

Understanding Journal Bearings

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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. Scallop marks are cut at the splitline to admit oil. Figure 2 is a computer model of this same bearing.

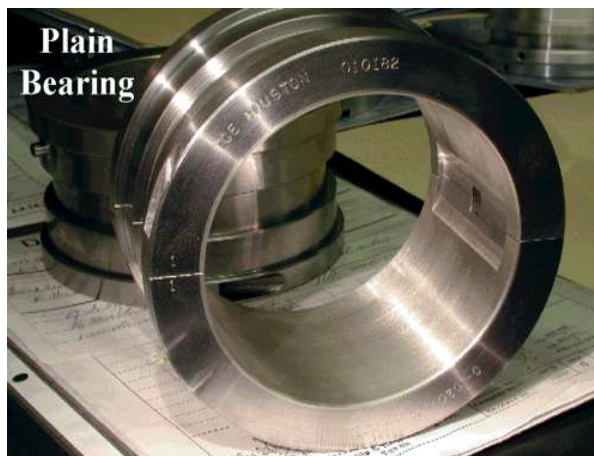


Figure 1 - Typical Plain Journal Bearing

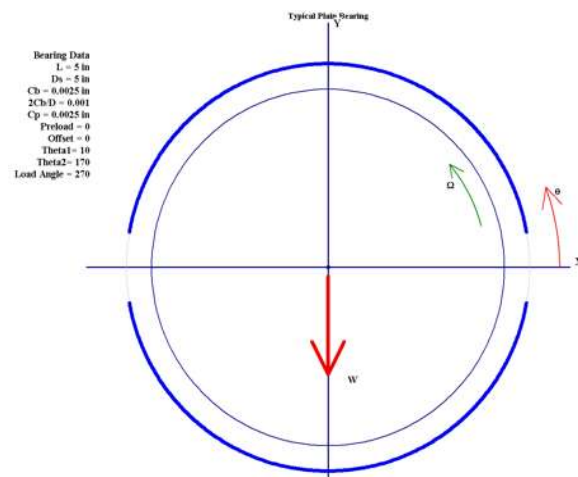


Figure 2 - Computer Model of Plain Journal Bearing

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.

