

Understanding Journal Bearings

Malcolm E. Leader, P.E.
Applied Machinery Dynamics Co.
Durango, Colorado

ABSTRACT

This paper covers the basic aspects of journal bearings including lubrication, design and application. Descriptions of various types of journal bearings are presented. Guidance is given for choosing the proper bearing type and keeping your bearings healthy. A section on do's and don'ts gives practical information.

INTRODUCTION

Bearings are used to prevent friction between parts during relative movement. In machinery they fall into two primary categories: anti-friction or rolling element bearings and hydrodynamic journal bearings. The primary function of a bearing is to carry load between a rotor and the case with as little wear as possible. This bearing function exists in almost every occurrence of daily life from the watch on your wrist to the automobile you drive to the disk drive in your computer. In industry, the use of journal bearings is specialized for rotating machinery both low and high speed. This paper will present an introduction to journal bearings and lubrication. Lubrication technology goes hand-in-hand with understanding journal bearings and is integral to bearing design and application.

Since they have significant damping fluid film journal bearings have a strong impact on the vibration characteristics of machinery. The types of machinery we are concerned with range from small high speed spindles to motors, blowers, compressors, fans, and pumps to large turbines and generators to some paper mill rolls and other large slow speed rotors.

Not covered here is the topic of bearings for reciprocating machinery. While some of the same principals apply, engine bearings have special needs and design considerations and deserve a more complete study. Reciprocating machinery bearings tend to be simpler in geometry and much more complicated in application than turbomachinery bearings. For example, the typical turbomachinery journal bearing consists of a thin layer of babbitt on steel while a connecting rod bearing may have numerous different layers of copper, steel, nickel, or other metals with a thin layer of babbitt on top. This layering is done for fatigue resistance to the pounding loads encountered in such machinery. Engine bearings are often required to withstand peak specific loads in excess of 3,000 PSI or about ten times a typical motor or turbine bearing. Reciprocating machines rely primarily on the squeezing of the oil film for load support.

WHEN TO USE FLUID FILM BEARINGS

There are applications where anti-friction bearings are the best choice. Commonly, smaller motors, pumps and blowers use rolling element bearings. Paper mill rolls often use large specialized spherical roller bearings. Clearly, anti-friction bearings are best for these applications. However, once the size of a pump (or fan or motor, etc.) gets large enough and fast enough, a gray area is entered. Here you will still find rolling element bearings used successfully but as speeds increase and temperatures rise, rotor dynamics often become a concern and critical speeds are encountered. This is when damping is required and fluid film bearings become increasingly necessary. My experience is that turbomachinery designers (and users) should consider using fluid film bearings if running above 3,000 RPM or the machine exceeds 500 HP. In my opinion, at 1,000 HP and up, all machines except very special cases should be on journal bearings specifically designed for that service. There are exceptions of course, and the decision where to apply what type of bearing is ultimately done for every machine individually based on good engineering practice and experience. Unfortunately, this decision is sometimes based on economics which keeps maintenance engineers and consultants employed.

ADVANTAGES OF FLUID FILM BEARINGS

The primary advantage of a fluid film bearing is often thought of as the lack of contact between rotating parts and thus, infinite life. In a pure sense, this is true, but other complications make this a secondary reason for using these bearings. During startup there is momentary metal-to-metal contact and foreign material in the lubricant or excessive vibration can limit the life of a fluid film bearing. For these reasons, special care must be taken when selecting and implementing a lubrication system and special vibration monitoring techniques must be applied. The most important aspects of the health and longevity of a fluid film bearing are proper selection, proper installation, proper lubrication, and the alternating hydrodynamic loads imposed on the bearing surface by relative shaft-to-bearing vibration.

Some of the primary advantages of fluid film bearings are:

- Provide damping. Damping is required in order to pass through a critical speed. Damping is also required to suppress instabilities and subsynchronous vibration.
- Able to withstand shock loads and other abuse.
- Reduce noise.
- Reduce transmitted vibration.
- Provide electrical isolation of rotor to ground.
- Very long life under normal load conditions.
- Wide variety of bearing types for specific applications

The lubricant used provides these functions to all bearings:

- Remove heat generated in the bearing.
- Flush debris from load area.

Some disadvantages to fluid film bearings are:

- Higher friction (HP loss) than rolling element type.
- Susceptible to particulate contamination.
- Cannot run for any length of time if starved for lubricant such as a lube system failure.
- Radial positioning of rotor less precise.

Use of journal bearings is also an advantage in many applications when it comes to maintenance. Most fluid film bearings are split and rotor removal is not required to inspect and replace. While split rolling element bearings are also available they are costly and not common. Journal bearing fatigue damage is usually visible at an early stage and allows for better diagnostics of failure modes so that corrective action can be taken to prevent recurrence.

PLAIN BEARINGS AND BASIC CONCEPTS

In order to illustrate the basic nomenclature, geometry, and introduce the ideas of how fluid film bearings work, the simplest bearing called a plain journal bearing will be examined. Figure 1 is a photograph of a plain bearing. A steel base material is overlaid with a babbitt material and bored to a circular diameter equal to the shaft diameter plus the desired clearance. Scallops are cut at the splitline to admit oil. Figure 2 is a computer model of this same bearing.

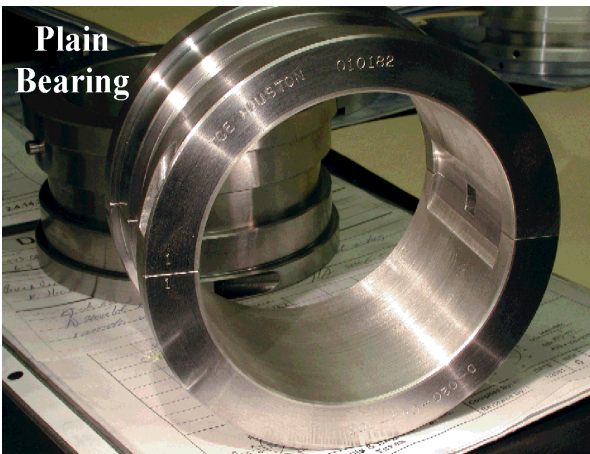


Figure 1 - Typical Plain Journal Bearing

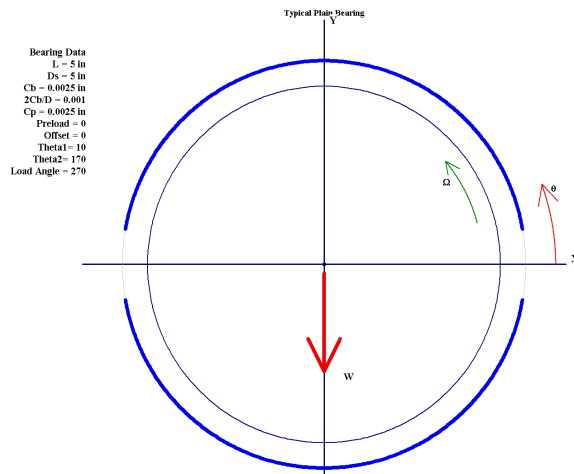


Figure 2 - Computer Model of Plain Journal Bearing

At zero speed, the shaft rests on the bearing at bottom dead center. As soon as shaft rotation begins the shaft “lifts off” on a layer of oil. In fluid film bearings, lubrication is required between a pair of surfaces with relative motion between them. There is always a convergent wedge developed that is formed due to the relative surface speeds and the lubricant viscosity to carry the applied load. An oil pressure film develops with equal and opposite force vectors to the applied load. One surface drags the lubricant, usually an oil, into a converging gap. As the space available in this gap decreases, the fluid develops a pressure gradient, or pressure hill. As the fluid leaves the gap, the high pressure helps expel it out the other side. A simple diagram of this is shown in figure 3.

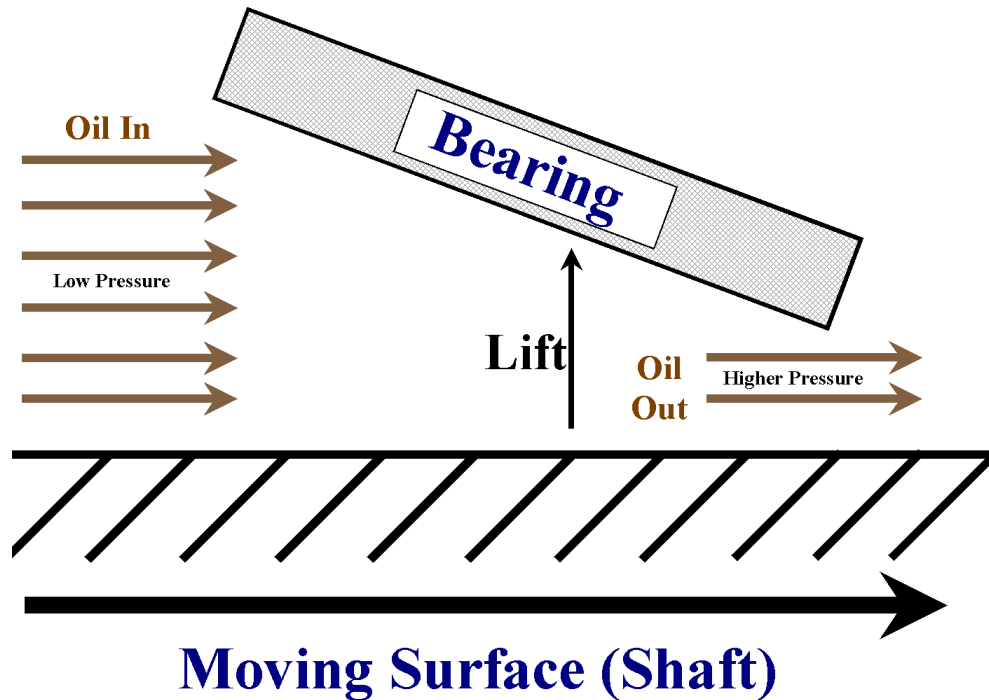


Figure 3 - Basic Development of an Oil Wedge

Lubricants can be any fluid, including gasses. In early reference books (1,2,3) some of the lubricants discussed are tallow, lard (animal fat), vegetable oils, and whale and fish oils. Obviously, sometimes you used what was available! Even water can be used under some conditions. Mineral oils from petroleum have evolved from straight distillates to complex formulations with special additives today. Synthetic lubricants have also been developed, primarily polyalphaolefins and esters. Silicones, glycols and other fluids are also used in special applications. There is no ideal or universal lubricant, all are compromises to fit any given situation. Applications range from heavy low speed loads to light high speed loads. At one extreme solid lubricants may be necessary and at the other, gas bearings may be required. Obviously, most applications fall in the middle where grease and oil lubricants are used. In this discussion of journal bearings we will limit ourselves to light oil lubrication found in the majority of turbomachinery.

Bearing Nomenclature

The shorthand that bearing analysts use with regards to journal bearings can be confusing and is certainly inconsistent from one analysis program to another. The terminology used in this paper is shown diagrammatically in figure 4. The symbolic notation and the definitions are as follows:

R_j = Radius of Journal

R_b = Radius of the Bearing

C_b = Radial Clearance of the Bearing = $R_b - R_j$

h = Radial clearance as a function of the angular position where the clearance is measured

h_{min} = Minimum oil film clearance

e = Eccentricity - the distance between the center of the bearing and the center of the shaft

$e/c_b = e/C_b$ = Eccentricity Ratio - if zero, shaft is centered; if 1 then shaft touches bearing

Line of Centers = Line connecting the center of the bearing and the center of the shaft

ϕ = Attitude Angle = Angle from -Y axis to Line of Centers

Ω or ω = Rotation Direction and Speed in RAD/SEC

W = Gravity Load

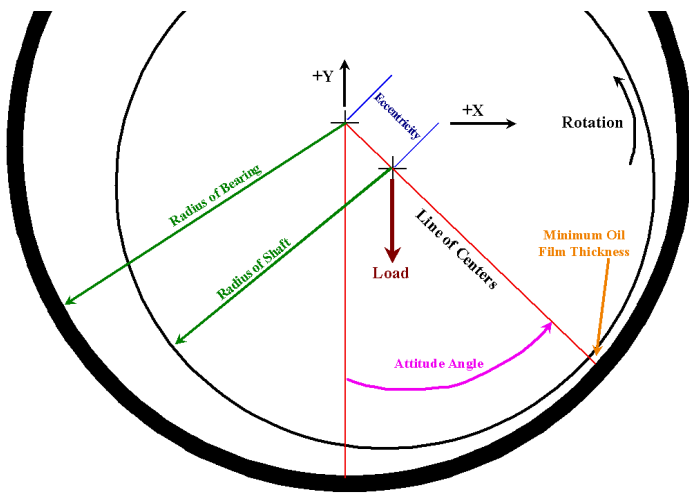


Figure 4 - Bearing Nomenclature

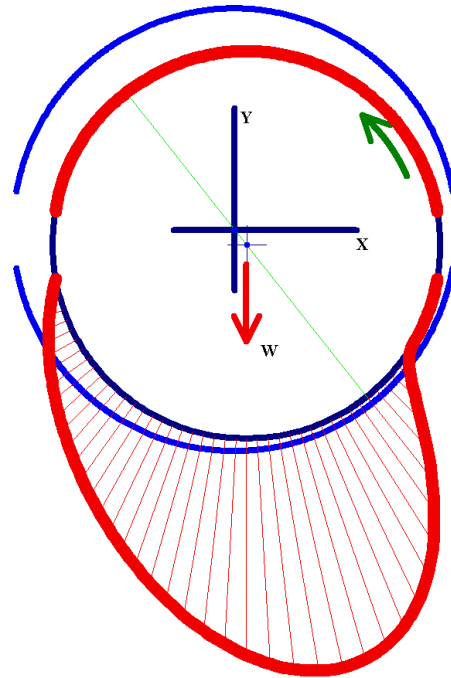


Figure 5 - Pressure Profile in a Journal Bearing

In our plain journal bearing example the load is supported by a high pressure oil region as shown in figure 5. Each line in the pressure profile represents an oil pressure vector at the centerline of the bearing. The sum of the vertical components add up to the applied load and the horizontal components cancel out for equilibrium. Oil inlet ports are placed in areas of minimum pressure. The pressure profile can also be examined in a three-dimensional format as shown in figures 6 (low load) and 7 (for a highly loaded bearing).

