot and rotate in

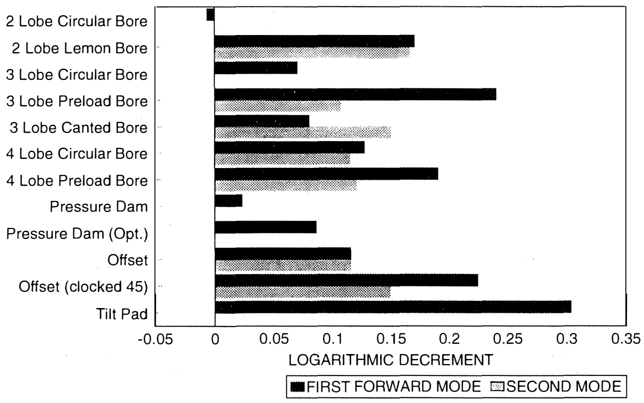

Figure 20. Stability Comparison for Various Fixed Geometry Journal Bearings.

response to the load applied by the journal, and always produce a symmetric reaction force on the journal. The attitude angle is zero, provided the inertia of the pad and pivot friction are neglected. These are the assumptions used in most computer programs developed for the computation of tilt pad bearing coefficients and performance. However, recent experimental investigations have shown that there is a significant amount of cross coupling present in tilt pad bearings.

Conventional tilt pad bearings achieve low cross coupling by pivoting on a point contact (point pivot), line contact (rocker back), or a spherical contact (ball-in-socket). The first design is the spherical point pivot design shown in Figure 21, in which a spherical button, mounted in either the pad or the housing, pivots on a hardened flat disc in the opposite member. This allows tilting in all directions, but because of high pivot stresses, this design is subject to pivot flattening and rapid increase in clearance. This may be minimized by using hardened mating surfaces. However, cracking of these supports has been observed under high impact loads.

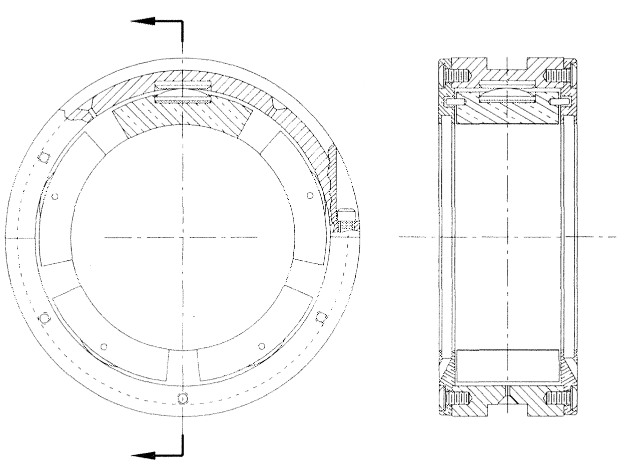

Figure 21. Spherical Point Pivot Tilt Pad Bearing.

The second design is the line contact rocker back pad design shown in Figure22. This design, which is the simplest and least expensive to manufacture, allows tilting motion in the circumferential direction only but none axially. Since the support contact is a line, the pivot stresses may be high, especially if good alignment is not achieved. Some rocker back bearings are supported by a key that rocks directly on the bearing housing, as shown in Figure 23. In this case, fretting and wear will occur on

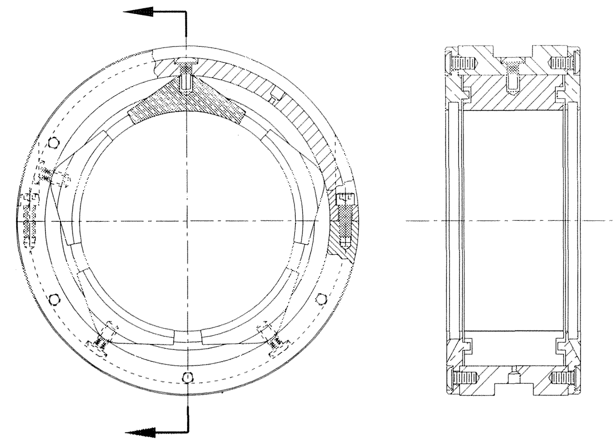

Figure 22. Rocker Back Tilt Pad Bearing.

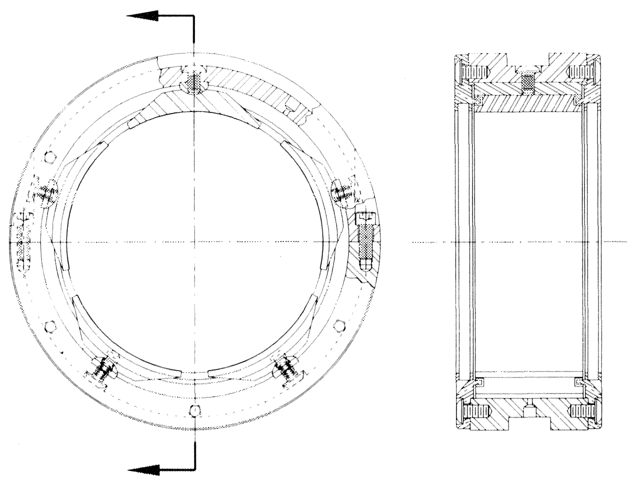

Figure 23. Rocker Back Tilt Pad Bearing with Key Supports.

both the pad pivots and the housing. Replacement of the pads will not be sufficient to restore performance, since the wear in the housing cannot be easily remedied.

The third design is a spherical surface pivot design with a matching spherical socket in the pad. As shown in Figure 24, the pad load is transmitted into the housing through the ball and socket arrangement. Under normal conditions the ball and socket size can be selected to quite easily control pivot stresses to a low level. At high loads and speeds special considerations in the design and manufacture of the spherical pivot and pad socket have to be addressed to reduce fretting and wear.

The contact stresses with conventional style tilt pad bearings are typically very high and can often lead to pivot wear and brinelling of the pad and bearing retainer surface as shown in Figure 25. The result is an increase in the set or assembly clearance, and consequently a reduction in the bearing preload factor. A lower preload leads to a reduction in the bearing stiffness, which in turn leads to a drop in the critical speed value. Since the majority of high performance turbomachinery operates just below the second critical speed, the pivot wear tends to drop that critical closer to the operating speed range, as shown in Figure 26. This phenomenon is generally self-propagating; as the vibrations increase, pivot wear will also increase, causing further increase in vibrations. This phenomenon and the bearing solution to reduce or eliminate its effect will be discussed in one of the case studies in a later section.

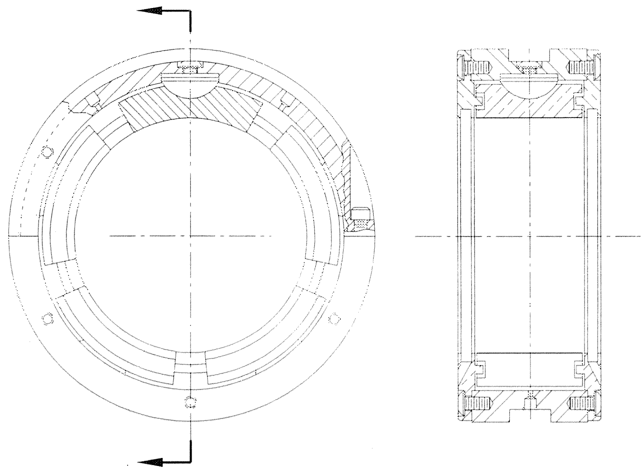

Figure 24. Ball-in-Socket Tilt Pad Bearing.

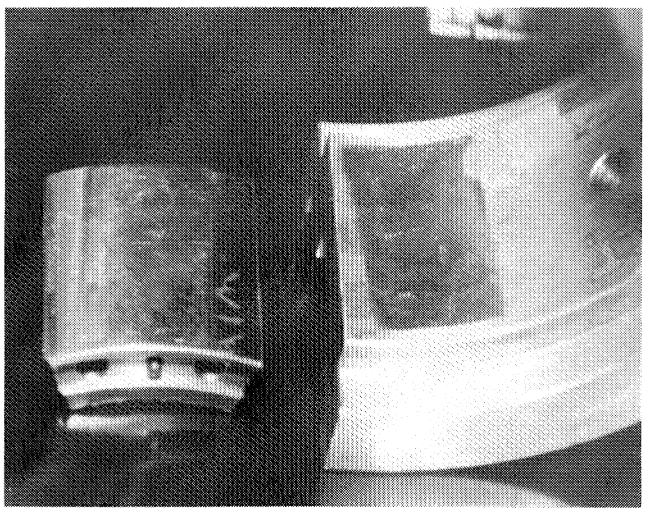

Figure 25. Wear and Brinelling of the Pad and Bearing Retainer.

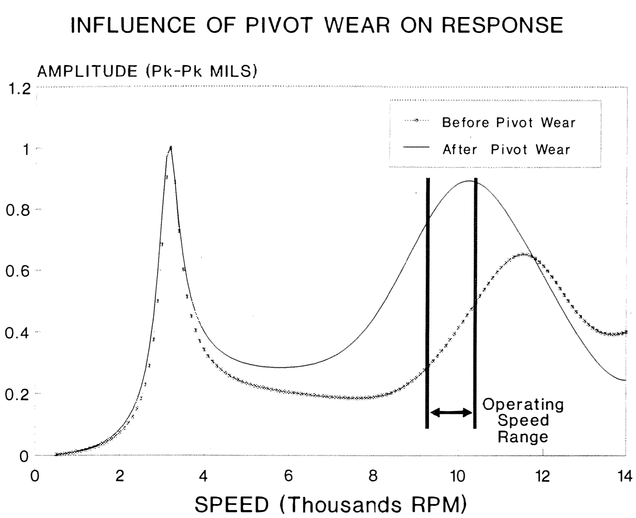

Figure 26. Effect of Pivot Wear on Location of Critical Speed.

Another problem with conventional type tilt pad bearings is their inherent susceptibility to pad flutter. This particularly affects the unloaded pads and often leads to babbitt fatigue at the leading edge of the pad, as demonstrated by Adams and Payandeh [4]. In some higher speed applications, pad flutter is often noted by the fretting and wear on the pad pivots and the corresponding contact area in the bearing shell. When wear is greater on the unloaded pads, flutter is the most likely cause. This phenomenon and some of the design considerations that are used to counter or limit the damage caused by pad flutter will be discussed in a later section.

Conventional type tilt pad bearings are more complex than fixed geometry bearings. The multipiece assembly, each piece of which is made with a set manufacturing tolerance, can lead to a significant tolerance stackup. The tolerance stackup is often at the heart of the poor performance and problems associated with conventional tilt pad bearings. Examples will be shown where the tolerance stackup in a particular bearing can make the difference between acceptable stability and unbalance response performance or high subsynchronous and synchronous vibrations.

FLEXURE PIVOT™ TILT PAD BEARINGS

Flexure Pivot™ (FPB) tilt pad bearings achieve the low cross coupling characteristics through flexural rotation of the web support, as shown in the schematic of Figure 27. The rotational flexibility of the web support should not be confused with the radial flexibility that tends to degrade the oil film stiffness and damping coefficients. The low rotational stiffness reduces the magnitude of the cross coupled coefficients. The patented [5], one-piece design of this particular tilt pad bearing eliminates the multipiece construction typical of conventional style tilt pad bearings. This also significantly reduces the manufacturing tolerance, as shown in the schematic of Figure 28. The one piece construction does not limit the ability to provide a FPB tilt pad bearing in a split design.