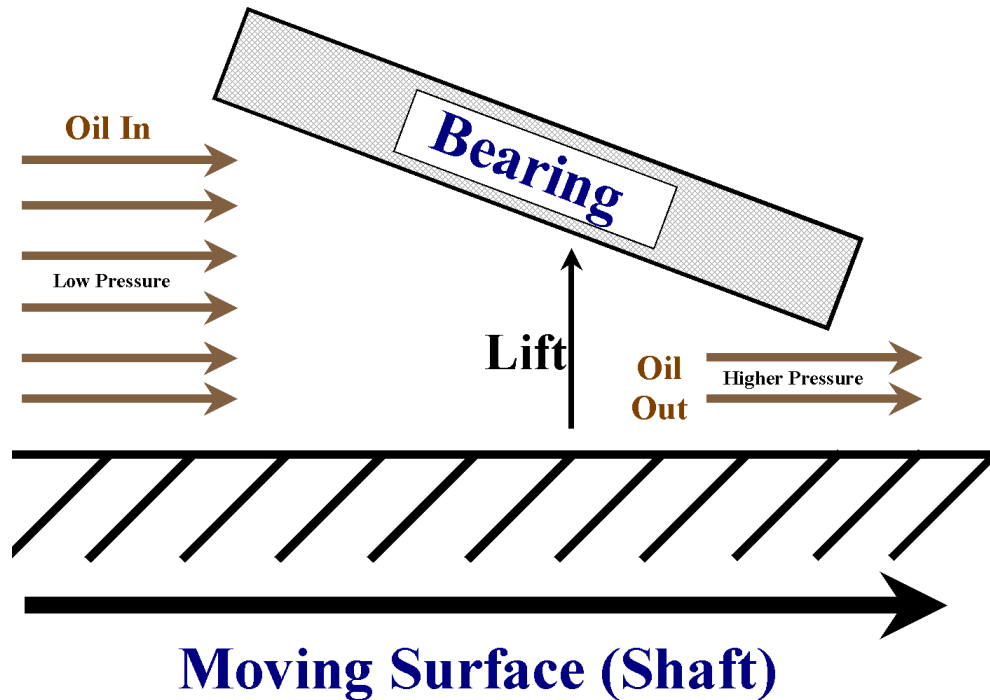


Figure 3 - Basic Development of an Oil Wedge

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

$ecb = e/C_b$ = 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

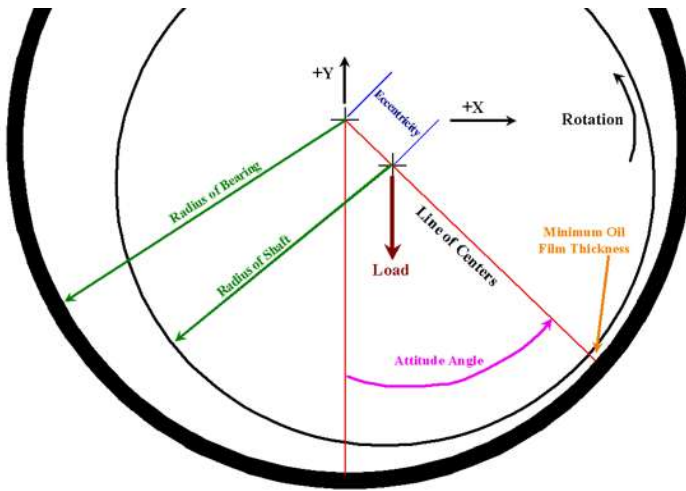


Figure 4 - Bearing Nomenclature

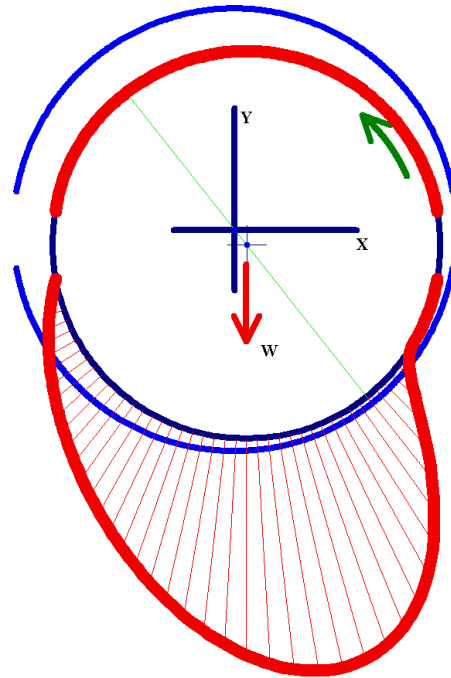


Figure 5 - Pressure Profile in a Journal Bearing

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).

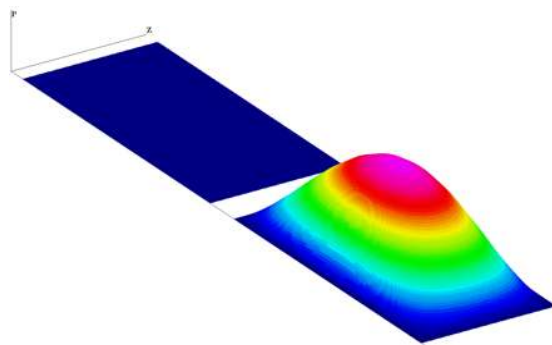


Figure 6 - Hydrodynamic Pressure Profile - Low Load

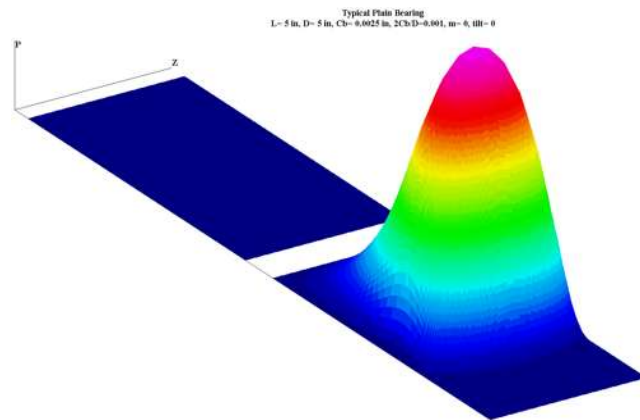


Figure 7 - Hydrodynamic Pressure Profile - High Load

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.

