PRESSURE DAM & ELLIPTICAL BORE BEARINGS

Why were pressure dam and elliptical bore bearings developed from circular bore bearings? To control or suppress rotor instability.

TRI designs, analyzes, and manufactures circular bore, pressure-dam, and elliptical bore bearings for users, as well as for other OEMs. TRI will manufacture bearings to existing specs or will analyze applications to provide upgraded designs that will improve bearing performance and control rotor stability. If appropriate for specific applications, TRI offers TRI Align-A-Pad® Bearings. Over 55,000 MW of electrical generation depend upon TRI Journal Bearings.

Sub-synchronous rotor vibrations in rotating machinery have long been the source of many vibration and bearing damage problems. Initially, circular bore bearings were used, and then as higher speeds were introduced, sub-synchronous rotor vibrations occurred with seriously damaging results. To suppress sub-synchronous rotor vibrations, that is, to

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increase rotor stability, changes were made to the bearing designs. In the early 1900s, pressure dam bearings were introduced with success. Later, in the 1930s, as designers made machines with even more challenging rotor stability issues, elliptical bore bearings were developed for the same reasons. This Tech Note will explain the fundamentals of how sub-synchronous rotor vibrations occur for rotors supported by simple circular bore bearings and then address the differences in the designs and stability characteristics of circular bore bearings, pressure dam bearings, and elliptical bore bearings.

**Circular Bore Journal Bearing Designs and Associated Rotor-Bearing Stability Characteristics:**

To understand the advantages of pressure dam and elliptical bore bearings, one must first understand the principals of circular bore bearings. Circular bore bearings are the simplest form of film bearing. Such a bearing with the journal rotating in the counter-clockwise direction is shown in Sketch 1. The lube oil flow rate and the viscosity of the lubricate are sufficient to create a “wedge film” in the “converging” section of the oil film. This wedge of oil creates a pressure capable of lifting the journal off the bearing surface. In typical service, there is no contact between the journal and the bearing bore.

With the journal rotating in the counter-clockwise direction, the pressure distribution in the axial center-plane of a circular bore journal bearing and going 360 degrees around the bearing is depicted in Sketch 2. A wedge film with positive
The area of a bearing is the axial length of the bearing surface multiplied by the diameter of the bearing bore.

Circular bore journal bearings work well for very heavily loaded bearings that never become unloaded or shafts that rotate very slowly, but these are not the subject of this Tech Note.

This Tech Note focuses on circular bore bearings that have a more reasonable loading, that is, when the specific loading is in the range of 100 to 175 pounds per square inch, and when the rotational speeds are slow to moderate. However, as the rotational speed increases, or the diameters become larger, or the specific loads become smaller, the journal tends to experience sub-synchronous vibration, that is, the frequency of rotor vibration is at or below half of the rotational frequency of the rotor. This phenomenon can happen when the rotor is well balanced or not well balanced. In other words, sub-synchronous rotor vibration has very little to do with whether the rotor is balanced or unbalanced. Rotor vibration due to unbalance occurs at the rotational frequency of the rotor, not a sub-synchronous frequency phenomenon. In very unique situations, high levels of rotor unbalance can lead to sub-synchronous rotor vibration in circular bore bearings, but this very specific circumstance is not a part of this Tech Note.

**S-Omega Stability Chart**

Sub-synchronous rotor vibration, or “rotor instability” characteristics of rotors using circular bore bearings are generally related to two principal parameters of the journal bearing designs: The Sommerfeld pressure (up to 800 psia or higher) occurs in the “converging” section and terminates at the “line of centers”. The pressure distribution is parabolic, from end to end in the axial direction within the converging wedge, Section A-A, as shown in Sketch 3. In the divergent portion of the film, the pressure is sub-ambient (between 14.7 psia and near zero psia) and actually cavitates forming circumferential streams of oil vapor bubbles. A 3-D image showing the film pressure is shown in Sketch 4. The red area is the high oil pressure or oil wedge.