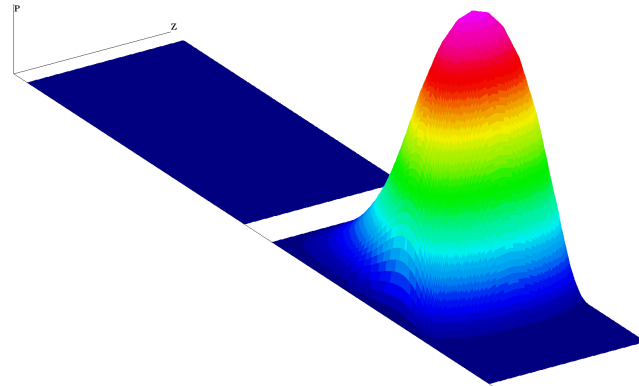
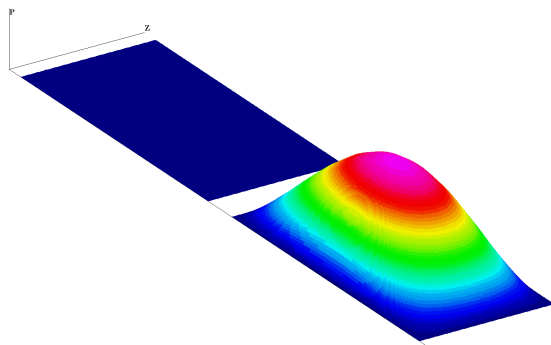


Figure 6 - Hydrodynamic Pressure Profile - Low Load Figure 7 - Hydrodynamic Pressure Profile - High Load

These figures show the bearing “unwrapped” with the top half on the left and the loaded bottom section on the right. It is important to note several things about the hydrodynamic pressure profiles. First, the peak pressure is significantly higher than the specific load (W/LD). Secondly, the pressure at the margins always returns to the boundary condition which is usually atmospheric pressure. The unloaded top half, even though it is cavitated is essentially at atmospheric pressure which is why no cavitation damage ever occurs in this type of bearing.

Bearing Performance

While the stiffness and damping provided by a journal bearing are crucial, there are other design factors that must be considered in order to understand how bearings work. For example, if the eccentricity is too high there is a risk of metal-to-metal contact and higher dynamic loads being imparted to the babbitt causing premature fatigue. If the eccentricity is too low (journal is nearly centered) then the machine could more easily become unstable. Eccentricity is a function of both speed and load. Figure 8 indicates that, with a constant load, as speed increases, the eccentricity decreases.

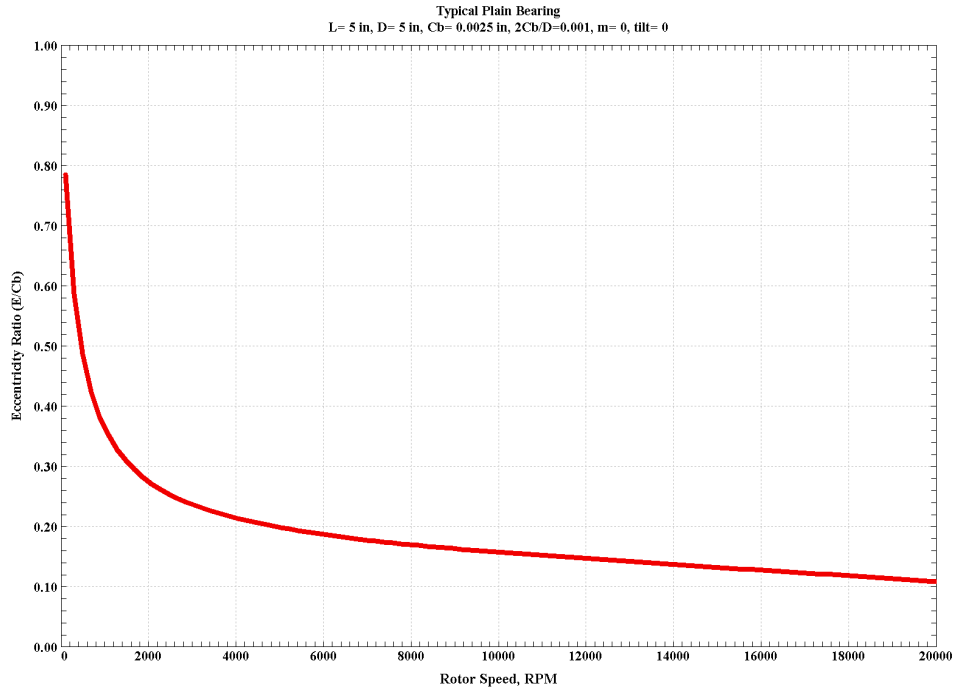


Figure 8 - Plain Bearing Eccentricity versus Speed with a Constant Load

The attitude angle between the vertical axis in the load direction and the line of centers also changes with speed. A plot of this angle with eccentricity describes a plot of the locus of the shaft centerline as speed changes as seen in figure 9.

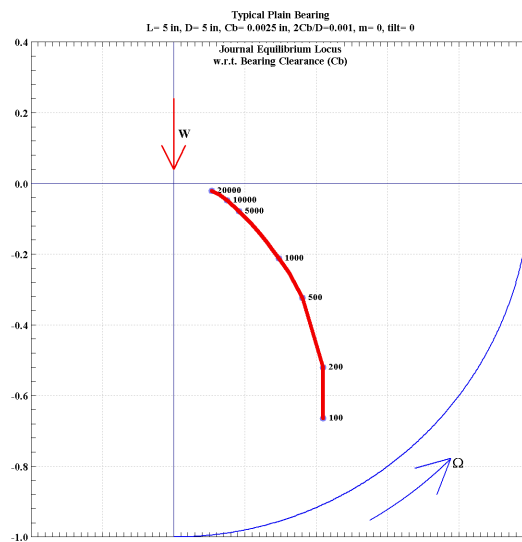


Figure 9 - Shaft Centerline Position in a Plain Bearing as a Function of Speed

If you combine the effect of speed and load on the bearing eccentricity, figure 10 tells the complete story. At low speeds the eccentricity ratio e/C_b is high. At low loads, the eccentricity is high.

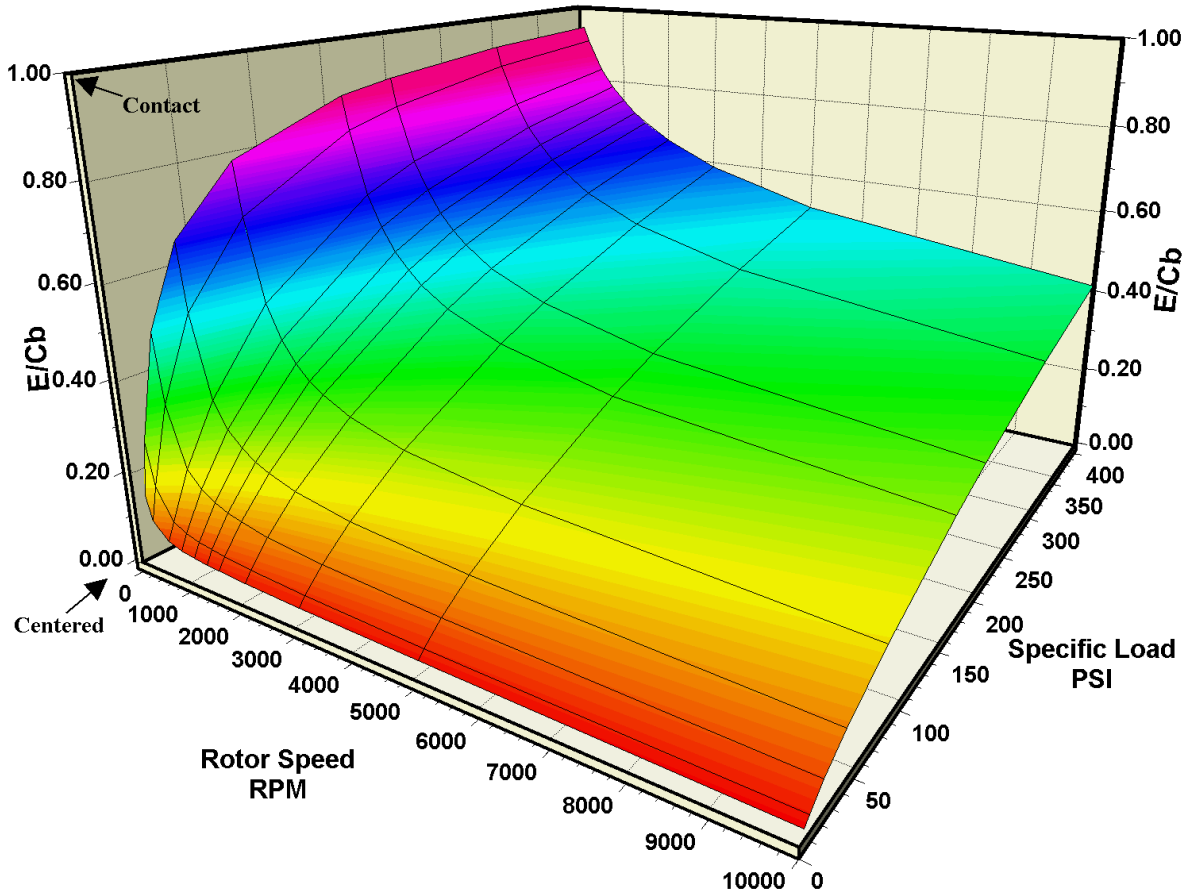


Figure 10 - Effect of Load and Speed on Plain Journal Eccentricity

There are additional parameters that must also be examined when analyzing fluid film bearings. One of the most important is the maximum temperature that will be generated in the fluid film. This is consistent with the horsepower losses for a given bearing and will be somewhat higher than the actual temperature measured with an RTD or thermocouple in the bearing shell. Figure 11 is a plot of the inlet oil temperature (kept at a constant 120°F in this case) and the predicted measured temperature and the predicted maximum oil film temperature for a constant load as speed is varied. While it is possible to operate journal bearings above 200°F, typically a bearing designer will seek to keep the maximum oil film temperature below that due to loss of babbitt fatigue strength. If possible, a good bearing design will have the maximum oil film temperature less than 175°F to allow for some margin for transient events. It is also important to analyze the minimum oil film thickness as seen in figure 12. Here the ratio h_{\min}/C_b is plotted as a function of speed with a constant load. This ratio is the minimum film thickness as a percentage of bearing clearance. If we knew the radial clearance was 5 mils then we can calculate the minimum film thickness at any speed. For 5 mils radial clearance in this case the h_{\min} would be about 1 mil at 100 RPM and more than 4 mils above 5,000 RPM.