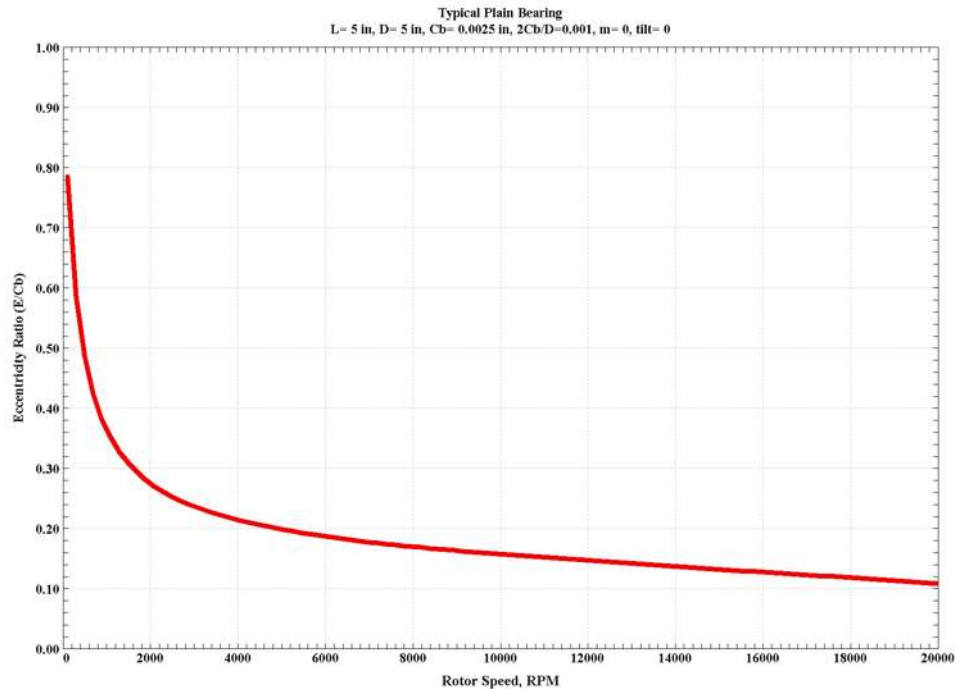


Figure 8 - Plain Bearing Eccentricity versus Speed with a Constant Load

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.

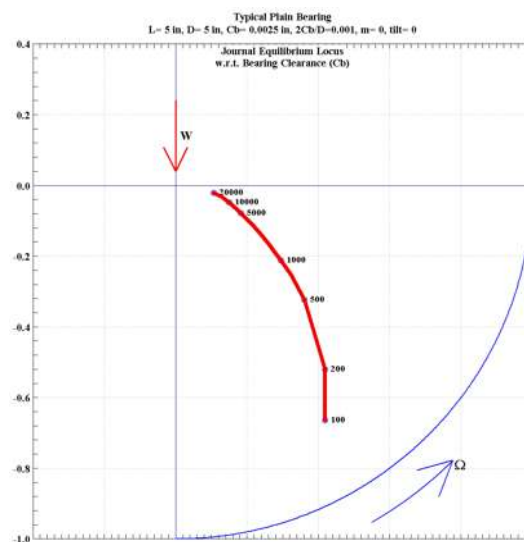


Figure 9 - Shaft Centerline Position in a Plain Bearing as a Function of Speed

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.