The “specific loading” of a bearing is a critical parameter for determining the load carrying capabilities of a bearing. The specific loading of a bearing is the ratio of the downward force to the cross sectional area of the bearing. In a simple case, the downward force is equal to the rotor weight that is supported by the bearing. The cross-sectional

\[
\text{Specific Loading} = \frac{W}{L \times D}
\]

Equation 1: Specific Loading
A demonstration of the sub-synchronous vibration of an unstable rotor is shown in Sketch 5. The journal center moves in an unstable orbit, or trajectory, with the position of the journal center marked sequentially at points 0, 1, 2, 3, 4, and 5. There is one full rotation of the rotor between each successive pair of points as depicted by the key phasor notch. The notch is always on top at the end of every journal rotation, while the journal center is in different locations along the unstable trajectory at the end of every journal rotation. For instance, at 3600 rpm, or 60 hertz, there is 16.667 sec between successive points.

**Bearing Parameters:**

\[ S = \mu f \frac{LxD}{W} \left( \frac{r_j}{c_r} \right)^2 \]

Equation 2: Sommerfeld Number

\[ \Omega = f \left( \frac{c_r}{g} \right)^{1/2} \]

Equation 3: Omega Frequency

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Sketch 5: Unstable Orbit
Sketch 6: Stable Operating Point
S-Omega Chart

\[ S = \mu f \left( \frac{L D}{W} \right) \left( \frac{r_j}{c} \right)^2 \]

\[ \Omega = f \left( \frac{c}{g} \right)^{1/2} \]

Sketch 7: Circular Bore
Increasing Rotation Frequency

Sketch 8: Circular Bore
Decreasing Specific Load
or Increasing Viscosity

One example that may be used to stabilize this rotor-bearing system as shown in Sketch 6, is to add a heavier weight, \( W_{\text{add}} \), to the rotor, that is, to the journal. With this additional weight, the Sommerfeld number becomes smaller and the rotor becomes stable as evidenced by the journal center staying at one “Steady-State” location, SS, with a higher eccentricity in the bearing than exists for the unstable trajectory in Sketch 5.

In order to generalize the predictability of the rotor stability characteristics of a range of circular bore bearing designs, a stability chart is drawn with the Sommerfeld number (dimensionless) on the y-axis and the Omega frequency parameter (dimensionless) on the x-axis. On this chart, a stability boundary is drawn which separates the region within which the journal bearing will be stable from the region where it will be unstable. Journal Bearing designs that lie to the left or below the boundary are stable. Journal Bearing designs that lie to the right or above the boundary are unstable.

This stability chart is developed using numerical computer programs to solve the many equations related to the mathematical models of film bearings for the bearing design being analyzed along with the basic equation of motion of the journal mass to which the film pressure distributions and resulting film forces are applied. A very successful technique to solve these equations is to use a time-transient process to create a trajectory, or orbit, of the motion of the journal center through a sequence of small time increments. At the end of every time increment, the solution of the hydro-dynamic film model produces forces on the journal from which the next location of the journal center along the trajectory is predicted. Changing the parameters of the bearing model (enumerated after Equation 3 above) and making many computer runs provides the data to generate the stability boundary for the bearing model. This data can be verified by actual tests, including operation of real machinery, as well.

Using the chart shown in Sketches 7 and 8, let’s examine a few scenarios:

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*Heavy duty bearings for excellent vibration control*
Consider a circular bore bearing operating in the stable region and then increase the speed of the rotor. As the rotor speed increases, both the Sommerfeld number and the Omega frequency parameter increase linearly. Eventually, the bearing will cross the stability boundary and go from the stable region to the unstable region resulting in unstable sub-synchronous rotor vibration.

Another typical scenario machine designers and developers often face is increased power or torque for a new machine that is generally the same size as a previous machine in that product line. If a design is required to handle more torque, a larger diameter shaft may be required, for instance, to keep the shaft surface stress at a safe limit, say, 6,000 psi. Keeping the rotor weight constant and keeping the clearance ratio \(c_r/r\) or \(C_d/D\) constant, both of which are common, an examination of the stability chart shows that increasing the diameter increases the Sommerfeld number, and the Omega frequency parameter changes by \((c_r)^{1/2}\). The conclusion is that increasing the shaft diameter can be a source of rotor instability.

External forces that lift the rotor, and simultaneously decrease loading on a bearing can drive a rotor from stability to instability. An excellent example is a steam turbine under certain operating conditions for which inlet nozzle steam forces lift the rotor upward unloading one or both bearings. On the S-Omega chart, the Sommerfeld number increases with decreased loading, while the Omega frequency parameter is not affected. This causes the operating point to move vertically upward and the rotor becomes unstable.

