Rotor Dynamics as a Tool for Solving Vibration Problems
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Introduction
This paper continues the series begun in 2001 for the Vibration Institute annual meeting proceedings. The two companion papers, *Understanding Journal Bearings* published in 2001 and *Practical Rotor Dynamics*, published in 2002, discussed the theory behind lubrication, fluid film bearings and rotor dynamics. This paper combines the theory of those two works into a guide for implementing rotor dynamics into the process of machinery evaluation. The two areas that can be resolved with rotor dynamics evaluation and modification are critical speed response and rotor system stability.

In order to use rotor dynamics as a tool to solve machinery problems, one must first look at what factors can be altered to make the machinery operate with less vibration and stress. In most cases this means changing or altering different machine components since replacing a machine is often too costly or impractical. For example, if a shaft keeps breaking, a change to high strength material may eliminate the failures. However, changing the shaft diameter may require significantly more effort and expense.

In an effort to limit the scope of this paper, a typical rotor was constructed that might represent a multi-stage turbine or compressor. Figure 1 is a cross-section of the computer model of this rotor as constructed in the DyRoBeS finite-element based analysis software.

![Figure 1 - Idealized Rotor Cross-Section](image)

This rotor will be used throughout this paper as a benchmark that will be evaluated as various bearing factors are adjusted. The rotor is exactly symmetrical with negligible overhung weight. The rotor weighs 1000 pounds, has a principal diameter of 5.0 inches and a bearing span of 50 inches. The bearing journals are 3 inches in diameter. Speeds from 1,000 to 10,000 RPM will be evaluated. The equation for calculating shaft stiffness is:

$$K_s = \frac{48 \ E \ I}{L^3}$$

where

$$I = \frac{\pi \ D^4}{64}$$
So, for this case the shaft stiffness is 348,000 LB/IN which is reasonably flexible for this type of machine. The guideline previously expressed by the author is that a span-to-diameter ratio of 10 or more usually signals a potential problem machine with high amplification factor and low stability.

Let’s assume that this is a problem machine. What could be done to make it run better with the least effort and cost? In order of increasing difficulty, possible modifications would be:

1. Change bearing clearance
2. Change bearing axial length (width)
3. Modify or change bearing type
4. Modify overhung moment
5. Strengthen foundation
6. Change bearing span
7. Change shaft diameter
8. Alter total rotor mass

Unfortunately, only the first 3 items can be done relatively inexpensively and reversibly. The other changes can be very costly and may be irreversible except at additional cost.

Figure 2 - Plain Journal Bearing
For comparison, let’s also assume that the original bearings (figure 2) are plain circular bush type with 2 axial grooves for lubricant entry. A photo of this type of bearing is shown in figure 3. A clearance ratio of 1.5 mils/inch diameter (4.5 mils on diameter for a 3 inch shaft) is considered standard. If we choose a babbitted length of 1.667 inches, the resulting specific load (W/LD) of 100 PSI will also be a good nominal starting value. In all cases, ISO 32 lubricant was used at 120°F inlet temperature. Babbitt and steel were the assumed bearing materials. DyRoBeS software was used for all bearing characteristic evaluations.

Figure 3 - Plain Journal Bearing Picture
**Journal Bearing Characteristics**

There are 8 stiffness and damping characteristics that are important - see figure 4. The designations used in this paper are:

- **Kxx** - Principal Horizontal Stiffness
- **Kxy** - Cross-Coupled Stiffness
- **Kyx** - Cross-Coupled Stiffness
- **Kyy** - Principal Vertical Stiffness
- **Cxx** - Principal Horizontal Damping
- **Cxy** - Cross-Coupled Damping
- **Cyx** - Cross-Coupled Damping
- **Cyy** - Principal Vertical Damping

The principal stiffness and damping characteristics are easy to understand. Shaft motion in the horizontal direction is met with resistance by the horizontal stiffness. The horizontal damping reacts to the horizontal velocity of the shaft. In essence, the damping is a frequency dependent dynamic stiffness.