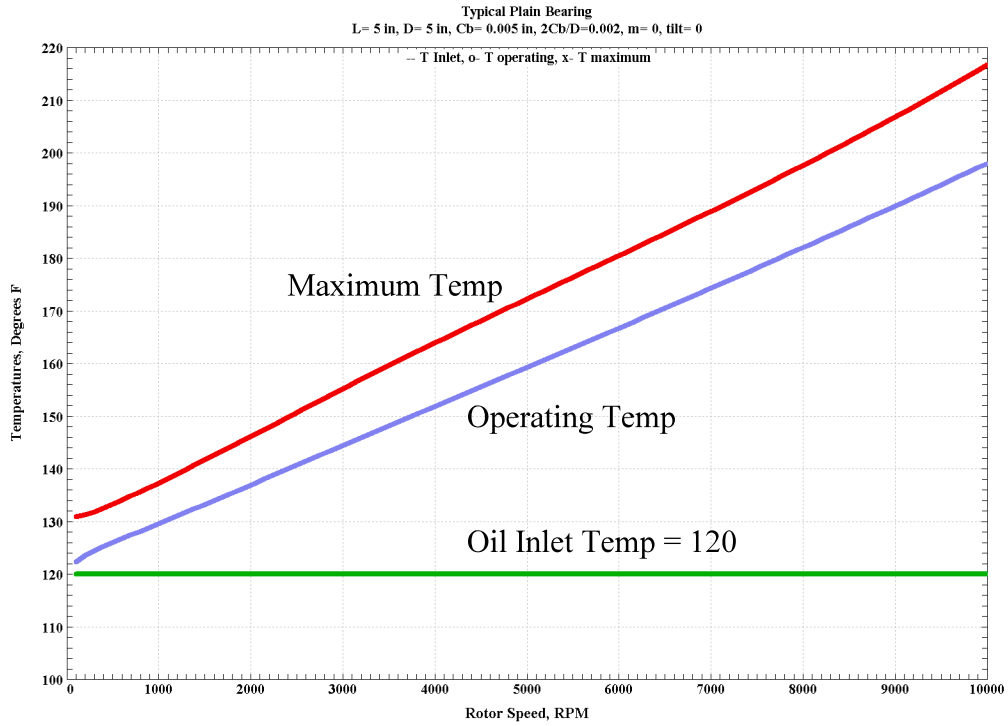


Figure 11 - Temperatures in a Plain Bearing versus Speed at a Constant Load

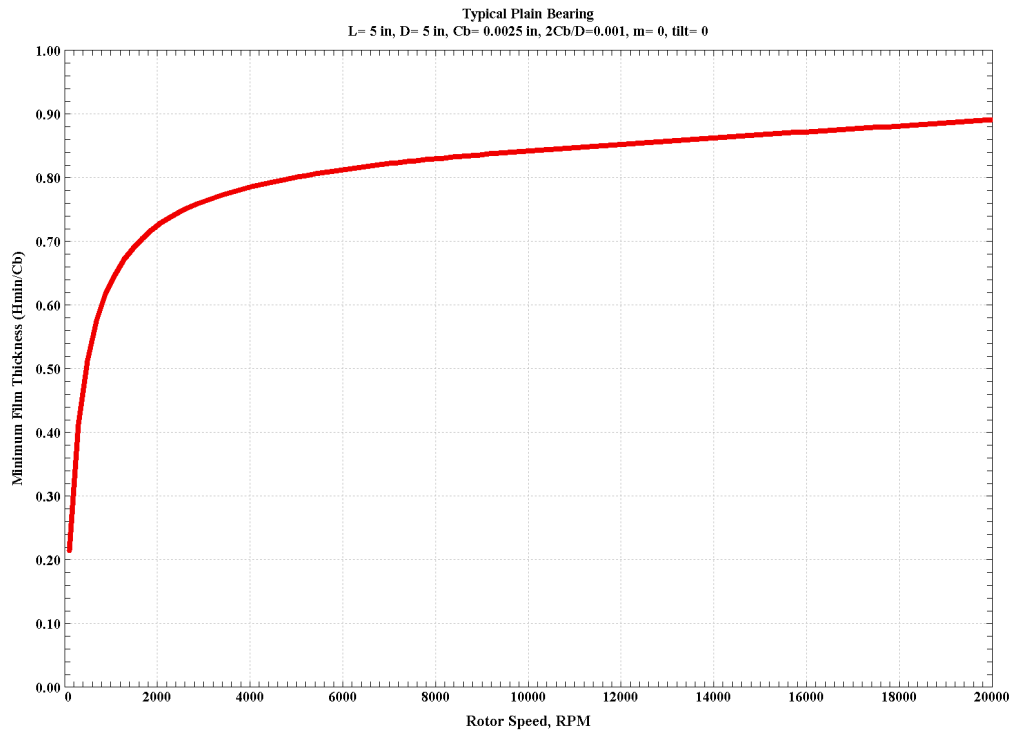


Figure 12 - Minimum Oil Film Thickness in a Plain Bearing versus Speed at a Constant Load

LUBRICATION

For fluid film bearings viscosity is the most important factor. Unfortunately, there are two forms of viscosity terminology and numerous units associated with these measures. Absolute or dynamic viscosity is the ratio of the shear stress to the resultant shear rate as a fluid flows. The more a fluid resists shear, the thicker it is and the higher the absolute viscosity. This is measured in Poise (or Centipoise) or Reyns. Some reference books use different terminologies, so read carefully. Kinematic viscosity is the absolute viscosity divided by the specific gravity. The most common unit of measure is the Centistoke, abbreviated cSt. The following tables help define these two measures of viscosity. Figure 13 relates SUS to these measures.

Table 1 is a multiplying factor chart for converting between the various viscosity units. Note that since the specific gravity (ρ) is usually between 0.8 and 1.2, the absolute and kinematic viscosities are nearly the same for all practical purposes. Table 2 is a list of typical applications and the viscosity ranges found in these applications. This list is by no means complete nor meant to preclude other oils in similar services. Viscosity varies significantly with temperature and the variation is highly non-linear. Often blending of oil bases is done to reduce these effects.

CONVERT FROM/TO	Poise (P)	CentiPoise (Z) cP	Reyn (μ)	Stoke (S)	CentiStoke (ν) cSt
Poise (P) Dyne-S/cm ²	—	100	1.45×10^{-5}	$\frac{1}{\rho}$	$\frac{100}{\rho}$
CentiPoise (Z) Dyne-S/100cm ²	0.01	—	1.45×10^{-7}	$\frac{0.01}{\rho}$	$\frac{1}{\rho}$
Reyn (μ) LB _r -S/IN ²	6.9×10^4	6.9×10^6	—	$\frac{6.9 \times 10^4}{\rho}$	$\frac{6.9 \times 10^6}{\rho}$
Stoke (S) cm ² /SEC	ρ	100ρ	$1.45 \times 10^{-5} \rho$	—	100
CentiStoke (ν) cm ² /100 SEC	0.01ρ	ρ	$1.45 \times 10^{-7} \rho$	0.01	—

Table 1 - Viscosity Conversion Factors

Application	Viscosity Range, cSt
Clocks, Fine Instruments, High Speed Spindles	2 - 20
Turbomachinery - Turbines, Compressors, Etc.	4 - 30
Rolling Element Bearings	8 - 300
Low Speed, Heavy Load Journal Bearings	8 - 100
Reciprocating Engines and Pumps	10 - 300
High Speed Gears	50 - 150
Low Speed Gears	50 - 600
Worm Gears	200 - 1000

Table 2 - Typical Viscosity Ranges for Various Applications

Figure 13 shows how temperature affects three common ISO turbomachinery oils. At lower temperatures the differences are much more significant than at a typical operating temperature.

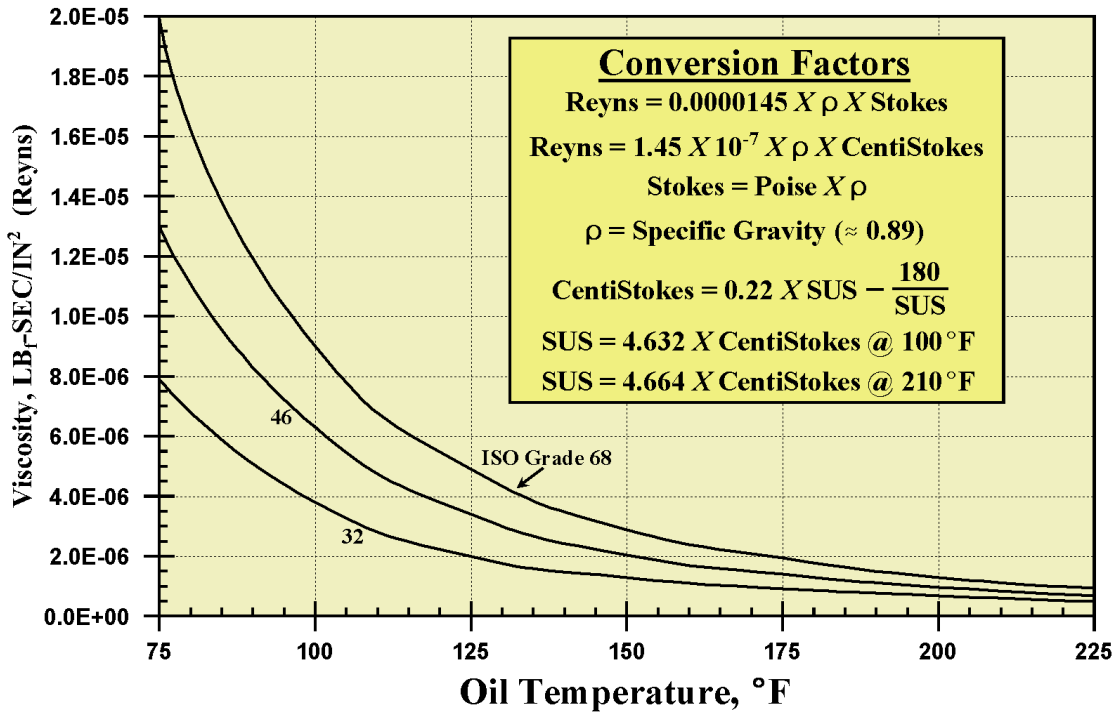


Figure 13 - Viscosity Variation with Temperature and Conversion Factors

The friction factor (and thus the horsepower loss) is a function of viscosity, load, and speed. This was quantified by Richard Stribeck, a German engineer who did extensive friction testing in 1902. There was a great deal of research at that time trying to find the best combinations of materials and lubricants that would give the lowest coefficient of friction. Figure 14 is the original Stribeck curve from Reference 2. The friction factor is plotted as a function of ZN/P where Z is the viscosity, N is the speed and P is the load. This is a non-dimensional equation.

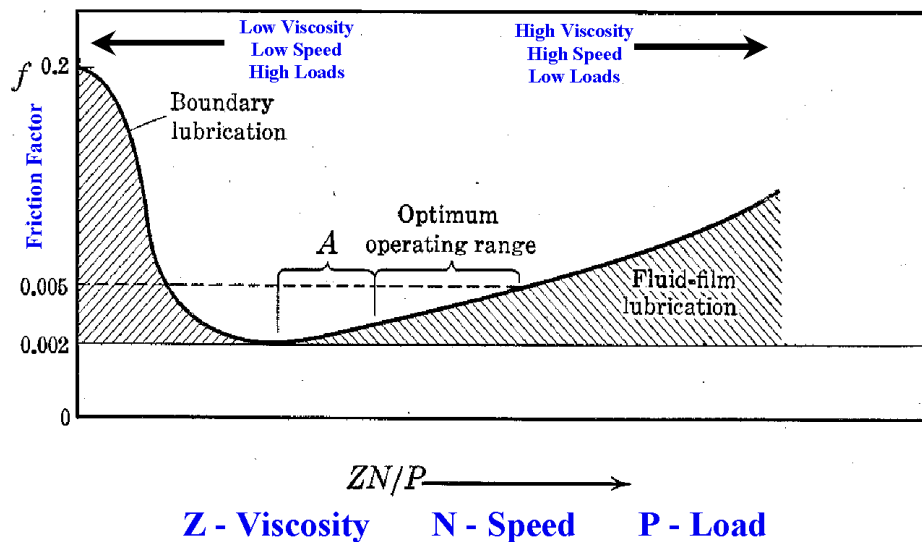
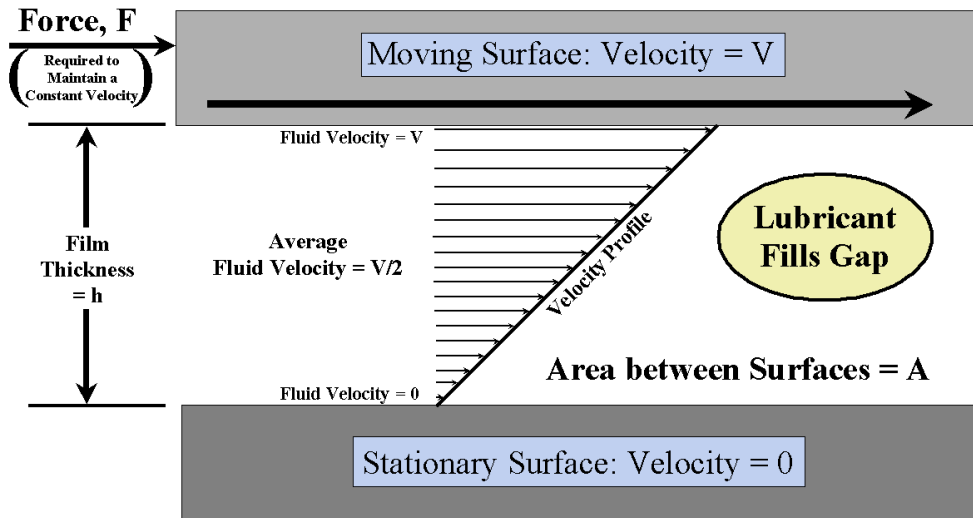