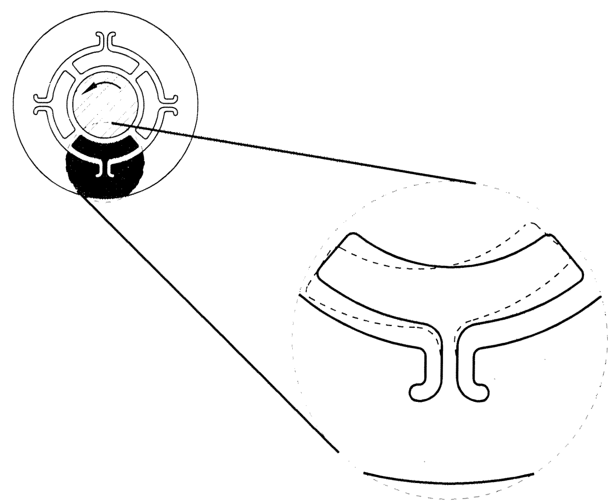

Figure 27. Flexural Rotation of Flexure Pivot Tilt Pad Bearing.

Besides eliminating the manufacturing tolerances, the simple construction also eliminates the pivot wear and contact stresses. Other advantages of these bearings will be demonstrated using cases where they were used to help solve and eliminate some of the problems characteristic of fixed geometry and conventional style tilt pad bearings.

The Sommerfeld Number

In order to understand how different bearing design parameters influence rotordynamics, it is essential to transform the

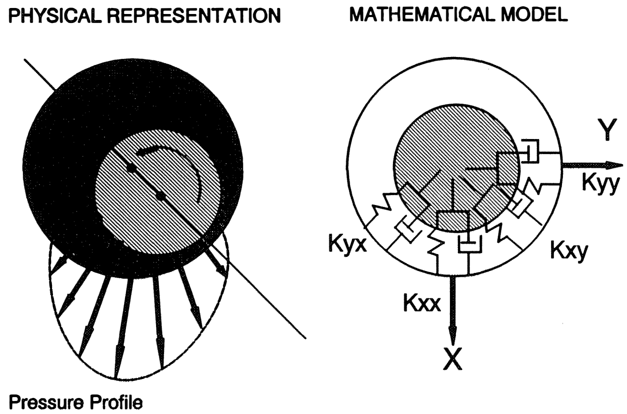

Figure 28. Comparison of Manufacturing Tolerances for Conventional and Flexure Pivot Tilt Pad Bearings.

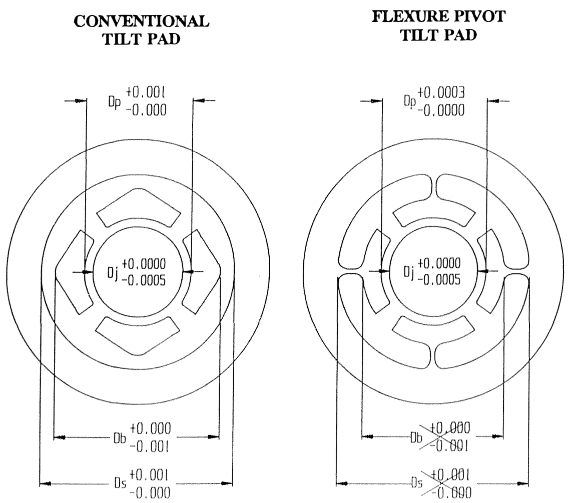

Figure 29. Physical and Corresponding Mathematical Model of a Fluid Film Bearing.

physical model in the fluid film to a mathematical model, as shown in Figure 29. The pressure generated in a fluid film bearing is determined using many of the computer programs which solve some form of the Reynolds lubrication equation. The pressure is then integrated to obtain the bearing reaction forces. These forces are a function of the dynamic stiffness and damping coefficients in the oil film. Dynamic stiffness and damping coefficients are generally presented as a function of a dimensionless parameter known as the Sommerfeld number. There are many ways of expressing the quantities that form this dimensionless variable. Sommerfeld himself presented this in two different ways in his papers. Therefore, it is very important to recognize which form of this expression is being used. The more common way of expressing this variable is

$$S = \frac{\mu NLD}{W}\left(\frac{R}{C}\right)^2 \qquad (2)$$

The variation of the stiffness and damping coefficients as a function of the Sommerfeld number is shown in Figures 30 and 31 for a four pad tilting pad bearing with a load between pad configuration. From these figures, one sees how the different variables that constitute the Sommerfeld number influence the stiffness and damping provided by the bearings. Higher loads and lower speeds will result in a lower Sommerfeld number for

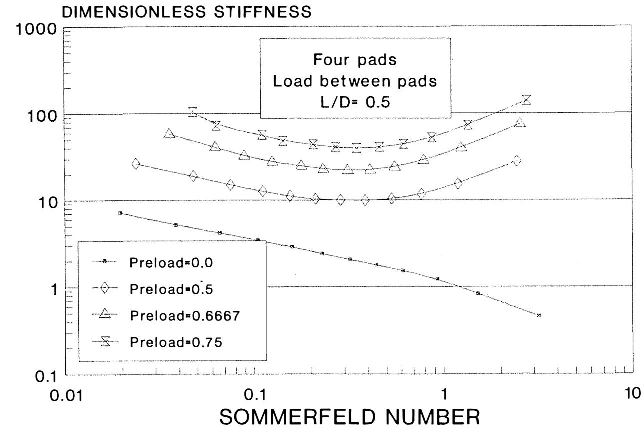

Figure 30. Stiffness Vs *Sommerfeld Number.*

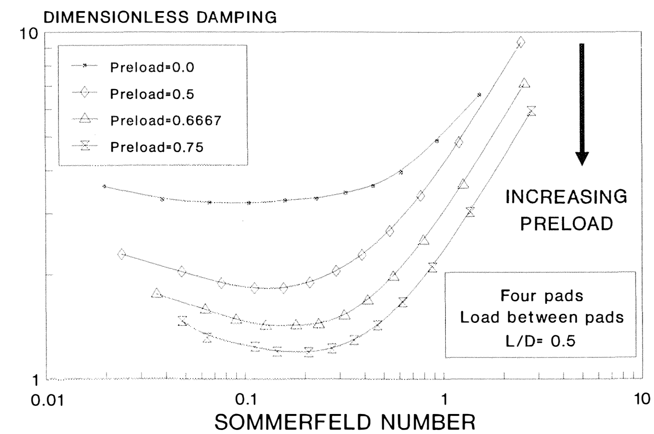

Figure 31. Damping Vs *Sommerfeld Number.*

a given bearing configuration. On the other hand, lighter loads and higher speeds result in higher Sommerfeld numbers.

Influence of Bearing Length

How the variables in the Sommerfeld number affect the stiffness and damping coefficients depends to a great extent on what portion of the curve the particular machine operates at. Most of the curves for different preloads possess a minimum. Whether the operating regime of a given configuration resides to the left or right of the minimum determines how the changes will reflect on the stiffness and damping values. If a certain application lies to the left of the minimum on the stiffness curve, then decreasing speed or increasing load will shift the Sommerfeld number to the left resulting in increased bearing stiffness. Decreasing the length of the pad for an application that lies on the left of the minimum point, will also result in increasing the stiffness. The length of the pad is a variable that the designer has more control over than the speed or load. If the objective is to stiffen the bearing and shift a critical further away from the operating speed range, then this will constitute a viable alternative. Note in this case the eccentricity will also increase, thus increasing the operating temperature of the bearing and reducing the minimum film thickness. An increase in the pad length will reduce the stiffness and damping of the bearing. On the other hand, if the application lies to the right of the minimum, then increasing the length will increase both the stiffness and damping coefficients. One can see from this that the effect the length has on the bearing cannot be established until the operating region for the particular application is known. Thus, the length can increase or decrease the stiffness and damping coefficients for a given bearing.

Influence of Bearing Preload

Preload can be explained by the graphic presentation shown in Figure 32. Preload in most applications is positive, but in some applications is set to zero if temperature limitations are present. It is a very useful parameter which is often used by the bearing designer to alter the characteristics of a certain rotor-bearing system. A good indication of how preload affects the stiffness and damping coefficients is also apparent in Figures 30 and 31. An increase in preload will increase the stiffness, but on the other hand will result in a reduction in the damping available from a given bearing configuration. This is where a bearing optimization study is often required to determine the most desirable characteristics for a given application.

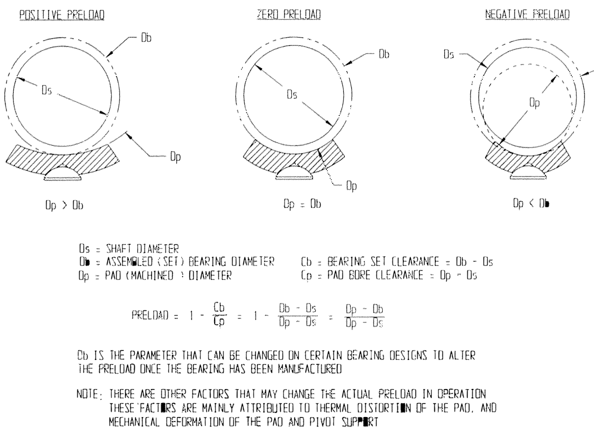

Figure 32. Graphic Illustration of Preload in a Tilt Pad Bearing.

Preload is a critical bearing parameter. Without preload, some pads (top pads) might operate completely unloaded. Unloading of the pads not only reduces the overall stiffness of the bearing, but also affects stability, because the upper pads do not aid in resisting cross coupling influences. Unloaded pads are also subject to flutter instability; a phenomenon often referred to as leading edge lockup or spragging. This is where the leading edge is forced against the shaft and is maintained in that position by the frictional interaction of the shaft and the pad. A detailed description of this phenomenon and the means to reduce or eliminate such an effect is discussed in detail in the section on pad flutter.

Influence of Manufacturing Tolerances

Tolerance stackup in tilt pad bearings has a significant influence on the bearing clearance and preload. This effect is further magnified for smaller shaft diameters. The latest API specifications (API 617 5th Edition and API 612 3rd Edition) recognize the importance of manufacturing tolerances in conventional tilt pad bearings. The specifications explicitly call for examining the predicted rotordynamic performance of the rotor-bearing system at both extremes of the manufacturing tolerances.

The preload in a tilting pad bearing is very important in maintaining the desired stiffness and damping coefficients for a specific application. Preload also helps reduce the possibility of pad flutter and ensures the presence of a converging wedge in the bearing. The larger the spread in the preload for a specific bearing, the greater the chances for unpredicted and undesirable performance. The design limits of advanced turbomachinery applications and the additional requirements for verification of the predicted analysis have increased the importance of maintaining a narrow range for the preload and clearance in bearings.

The following example shown in Table 2 demonstrates how the minimum and maximum preload of a conventional tilt pad bearing (TPB) compare to that obtained for a FPB bearing. In both cases, a desired preload value of 0.25 was used as the goal.

Table 2. Minimum and Maximum Preload for Conventional Bearing and Flexure Pirvot Bearing.