![Figure 4 - Schematic of Bearing Stiffnesses and Dampings](image-url)
Less obvious are the cross-coupled stiffnesses. What this means is that when a shaft moves in one direction, a stiffness or reaction force is generated at 90 degrees to the direction the shaft is moving. For example, a positive $K_{xy}$ means that shaft movement in the $+x$ direction results in a reaction force in the $+y$ direction. Unfortunately, for most fixed-lobe journal bearings, the $K_{yx}$ value is negative. This means that a $+y$ motion of the shaft results in a $-x$ reaction force. Vectorially, this causes a net force in the direction of rotation that often leads to forward unstable shaft whirl. It is very important to study and compare the $K_{yx}$ values when evaluating different bearing designs because this factor often dominates the stability calculations.

While all 8 bearing coefficients are needed to do a complete analysis, and all 8 were used in all calculations performed for this paper, the cross-coupled damping has minor effects in most cases and is ignored in this paper. Also, in an effort to reduce the number of figures, only the vertical stiffness and damping plots will be presented.

The other consequences of changing bearing geometry that will be examined are bearing maximum temperature, rotor synchronous unbalance response and rotor stability. Several other factors that the designer should consider but are not addressed here are maximum hydrodynamic film pressure (always much higher than the specific load) and minimum oil film thickness. This author recommends avoiding peak film pressures above 1000 PSI and minimum oil film thicknesses less than $\frac{1}{2}$ mil.
**Bearing Clearance Changes**

So, what happens as the bearing clearance is varied from 1 mil/inch (3.0 mils) up to 3 mils/inch (9.0 mils)? Since stiffness plays such an important role in controlling rotor motion, we would be very interested in the effect clearance has on bearing stiffness. In a horizontal rotor with only gravity load, the vertical stiffness will almost always be greater than the horizontal stiffness. In the horizontal direction, the tighter the bearing clearance, the higher the stiffness. In fact, doubling the clearance halves the horizontal stiffness. The vertical stiffness variation with clearance, figure 5 shows the relationship is similar but not as strong. At low speeds the clearance doesn’t make much difference. Figure 6 shows a similar relationship for the damping changes with clearance variation.

Figure 7 indicates that increasing clearance reduces the absolute value of the Kyx cross coupling. This seems to say that opening the clearance will increase the rotor stability. Unfortunately, the increased clearance also means less direct damping. So it is not always clear what changing bearing clearance will do to the rotor dynamics.

One factor which must always be considered in the bearing design is babbitt temperature. Figure 8 shows that one mil per inch diameter bearing clearance is probably too tight from a temperature consideration alone.
Figure 5 - Plain Bearing Vertical Stiffness Variation with Clearance

Figure 6 - Plain Bearing Vertical Damping Variation with Clearance
Figure 7 - Plain Bearing Cross-Coupled Stiffness Variation with Clearance

Figure 8 - Plain Bearing Babbitt Temperature Variation with Clearance
**Bearing Load Changes**

Another easily implemented option is to increase or decrease the axial length of a bearing. Often there are physical dimensional limits on how wide the bearing can be made. In this section we'll examine the effects of different bearing widths from 3.333 inches to 0.833 inches. With the 500 pound load, this corresponds to specific loads of 50 to 200 PSI. Although some machines have bearings loaded less or more heavily than this range, this covers the most common situations.

In the horizontal plane, as load decreases from 200 PSI to 100 PSI, the horizontal stiffness increases. However, further load reduction to 50 PSI results in a rapid drop in horizontal stiffness. The vertical stiffness, figure 9, increases significantly as loading increases. Since the critical speeds are most sensitive to vertical stiffness, this may be a way of controlling the frequency of a rotor’s critical speeds.

Examining the load effect on damping, the horizontal damping increases as loading decreases. The vertical damping, figure 10, is much less sensitive to load but the 50 PSI case has the highest damping.

The cross-coupling variation with load, figure 11, shows that the 100 PSI case is the best below 7,500 RPM. Both the lower and higher loads have more cross-coupling and, therefore, less stability.