Figure 14 - Stribeck Curve Relating Friction Factor to Viscosity, Speed, and Load

To define what viscosity is mathematically, we must go back to the early history of lubrication theory. As a shaft turns in a journal bearing the moving surface shears the oil film between it and the stationary bearing surface. This shearing action creates a stress field in the lubricant film in PSI. The relative velocity between the surfaces defines the shear rate. The dynamic viscosity is defined as the ratio between the shear stress and the shear rate. Higher viscosities increase the shear stress while lower viscosities increase the shear rate due to a smaller minimum film thickness. Figure 15 shows this diagrammatically with a full unit derivation in English and SI units.



$$\text{Shear Stress} = \frac{F}{A} = \frac{LB_f}{IN^2} \quad \text{Shear Rate} = \frac{V}{h} = \frac{IN/SEC}{IN} = \frac{1}{SEC}$$

$$\text{Dynamic Viscosity} = \frac{\text{Shear Stress}}{\text{Shear Rate}} = \frac{F h}{V A} = \frac{\frac{LB_f}{IN^2}}{\frac{1}{SEC}} = \frac{LB_f \cdot SEC}{IN^2} = 1 \text{ Reyn}$$

Or in SI units:

$$\text{Shear Stress} = \frac{F}{A} = \frac{\text{Dynes}}{CM^2} \quad \text{Shear Rate} = \frac{V}{h} = \frac{CM/SEC}{CM} = \frac{1}{SEC}$$

$$\text{Dynamic Viscosity} = \frac{\text{Shear Stress}}{\text{Shear Rate}} = \frac{F h}{V A} = \frac{\frac{\text{Dynes}}{CM^2}}{\frac{1}{SEC}} = \frac{\text{Dynes} \cdot SEC}{CM^2} = 1 \text{ Poise}$$

$$\text{Kinematic Viscosity} = \text{Stoke} = \frac{\text{Dynamic Viscosity}}{\text{Density}} = \frac{\frac{\text{Dynes} \cdot SEC}{CM^2}}{\frac{\text{Grams}}{CM^3}} = \frac{\text{Dynes} \cdot SEC \cdot CM}{\text{Gram}} = \frac{CM^2}{SEC}$$

Figure 15 - Defining Dynamic Viscosity - English and Metric Units

The Reyn is named after Osbourne Reynolds famous for the “Reynolds number” that defines laminar and turbulent flows. However, a Reyn is much too large a unit and typical lubricants are in the micro Reyns region. Likewise, Poise is named after Jean Louis Marie Poiseuille a French physician who worked on fluid flow in the early 1800's. This unit is commonly used as centiPoise or cP which is 0.1 Pascal-second. The Stoke, after George Stokes, is a Poise divided by the fluid density. Usually used as centiStokes, or cSt. Water at room temperature has a viscosity of about 1 cSt.

Usually we want to minimize friction so we want to be somewhere in the center region of the Stribeck curve. Since speed and load are often fixed for a given machine, the bearing designer is left with choosing an appropriate lubricant viscosity. Other factors that affect this decision include the supply temperature and pressure, heat generation and removal, corrosion resistance and material compatibility.

Oil formulation is a complicated science and includes many additives to the base oils. These additives include high temperature breakdown resistance, extreme pressure, anti-foam, anti-sludge, corrosion resistance, anti-oxidation, and others. Often the formulations are proprietary depending on the supplier. Sometimes you have to tell your supplier what conditions and materials the oil will contact and ask them if their oil will react or degrade in such an environment. In some systems with high surface speeds, additive components have been known to centrifuge out of the oil and collect in areas like the inside of gear-type couplings. Here these materials build up and can restrict motion leading to increased vibration and wear.

JOURNAL BEARING MATERIALS

Since the bearing is usually much less costly than the shaft, it is considered sacrificial if necessary. In addition, a low dry sliding friction coefficient between the shaft and bearing material is important for startup and shutdown conditions.

Virtually every conceivable engineering material has been tried as a bearing material. Early man used wood or even stone. Later iron, copper and leather were applied to relatively low speed shafts. In the 17th century Robert Hooke advocated steel shafts with “bell metal”, essentially bronze, bushes to replace the practice of using wood blocks on cast iron. There has been extensive research on the best bearing material to use.

In 1839 Isaac Babbitt patented several high tin and high lead alloys which are similar to modern formulations. Of course, his name is commonly used to describe many different bearing materials. Carbon, graphite, ceramics, plastics and composite materials are all used today in some applications. Lead alloys have been virtually eliminated in modern applications due to lower strength and environmental considerations. The most common bearing lining material used in turbomachinery today is high tin babbitt. The formulation and characteristics of the two most common babbitt alloys are given in table 3. There are other alloys with specialized properties and often proprietary composition.

Parameter	Type 2	Type 3
Tin	89%	84%
Antimony	7.5%	8%
Copper	3.5%	8%
Liquidus Point	669 °F	792 °F
Brinell Hardness	24	27
Tensile Strength	12,600 PSI @ 68°F	10,000 PSI @ 68°F
% Strength at 212°F	52	52

Table 3 - Babbitt Material Characteristics

These formulations are applied to a base material, typically steel, by either molding in a stationary fixture or spin casting which is a messy and exciting thing to experience. Spin casting is supposed to create a better bond with the substrate but it has been determined that impurities in the molten babbitt can migrate to the babbitt-base metal bond line during this process and weaken that bond. While babbitt won't melt until the temperature is very high, it has lost almost half of its room temperature tensile strength at 212°F.

BEARING FAILURE MECHANISMS

There are many reasons a journal bearing might fail. A common mechanism is loss of lubricant and is not so much a bearing failure as a system failure. The next most common mechanisms is fatigue damage and one of the most important considerations for the material lining bearings is fatigue resistance. Thin babbitt has significantly more fatigue resistance than babbitt thickness greater than 15 mils as seen in figure 16. This is data from an extreme test conducted with about ten times the normal static load to speed up the test results.

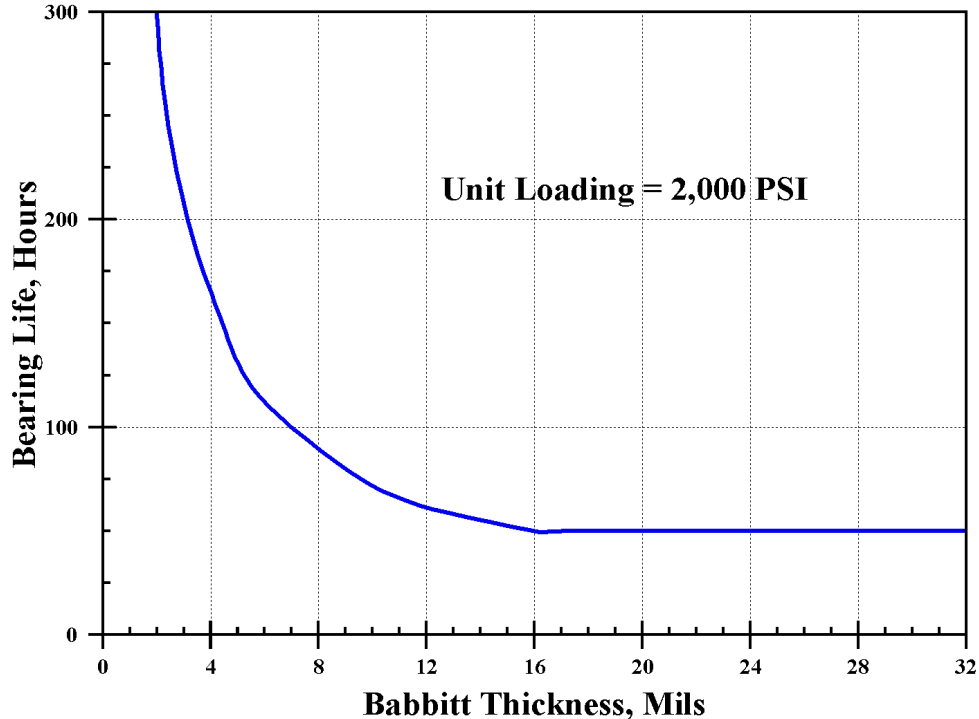


Figure 16 - Bearing Fatigue Life as Babbitt Thickness Varies

Rotor imbalance and other conditions cause the journal to orbit in the bearing. This causes an oscillatory dynamic pressure to act on the babbitt surface. It is not uncommon for the *peak* static hydrodynamic pressure to approach 3 to 5 times the specific W/LD load. Oscillatory shaft motion adds an alternating pressure on top of this that impinges on the babbitt surface. Babbitt fatigues in a manner similar to the way potholes develop in a road surface. Surface cracks develop which propagate to the bond line below as shown in figure 17. The piece of babbitt detaches but is unable to leave the area because of the close clearance to the shaft. This piece of babbitt rattles around the pit, breaking into smaller bits which are carried away by the oil film. This action smooths the sides of the pit. Finally all the pieces of loose babbitt are gone leaving a rounded smooth hole. Figures 18 and 19 are micrographs of this process and figures 20 and 21 are photographs of a failing bearing. This type of damage is often mistaken as cavitation or erosion, which it is not. Virtually all journal bearings cavitate. Typically oil contains dissolved gasses and in the divergent unloaded area of the oil film it is normal for streamers to form. As the pressure wedge re-forms, the gasses dissolve back into the oil. This is a gradual process and no damage to the bearing occurs. There are cases where rapid rupture of the oil film is normal such as with high speed reciprocating machinery. Here vapor bubbles can form and collapse explosively. These impacts may locally fatigue the bearing surface. Entrained free air will aggravate this problem. The most common failure mechanisms are a result of operating conditions outside of the intended design. These include foreign material (dirt) or contamination (water, etc.) in the oil, overload, lube oil viscosity degradation, or, far too often, lubrication failure. Electrostatic and electromagnetic discharge across the oil film will erode the babbitt surface over time. Corrosion of the bearing babbitt is relatively rare especially if a regular oil analysis program is in effect. The steel backing can be attacked by water in the oil and copper alloys are more susceptible to corrosion from ammonia and other contaminants.