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

C_b = Bearing Set Clearance C_p = Pad Machined Clearance

D_{smax} = Maximum Shaft Diameter = 5.000

D_{smin} = Minimum Shaft Diameter = 4.9995

Required M = 0.25

Conventional Tilt Pad Bearing	*Conventional Tilt Pad Bearing*
D_{bmin} = Bearing Set Bore$_{min}$ = 5.0070	D_{bmin} = Bearing Set Bore$_{min}$ = 5.0073
D_{bmax} = Bearing Set Bore$_{max}$ = 5.0080	D_{bmax} = Bearing Set Bore$_{max}$ = 5.0077
D_{pmin} = Pad Machined Bore$_{min}$ = 5.0095	D_{pmin} = Pad Machined Bore$_{min}$ = 5.0098
D_{pmax} = Pad Machined Bore$_{max}$ = 5.0105	D_{pmax} = Pad Machined Bore$_{max}$ = 5.0102
$\text{Preload}_{min} = \frac{D_{p_{min}} - D_{b_{max}}}{D_{p_{min}} - D_{s_{min}}}$	$\text{Preload}_{min} = \frac{D_{p_{max}} - D_{b_{max}}}{D_{p_{max}} - D_{s_{min}}}$
$\text{Preload}_{max} = \frac{D_{p_{max}} - D_{b_{min}}}{D_{p_{max}} - D_{s_{max}}}$	$\text{Preload}_{max} = \frac{D_{p_{min}} - D_{b_{min}}}{D_{p_{min}} - D_{s_{max}}}$
$M_{min} = \frac{5.0095 - 5.0080}{5.0095 - 4.9995}$	$M_{min} = \frac{5.0102 - 5.0077}{5.0102 - 4.9995}$
M_{max} M_{min} = 0.1500 M_{max} = 0.3333	M_{max} M_{min} = 0.2336 M_{max} = 0.2551

A low preload in most cases will result in lower stiffness and higher damping. However, at very low preload values, manufacturing tolerances can result in preload values that can range from very low to a negative value. Many high speed centrifugal compressors can be very sensitive to the preload, particularly if the operating conditions are very close to the stability threshold of the unit. Crease [6] showed that stable operation of the compressor was very much dependent on the preload value, which directly resulted from tolerances on the pads and the housing manufacturing tolerances.

The stackup of tolerances in a conventional tilt pad bearing also has a significant influence on the synchronous response and the location of the critical speeds. The undamped critical speed map shown in Figure 33 was generated for a high speed compressor supported on a one inch diameter bearing. The location of the critical speed, as determined by the intersection of the bearing dynamic stiffness coefficients and the critical speed lines, can vary over a wide range, due to the wide spread in the preload value. This makes it very difficult to predict the location of the critical speed. Furthermore, the critical will be significantly different from one bearing build to another.

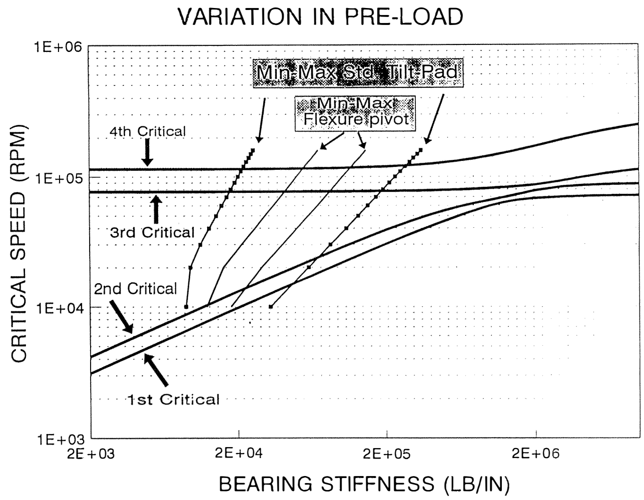

Figure 33. Undamped Critical Speed Map Showing Effect of Manufacturing Tolerances.

Load On Pad (LOP) Vs *Load Between Pad (LBP)*

When higher load capacity and better synchronous response characteristics are the predominant factors, the load between pad configurations are favored over the load on pad tilt pad bearings. This is because the damping resulting from the larger effective support area is higher for a load between pad configuration. Consequently, heavy rotors running at relatively low speeds (low Sommerfeld number applications) have predominantly tilt pads with a load between pad configuration. On the other hand, light rotors running at high speeds do not require the load capacity of a load between pad configuration. Furthermore, applications that fall in this category, (i.e., light rotors running at high speeds) often have higher aerodynamic cross coupling and thus stability is more of a concern than synchronous response. In these applications, a load on pad configuration provides asymmetry in the support. Asymmetry was shown to enhance stability as described in a previous section. The stiffness coefficients for a four pad bearing with load on pad (LOP) and load between pad (LBP) are compared in Figure 34. The load on pad configuration has a much higher asymmetry, while the load between pad vertical and horizontal stiffness are identical and, therefore, symmetric.

The following example will help show the advantages of each of the bearing load configurations. The rotor model shown in Figure 35 is for an industrial centrifugal compressor. The stability of the rotor was analyzed with a four pad bearing in a load on pad and load between pad configurations. In the original analyses no aerodynamic cross coupling was introduced in the model. The logarithmic decrement for the first forward mode was higher for the load between pad configuration as shown in Table 3. If high aerodynamic cross coupling is present, the load on pad configuration produces a positive logarithmic decrement (stable). On the other hand, the load between pad configuration results in a negative logarithmic decrement, and therefore an unstable rotor bearing system. The first forward mode for each case is shown in Figures 36 and 37 for the load between pad and load on pad cases.

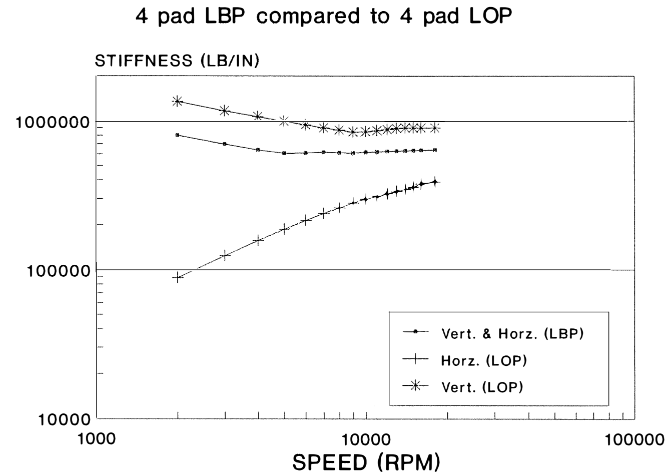

Figure 34. Stiffness Coefficients for a Four Pad Bearing with LBP and LOP.

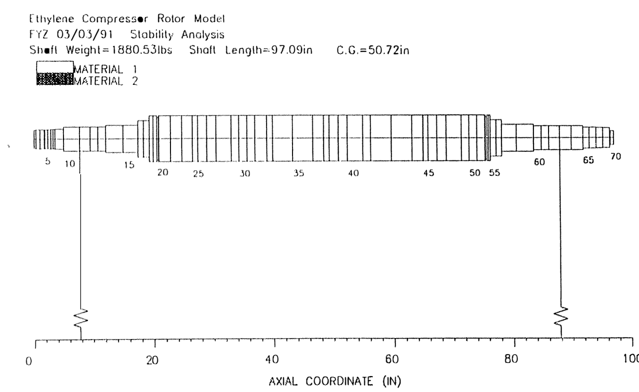

Figure 35. Rotor Model for an Industrial Centrifugal Compressor.

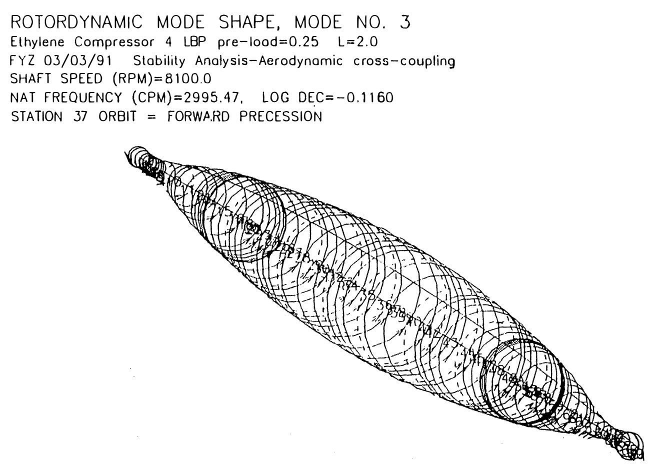

Figure 36. Forward Mode Log Dec for LBP Bearing Configuration.

This example showed how the load on pad (LOP) configuration, which possesses less damping, but more asymmetry, can be a more appropriate selection for a high speed, light weight compressor, compressing a dense gas. Although asymmetry enhances stability, it will have other undesirable side effects. The major axis response amplitude as the rotor traverses the

Table 3. Comparison of Stability and Unbalance Response for Load On Pad (LOP) and Load Between Pad (LBP) Bearings.

Bearing Config.	Logarithmic Decrement		Amplification Factor
	With Aerodynamic Cross-Coupling	W/out Aerodynamic Cross-Coupling	
4 Pad LBP	-0.116	0.467	6.6
4 Pad LOP	0.073	0.267	11.0

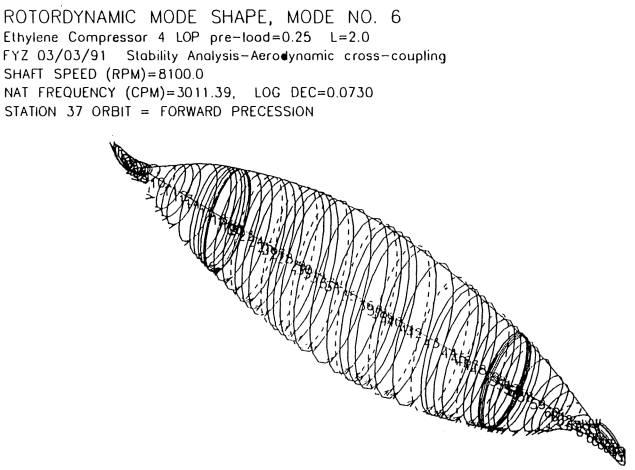

Figure 37. Forward Mode Log Dec for LOP Bearing Configuration.

critical speed will be higher. This is demonstrated graphically in Figure 38. The amplification factor at the first critical is much higher for the load on pad (LOP) configuration. The asymmetry may also result in a pronounced split critical. The critical speed range in such a case will become wide resulting in a reduction of the required separation margin.

Effects and Influence of Number of Pads

The number of pads in a tilt pad bearing has a significant influence on the load capacity and the rotordynamic characteristics of the bearing. In some instances, the radial envelope also plays a role in determining the number of pads to use. This is because in a narrow radial space, the use of a fewer number of pads will result in a relatively long and thin pad structure. The pad can experience excessive deflections under load, which can significantly increase the stresses in the pad and increase the operating preload. The load and temperature might also be very high causing a significant elastic and thermal deformation in the pad profile such as in a gear loaded bearing. In these instances, the choice might favor a larger number of pads (shorter pad arc length) to keep the deflections within a reasonable limit. Another factor that favors the use of a larger number of pads are applications where the load can have a wide swing from load on pad to load between pad. In such instances, the dynamic bearing coefficients can vary considerably as the load swings with fewer number of pads. The larger the number of pads the smaller are the variations in stiffness and damping, as shown in Figures 39 and 40.