Lube oil viscosity is dependent on the temperature of the oil. Hotter oil is less viscous. The rotating shaft causes shearing which heats up the oil. As the operating temperature increases and the viscosity decreases, the operating point tends to move downward to a more stable position.
Similarly, an increase in the temperature of the supply of oil flowing to a bearing causes the oil temperature to rise, which reduces the lube oil viscosity, and this moves the journal bearing toward the stable region of the stability chart.

Making the bearing shorter will indeed reduce the Sommerfeld number and this can lead to stabilizing a rotor-bearing system, but this can only be done when it is certain that loading in every operating condition will not overload and damage the bearing. Conditions such as wear on turning gear, thermally induced misalignment during start-up, high amplitude vibration forces when going through critical speeds, and shutdowns with very high temperature oil, for example, may dictate that the bearing length cannot be shortened sufficiently to create adequate rotor-bearing stability at normal operating conditions. In this situation, another method must be used to create stable rotor bearing operation.

The reason for so much concern about rotor stability is this: If rotor instability is not controlled, it can be very dangerous, even totally destructive to a rotating machine of any size, small, medium or gigantic, because the amplitude of the orbit of vibration can increase until the journal rubs the bearing wall with great force, destroying the bearing, and then the vibration amplitude can get even larger, resulting in catastrophic failure. There are examples where large steam turbine-generators with circular bore bearings became unstable and destroyed themselves within only a few rotations of the rotor. This is indeed a very serious matter.

In its simplest form, this form of rotor vibration is called “oil whip”, “oil whirl”, “rotor instability”, or “bearing instability”. There are many euphemisms that people use for the basic phenomena because, apparently, they wish to invent new names for this old and well recognized problem, presumably to “show original thought”, though, in reality, most of these names are forms of rotor instability induced by the bearing design itself.

On the other hand, there are other more complex versions of sub-synchronous rotor vibration that are induced by rotor flexibility, fluid flow forces along a rotor, or magnetic forces, that are not involved with the simple forms being discussed here. In addition, the most complex forms of rotor instability involve a combination of rotor natural frequencies that match the frequencies induced by the journal bearing design itself, and these usually are solved only by the use of tilting pad bearings of sophisticated design, not discussed here.

Rotor instability is often considered to be the most prominent factor in limiting the “power density” (hp per lb) of rotating machinery. Due to the highly destructible results of “oil whip” (or use any other name you wish for sub-synchronous rotor vibration), great efforts have been made to suppress this phenomenon since it first appeared approximately a century ago.

One of the earliest methods to suppress “oil whip” was the “pressure dam” bearing, and later, the “elliptical bearing” was developed. Since perhaps the late 1940s / early 1950s, “tilting pad” bearings have been developed and used with great success for controlling rotor instability in the most difficult cases.

While other Tech Notes have focused on TRI Tilting Pad Bearings, this Tech Note is
focused on Pressure Dam Bearings and Elliptical Bearings primarily for these reasons:

1. These bearing designs are created from a circular bore bearing via small changes in bearing bore geometry that achieves substantial increased downward loading, \( W_{\text{add}} \);

2. The increased downward loading decreases the Sommerfeld number, which stabilizes the rotor bearing design;

3. In many cases, using pressure dam or elliptical bearings instead of circular bore bearings produces stable operation, and then there is no need to change the rotor design or to change the axial length of the bearing; and

4. The modified bearing bore geometry for either design is relatively easy to manufacture.

Consequently, for certain applications where pressure dam or elliptical bore bearing designs can be properly applied, they are still in substantial use today, and will be for the foreseeable future.

**Pressure Dam Bearings:**

Consider this example: A 3600 rpm application with a high transmitted torque and a journal diameter of 7 inches might have a diametral clearance on the order of 0.007 inches to 0.010 inches. If the weight of the rotor is low or moderate, the values of specific loadings \( W/(LxD) \) for these bearings might be in the range of 50 psi to 150 psi. That is, the bearing is not heavily loaded, leading to rotor instability.