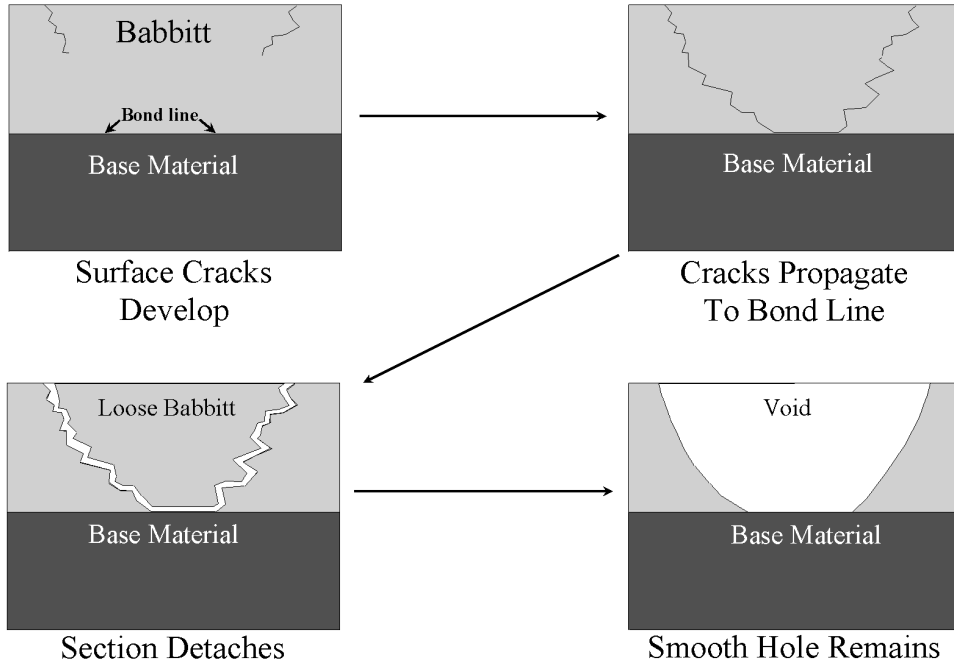


Figure 17 - Babbitt Failure due to Fatigue Pounding Damage

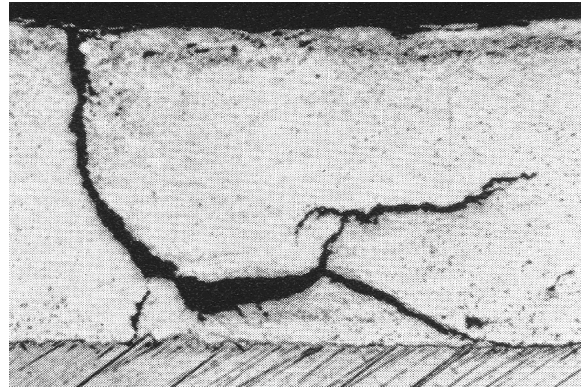
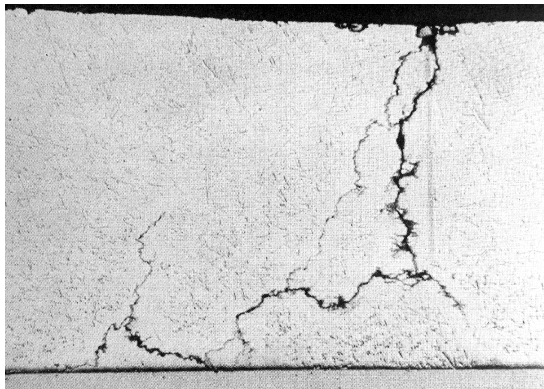
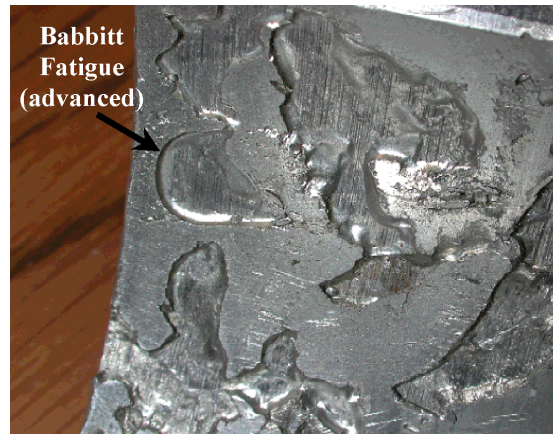
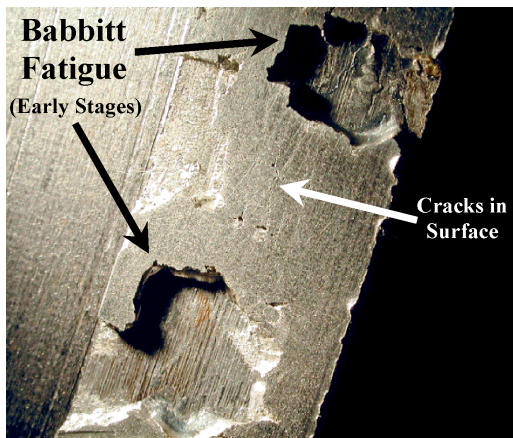


Figure 18 and 19 - Fatigue Cracks in Babbitt



Figures 20 and 21 - Early and Late Stage Babbitt Fatigue

JOURNAL BEARING DESIGN PARAMETERS

The design of a journal bearing for a particular application involves many considerations.

- Available Space

Many machinery manufacturers purchase bearings from specific bearing companies. It is advantageous to use standard bearing form factors for cost reduction. A machine that was designed for thin shell bearing liners is not usually a good candidate for conversion to tilting pad bearings simply because there is insufficient radial room in the bearing housing. Sometimes new housings can be made, but this involves additional cost.

- Friction and Heat Generation

The basic equation that bearing designers use to evaluate friction and heat generation is Petroff's equation (9) which assumes a centered shaft in a plain bushing. Notice that journal load is not part of this calculation. Only the oil shear forces are used. The viscous shear force in this case is:

$$F = \mu \frac{\pi^2 D^2 L N}{30 C}$$

where μ = viscosity in Reyns

$$\pi = 3.14159\dots$$

D = Diameter (Inches)

L = Axial Length (Inches)

N = RPM

C = Diametral Clearance (Inches)

and the Horsepower Loss is:

$$Hp = 2.61 \times 10^{-6} \mu \frac{D^3 L N^2}{C}$$

from this equation, an estimate of the temperature rise for a given oil flow rate is made using either empirical equations available in some of the references (10) or various computer programs. It is very important to understand the assumptions used in the empirical equations, particularly the heat capacity of the lubricant. This may vary significantly if using synthetic lubricants. As radial load on a bearing increases, the Petroff equation is not sufficient to predict the temperature rise and relatively complex computer programs are used. Getting sufficient oil into the bearing and subsequently draining it out are very detailed topics not covered here.

- Specific Steady Load

A very important concept is the specific load, P , is defined as:

$$P = \frac{W}{L D}$$

or simply the journal load divided by the active bearing length times diameter. In English unit this is in PSI. The loading classification ranges (by this author) for this variable are:

0 to 50 PSI - Very Light
50 to 100 PSI - Light
100 to 200 PSI - Moderate
200 to 300 PSI - Heavy
more than 300 PSI - Special Design

There are problems associated with bearings that are too lightly loaded, specifically oil whirl and instability. For heavily loaded bearings heat generation and babbitt fatigue can become significant. One should always consider both the

minimum film thickness and the peak hydrodynamic load when evaluating a design. Special evaluation is required if the peak pressure exceeds 1,000 PSI or the minimum film thickness is less than 1.0 mils.

- Surface speed

The larger the journal diameter, the higher the surface speed for a given RPM. To calculate feet-per-minute (FPM) from RPM the relationship is: (use inches for D)

$$FPM = RPM \frac{\pi D}{12}$$

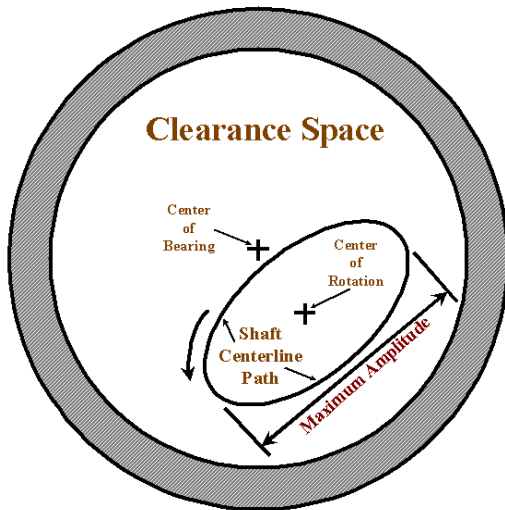
some typical examples are:

600 RPM with 8 Inch Journal = 1,257 FPM
3,600 RPM with 4 Inch Journal = 3,770 FPM
12,000 RPM with 6 Inch Journal = 18,850 FPM

At some point, which depends mainly on the lubricant and clearance, the oil film becomes turbulent. Turbulence increases the friction, reduces oil flow and generates more heat. For typical turbomachinery, surface speeds above 15,000 FPM should receive special scrutiny.

- Dynamic Load

Dynamic load is the result of shaft orbital motion in the oil film clearance space due to imbalance, misalignment, and other non-static forces. Thus, an alternating hydrodynamic pressure fluctuation is superimposed on the steady pressure being exerted on the babbitt. It is this alternating force which fatigues the babbitt. The easiest way to give guidelines for this is to limit maximum orbit vibration to less than 50% of the diametral clearance of the bearing. For example, if the bearing clearance is 6 mils, then the maximum acceptable displacement amplitude is 3 mils peak-to-peak measured on the orbit from two orthogonal proximity probes. Once the orbit exceeds 75% of the clearance, short-term damage is almost always happening and bearing life is likely limited to a few hours if the orbit size exceeds 90% of the available clearance. This author defines a dynamic load of 300 PSI (on top of the static load) as the upper limit of acceptability.



1. **Must measure with 2 Proximity Probes**
2. **Must Know Diametral Clearance**
3. **Maximum Peak-to-Peak Orbit Amplitude**
Normal - Less than 25% of Clearance
Alert - 50% of Clearance
Danger - 75% of Clearance
>90% - Failure is Imminent
4. **Fluctuating Hydrodynamic Load is Superimposed on Steady Load and may cause Fatigue of Babbitt**

Figure 22 - Dynamic Load - Vibration and Alternating Hydrodynamic Load

- L/D Ratio

The length to diameter ratio is one of the first things a bearing designer considers. Since the shaft diameter is often determined by other factors (torque and bending strength, for example) it is usually the active babbit axial length that is controlled. This factor is “tuned” to give sufficient steady and alternating load capacity. However, it also significantly affects the stiffness and damping characteristics of the bearing. The longer the bearing, the lower the specific load and the lower the stiffness. A longer bearing also tends to have higher effective damping. Since damping is a frequency dependent stiffness, the speed of the machine must be considered as well. While, from a rotor dynamics standpoint, softer bearings are almost always a benefit, a too lightly loaded bearing may have stability problems. In the experience of this author, an L/D ratio less than 0.3 has poor damping and an L/D ratio greater than 0.75 usually shows little gain in effective damping. Heavily loaded bearings such as gears often exceed an L/D ratio of 1.0 and may need special care in aligning the bores to the shafting.

- Clearance

The basic guideline universally used for diametral journal bearing clearance is 1.5 mils-per-inch of journal diameter. That is, a 4 inch diameter shaft would need about 6 mils of diametral clearance. Always check if the specifications you see are for diametral clearance. Some manufacturers specify *radial clearance* - which is really difficult to measure! If the application has atypical loads and/or speeds, then this clearance rule may need to be adjusted. The bearing should be held in the housing with a zero to 1 mil interference fit. A loose bearing will have vibration problems and too much housing interference can reduce the clearance in the bearing. The issue of tolerances should also be considered. Clearance values are usually given as a range. The limits suggested here for minimum allowable clearance is 1.0 mil-per-inch shaft diameter plus one mil. The maximum allowable is 2.0 mils-per-inch. Remember tolerance costs money.

- Base Material

Normally steel is used as the backing material for babbit bearings and is preferred for strength. Cuprous alloys are about half the strength of steel and have virtually no endurance limit. Bronze or even copper may be used where extra heat conduction is needed for a successful design. Caution should be used with new machinery. API 612 (Special Purpose Turbines) and API 617 (Centrifugal Compressors) both specifically disallow anything other than steel for a backing material. The reason for this is that it allows a relatively inexpensive and convenient path to fix a hot bearing or upgrade a design. If the original design includes copper pads, you are limited on the redesign.

- Grooving

On some bearings grooves are cut into the surface of the babbit for a variety of reasons. The most often cited reasons for grooving are to direct lubricant to the loaded areas and for cooling. A well-designed bearing with steady loads (not reciprocating) does not need any grooves and, in fact, the sole effect is to increase stiffness, reduce load capacity, and damping. In some cases a circumferential groove is used (called a *load* groove) to increase the specific loading and increase stability.

JOURNAL BEARING DESIGN

The simplest form of journal bearing is a plain circular bushing with an ID that is slightly larger than the shaft OD. Early bearings were often made by pouring the babbit or white metal directly into a sleeve containing the actual shaft or a same-size mandrel. Then, the surface of the babbit was scraped to provide the desired clearance. This success of this procedure was highly dependent upon the skill of the mechanic. Scraping of bearings is strictly forbidden in most plants today and is not recommended. Many different bearing profile designs have been invented in the last century or so, but the basic geometries are primarily constrained by cost and ease of manufacturing. Special circumstances may require unique solutions.

Preload

Besides clearance the next most important factor of a bearing is preload. A preloaded bearing simply means that the center of curvature of the pad is not coincident with the geometric center of the bearing. Preload can be thought of as the percentage of clearance reduction. No preload means, with a centered shaft, even clearance all the way around and 50% preload would reduce the clearance by half at the closest point. Typical industrial bearings have preloads in the zero to 50% range. Preload is easily and cheaply added to a plain circular bearing. It is good to avoid getting too close to zero preload so that tolerances will not cause you to end up with negative preload. Negative preload pinches the inlet and outlet of the pads causing a scraper effect that will starve the bearing of lubrication.

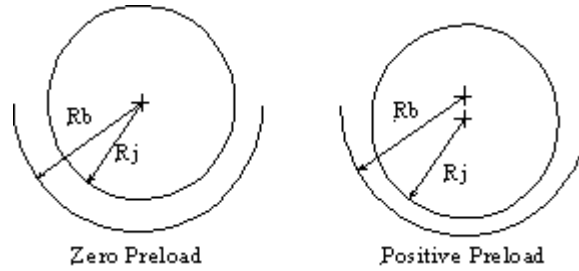


Figure 23 - Pad Preload Definition

Offset

The other geometric parameter that is controlled is offset. When the closest clearance is half-way along the arc of a pad, this is the most common and is called 50% offset. Offset can be increased past 50% but offsets less than 50% are not normal and not recommended. Offset is easily manufactured and difficult to detect. An offset condition can be very useful both in controlling the stiffness and damping characteristics as well as reducing operating temperatures. The increase in offset opens the inlet side of the bearing pad admitting more fresh oil. Bearings with offset greater than 50% should not be run backwards since this essentially creates an offset less than 50%. In practice, offset bearings always include positive preload as well.