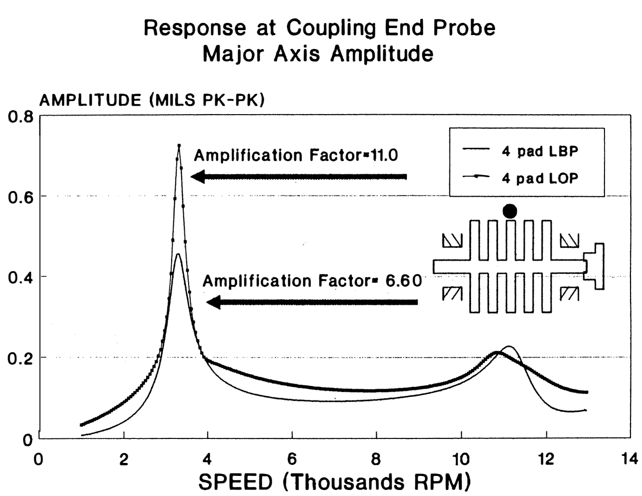

Figure 38. Unbalance Response for LBP and LOP Bearing Configurations.

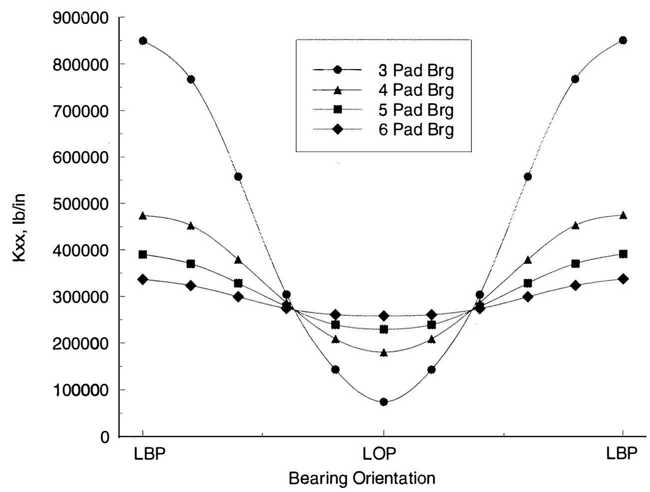

Figure 39. Variation in Stiffness Vs *Number of Pads.*

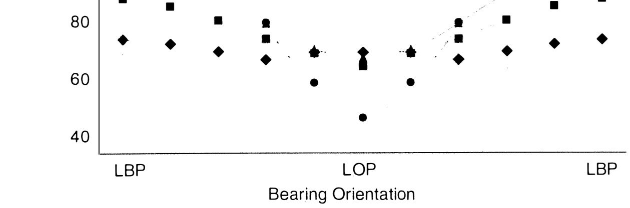

Figure 40. Variation in Damping Vs *Number of Pads.*

Increasing the number of pads also tends to reduce the effective bearing load area. Therefore, a smaller number of pads generally results in a higher load capacity and higher damping. Another criterion that can influence the choice of number of pads is the startup torque. A four pad bearing will generally have a higher startup torque than a five pad bearing. A comparison of the torque required for different bearing configurations is shown in Figure 41. Pad flutter is covered in a following section, but it is important to mention here that pad flutter considerations could influence the number of pads selected for a given bearing. It is generally accepted that the higher the number of pads, the higher the threshold for pad instability and flutter.

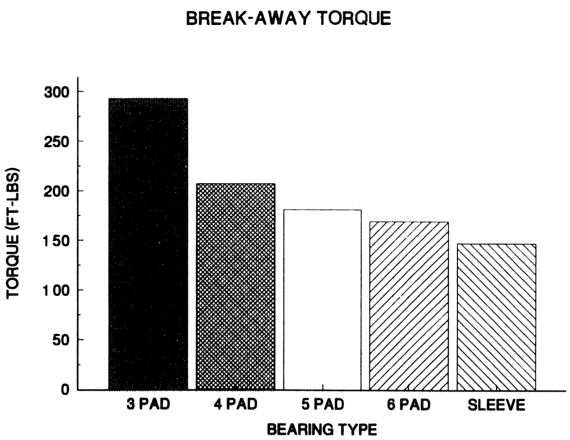

Figure 41. Startup Torque Vs *Number of Pads.*

Effects of Pivot Offset

Most tilt pad bearings have the pivot located in the center of the pad arc. When the pivot is in the center of the pad, the pivot offset is 0.5. The offset is a dimensionless number arrived at by dividing the distance from the pad leading edge to the pivot by the total pad arc length. Increasing the pivot offset to 0.6, for example, means that the pivot is moved in the direction of rotation to 60 percent of the pad arc length from the leading edge of the pad. In a thrust bearing, an offset pivot value between 0.58 and 0.65 generally provides a higher load carrying capacity and a more optimum wedge ratio. The startup torque on the bearing is lower in this case, and a wedge is formed without the necessity for pad crowning. An offset pivot will also result in lower operating temperatures. In the case of a thrust bearing, the existence of an offset pivot is more important than in the case of a journal bearing. This is because in thrust bearings a converging wedge does not exist at startup.

An offset pivot can also be advantageous under certain conditions in journal bearings. The offset can become essential if the rotordynamic performance requires a higher stiffness in the bearing. Although preload can be used to increase the stiffness, it results in lower damping. On the other hand, the offset pivot increases the stiffness without the penalty of lower damping and higher bearing temperatures generally experienced with increased preload. An example of the use of offset pivot to provide better rotordynamic performance and cooler operating temperatures is discussed by Chen, et al. [7], and will be summarized in this section.

A computer generated rotor model for the high speed pinion shaft is shown in Figure 42. The undamped critical speed map as a function of support stiffness is shown in Figure 43. The equivalent dynamic stiffness coefficients for the conventional five pad tilting pad bearing (TPB) are shown as a function of speed and are superimposed on the undamped critical speed map. The coefficients intersect the first two rigid body critical speeds and the first flexible mode in the sloping section of the curves. This ensures that there is significant motion at the bearings to help damp the motion as the shaft traverses the critical speeds when accelerating to the operating speed. The coefficients for the optimized four pad FPB are also superimposed on the critical speed map shown in Figure 43. This bearing has an offset pivot which was chosen for its superior stability characteristics over a wider range of bearing clearances. The offset pivot configuration also provides higher stiffness in the bearing and helps shift the third mode further away from the running speed. This is also verified through the stability and unbalance response analysis.

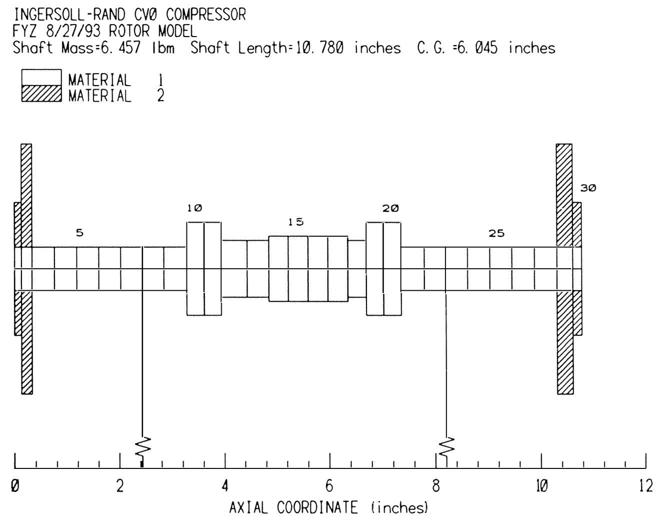

Figure 42. Rotor Model for a High Speed Pinion Shaft.

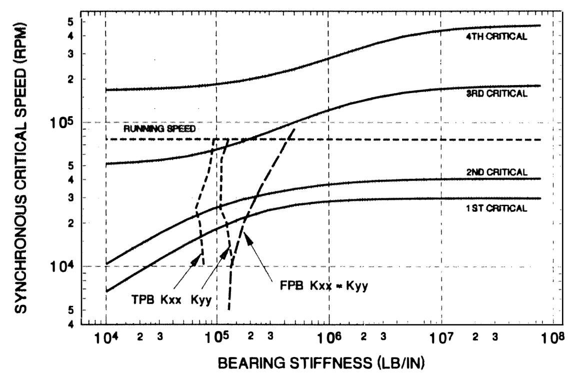

Figure 43. Undamped Critical Speed Map.

The vibration characteristics with the FPBs bearings showed improvements over the conventional style tilt pad bearings during normal steady state operation and under surge conditions. The precision with which the FPB bearing is manufactured gives it an edge since the stackup in tolerances with conventional style tilt pad bearings results in a very wide range for the clearances and preload.

The use of an offset pivot resulted in a 20 to 30°F decrease in the bearing operating temperature. The offset pivot design also showed advantages that were not previously identified in the literature. One of these advantages is the reduced bearing sensitivity to clearance variations as compared to the center pivot (0.5 offset) bearing. The cross coupled stiffness for the 0.5 and 0.6 offset bearing, as a function of the flexure pivot rotational stiffness, is shown in Figure 44. Once the transition region is passed going from the high rotational stiffness (right side) to the lower rotational stiffness (left side), there is not much difference between the two bearings offset configurations. The distinct difference, however, can be seen when the principle stiffness is plotted for both offsets (0.5 and 0.6) as a function of rotational stiffness, as shown in Figure 45. As the pads reach the tilting region, there is a drastic increase in principle stiffness with the 0.6 offset pads. What is of even more significance is the increase in damping with the 0.6 offset pads, as shown in Figure 46. The increase in the direct or principle stiffness and damping is the reason for the reduction in the stability sensitivity to the variation in bearing clearance.