Another non-obvious result is shown in figure 12. The trend for 100 up to 200 PSI tracks increasing temperature. However, the 50 PSI case has higher temperatures. This is due to the longer bearing having more surface area where the oil is being sheared creating more heat loss.
Figure 9 - Plain Bearing Vertical Stiffness Variation with Load

Figure 10 - Plain Bearing Vertical Damping Variation with Load
Figure 11 - Plain Bearing Cross-Coupled Stiffness Variation with Load

Figure 12 - Plain Bearing Babbitt Temperature Variation with Load
**Changing Bearing Type**

Five different fixed-lobe bearing designs were considered. These are all fairly common designs when a wide range of machinery is considered. Certain industries and manufacturers will use one or more types that they have experience with and avoid other types. There is no best bearing for every situation. As before, we will look at the effect of bearing type on the bearing stiffness and damping characteristics. It should be noted that these comparisons are only valid for the configurations shown. There are many different ways to manipulate the geometry of bearings to produce different dynamics characteristics.

The five types are shown in figure 13. Obviously, for each type, there are more than a few geometry choices. In all cases, the set clearance was held at 4.5 mils, the load at 500 pounds and the length at 1.667 inches for a 100 PSI specific load. For the elliptical bearing, a preload of 50% was chosen as typical. This means that the horizontal clearance is twice the vertical clearance or 9.0 mils. The offset half bearing is also preloaded 50% with 100% offset. That is the trailing edge (radial) clearance is 2.25 mils while the leading edge (radial) clearance is 4.5 mils. The 3-lobe bearing consists of 3 tilted lobes with minimum clearance on the trailing edge. The pressure dam bearing has been optimized with a 1.0 inch wide dam that is 5 mils deep at 155° from the horizontal axis in the direction of rotation. Of these five, all are relatively simple and inexpensive except the 3-lobe which requires special machining techniques.

![Fixed Profile Bearings - Cross Section Comparison](image)
The horizontal stiffness, figure 14, varies greatly with type. Due to the large horizontal clearance, the elliptical bearing has the lowest horizontal stiffness. The pressure dam stiffness increases with speed as the pressure in the pocket increases and presses down increasing the effective bearing load.

The vertical stiffness comparison, figure 15, indicates that above 4,000 RPM the elliptical bearing has the highest stiffness while the offset-half and pressure dam the lowest. Thus, critical speed frequencies can be changed by changing bearing type.

The plain bearing has the highest Kyx cross-coupling as the comparison, figure 16, shows. The pressure dam gets better with speed while the 3-lobe has the least cross-coupling.

The temperature comparison, figure 17, shows that any bearing runs cooler than the plain design. This can be very important when temperature is a problem. The offset-half bearing is cost effective and a very efficient method of cooling a hot bearing. However, it won’t run backwards!

For horizontal damping, figure 18, the plain bearing and pressure dam have the most while the offset-half the least. In the vertical plane, figure 19, there is a similar trend except the 3-lobe has the least vertical damping.
Figure 14 - Horizontal Stiffness Variation with Bearing Type

Figure 15 - Vertical Stiffness Variation with Bearing Type
Figure 16 - Cross-Coupled Stiffness Variation with Bearing Type

Figure 17 - Babbitt Temperature Variation with Bearing Type
Figure 18 - Horizontal Damping Variation with Bearing Type

Figure 19 - Vertical Damping Variation with Bearing Type
**Unbalance Response**

There are two primary evaluation speeds that need to be considered when evaluating a bearing geometry change: critical speed performance and operating speed performance. For our test rotor, there is only one critical speed. At the critical speed we want low amplitude and a low amplification factor. At operating speed, we want the lowest possible amplitude. Unfortunately, these two traits are sometimes mutually exclusive since damping controls the critical speed amplitude but can increase the synchronous amplitude above the critical speed. For every situation in this evaluation, the imbalance placed on the subject rotor was 2.0 ounce-inches at the center of the rotor. This is equivalent to 16W/N imbalance at 8,000 RPM for this 1,000 pound rotor. The unbalance response analysis is linear so the results can be scaled for any level of center span imbalance.