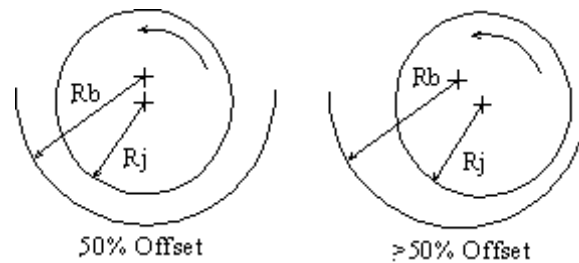


Figure 24 - Pad Offset Definition

JOURNAL BEARING TYPES

As mentioned above, the simplest bearing geometry is a circular bushing. It turns out that this is adequate for many machines and a poor bearing for many other applications. Some of the other common geometries of bearings are:

● **Partial Arc Bearing**

This is a bearing without a top half. Used with known unidirectional loads and to save energy. Has reduced damping and less ability to withstand vibration than a whole bearing. This type is not normally a good choice. Sometimes these are used to minimize friction and most often found in electric motors.

● **Axial Groove Bearing**

In its simplest form this is a plain circular bush with 2 or more axial grooves for oil distribution. Typically the 4-pad design is found but higher number of pads may be used. May be used when the load angle changes as operational variables change. This design has slightly better stability than a plain bearing without grooves.

● **Elliptical (or lemon bore) bearing**

Elliptical bearings are very common. In some applications bearing “crush” is specified which changes a circular bush into an elliptical arrangement. This bearing has better dynamic characteristics over a plain bearing and is more stable. Usually the horizontal clearance is 1.5 times to twice the vertical clearance (33 to 50% preload). The angle of the split between halves can be re-oriented for the best load capacity and damping. Dynamically this is a highly asymmetric bearing. The stiffness in the direction of the greater clearance is often an order of magnitude less than across the tight clearance direction. Damping is also significantly less in the direction of the maximum clearance.

●**Multi-Lobe Bearings**

Multi-lobe can refer to any number of lobes with the designer taking advantage of the ability to control the geometry of each pad. This usually means adjusting the preload and offset for each pad. The load angle can also be selected for optimum conditions. The three-lobe bearing is the most popular configuration but can be expensive to manufacture. Figure 25 shows a typical 3-lobe bearing and figure 26 shows the hydrodynamic pressure profile created by the individual lobes. The bearing designer can select the clearance, preload, and offset of each pad as well as the load orientation on-pad, between pad or somewhere in between. The optimum configuration will depend on the rotordynamics. Multi-lobe bearings like these are very useful in vertical machines.

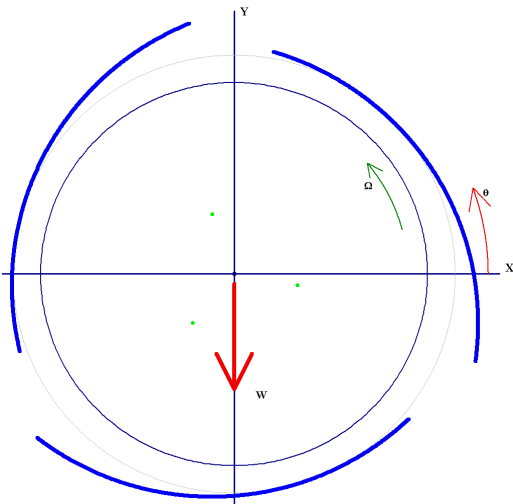


Figure 25 - Multi-Lobe Bearing

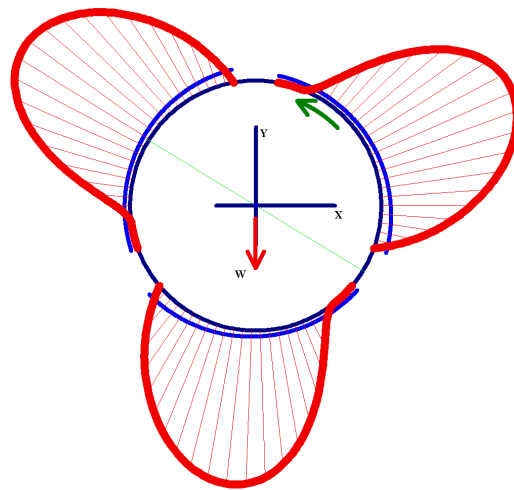


Figure 26 - 3-Lobe Bearing Pressure Profile

●**Offset half bearing**

This is a very simple and very effective bearing in some applications. It is a 2-pad multi-lobe with 100% offset. The pads are almost always preloaded as well. It is easy to make by boring a plain bushing off-center and the flipping the top half around. This bearing will not tolerate reverse rotation.

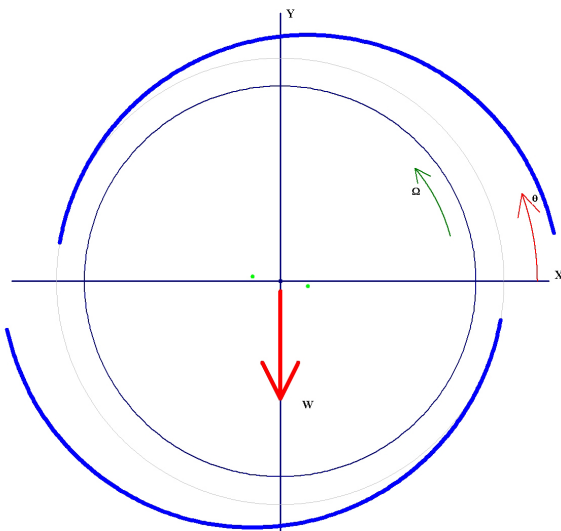


Figure 27 - Offset Half Bearing

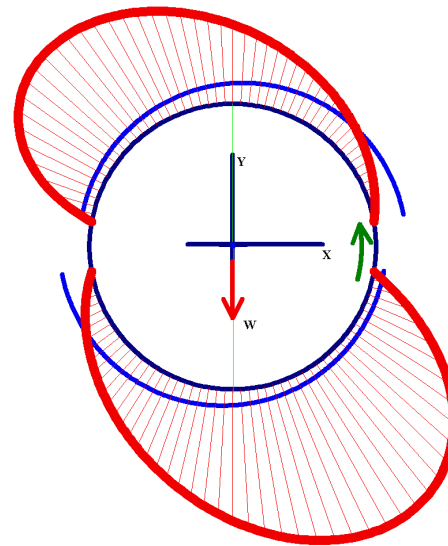


Figure 28 - Offset Half Bearing Hydrodynamic Pressure Profile

● Pressure Dam Bearing

Pressure dam bearings, sometimes called pocket or step bearings, are plain bearings that may (rarely) have preload and even offset. Typically they are two-lobe bearings with a pocket cut out of the top half (or unloaded half) that creates an oil dam. The circumferential velocity head of the oil suddenly being impeded by the dam is converted to pressure head. This pressure head imposes a downward load on the shaft that will increase the eccentricity and may stabilize the system. This design also significantly increases the bearing stiffness and may cause the critical speed amplitude and amplification factor to increase. This is often an excellent first design modification if oil whirl is encountered with plain bearings. The height of the dam should be about the same as the diametral clearance of the bearing and up to 3 times that clearance. A deeper pocket will have significantly reduced effect. If this bearing wears and clearance increases, the effectiveness of the dam will be diminished. Some designs now use multiple pockets, particularly in vertical machines where there is no gravity load. Computer programs are necessary to evaluate such cases. As always, a system design approach will be necessary to evaluate whether this bearing is appropriate. This bearing is also unidirectional in rotation.

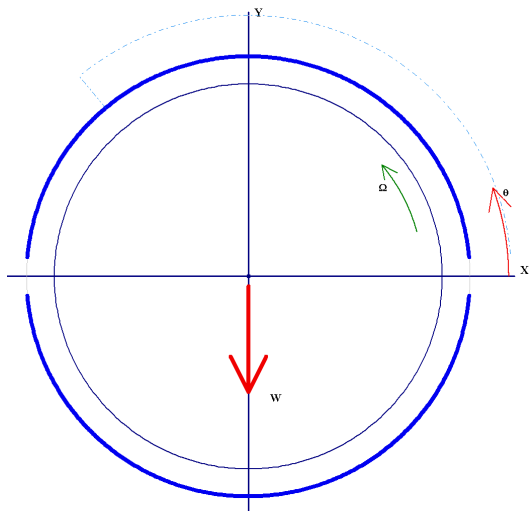


Figure 29 - Pressure Dam Bearing

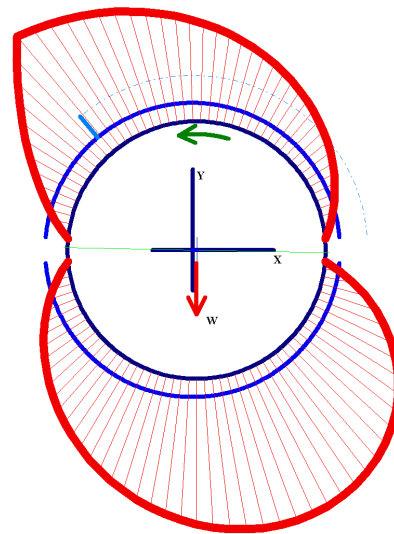


Figure 30 - Pressure Dam Bearing Hydrodynamic Pressure Profile

● Tilting Pad Bearings

Tilting pad bearings are becoming the design of the turbomachinery industry. They are very robust, able to be designed to optimize the rotor dynamics and widely accepted as the very best bearing you can get. Tilting pad bearings also have no significant destabilizing cross-coupling. They can be costly and may not always be the best choice when all factors are considered. Figure 31 shows a typical modern tilting pad bearing. Figure 32 is a typical hydrodynamic pressure profile for two tilting pad bearings with different load orientations. Besides clearance, preload and offset, the bearing designer can choose the number of pads, different pivot arrangements, different types of end seals and load orientation. By choosing a 5 pad bearing with load directly over one pad the horizontal and vertical stiffness will be significantly different than a 4-pad design with load between pivots which will have symmetrical characteristics. Figure 33 shows this comparison. Note that K_{xx} is shorthand for the horizontal stiffness and K_{yy} means vertical stiffness. Thus, for a particular machine, the various factors may be manipulated to achieve the frequency of critical speeds, the lowest critical speed amplitude response, the lowest operating speed vibration, and the best rotor stability. Figure 33 indicates that the stiffness of the 5-Pad bearing with load between pivots will have a horizontal stiffness less than 1,000,000 LB/IN and a vertical stiffness in excess of 2,500,000 LB/IN. However, if a 4-Pad bearing is selected with load between pivots, the horizontal and vertical stiffnesses are equal at around 2,000,000 LB/IN throughout the speed range.