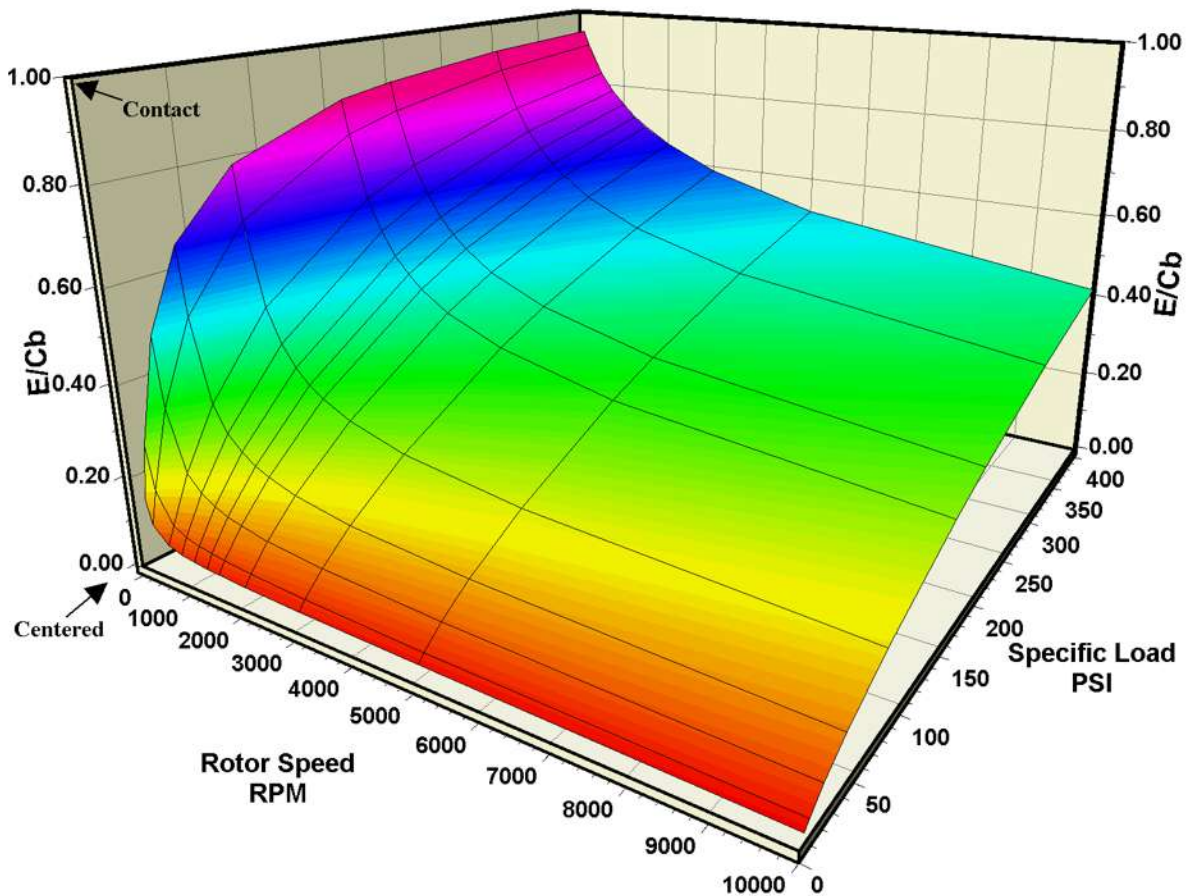


Figure 10 - Effect of Load and Speed on Plain Journal Eccentricity

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.