A solution providing for a stable design might be obtained by the use of a pressure-dam bearing. A pressure dam bearing differs from a circular bore bearing by the addition of a shallow groove in the upper half and the axial length of this groove is usually over half of the axial length of the bearing. The groove starts at the horizontal joint on the up-coming side and terminates about 30 degrees past top dead center as shown upside down in the right hand side of Sketch 9. This groove creates a pressure

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**Sketch 9: Disassembled Pressure Dam Bearing**
distribution in the oil film of the top half that produces a significant downward force on the journal.

The downward force created by the pressure dam adds to the rotor weight supported by the lower half of the bearing. If this total downward force is sufficient to move the operating condition on the Sommerfeld-Omega chart from a point in the unstable region to a point within the stable region, the sub-synchronous vibration is suppressed, and the rotor-bearing system operates stably, that is, operates successfully without sub-synchronous rotor vibration.

The actual dimensions of the bearings can be optimized for each application using computer programming techniques that have been well understood since the mid-1960s when computers with very large memory became available. In other words, the reason why these computer-based analytical methods for solving rotor-bearing instability problems became available at that time in history was not the mathematics, models, or computational techniques, because they were known. However, simply before that time the data storage capabilities of the commercially available digital computers were insufficient to handle the very large data storage requirements for modeling film bearings using time-transient solutions with any degree of efficiency.

**What are the limitations of the Pressure Dam bearing design?**

There are many legitimate reasons for high amplitude rotor vibrations to occur, such as a crank in a coupling between adjacent shafts or a pressure dam pressure distribution that is not well developed at low speed permitting the journal to rub the bearing; thermally induced misalignment can cause a journal to rub the top of a low-clearance bearing; a rotor may become highly unbalanced with a corresponding high synchronous vibration which can lead to rubs

Sketch 10: Leading Edge Erosion of Pressure Dam Bearing
Babbitt chunks can enter the thin film of the lower half leading to a smeared or wiped bearing. At 3600 rpm, it does not take many minutes of this contact before severe damage results.

There must be other bearings that are less sensitive to damage at the leading edge or adjacent to the oil supply groove, such as discussed just above. Indeed, there are: elliptical bearings are much more robust than circular bore or pressure dam bearings, and they will be discussed in the next section:

**Elliptical Bearings:**

Elliptical bearings are made by installing a shim in the horizontal joint on each side of the bearing before the bearing is bored with a circular bore. The shim on each side is the same thickness. After being bored, the bore diameter is measured and recorded. If bored correctly, i.e., no taper, the bore will have the same diameter at every location that a measurement is taken: near the horizontal joint, on either side of the horizontal joint, and in the direction perpendicular to the horizontal joint.

Then, the bearing is disassembled and the shim is removed.

Regarding the bottom half, if the journal hits the leading edge of the bottom half of any circular bore or pressure dam bearing, the Babbitt can be damaged and spalled, and
What is important to note is that while it is common to measure the bore diameter just above and below the horizontal joint, it is a totally useless dimension because this bore dimension will be less than the bore measured with the shim in place and will be more than the vertical diameter. This dimension that is measured near the horizontal joint cannot and definitely should not be used as the dimension to bore this bearing again at a later date because it will be too small and may lead to overheating. The only reason that I have found for a mechanical technician to take this measurement is, “I am doing what I was told: take the ‘horizontal bore’ measurements and fill in the blanks”. A waste of precious time.

If the real diameter that was used to machine the bore is wanted, do not take the bore dimensions near the horizontal joint, but install the shims of the proper thickness used in the manufacturing of the bearing. Then the bore measurements in multiple directions may be taken and, if the bore is round, these measurements will be the same and will be the previous diameter of the bore. Sometimes this may be a process of installing a first shim thickness, and finding that the bore is not round, then repeating the process with another shim thickness. In a few steps, the bore will be round, or nearly round, perhaps with the exception of where a wear pattern in the bore exists, and the previous machined bore diameter will be known with certainty. It should also be noted that the bore diameter that was found with this process may not be the correct bore diameter for that application. Nevertheless, the previous bore diameter will be revealed and a decision can then be made with proper information, not guessed.