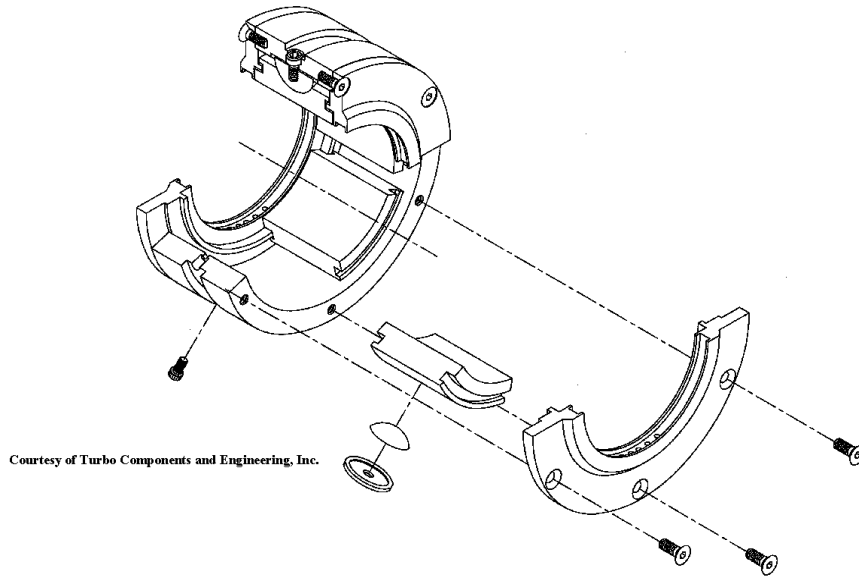


Figure 31 - Exploded View of Tilting Pad Bearing

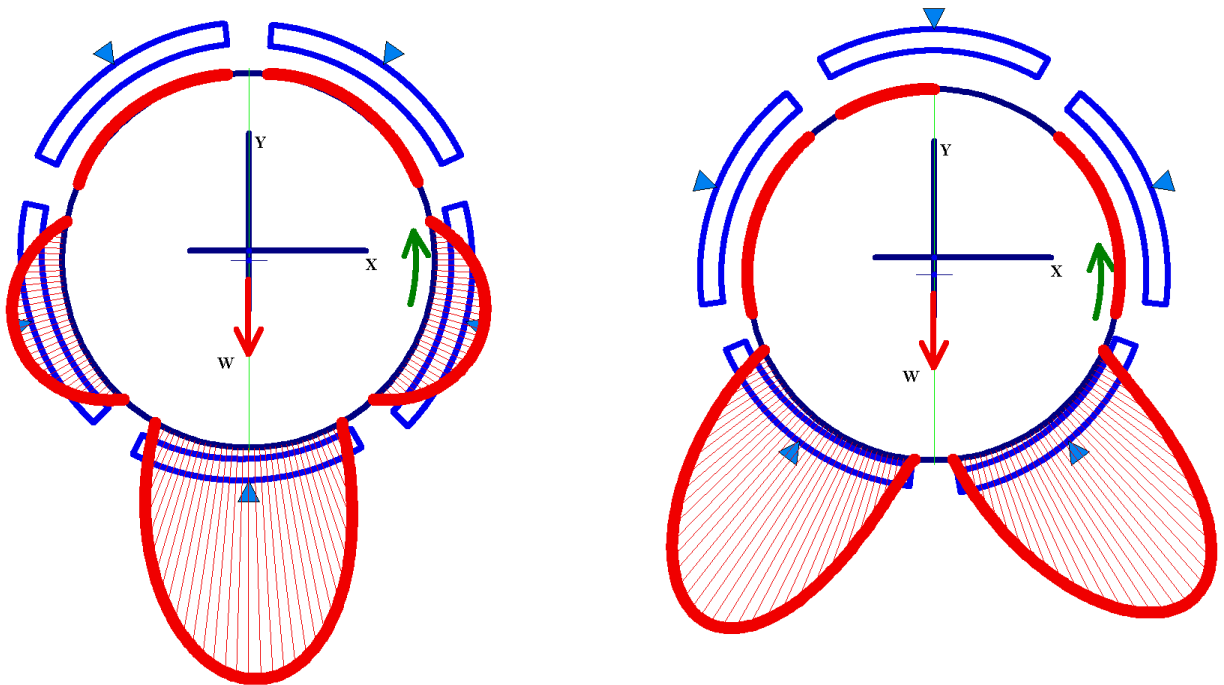


Figure 32 - Hydrodynamic Pressure Profile for Two Tilting Pad Bearings

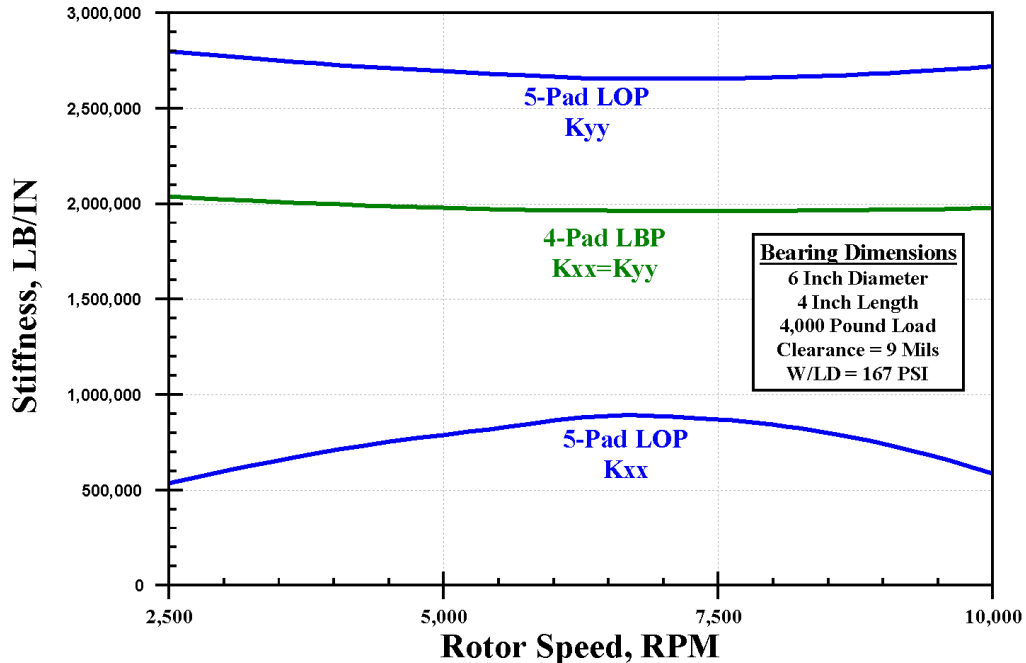


Figure 33 - Using Load Orientation and Number of Pads to Fine-Tune Bearing Stiffness

● Specialty Bearings

There are many other types of bearings. Even today “new” bearing advances are announced with some regularity. Many of these are variations of concepts that are quite old. Specialty bearings fill niche markets where certain dynamic characteristics are required or space is limited, etc. A few of the more common types are:

Hydrostatic Bearing - This bearing uses high pressure oil supplied to a number of ports in the bearing. This creates a very stiff oil film and is often used in high precision spindles and high speed turboexpanders. By varying the supply pressure, active control of the bearing dynamics has been demonstrated. A complex hydraulic system is required as well as a sealed environment to control leakage.

Flexible Pivot Bearing

This is a fixed pad bearing which has a flexible beam attachment of the pad to the shell instead of a separate pivot like the tilting pad bearing. It has less cross coupling than a fixed pad bearing but more than a tilting pad bearing. Useful where limited radial space precludes the use of a tilting pad bearing.

Hydraulic Lift Bearing

This is usually a plain bearing with high pressure oil injected at the bottom to “float” the shaft before rotation begins. Used for very heavy shafting like large generators to avoid damaging the babbitt during slow-roll, startups and shutdowns. This design also makes turning the shaft for alignment much easier. Sometimes requires maintaining high pressure during the run or load capacity can be severely reduced.

Hybrid Bearings

Combining several different designs into one bearing can create unique bearings for solving difficult rotordynamics problems. For example, the author once combined a 3-lobe bearing with two pressure dams that successfully stabilized a machine that could not be fitted with tilting pad bearings.

Comparing Bearing Performance - Critical Speeds

The first critical speed of a machine must often be passed on the way up to operating speed. Thus, the rotor amplitudes and amplification factor are important especially at those locations where the highest amplitudes occur and rubs are most likely. Figure 34 is symbolic of a rotor that would be sensitive to rubs at the first critical speed in the center of the rotor.

Fictional 9-Stage Compressor Rotor

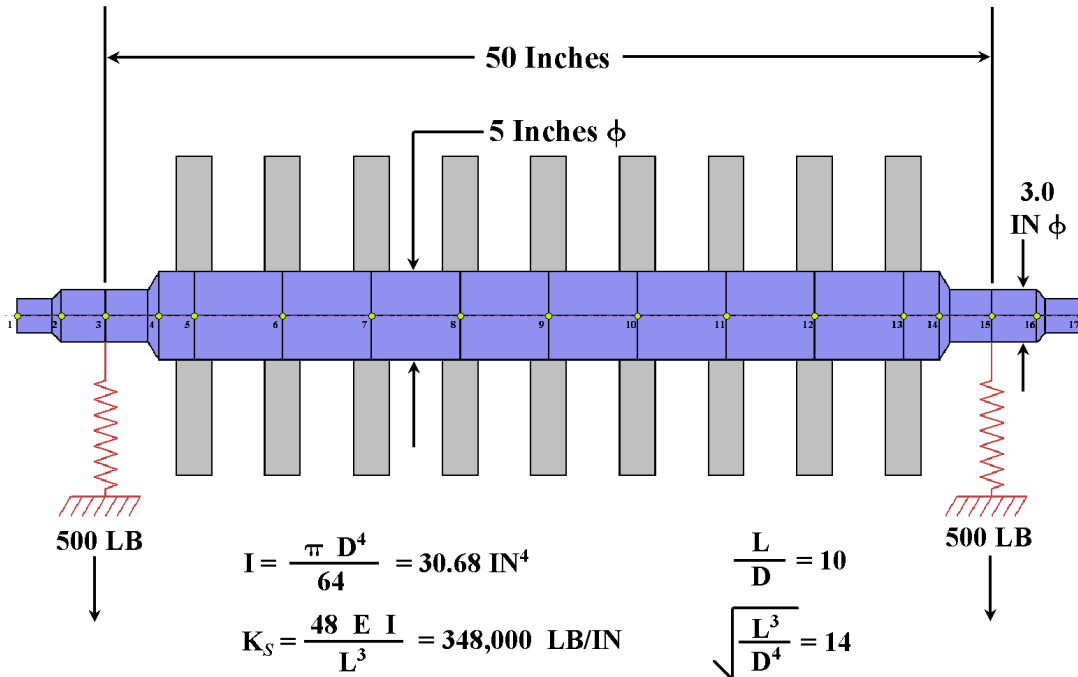


Figure 34 - Compressor Rotor Used for Bearing Comparison

To illustrate how different bearings would perform in such a machine, each of the common fixed-profile bearing types was analyzed in the vicinity of the first critical speed and the results are plotted in figure 35. In all cases the bearing L/D ratio, clearance, and load were held constant. The plain, elliptical, 3-lobe and offset half bearings showed similar amplitudes while the much stiffer pressure dam bearing did not do well at all. Table 3 is a summary of the results for the fixed profile bearings.

Bearing Type	Critical Speed, RPM	Maximum Amplitude, Mils P-P	Amplification Factor
Plain Journal	4,200	3.3	11.6
Elliptical	4,100	2.8	9.2
Offset-Half	4,075	2.9	9.2
3-Lobe	4,000	3.2	11.4
Pressure Dam	4,250	9.0	30.0

Table 4 - Fixed Profile Bearing Critical Speed Performance

With a rotor as flexible as the one in this example, changing bearing type results in a change in the critical speed frequency of only about ± 3 percent.