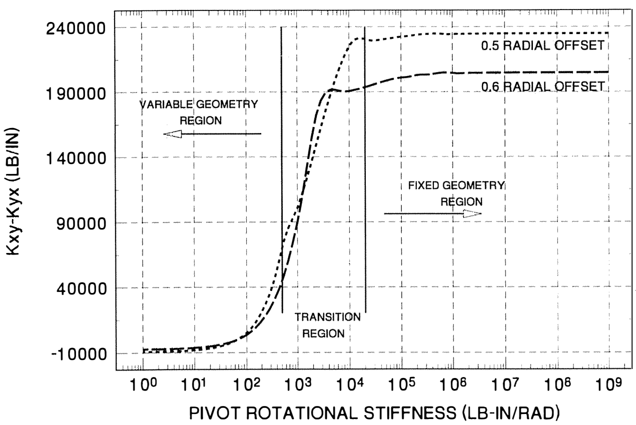

Figure 44. Cross coupled Stiffness Vs *Pivot Rotational Stiffness.*

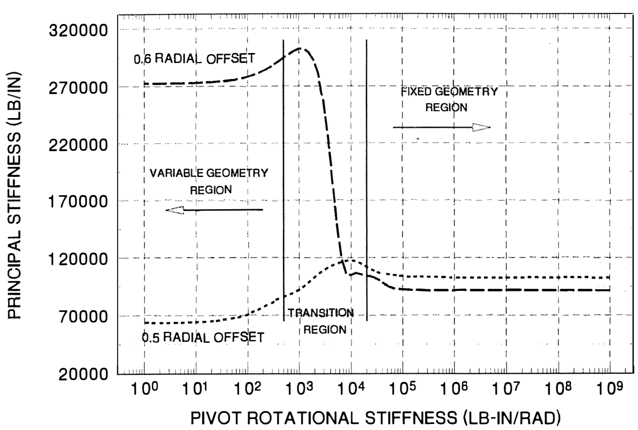

Figure 45. Principle Stiffness Vs *Pivot Rotational Stiffness.*

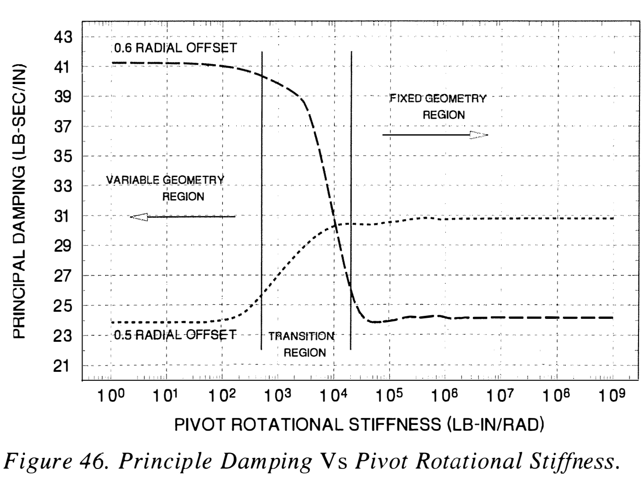

Figure 46. Principle Damping Vs *Pivot Rotational Stiffness.*

Variation in the Load Direction and Magnitude

Many bearings have resulted in failures and poor performance because the variation in load direction and magnitude were neglected or not properly accounted for. While the load in most bearings is simply due to the weight of the rotor, in gear applications the load is further complicated by gear reaction forces, which can be orders of magnitude larger than the weight of the rotor. This load can also change in magnitude and direction depending on the torque, speed, and weight of the rotor, in addition to any external forces applied to the shaft. It is, therefore, essential to adequately evaluate the bearing configuration for such an application, taking into consideration the change in the magnitude of the load and the swings in direction from the unloaded to the loaded condition.

Determining the load direction is essential for stability considerations, as well as for proper location of the feed grooves. Shelly and Ettles [8] have shown that a feed groove located along the load line can result in a significant reduction in the load capacity of the bearing. This reduction can be as high as 60 percent when operating at high eccentricities.

Another factor that should be considered is the influence of the change in load orientation on the damping and unbalance response. This is because the bearing stiffness and damping coefficients will also change with load. Some bearing designers prefer to increase the number of pads in order to reduce the variation in the bearing coefficients as the load swings from load on pad (LOP) to load between pads (LBP) or visa versa. This was shown before in Figures 39 and 40 for different number of pads. Although this seems justified, generalizing is not recommended without considering each application on a case by case basis, since increasing the number of pads results in a penalty on the load capacity. Furthermore, increasing the number of pads, particularly with gear bearings which are generally long, can result in pad dimensions that are out of proportion or impractical.

One of the major characteristics of tilt pad bearings that has favored their use over other bearing configurations is the fact that they have no cross coupled bearing coefficients. However, this is only true when the loading direction is symmetric with respect to the bearing geometry; i.e., the load is directly on pad or between pads. In the case of gear loading where there is a swing in load direction from no-load to full load, conditions will exist where asymmetric loading will occur and cross coupled oil film coefficients of significant magnitude will result. Although cross coupled bearing coefficients are worrisome, because they are usually associated with rotordynamic instability, in this case they should not be of great concern. This is because the stiffness coefficients Kxy and Kyx are equal in magnitude and have the same sign. Tripp and Murphy [3] have shown that cross coupling of this nature is not destabilizing. This can be deduced from computation of the energy generated on a per cycle basis for the Kxy and Kyx represented by Equation (2) which was shown in a previous section. A positive (E) results in energy addition to the system, and therefore is regarded as destabilizing. Since both cross coupled coefficients are of the same magnitude and sign, we can see that (E) will be zero and, therefore, this cross coupling has no effect on stability.

The use of bearings for gears requires a complete rotor-bearing dynamic analysis, in addition to the static bearing load analysis. This would ensure arriving at an optimum bearing configuration for the particular application. Although in terms of the stability analysis, a conservative approach would be to analyze the bearings at partial or no load conditions. The bearings should also be analyzed at full load conditions. This is because at full load, the stiffness in the bearings will be relatively high making the damping very ineffective, which may result in unacceptable synchronous response characteristics. Therefore, the analysis should be performed at all possible loading conditions to adequately define the operating characteristics map.

Influence of Cavitation on Stability and Bearing Performance

Vapor cavitation is defined as the process of boiling in a liquid as a result of pressure reduction rather than by external heat addition. However, the basic physical and thermodynamic processes are the same in both cases. Severe erosion damage to the adjoining metallic surfaces can occur upon the vapor bubble collapse as a result of the large implosive forces that are confined to a small area. This type of cavitation is not generally associated with steadily loaded journal bearings. These normally experience gaseous cavitation, more appropriately called ventilated cavitation, in which air from the atmosphere or dissolved gases are drawn into the divergent clearance space where the pressure is close to ambient pressure. In certain conditions, however, a misapplication or inadequate placement of the feed groove can lead to conditions favorable for the formation of vapor type cavitation resulting in cavitation-erosion damage. Cavitation damage to a sleeve bearing in which the feed groove was located very close to the load vector is shown in Figure 47. A relatively high load carried by a reduced bearing area (due to the presence of the groove in the load region) results in a very thin oil film and high localized pressure. Consequently, the oil cavitates in the diverging section of the bearing and generates vapor bubbles. The bubbles appear to have travelled a short distance downstream and collapsed as the pressure started to build and increase above the vapor pressure of the oil.

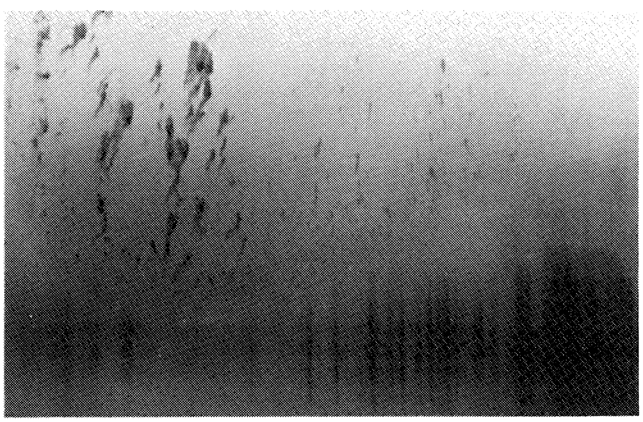

Figure 47. Sleeve Bearing with Cavitation Damage.

A typical pressure wave for a squeeze film damper bearing (dynamically loaded bearing) experiencing vapor type cavitation is shown in Figure 48. It is important to note the instantaneous increase in pressure as the vapor bubble collapses. While in a steadily loaded bearing the damage is confined to a small region in the diverging section of the bearing, in dynamically loaded bearings the cavitation damage can, depending on operating eccentricity, extend circumferentially all around the bearing surface since the pressure wave rotates with the journal.

Cavitation in journal bearings can be controlled and minimized by proper location of the feed grooves, or by increasing the supply pressure to the bearing. The cavitation resistance of bearing materials is very much like fatigue resistance. The harder the alloy material or the thinner the babbitt, the more resistant it is to cavitation damage and erosion. Cavitation conditions are enhanced by the presence of contamination in the oil, which could be in the form of entrained air or water. This

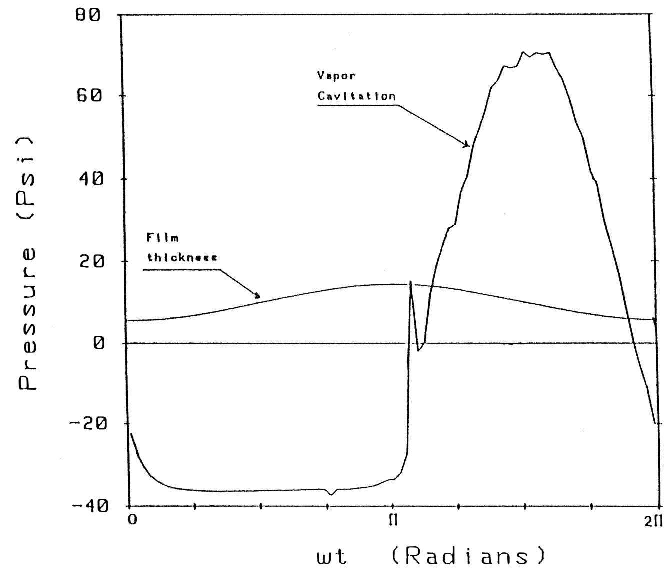

Figure 48. Pressure Wave for a Squeeze Film Damper Bearing Experiencing Vapor Type Cavitation.

entrained water or air acts as a nucleation site for the vapor cavitation bubbles. The presence of water in the lube oil reduces the vapor pressure and allows cavitation to occur more readily. The effect of air bubbles can have opposing consequences depending on whether the air bubbles are dissolved or entrained in the oil. It is a proven fact that injection of air bubbles is one of the effective methods utilized for controlling cavitation damage in hydraulic machinery [9].

High speed and highly loaded three lobe and canted bore bearings that are commonly used in high speed gears and in integrally geared centrifugal compressors, are susceptible to vapor type cavitation. A typical cavitation damage on a three lobe bearing used in an integrally geared compressor is shown in Figure 49. Cavitation damage extended over half the area of the loaded lobe, and over one third of the other two lobes. The pinion speed is about 38,000 rpm. This operating speed and load results in vapor type cavitation on all three lobes. The circumferential pressure profile at the bearing centerline is shown in Figure 50 for the loaded lobe. Cavitation is predicted over approximately one third of this lobe. A fixed geometry bearing like this three

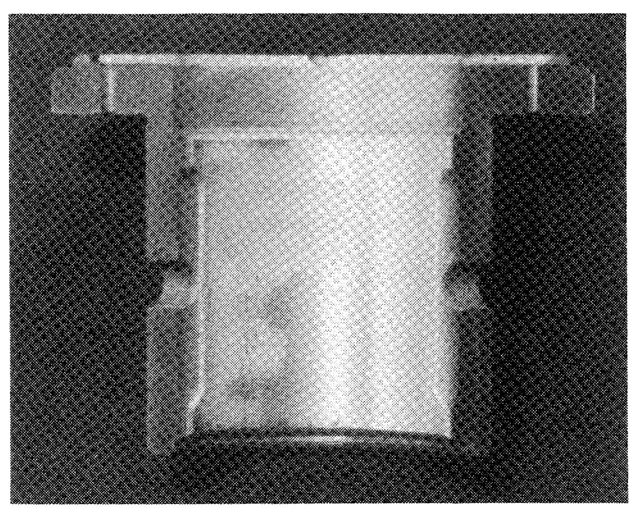

Figure 49. Three Lobe Bearing with Cavitation Damage.

lobe canted bore bearing cannot tilt and, therefore, is incapable of reducing the extent of the negative or vapor cavitation region. The bearing surface erosion due to cavitation continues, and will eventually change the operating bore profile. This will degrade the bearing performance, reduce damping, and increase vibrations. A FPB bearing operating under the same condition will have the pressure profile shown in Figure 50. The ability of the pads to tilt and the location of the pivot provide much better performance, reduce and eliminate cavitation, and maintain a very good level of damping throughout the service life of the machine.