**Rotor Mode Shape**

There is only one critical speed for this rotor below 10,000 RPM; it is around 4,100 RPM. At resonance, the mode shape is dependent on the stiffness of the bearings. At low bearing stiffness, where the bearing stiffness equals the shaft stiffness, the mode shape shows little bending as seen in figure 20.

![Figure 20 - Rotor Mode Shape $K_B = K_s$](image)

If the bearings are 5 times stiffer than the rotor, the first critical speed mode shape will look like figure 21 with significant rotor bending.

![Figure 21 - Rotor Mode Shape $K_B = 5K_s$](image)
Bearing Clearance Effects

As bearing clearance is reduced, vertical stiffness and damping both increase. In order to fairly compare all conditions, the amplitude and amplification factor predicted for the rotor center will be evaluated for all clearance cases. Figure 22 is a plot of the rotor center displacement amplitude versus speed for the subject rotor (often called a Bodé plot) with different clearance plain bearings. As clearance is reduced, the frequency of the critical speed increases and the amplitude at resonance is also increases. The exception is the tightest case, 1 mil/inch clearance which has increased frequency but slightly reduced amplitude. We have already seen that the bearing temperature factor may eliminate having the bearing at 1 mil/inch clearance. However, if we consider the operating speed amplitude, the tight bearing will have the highest vibration. Table 1 is a listing of the results for this evaluation assuming a 9,000 RPM operating speed.

<table>
<thead>
<tr>
<th>C Mils</th>
<th>N&lt;sub&gt;CR&lt;/sub&gt; RPM</th>
<th>AMP (mils pk-pk)</th>
<th>AF</th>
<th>9,000 Amp</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.0</td>
<td>4,224</td>
<td>3.1</td>
<td>9.5</td>
<td>0.52</td>
</tr>
<tr>
<td>4.5</td>
<td>4,200</td>
<td>3.3</td>
<td>11.7</td>
<td>0.40</td>
</tr>
<tr>
<td>6.0</td>
<td>4,162</td>
<td>2.9</td>
<td>10.2</td>
<td>0.31</td>
</tr>
<tr>
<td>9.0</td>
<td>4,100</td>
<td>2.6</td>
<td>8.4</td>
<td>0.20</td>
</tr>
</tbody>
</table>

Table 1 - Clearance Comparison
Figure 22 - Critical Speed Performance Variation with Clearance

Figure 23 - Critical Speed Amplitude Variation with Clearance
Figure 23 shows that the critical speed amplitude peaks near the 1.5 mils/inch clearance ratio. However, since there is only 0.7 mils difference (about 20%) from best case to worst, clearance is probably not a very effective way to control critical speed amplitude. The amplification factor variation with clearance, follows a similar pattern with the open clearance having the lowest value. Likewise the synchronous amplitude at 9,000 RPM seems to favor the large clearance ratio. Critical speed frequency variation with clearance shows a change of 124 RPM from minimum to maximum clearance. This is probably not significant in most real machines. Although these figures suggest that a more open clearance is desirable to lower critical speed and operating speed amplitude, there are other factors to consider, as we shall see.

**Bearing Loading Effects**

Increasing the bearing load from 50 PSI to 200 PSI has been shown to increase vertical stiffness but not damping significantly. In fact high stiffness makes available bearing damping less effective. This is demonstrated in figure 24, the predicted critical speed response of the subject rotor as the bearing specific load is altered. Clearly the lower the specific load, the lower the critical speed amplitude and the lower the amplification factor. Table 2 is a summary of these results.