The ellipticity ratio is a parameter that is helpful for ascertaining the stability characteristics of a journal bearing.

\[
\text{Ellipticity ratio} = \frac{C_{da} - C_{dm}}{C_{dm}} = 1 - \frac{C_{da}}{C_{dm}}
\]

Wherein:

- \[C_{dm} = D_m - D_j = D_{\text{machined}} - D_{\text{journal}}\]
- \[C_{da} = D_{\text{assembled}} - D_{\text{journal}}\] which is the same as
- \[C_{dv} = D_{\text{vertical}} - D_{\text{journal}}\]
- \[T_{\text{shim}} = C_{dn} - C_{dv}\]
that the ellipticity ratio directly affects the stability of the rotor very significantly, as shown in the Stability Chart of Sketch 13.

Typical stability boundaries for circular bore bearings and for elliptical bore bearings with a particular ellipticity ratio are shown in this chart. The stability boundary for the circular bore bearing is on the left and the one for the elliptical bore bearing of the same length has moved to the right and up. From this chart, it is clear that ellipticity greatly enhances the stability characteristics. The fact that ellipticity moves the stability boundary to the right as well as upwards is very helpful because it means that the use of elliptical bearings permits a rotating machine to operate with higher speeds, larger clearances, lighter loads, larger diameter

Different bearing manufacturers use different values for $C_{dm}$ and $C_{da}$ ($C_{dv}$). They depend upon their own experience with their machines as to what values they wish to use for these parameters.

Different manufacturers also tend to use different arc lengths for oil supply grooves, and whether they wish to supply oil on only one side or both sides. All of these factors do have influence on the stability characteristics of the rotor-bearing system.

As shown in Sketch 12, one of the great advantages that elliptical bearings enjoy is that when properly designed, it is not possible for the journal to contact either the leading edge or the trailing edge of the top arc or the bottom arc.

Another advantage of elliptical bearings is

Sketch 12: Elliptical Bore Bearing Showing Non-contact Areas At The Joints
shafts, longer bearings, and lubricants with higher viscosities. All of these trends make it much easier to design machinery with increased power densities.

There are factors that complicate these trends, and sometimes the complications are very substantial. For instance, increasing the clearance usually increases the lube oil flow through the film, and this causes a reduction of temperature which increases the effective viscosity, a destabilizing influence and an unintended consequence. Furthermore, with increased surface velocity due to increased diameter, the Reynolds number for the film increases and the turbulence in the film increases in a non-linear manner. Increased turbulence dramatically increases the effective viscosity in the film, leading to an increased tendency for rotor instability and, incidentally, to much higher power loss. The only method for accurately modeling film bearings, particularly for large diameter, high speed bearings, is to use very complex mathematical models and associated numerical computer programs that can simultaneously handle the influences of all of these parameters as well as variable viscosity and variable turbulence throughout the films. TRI has such computer programs and a great deal of successful experience applying them.

From a separate viewpoint, a principal reason that elliptical bore bearings have improved stability characteristics is that there is a separate wedge oil film in the top arc in addition to the usual wedge oil film in the bottom arc. This wedge film in the top arc is very important because it produces a significant downward film force that is stabilizing. With two wedge films in a bearing, when the viscosity of the oil film in the bottom half is increased, the viscosity in the oil film in the top half is also increased in a similar manner, and these tend to offset or balance each other to some degree. Other parameters have similar offsetting characteristics: they affect the film force of
stabilize the rotor-bearing system, whereas, for an elliptical bore bearing with bottom and top films, increasing the lube oil temperature reduces the film viscosity in both bottom and top wedge films and this does not always assure improved rotor stability: The offsetting effects are not always fully predictable. Nevertheless, on balance, elliptical bore bearings are a great improvement over both circular bore bearings and pressure-dam bearings.

While TRI uses pressure dam bearings in specific applications, TRI prefers elliptical bore bearings over pressure dam bearings for most applications because, in almost every case, elliptical bore bearings keep the journal from contacting and damaging the leading edges of the arcs. Further, without having the narrow seals alongside the pressure dam groove, and having an upper arc with a much longer axial length, the wedge film of the top arc of an elliptical bearing produces a larger downward force than a pressure dam bearing, particularly as the journal approaches the bearing surface and the film gets thin, and for this reason, an elliptical bearing is more difficult to damage or erode than is a pressure dam bearing.

Because elliptical bearings have a top arc with a stronger downward force than can be achieved with a pressure dam bearing, elliptical bearings have greater stability over a wider range of design parameters, and therefore, they are preferred.