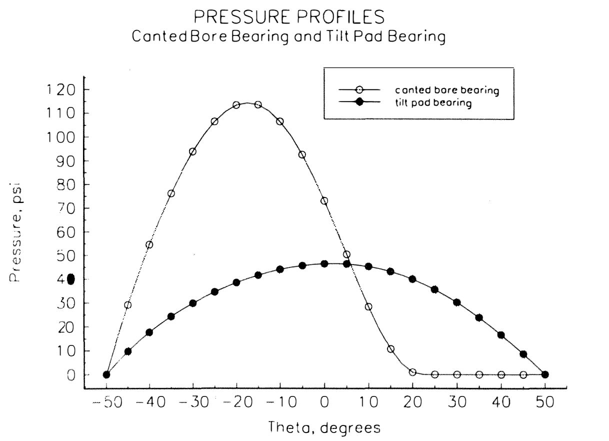

Figure 50. Predicted Pressure Profile for Canted Lobe and Flexure Pivot Bearing.

In general, cavitation in a bearing tends to enhance stability. Increasing the supply pressure in the bearing to suppress cavitation may produce negative pressures in the diverging section of the bearing, which will reduce the stability of the bearing. An understanding of how this comes about can be obtained by examining the schematic shown in Figure 51. The cross coupling generated by the positive pressure region tends to produce a reaction force that pushes the shaft in a direction tangential to the whirl orbit. Similarly, the negative pressure (due to the high supply pressure that suppresses cavitation) tends to generate a force that pulls the shaft in the same tangential direction to the whirl orbit thus algebraically adding to the net destabilizing force. Therefore, suppressing cavitation by increasing the supply pressure may solve the erosion resulting from cavitation, but will compromise the stability of the bearing. Note that in some starved bearings or some that are suffering from excessive hot

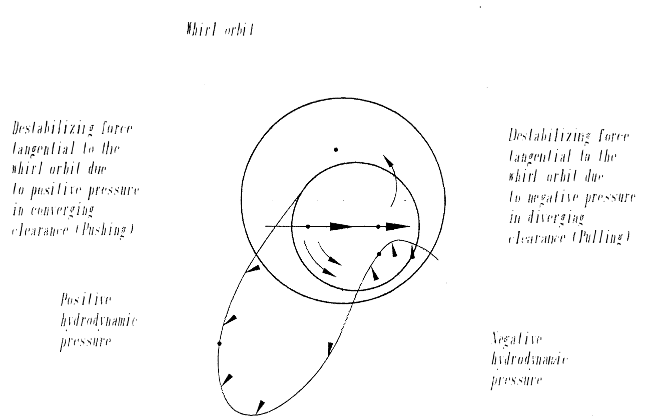

Figure 51. Additional Destabilizing Force Due to Suppression of Cavitation.

oil carryover, the increase in supply pressure will introduce cooler oil, increasing the effective viscosity of the oil film and subsequently the damping. The FPB bearing is inherently resistant to pad flutter problems. This because the pad by virtue of its design is restricted from moving radially inward to follow the shaft motion. Furthermore, the rotational damping provided by the small clearance space that exists between the underside of the pad and the bearing shell prevents and dampens flutter motion.

There has been extensive work and research on the cavitation phenomenon in steadily loaded journal bearings, and their performance to a certain extent can be accurately predicted. On the other hand, cavitation behavior in dynamically loaded journal bearings and squeeze film damper bearings is not complete and lags further behind. Cavitation in these types of bearings can have an adverse effect on the film pressure and load capacity of the bearing [10]. The subsequent effects on the stiffness and damping coefficients are also considerable. The patented squeeze film damper design shown in the schematic of Figure 52 was specifically developed to overcome the cavitation and other problems associated with conventional style squeeze film dampers.

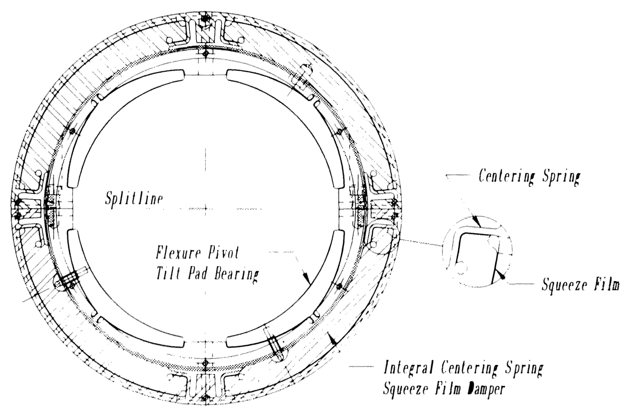

Figure 52. Integral Centering Spring Squeeze Film Damper.

Spragging and Pad Flutter

The "spragging and pad flutter" term is used to describe the damage often found on the leading edge of unloaded pads in large turbine or generator units. In severe cases, the damage could show signs of fatigue cracking or wiping. A schematic from Adams and Payandeh [4] that describes this phenomenon is shown in Figure 53. The pad floats back and forth between the pivot point and the journal. In some cases, momentary contact with the pivot occurs once per cycle. In most cases however, the frequency of the motion is between 0.4 and 0.5 times the rotational speed of the journal. The motion appears to be similar to the oil whirl phenomenon, except that the unloaded pad vibrates instead of the rotor. This self-excited vibration can be thought of as simply the absence of a stable static equilibrium position.

The Adams and Payandeh investigation [4] identified the sprag relief angle as one of the most influential parameters. For a sprag relief angle of 10 degrees or larger, the self-excited vibration is not present, since a stable static equilibrium position is found. The effect of sprag relief depth was found to be less significant a variable as long as the depth is sufficient to ensure a converging film geometry as the pad leading edge approaches the journal. The offset was not found to be significant in this investigation, however, if the pad is centered (offset of 0.5) for the purpose of accommodating pads installed backwards, then

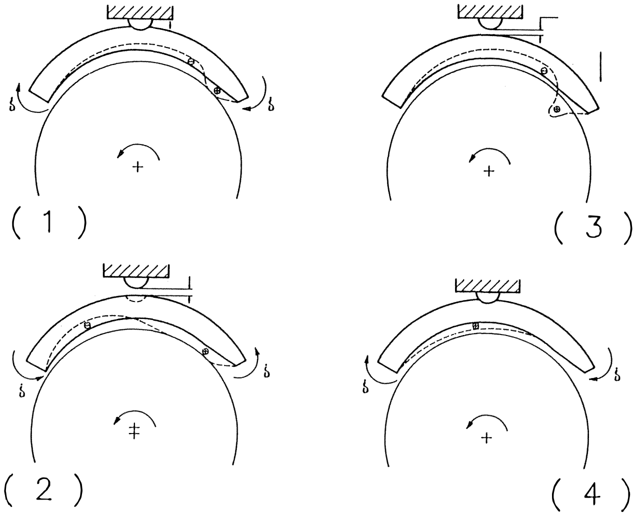

Figure 53. Schematic Showing Pad Flutter.

the pads should have a sprag relief on both the leading and trailing edges. The pad arc length was also found to be more influential than expected. The smaller the arc angle, the more stable the pad motion. Therefore, a four pad bearing is more stable than a three pad bearing. Likewise, a five pad is more stable than a four pad bearing. The variation in the arc length to counter the sprag problem can be seen reflected in the four pad bearing design shown in Figure 54 from Zeidan [11]. In this configuration, the lower loaded pads have a full arc length, while the unloaded pads have a smaller arc length.

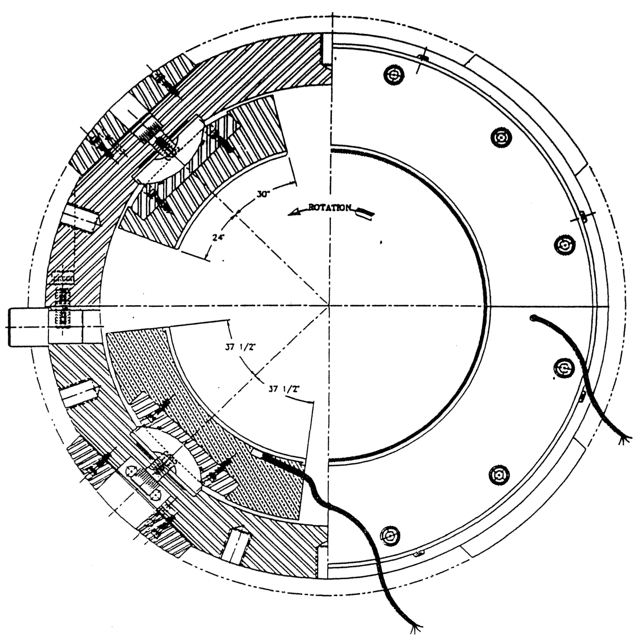

Figure 54. Bearing with Reduced Arc Length Upper Pads to Counter Pad Flutter and Spragging.

Another means to counter the spragging problem is through the use of preload. A positive preload is an essential element for the top pads to ensure a stable pad motion. Preload assures that all pads remain statically loaded under all operating conditions. However, preloading is not used on some large turbine/generator sets due to temperature limitations. Another means of suppressing this instability in such a case would be through introducing damping by providing the type of support shown in Figure 55. In this configuration, an elastomeric ring is located in a groove between the ball and socket of the pad. The damping provided by the elastomer ring has been successful in suppressing this instability phenomenon.

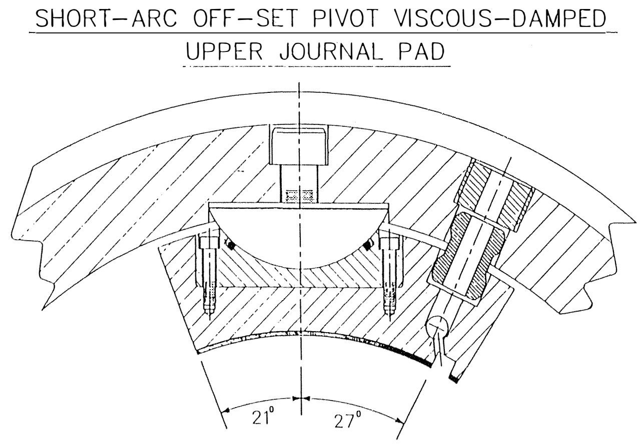

Figure 55. Bearing with Elastomeric Ring Supports to Counter Pad Flutter and Spragging.

The calculated load on each pad in a four pad tilt pad journal bearing is shown in Table 4 as a function of speed. The load is determined by integrating the pressure profile for each pad. Note that at speeds below 9000 rpm, the load on pad one is zero. The absence of a stable static equilibrium position allows periodic vibration (pad flutter) of the unloaded pad. At 9000 rpm and higher, the journal rises high enough in the bearing clearance so that positive oil film pressures are developed at pad one. Therefore, pad flutter is not likely to occur above 9000 rpm.

Pivot Brinelling and Wear in an Integrally Geared Compressor

In certain applications, the dynamic loading in addition to a high static loading can lead to high contact stresses, which result in local yielding of the radial pad's outside diameter and the corresponding contact area on the shell's inside diameter. This failure is usually referred to as brinelling. As a result of the local

Table 4. Pad Loads Vs *Speed for a Four Pad Tilt Pad Bearing.*

PAD LOADS, LB				
Pad 1	*Pad 2*	*Pad 3*	*Pad 4*	*Speed*
0.0	58.3	456.5	524.4	3000
0.0	77.1	457.2	543.7	4000
0.0	95.9	458.1	562.6	5000
0.0	113.6	459.0	580.6	6000
0.0	129.9	459.4	596.8	7000
0.0	144.1	460.7	611.9	8000
129.5	159.8	587.8	622.8	9000
144.8	173.2	603.2	635.6	10000
157.5	184.5	616.1	646.8	10946
180.2	205.0	638.7	667.7	12000
197.9	223.0	657.0	685.2	13000
190.4	220.0	649.2	682.5	14000
169.1	202.6	627.4	665.1	15000
151.3	187.8	609.0	650.1	16000
139.4	178.0	596.7	640.2	17000

yielding and brinelling, the bearing clearance will increase, which in turn results in further increase in the vibration levels and further degradation of the bearing, as shown in Figure 56.