<table>
<thead>
<tr>
<th>Load PSI</th>
<th>$N_{CR}$ RPM</th>
<th>AMP (mils pk-pk)</th>
<th>AF</th>
<th>9,000 Amp</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>4,195</td>
<td>2.3</td>
<td>10.0</td>
<td>0.53</td>
</tr>
<tr>
<td>100</td>
<td>4,200</td>
<td>3.3</td>
<td>11.7</td>
<td>0.40</td>
</tr>
<tr>
<td>150</td>
<td>4,228</td>
<td>6.7</td>
<td>20.0</td>
<td>0.33</td>
</tr>
<tr>
<td>200</td>
<td>4,256</td>
<td>16.0</td>
<td>43.4</td>
<td>0.32</td>
</tr>
</tbody>
</table>

Table 2 - Specific Load Comparison

Figure 25 shows that the critical speed amplitude increases significantly with increasing bearing specific load. However, as the loading drops below about 125 PSI, there are diminishing benefits. Thus, a longer axial length bearing appears to be a good way to control critical speed amplitude. The amplification factor variation with specific load, figure 26, also indicates that keeping the bearing specific load below 125 PSI would be beneficial. Unfortunately the synchronous amplitude at 9,000 RPM, figure 27, favors the larger specific loads although load above 125 PSI shows marginal benefit. The critical speed frequency variation with specific load indicates only a 61 RPM gain from minimum to maximum specific load. This is even less significant than the clearance variation.

While the operating speed amplitude decreases with increasing load, the critical speed amplitude and amplification factor increase dramatically. Thus it can be concluded that lower specific load is very beneficial to decreasing the severity of critical speed response.
Figure 24 - Critical Speed Performance Variation with Load

Figure 25 - Critical Speed Amplitude Variation with Load
Bearing Type Effects

With different bearing types, the number of factors that can be adjusted increases. Because of this, these results must be tempered with the knowledge that additional “tweaking” of various geometric parameters could change the results significantly. Different rotor configurations may also show somewhat different results.

Figure 28 is a plot of the first critical speed response for nominal conditions for the five different bearing types. The elliptical and offset half bearings have the lowest amplitude while this rotor with pressure dam bearings has significantly higher center span vibration levels. Table 3 is a summary of these results.

<table>
<thead>
<tr>
<th>C Mils</th>
<th>( N_{CR} ) RPM</th>
<th>AMP (mil pk-pk)</th>
<th>AF</th>
<th>9,000 Amp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>4,200</td>
<td>3.3</td>
<td>11.7</td>
<td>0.40</td>
</tr>
<tr>
<td>Elliptical</td>
<td>4,108</td>
<td>2.8</td>
<td>9.1</td>
<td>0.37</td>
</tr>
<tr>
<td>Offset</td>
<td>4,078</td>
<td>2.9</td>
<td>9.1</td>
<td>0.18</td>
</tr>
<tr>
<td>Pressure Dam</td>
<td>4,246</td>
<td>9.0</td>
<td>27.2</td>
<td>0.37</td>
</tr>
<tr>
<td>3- Lobe</td>
<td>4,000</td>
<td>3.2</td>
<td>9.6</td>
<td>0.35</td>
</tr>
</tbody>
</table>

Table 3 - Bearing Type Comparison

Thus, the pressure dam bearing appears to be a poor choice for controlling critical speed amplitude. The reason for this is that the parasitic load created by the hydrodynamic pressure in the dam area increases the apparent specific load. As we saw in figure 24, higher specific loads cause higher critical speed amplitude. The pressure dam is more effective when applied to situations where the specific loading is very low.

In this study, the critical speed itself can be changed as much as 246 RPM by changing bearing type. This is about a 6% increase which may or may not be significant to any given system. It is difficult to alter the first critical speed of a very flexible rotor like this one with bearing changes.
Figure 28 - Critical Speed Performance Variation with Bearing Type

Figure 29 - Stability Variation with Plain Bearing Clearance
**Rotor-Bearing System Stability**

The stability of any rotor-bearing system is dependent on many factors. One of these is the rotor flexibility. In this case, with a fairly flexible rotor, stability will be a concern. Another issue is aerodynamic cross-coupling. While this is usually significant in all turbines and compressors, we will ignore it in this paper. As we have seen, this rotor has a critical speed near 4,100 RPM. Since our supposed operating speed is 9,000 RPM, or more than twice the first critical speed, stability is almost guaranteed to be a problem.