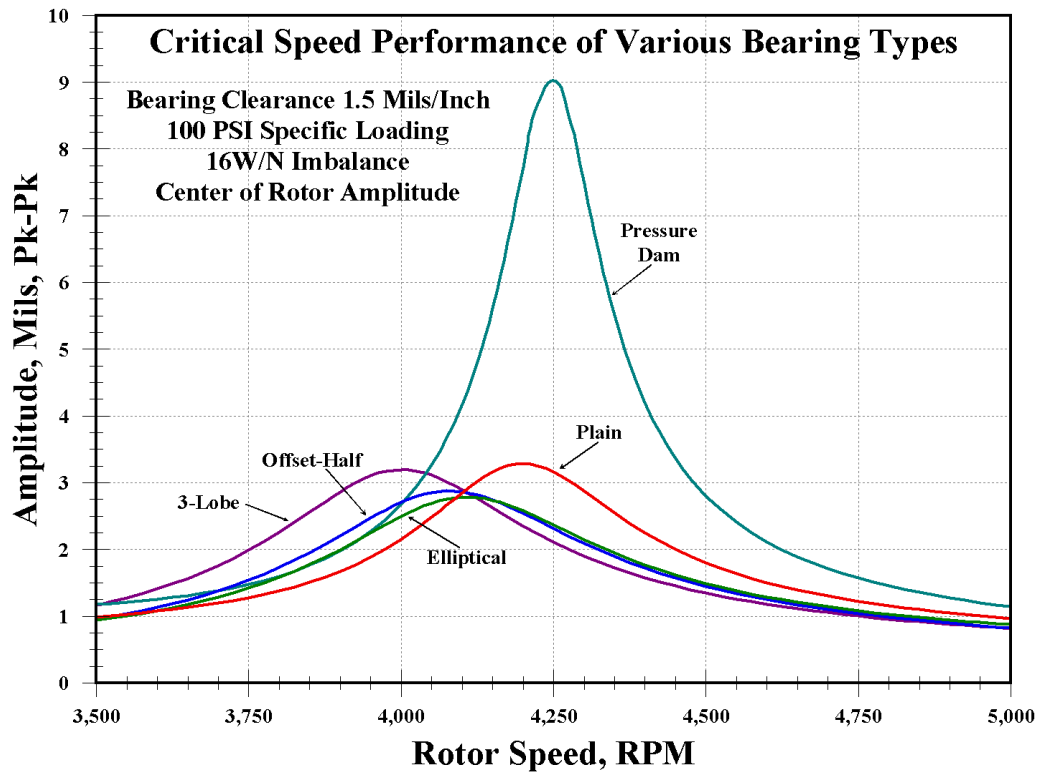


Figure 35 - Critical Speed Performance for Fixed-Profile Bearings

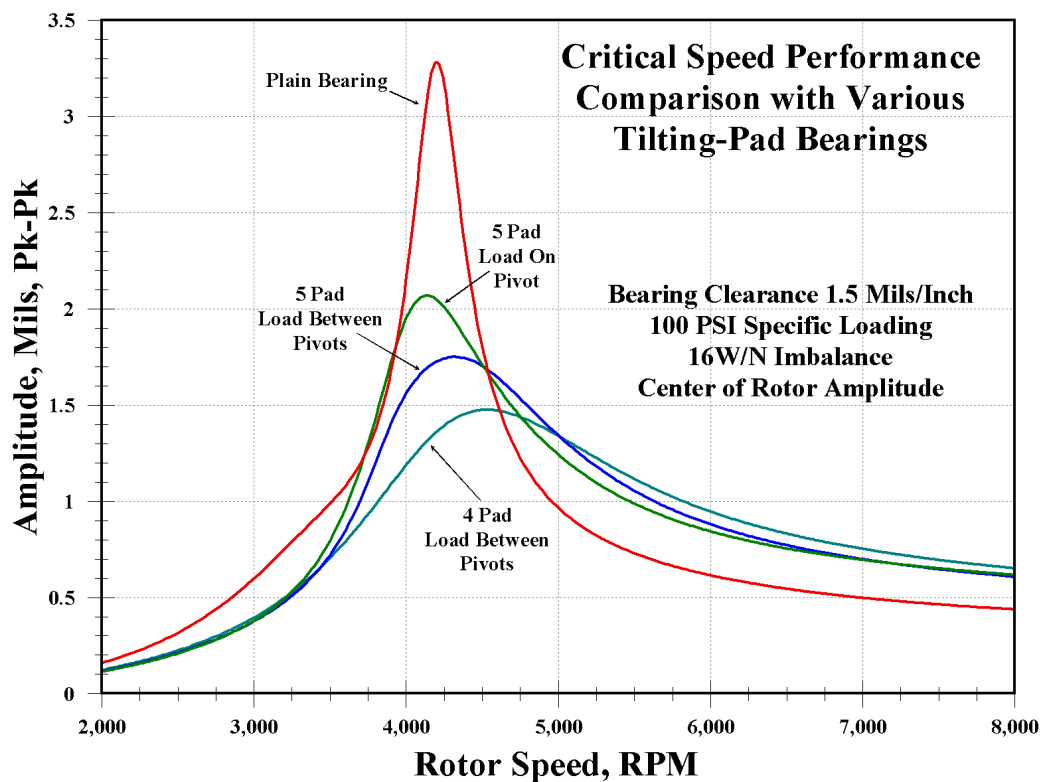


Figure 36 - Critical Speed Performance for Tilting Pad Bearings

Figure 36 compares the plain bearing critical speed performance with three variations in tilting pad bearings. As before the clearance and L/D ratio was fixed. Each tilting pad bearing had a nominal preload of 20 percent. The plain bearing response is included for comparison. Any of the tilting pad designs is significantly better than the plain bearing or any of the other fixed profile bearings. Table 5 is the summary of the results. The 4-Pad design actually has an amplification factor below the limit where API considers this response peak to be a critical speed.

Bearing Type	Critical Speed, RPM	Maximum Amplitude, Mils P-P	Amplification Factor
Plain Journal	4,200	3.3	11.6
5-Pad, LOP	4,125	2.1	4.5
5-Pad, LBP	4,325	1.8	3.2
4-Pad, LBP	4,525	1.5	2.4

Table 5 - Tilting Pad Bearing Critical Speed Performance

Rotor-Bearing Stability

It is important to discuss rotor dynamic stability in conjunction with journal bearings. Oil whirl and oil whip are relatively common phenomena. See references 13 through 20 for many useful stability articles. Instability usually means high levels of subsynchronous vibration. Often a tilting pad bearing is fitted to solve the problem but this may not always work. Tilting pad bearings do not contribute to the instability but a rotor may still become unstable if the available damping is not sufficient to counter the destabilizing forces generated by seals and aerodynamic effects. Any bearing change in a machine should be thoroughly evaluated for imbalance response, stability, temperature, and maintenance considerations. There are now API standard paragraphs in RP684 for stability of turbomachinery. If we assume the operating speed of our fictional compressor is 9,000 RPM and calculate the stability with each type of bearing discussed, the results in table 6 indicate that a tilting pad bearing is going to be required. With some tweaking the offset half and 3-lobe designs could possibly be made stable but only just and nowhere near the stability afforded by the tilting pad designs all of which are very stable and could handle significant cross coupling. Note that all of the tilting pad bearings listed here have their pivots in the center of each pad. If you were to introduce pivot offset as is sometimes done for bearing temperature control, the stability would decrease significantly because pivot offset significantly increases the bearing stiffness and reduces effective damping. All aspects of a machine's rotordynamics should be considered when selecting the bearings.

Bearing Type	Logarithmic Decrement
Plain Journal	-0.38
Elliptical	-0.13
Offset-Half	-0.03
3-Lobe	-0.07
Pressure Dam	-0.21
5-Pad, LOP	1.12
5-Pad, LBP	1.05
4-Pad, LBP	1.04

Table 6 - Stability Comparison for Different Types of Journal Bearings

FLUID FILM BEARING DO'S and DON'TS

DO:

1. Always use the proper lubricant as determined by the manufacturer or engineering.
2. Understand the additive package in your lubricants to avoid potential conflicts with process fluids and/or component materials.
3. Provide proper cooling to the bearing and the lubricant.
4. If applicable, provide proper filtration to the lubricant.
5. Implement a regular oil analysis program for all critical machinery. Monitor lubricants for viscosity changes, wear metals and contamination, especially water.
6. Stay within design guidelines on clearances. General rule is 1.5 mils (0.0015 inches) per inch of shaft journal diameter. 3.0 mils/inch diameter is excessive clearance in most cases.
7. Handle bearings carefully. Babbitt surfaces are very soft and thin liners are easily distorted.
8. Inspect bearings under magnification or have a professional evaluation before reuse. Early fatigue damage is usually invisible and other damage like electrostatic discharge may not be readily apparent. Replace bearings if any doubt exists as to the serviceability of the used bearing. Tilting pad bearings may have back-of-pad and pivot wear and brinelling concerns.
9. Lubricate bearings during replacement with a heavy prelude oil such as an ISO 460 especially if a lot of shaft rotation will occur as during alignment.
10. Monitor bearing temperatures, preferably metal temperature with embedded thermocouples or resistance temperature detectors (RTD). Calibrate transducers before installation in the bearing. Install dual sensors in case one fails - leave the spare unconnected. Drain temperatures, while useful, will not give an early enough indication of heat problems.
11. Carefully protect the temperature device leads during handling and installation.
12. If a bearing is spherically mounted, ensure line-to-line contact or a slight crush. These bearings ***do not self align!*** The user must manually align these bearings.
13. Inspect all bearings and inspect the shops making or repairing your bearings. Many rebabbiting shops are not qualified to repair all types of bearings. Ask where they get their babbitt and what quality controls are used. Is babbitt welding and babbitt casting done?
14. Use common sense. Treat the bearing with respect and you will get good service. Follow the rules specified by API and the guidelines proposed in this

article.

15. Fix damaged shaft journals with submerged arc welding or plasma spray - NOT chrome plating.
16. If you find any lead babbitt bearing material replace it with type 2 high tin babbitt which is much stronger and environmentally friendly.

DON'T:

1. Never use unapproved substitute lubricants.
2. Never mix different lubricants unless the combination has been evaluated and approved - different additive packages may react. Never mix mineral oils and synthetic lubricants. Using synthetic oil in a system previously using mineral oil may loosen old deposits.
3. Never bypass oil filters or coolers.
4. Installing bearings too tight or too loose is a recipe for disaster. Never set clearances in a tilting pad bearing in the field - this must be done in a qualified shop.
5. Do not use automotive viscosity enhancers (e.g. STP) when fitting bearings.
6. Never hand scrape bearings for proper fit. Take the time to have bearing properly machined by a qualified shop. No pocket knives touching babbitt!
7. Never pry or hammer a bearing liner out or back into place.
8. Throwing a babbitt bearing into the parts bucket usually means you ruined it.
9. Never install temperature sensors in the babbitt, rather 0.030" behind in the steel backing. Those wires sticking out of the bearing are not used to carry the part!
10. Don't use the low bid to buy or repair bearings. Reusing babbitt from old jobs or overflow is forbidden. Tiny contaminations can lead to early bearing failure.
11. If there is a choice, don't use spherically seated bearings, use cylindrical fits.
12. Do not roll shafts in Teflon[®] strips or "V" blocks due to micro embedment. This could result in the shaft being unable to properly "wet" and causes failure.
13. Never let oil reservoir or sump temperature exceed 200°F.
14. Don't expect thick babbitt (e.g. 0.060") to be better than "thin" babbitt (<0.015") The fatigue resistance of thin babbitt can be more than 10 times greater than thick babbitt.
15. Don't use copper backed pads in the initial design unless there is no other option. This method of increasing capacity should be "held-back" in case additional capacity is needed in the future. It is easier than increasing bearing size.
16. Don't ever disassemble a machine without measuring the bearing clearances "as-found". This is the only chance you get to obtain this data which is an invaluable diagnostic tool.

17. NEVER coat a fluid film bearing journal with chrome. Use almost any other coating. Chrome is porous and water may get trapped behind the chrome and pop off the chrome layer causing catastrophic failure.

REFERENCES

1. *Machine Design, Part III* by International Textbook Company, 1907

2. *Bearings and Their Lubrication* by L. P. Alford, McGraw Hill, The American Machinist, 1911

3. *Lubrication of Industrial and Marine Machinery* by C. L. Pope and W. T. Everitt, John Wiley and Sons, 1954, LC number 53-9023

4. *Lubrication Fundamentals* by J. George Wills, Marcel Dekker, Inc., 1980, ISBN 0-8247-6976-7

5. *Bearings and Lubrication* by Joseph E. Shigley and Charles R. Mischke, a Mechanical Designer's Workbook by McGraw Hill, 1990, ISBN 0-07-056928-2

6. *Basic Lubrication Theory* by Alistair Cameron. John Wiley & Sons, 1976, ISBN 85312-057-9

7. *Machinery's Handbook, 25th Edition*, Industrial Press, Inc., 1997, ISBN 0-8311-2424-5

8. *Design of Film Bearings* by Paul Robert Trumpler. The Macmillan Co., 1966, LC number 66-11212

9. *Bearing Design and Application* by Donald F. Wilcock and E. Richard Booser, McGraw Hill, 1957, 195, LC number 56-9641

10. *Applied Tribology - Bearing Design and Lubrication* by Michael M. Khonsari and E. Richard Booser, John Wiley and Sons, Inc., 2001, ISBN 0-471-28302-9

11. Nicholas, J. C., "Hydrodynamic Journal Bearings - Types, Characteristics and Applications," Mini Course Notes, 20th Annual Meeting, 1996, The Vibration Institute, Willowbrook, Illinois, pp. 79-100.

12. Nicholas, J. C. and Allaire, P. E., 1980, "Analysis of Step Journal Bearings - Finite Length, Stability," ASLE Transactions, **23** (2), pp. 197-207.

13. Nicholas, J. C., 1986, "Stabilizing Turbomachinery with Pressure Dam Bearings," *Encyclopedia of Fluid Mechanics*, **2**, Gulf Publishing Co.

14. Mehta, N. P. and Singh, A., 1986, "Stability Analysis of Finite Offset-Halves Pressure Dam Bearings", ASME Journal of Tribology, **108** (2), pp. 270-274.

15. Lanes, R. F. and Flack, R. D., 1982, "Effects of Three-Lobe Bearing Geometries on Flexible Rotor Stability", ASLE Transactions, **25** (3), pp. 377-385.

16. Nicholas, J. C., 1985, "Stability, Load Capacity, Stiffness and Damping Advantages of the Double Pocket Journal Bearing," ASME Journal of Tribology, **107** (1), pp. 53-58.

17. Lund, J. W., 1974, "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid Film Bearings," ASME Journal of Engineering for Industry, **96** (2), pp. 509-517.

18. Newkirk, B. L. and Taylor, H. D., 1925, "Shaft Whipping Due to Oil Action in Journal Bearing," *General Electric Review*, **28**, pp. 559-568.

19. Gunter, E. J., 1966, "Dynamic Stability of Rotor-Bearing Systems," NASA SP-113, pp. 153-157.

20. Nicholas, J. C., Gunter, E. J. and Barrett, L. E., 1978, "The Influence of Tilting Pad Bearing Characteristics on the Stability of High-Speed Rotor Bearing Systems," Report No. UVA/643092/MAE81/141, School of Engineering and Applied Science, University of Virginia, Charlottesville, Virginia, pp. 30-32. Also in *Topics in Fluid Film Bearing and Rotor Bearing System Design and Optimization*, an ASME publication, April 1978. pp. 55-78.

21. *Theory and Practice of Lubrication for Engineers* by Dudley D. Fuller, Wiley and Sons, 1984, ISBN 0-471-04703-1

22. API Standards 612 (Special Purpose Steam Turbines), 617 (Centrifugal Compressors), 670 (Vibration and Temperature Monitoring) and RP684 (Rotordynamics) are available from the American Petroleum Institute in Washington, D.C. or from WWW.API.ORG

More information:

RotorBearingDynamics.COM

or contact:

MLeader@RotorBearingDynamics.COM