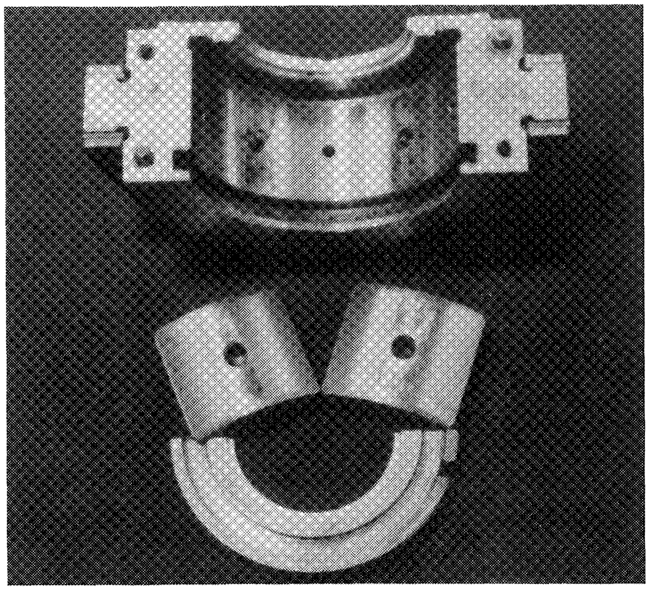

Figure 56. *Brinelling of Pads and Bearing Shell.*

Another fact that is often ignored or unaccounted for is the effect of pivot wear on the preload in the bearing. The pivot wear will reduce the preload and increase the possibility for a negative preload condition to exist as shown in Figure 57. This can explain the progressive degradation on some machines as the pivot wears and is accompanied by either a steady increase in the synchronous vibration or the formation of subsynchronous vibrations.

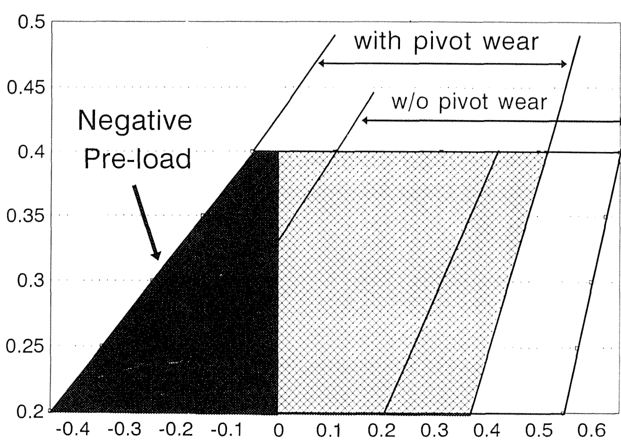

Figure 57. Effect of Pivot Wear on Bearing Preload.

The layout for an integrally geared centrifugal compressor is shown in Figure 58. This compressor experienced high vibrations and the bearing life did not exceed several months. The OEM bearings were replaced by a third party ball-in-socket bearings, but these bearings, in spite of the lower stresses, could not take the high unit loads and high frequency oscillations. The loaded pads and corresponding ball supports fretted very quickly, increasing the bearing clearance and leading to a further loss in damping. The ball-in-socket bearings were replaced by a

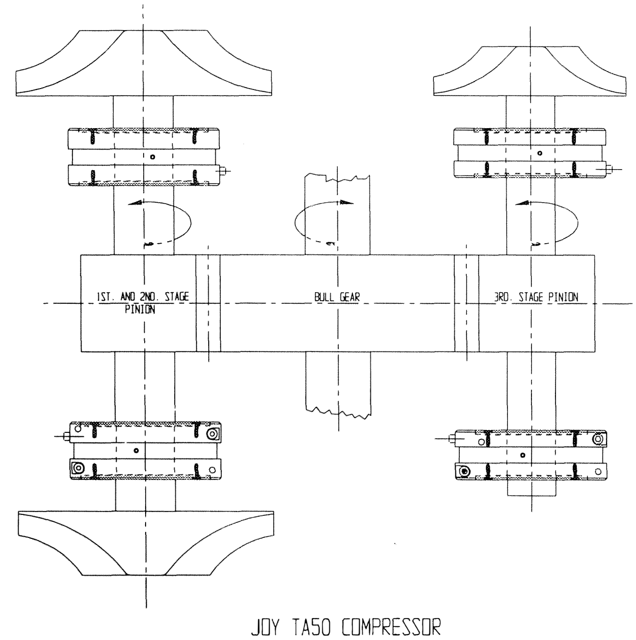

Figure 58. Layout of Integrally Geared Compressor.

rocker back set of bearings. These bearings did not fare much better. The vibrations kept steadily creeping up and resulted in frequent bearing replacements and a very low compressor online factor. The fretting on the back of the pads and on the contacting surface on the shell are shown in Figure 59.

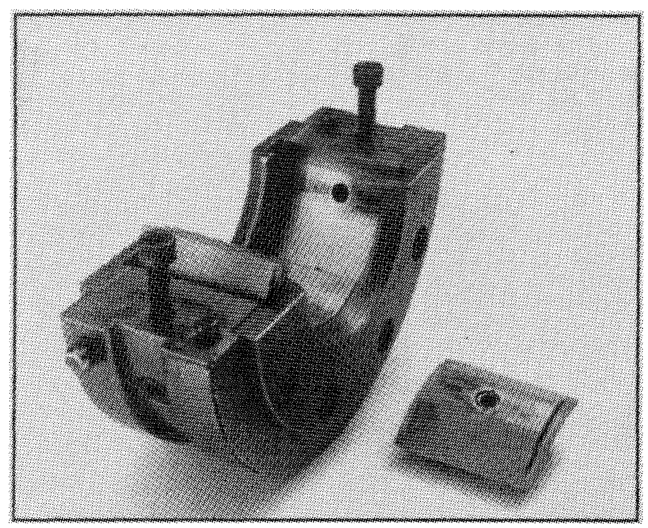

Figure 59. Fretting Damage.

In order to solve the problem and eliminate the pivot brinelling and fretting, a set of drop-in replacement FPB bearings was manufactured. The FPB bearing design used is shown in Figure 60 allowed for directed lubrication nozzles integral with the shell. The small space between the underside of the pad and the shell provides rotational damping due to the squeeze film effects. These bearings have been running with extremely low vibration levels for over two years.

High speed integrally geared compressors are particularly susceptible to brinelling and fretting. This is due to two factors that contribute considerably to the acceleration of wear in these

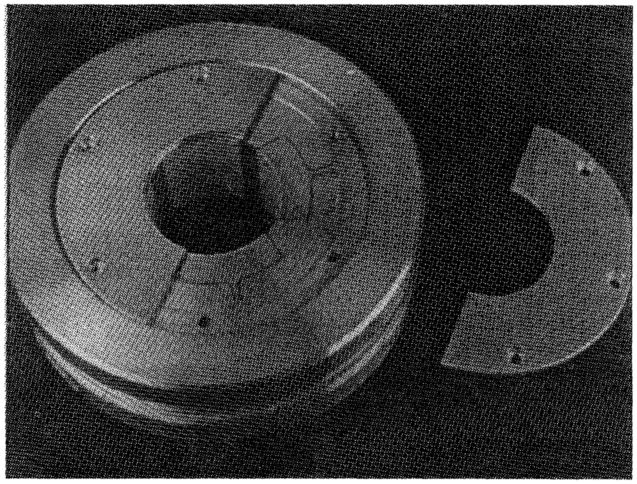

Figure 60. Drop-In Replacement Flexure Pivot Bearings.

applications: the relatively high loads (gear separating forces), and the high speeds (25,000 to 50,000 rpm). This combination of load and speed presents one of the most severe applications for a fluid film bearing.

Influence of Bearing Clearance and Oil Inlet Temperature

The ability to maintain a tight control of the bearing clearance is very critical in some of the high speed and high performance compressors. Satisfying the latest API specifications and the more stringent end user specifications has increased the importance of precision bearing set clearance and preload.

The unbalance response plot shown in Figure 61 demonstrates the influence of clearance on the synchronous response. A peak response appears with the larger bearing clearance very close to the operating speed of the compressor. The ability to manufacture a bearing with a very tight tolerance on the clearance and preload was essential in this application. The optimized set of bearings and the precision manufacturing enabled the suppression of the peak response and critically damped the mode close to the operating speed. This is a practical case where a conventional style tilt pad bearing could not satisfy the requirements dictated by the application due to the variations in manufacturing tolerances. Furthermore, even if the conventional tilt pad

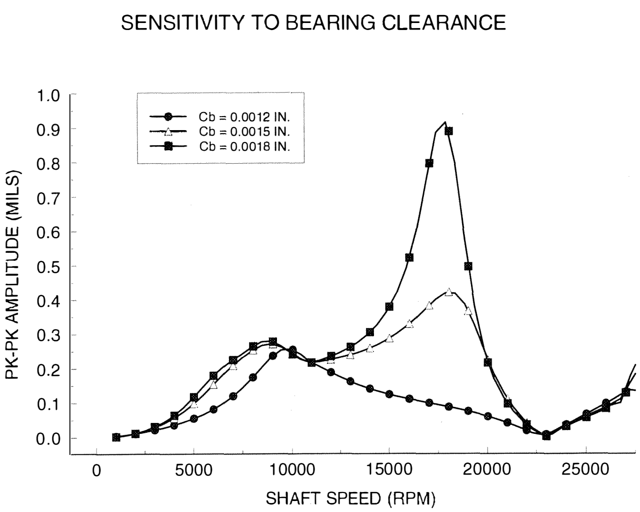

Figure 61. Influence of Bearing Clearance on Synchronous Response.

bearing, through tight control of clearances, resulted in a bearing configuration that fell within the specified limits, it is very likely that the compressor vibrations would cause pivot wear and quickly push the bearing out of the specified limits for smooth operation.

The oil temperature is generally not very critical and very little attention is paid to it as long as it is maintained within a certain range. More often the upper limit of the oil inlet temperature is of concern, since it has a more direct influence on the metal temperature in the bearing. The following case will demonstrate how the oil inlet temperature below the lower temperature set limit can also be alarming and may cause damage to the compressor. High speed compressors and turbines running through their first bending mode can be affected by the lower than normal oil temperatures. The compressor in this case experienced high vibration amplitudes while traversing the critical speed causing the rotor to rub against the labyrinth seals. The lower oil inlet temperatures resulted in higher oil film stiffness in the bearings. The stiffer oil film caused higher rotor deflections at the mid span. These high rotor deflections resulted in rotor rub against the labyrinth seals. The analysis results shown in Figure 62 confirmed this hypothesis.