The stability parameter calculated here is the logarithmic decrement or log dec. The log dec is dimensionless since it is defined as the log base 10 of two successive subsynchronous amplitude cycles. When positive, subsynchronous excitation of the first critical speed will not occur. The greater the log dec, the more stable the system will be and the more destabilizing aerodynamic cross-coupling can be tolerated. When the log dec is zero, this is defined as the stability threshold. As the log dec becomes increasingly negative, the more violent the subsynchronous vibration.

Let’s start by looking at the effect of clearance variation on the rotor stability. Figure 29 is a plot of calculated log dec for the plain bearing with different clearance ratios. There is no direct trend. This indicates that the most stable bearing clearance ratio for this particular rotor is 2 mils/inch. At 9,000 RPM this rotor with plain bearings is unstable regardless of clearance with the largest clearance being least stable.

Fixing the clearance and varying the bearing length results in the stability plot shown in figure 30. Again there is no direct relationship although at 9,000 RPM the lightest specific loading is the least stable by far. The 100 PSI case appears to afford the highest stability threshold. However, at 9,000 RPM the heavier loading increases stability.

In order to increase system stability past what can be accomplished with clearance and bearing specific load, the other four bearing types were evaluated. Figure 31 is a plot of log dec versus speed for each bearing type at nominal conditions. Table 4 lists the bearings in order of increasing stability and the threshold speed. None of these designs will allow stable operation at 9,000 RPM.

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Stability Threshold</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>7,655 RPM</td>
</tr>
<tr>
<td>Pressure Dam</td>
<td>8,002 RPM</td>
</tr>
<tr>
<td>Elliptical</td>
<td>8,144 RPM</td>
</tr>
<tr>
<td>3-Lobe</td>
<td>8,275 RPM</td>
</tr>
<tr>
<td>Offset-Half</td>
<td>8,775 RPM</td>
</tr>
</tbody>
</table>

*Table 4 - Bearing Type Stability Comparison*

It is quite possible that some of the other design factors of these more complicated bearings could be adjusted to yield a stable rotor at 9,000 RPM or above. The offset half bearing is not difficult to make and has been applied successfully by the author to stabilize turbomachinery.
Figure 30 - Stability Variation with Plain Bearing Specific Load

Figure 31 - Stability Variation with Bearing Type
Tilting Pad Bearings

Perhaps the most common bearing found in new compressors and turbines today, the tilting pad bearing is not a modern invention. The earliest tilting pad bearing design found so far by this author is in reference 2, page 538 (figure 32).

This bearing, called a Michell Journal Bearing was used in the main drive turbine of the Gouverneur-General Chanzy steamship in 1922.

The basic concept is a series of pads that are free to rock in at least one plane independent of the other pads. This allows each pad to self-adjust to the static and dynamic loads. Perhaps the biggest benefit of this is the virtual elimination of stability reducing cross-coupling. The only drawbacks are cost, increased complexity, and often the need for more radial room than the fixed profile type bearings. The designer has many factors to control. Table 5 lists the most common build options besides clearance and axial length and some typical values.
<table>
<thead>
<tr>
<th>Option</th>
<th>Typical Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Pads</td>
<td>4 to 7</td>
</tr>
<tr>
<td>Load Orientation</td>
<td>On or Between Pivots</td>
</tr>
<tr>
<td>Preload</td>
<td>0 to 50%</td>
</tr>
<tr>
<td>Pivot Offset</td>
<td>50 to 75%</td>
</tr>
<tr>
<td>Pivot Type</td>
<td>Line, Point, Spherical</td>
</tr>
<tr>
<td>Pad Arc Length</td>
<td>60° (5 Pad)</td>
</tr>
<tr>
<td>Lubrication</td>
<td>Flooded, Directed</td>
</tr>
</tbody>
</table>

Table 5 - Tilting Pad Bearing Design Factors

In addition to the geometry options, the analysis can include variable oil film viscosity, pad bending effects, and thermal growth effects.