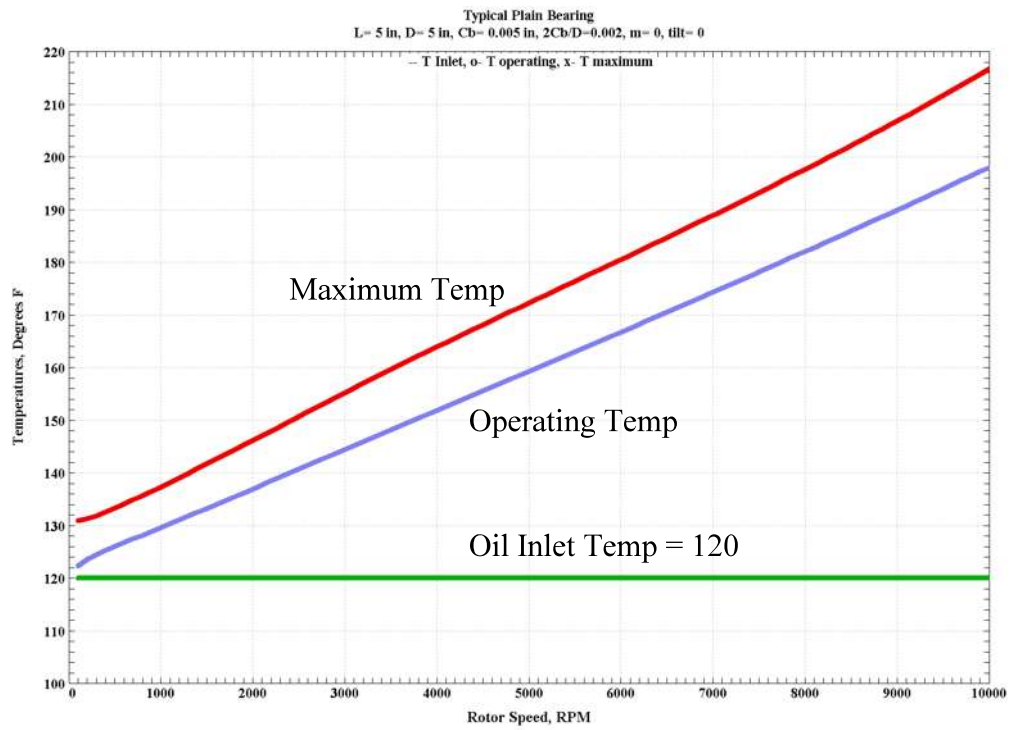


Figure 11 - Temperatures in a Plain Bearing versus Speed at a Constant Load

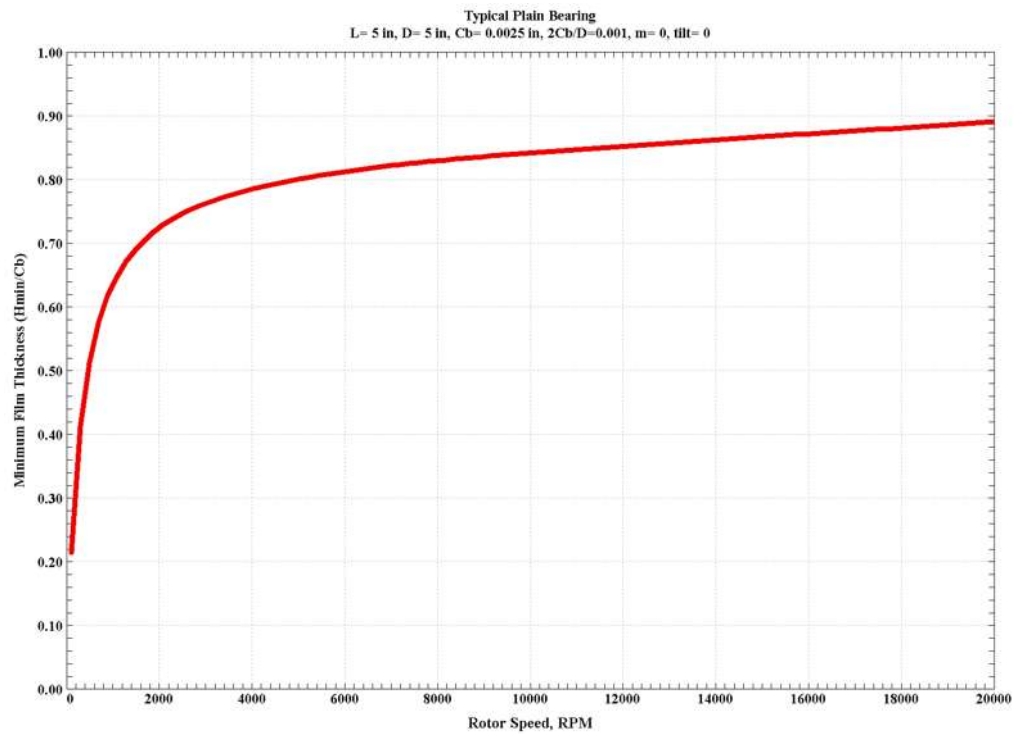


Figure 12 - Minimum Oil Film Thickness in a Plain Bearing versus Speed at a Constant Load

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	100 ρ	$1.45 \times 10^{-5} \rho$	—	100
CentiStoke (ν) cm ² /100 SEC	0.01 ρ	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.