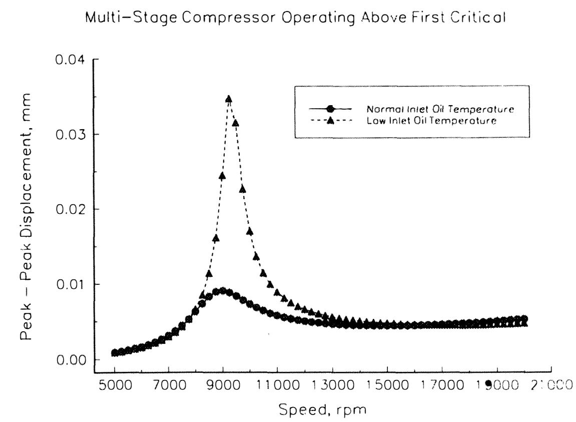

Figure 62. Influence of Inlet Oil Temperature on Synchronous Response of a Multistage Centrifugal Compressor.

This case should serve to alert us to the dangers of lower temperatures in high speed and relatively flexible compressors running above their first bending critical speed. The lower limits of oil inlet temperature have to be adhered to for safe and trouble free operation.

Influence of Support Flexibility

The support flexibility has a great influence on the bearing stiffness and damping coefficients. This is because the support stiffness is in series with the bearing oil film stiffness. The series arrangement tends to always degrade the resultant stiffness and damping coefficients. Nicholas [12] presented a formulation to account for the support flexibility in the analysis. Support flexibility effects tend to be more prominent in certain classes of machinery, while other types of machinery are not influenced by it. The support flexibility is significant in large steam turbines, motors, generators, and some of the larger and relatively heavier compressors. In these cases, the heavy rotor operating at high eccentricity, will generate high oil film stiffness, while the bearing pedestals or supports are relatively flexible. Cases where support stiffness is not significant include lighter turbines and compressors. In these applications, the rotor will generally operate at a low eccentricity, thus generating low oil film stiffness. The bearing supports on high speed turbines and high pressure compressors are relatively massive and very stiff in comparison to the oil film stiffness. Therefore, the influence of the supports can be ignored without sacrificing the accuracy of the analysis.

The following example shows the influence of the support flexibility on the unbalance response and the location of the critical speed. This analysis was performed on an electric motor running at an operating speed of 3600 rpm. The response assuming rigid supports compared to the response which included the support flexibility into the model is shown in Figure 63. Note that the effective stiffness has been reduced as evidenced by the shift in the critical speed from 5000 to 4000 rpm. The unbalance response amplitude more than tripled, indicating that the damping was also degraded by the support flexibility.

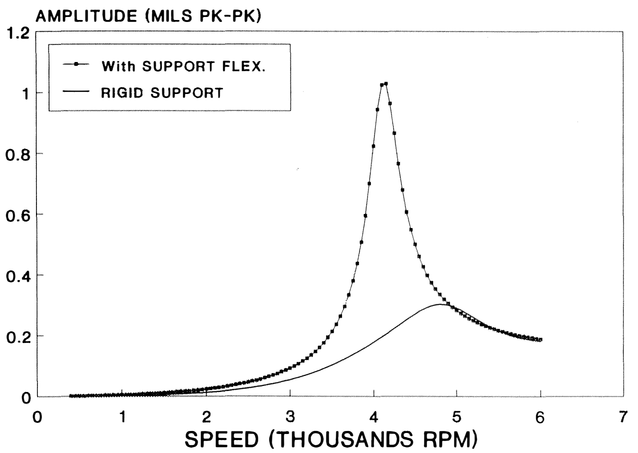

Figure 63. Influence of Support Flexibility on Unbalance Response.

Influence of Pivot Stiffness

Support flexibility is not limited to the bearing pedestals and supports. The pivots in tilt pad journal bearings are also a source of support flexibility. As shown by Kirk and Reedy [13], the stiffness of the pivot is strongly dependent on its geometry. The predicted pivot stiffness for a number of pivot configurations is shown in Figure 64. The stiffness is lowest for the spherical pivots that result in a point contact between the pivot and the bearing shell. The next lowest stiffness occurs with the rocker back tilt pad bearings that have a line contact between the pivot and the bearing shell. The ball-in-socket style pivot design has the highest stiffness among the conventional tilt pad bearings.

Note that for the conventional tilt pad bearing designs the pivot stiffness is a function of the bearing load. This is because as the pivot and bearing shell deform, the contact area expands, which increases the stiffness of the pivot.

The pivot stiffness for a flexure pivot tilt pad bearing is also shown in Figure 64. This bearing design has the highest pivot stiffness of all the designs. In addition, the stiffness of the pivot is independent of the bearing load. Therefore, the high pivot stiffness is achieved even at low bearing loads.

Applications of Squeeze Film Dampers

As the speed increases, the bearing stiffness increases, resulting in a very stiff oil film. The higher oil film stiffness in the bearings prevents the damping from being effective. This is because the stiff bearings become nodal points for the shaft operating mode. The absence of motion in the bearings will

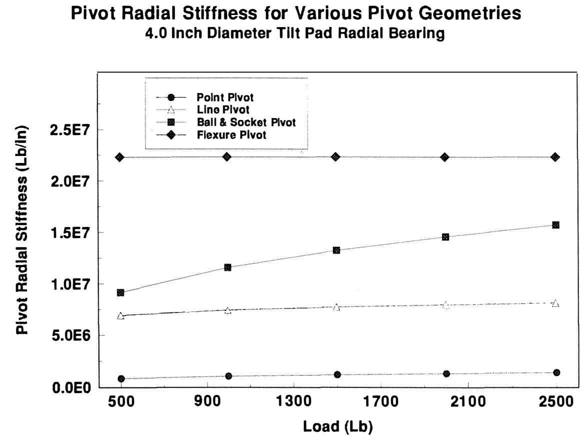

Figure 64. Pivot Stiffness of Various Pivot Designs.

increase the rotor's sensitive to unbalance and stability is reduced making the rotor-bearing system susceptible to subsynchronous type vibrations. Introducing a squeeze film damper in series with the bearing will reduce the forces transmitted through the bearings. The use of squeeze film dampers in series with a fluid film bearing can also reduce the sensitivity to unbalance, thus reducing the balance quality required for smooth operation. The lower stiffness in the squeeze film support spring allows more motion at the supports. This increased motion, combined with the squeeze film damping available in the damper, provides more stable and reliable operation.

The following example is for a pump that was experiencing rapid bearing wear. This was caused by operation very close to the second critical speed. The undamped critical speed for this rotor is shown in Figure 65. The running speed and the sleeve bearing stiffness coefficients were superimposed on this plot. Note that the intersection of the bearing stiffness with one of the critical speed lines identifies the location of the critical speed. The running speed line is very close to this intersection point, indicating that there will be excitation from the second mode or natural frequency. The bearing coefficients intersect the second critical speed line in the flat or horizontal section of the curve. Changes to the bearing configuration or type did not produce a significant shift in the stiffness in order to avoid this interference.

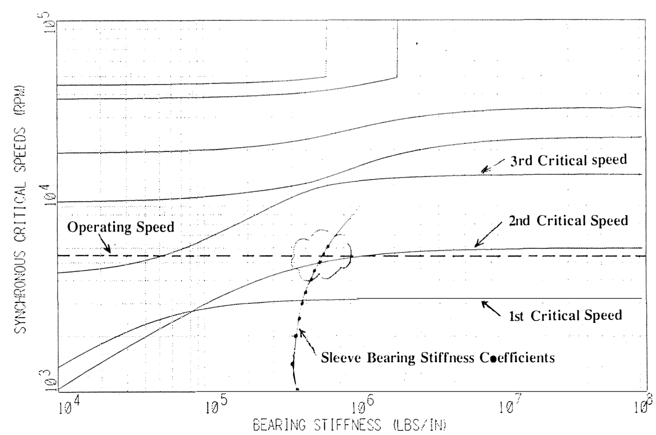

Figure 65. Undamped Critical Speed Map.

The unbalance response analysis also verified that the operating speed is encroaching on the second critical speed. This resulted in high forces transmitted from the shaft to the housing

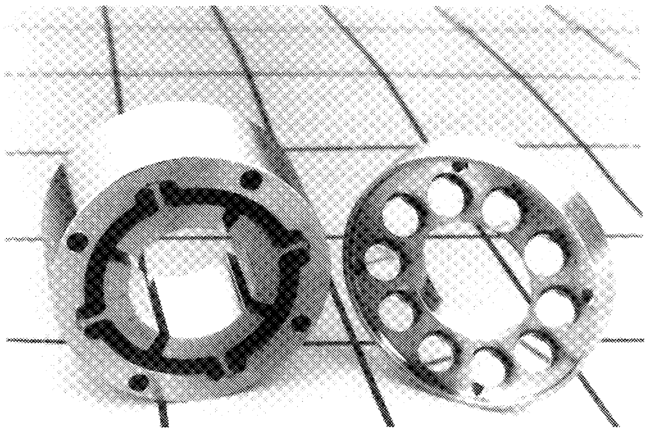

Figure 72. Water-Lubricated Flexure Pivot Bearing for Axial Steam Compressor.

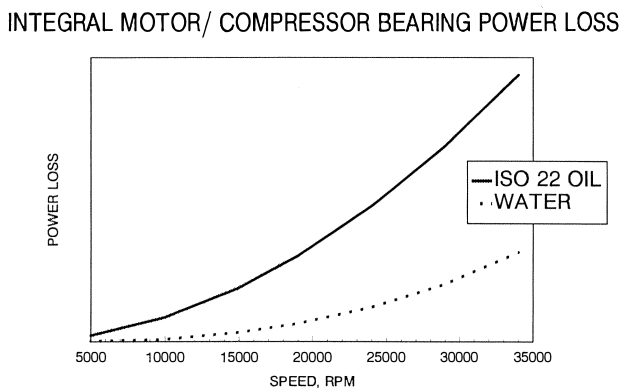

Figure 73. Comparison of Bearing Power Loss for Two Lubricants: ISO 22 Oil and Water.

CONCLUSIONS

Although bearings have not changed much over the last decades, the current trend towards higher speeds and smaller, more efficient, machinery is pushing the existing bearing technology to the limit. This trend has contributed towards the development of better and more reliable bearings. New manufacturing methods are being used to provide the precision and flexibility required from the new generation of turbomachinery. In the last few years, the high end of speeds in advanced turbomachinery progressed from the 20,000 to 40,000 rpm range to the 50,000 to 100,000 rpm range. The higher speeds increased and heightened the importance of the bearing performance and its influence on the stability and long term reliability.

The increasing use of process lubricated bearings has forced the changes in conventional bearing materials and continues to influence the development of more reliable bearing surface materials. In the case of process lubricated applications, which will continue to grow due to tighter pollution and emission regulations, the bearing is required to have a dual role. The bearing has to operate hydrodynamically with very low viscosity fluids, and is expected to survive long term operation under dry running conditions. The process fluids vary widely in properties and their lubrication characteristics are very poor. This increases the importance of the bearing surface finish and self lubricating properties.

This trend toward high speed and high performance bearings as well as the more nonconventional bearing applications will continue to grow. To meet the challenge, a significant amount of experimental verification and development will be required since the current analytical tools are not adequate to model these more demanding applications. It is, therefore, essential to have access to test rigs for verification of existing and new bearings designs at conditions close to the actual operating conditions. Such a test capability also enables the designer to test and troubleshoot existing and conventional style bearings to more accurately define their design and operating limits.

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