The most common design uses 5 pads with some preload and a 50% offset pivot for bidirectional operation. Load on pivot is often used because it creates large asymmetry between the horizontal and vertical stiffness and damping characteristics. This may enhance stability in some cases. Figure 33 shows cross sections of some typical tilting pad bearings.

![Figure 33 - Popular Configurations for Tilting Pad Journal Bearings](image-url)
Preload is an important concept in tilting pad bearing design. Starting with a pad that has equal clearance (which is zero preload) all along its arc, preload is defined as the percentage reduction of center-of-pad clearance. With zero preload the shaft center of curvature coincides with the pad center of curvature. As the pad is moved inward, the clearance at the center decreases faster than the edge of pad clearance. If the center of pad touches the shaft, then the preload would be 100 percent. In general, typical preloads in tilting pad bearings range from zero to 50 percent. From a practical standpoint, specifying zero preload can be a risk since machining tolerances could place the preload negative which is not desirable. Pad bending under load often occurs. With thin pads or heavy load, this can be significant and will increase the effective preload. Some computer programs allow for this effect to be included in the analysis.

Figures 34 and 35 show a comparison of the effects of different amounts of preload on the horizontal and vertical stiffness of a 5 pad load on pivot tilting pad bearing. In the horizontal plane, stiffness increases with increasing preload. In the vertical direction the lightly preloaded pad has the lowest stiffness. On both plots an additional case with 20 percent preload and a 60% pivot offset is included. The pivot offset has the effect of significantly increasing stiffness in both directions.

Examining the effects of preload on principal damping of the 5 pad load on pivot bearing, figures 36 and 37 show that lower preload results in higher damping. In the horizontal direction, pivot offset increases damping significantly; in the vertical direction it has a much less pronounced effect.

The maximum pad temperature, figure 38, is reduced as preload is increased up to 50 percent. The reason for this is that preload opens up the leading edge of the pad admitting more fresh cool oil and the open trailing edge promotes expulsion of the hot used oil. This can be take to extremes and very large preloads may cause higher pad temperatures. The offset pivot design at 20 percent preload causes a significant temperature reduction and this is one of primary uses for offset pivots. The drawback is that the offset pivot design is for one rotational direction only.

Another popular tilting pad bearing configuration is the 4 pad design with load between pivots. This arrangement results in symmetrical stiffness and damping characteristics. Figure 39 compares the principal stiffnesses of a 4 pad and a 5 pad bearing, both with load between pivots. Vertically, the 4 pad design has lower stiffness than the 5 pad and greater stiffness in the horizontal plane. The damping comparison, figure 40, is similar except the 4 pad bearing has higher damping at speeds above 5,000 RPM.
Figure 34 - Horizontal Stiffness of 5-Pad TPJ with Load On Pivot

Figure 35 - Vertical Stiffness of 5-Pad TPJ with Load On Pivot
Figure 36 - Horizontal Damping of 5-Pad TPJ with Load On Pivot

Figure 37 - Vertical Damping of 5-Pad TPJ with Load On Pivot
Figure 38 - Babbitt Temperature - 5 Pad TPJ with Load on Pivot

Figure 39 - Principal Stiffness Comparison of 4-Pad and 5-Pad TPJ
Figure 40 - Principal Damping Comparison of 4 Pad and 5 Pad TPJ

Figure 41 - Critical Speed Performance Comparison of TPJ Bearings
Tilting Pad Bearing Unbalance Response

Using the exact same rotor and imbalance used in the fixed profile comparisons, the critical speed performance of various tilting pad bearings was evaluated. Figure 41 shows that compared to the plain bearing, all of the tilting pad designs reduce critical speed amplitude and amplification factor. The 4 pad bearing is the best design for reducing critical speed response. Unfortunately, the increased effective damping of the 4 pad bearing causes a slight increase in operating speed amplitude. The opposite is true also with the plain bearing having the worst critical speed amplitude but the lowest high speed amplitude. Table 6 is a summary of the results. The 4 pad design has reduced the amplification factor so low that it is no longer classified by API as a critical speed.

<table>
<thead>
<tr>
<th>C Mils</th>
<th>(N_{CR}) RPM</th>
<th>AMP (mils pk-pk)</th>
<th>AF</th>
<th>9,000 Amp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain</td>
<td>4,200</td>
<td>3.3</td>
<td>11.7</td>
<td>0.40</td>
</tr>
<tr>
<td>5 Pad Load On</td>
<td>4,136</td>
<td>2.1</td>
<td>4.5</td>
<td>0.57</td>
</tr>
<tr>
<td>5 Pad Load Between</td>
<td>4,320</td>
<td>1.8</td>
<td>3.2</td>
<td>0.55</td>
</tr>
<tr>
<td>4 Pad Load Between</td>
<td>4,530</td>
<td>1.5</td>
<td>2.5</td>
<td>0.58</td>
</tr>
</tbody>
</table>

Table 6 - TPJ Unbalance Comparison
Tilting Pad Bearing Stability

All of the tilting pad bearings considered resulted in completely stable systems well past 10,000 RPM. Table 7 lists the log decs for each design with 20 percent preload at 9,000 RPM in order of decreasing stability. The primary factor was pivot offset. The increased dynamic stiffness from this arrangement was not beneficial to the stability of this very flexible rotor.

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Log Dec @ 9,000 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 Pad Load On Pivot 50% Offset</td>
<td>1.43</td>
</tr>
<tr>
<td>4 Pad Load Between 50% Offset</td>
<td>1.27</td>
</tr>
<tr>
<td>5 Pad Load Between 50% Offset</td>
<td>1.18</td>
</tr>
<tr>
<td>5 Pad Load On Pivot 60% Offset</td>
<td>0.37</td>
</tr>
<tr>
<td>5 Pad Load Between 60% Offset</td>
<td>0.37</td>
</tr>
</tbody>
</table>

Table 7 - TPJ Bearing Stability Comparison
**Conclusions**

1. Clearance adjustment of plain bearings can affect both critical speed frequency and amplitude.

2. For plain bearings, increased clearance results in lower critical speed amplitude.

3. Clearance adjustment of plain bearings can affect rotor stability.

4. Bearing loading has a very strong effect on the critical speed amplitude.

5. A longer axial length plain bearing appears to be a good way to control critical speed amplitude. However this will adversely affect rotor stability.

6. For each bearing type, bearing loading has an optimum value for maximum rotor system stability.

7. Fixed profile bearing type affects both critical speed performance and stability. Virtually any design discussed here is superior to the plain bushing.

8. Pressure dam bearings increase stability but may adversely affect critical speed response. Pressure dam bearings work best in lightly loaded situations at high surface speeds.

9. Clearance less than 1.5 mils per inch of shaft diameter may result in high bearing temperatures.

10. Offset half bearings run cool and are very stable.

11. Offset pivot tilting pad bearings can be used to reduce babbitt operating temperature but this design can increase critical speed amplitude and reduce stability.

12. For the case investigated here, the 4 pad, load between pad, tilting pad bearing had the lowest critical speed amplitude and amplification factor.

13. For the case investigated here, the 5 pad load on pad TPJ had the highest log dec at 9,000 RPM.

14. For a rotor where the bearing span is greater than 10 times the main diameter, altering the critical speed by more than a few hundred RPM with bearing modifications is virtually impossible.

15. The effects of changing clearance or bearing width are not obvious. Fixing a stability problem could result in excessive critical speed amplitude.

16. It is not wise to be limited to one geometric factor. Finding the optimum combination of clearance, load, preload, and orientation may take considerable time. It is likely a unique set of optimum factors exist for each different rotor system. All machines are different.
REFERENCES