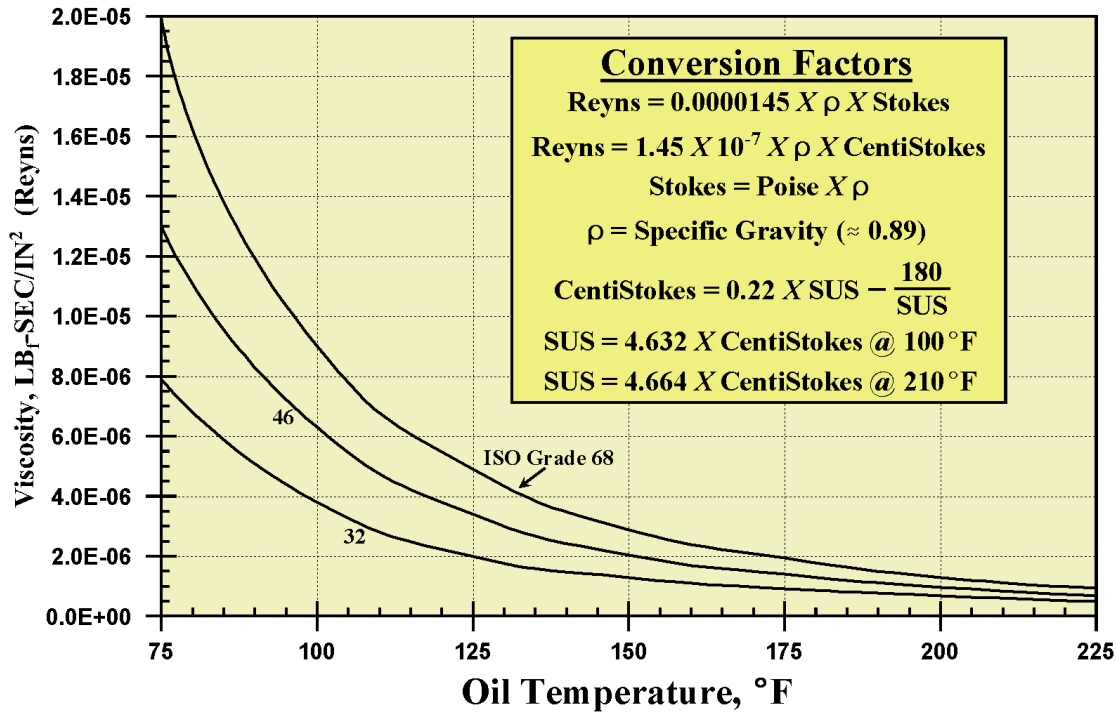


Figure 13 - Viscosity Variation with Temperature and Conversion Factors

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.

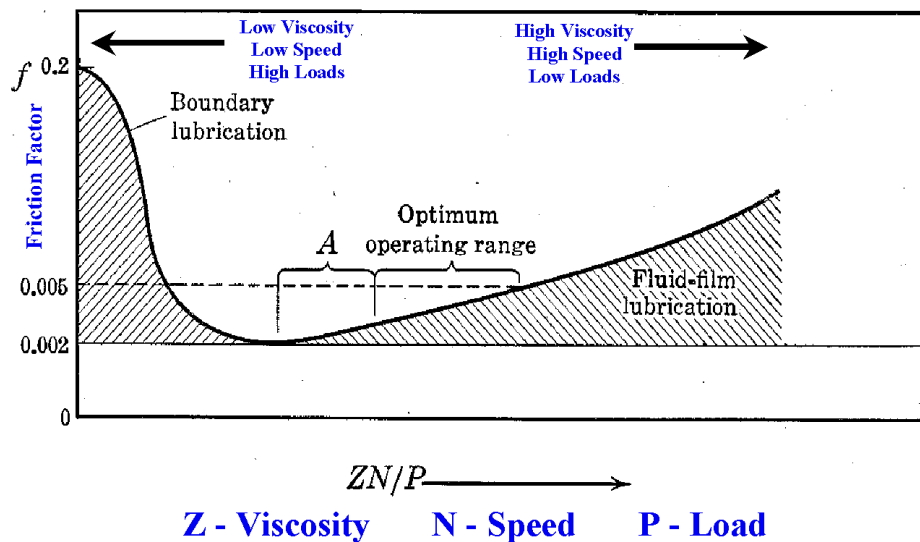
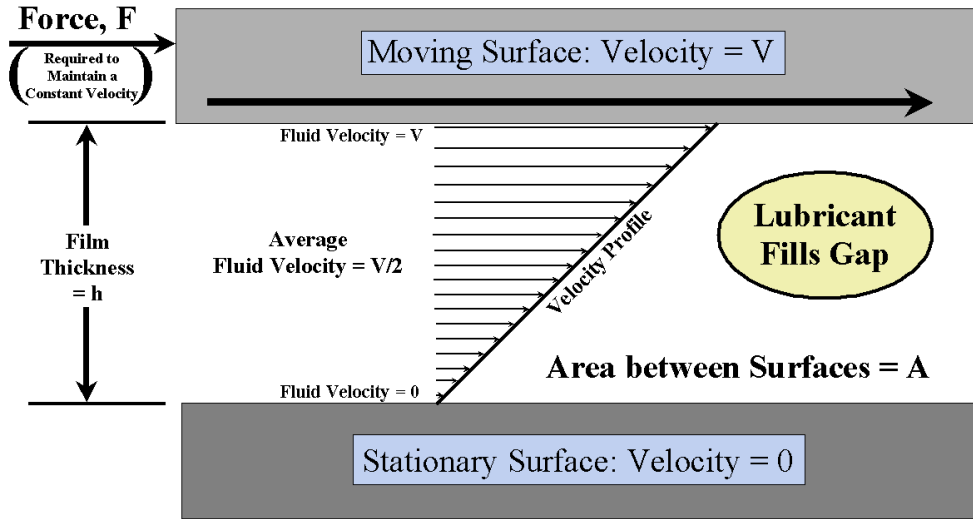


Figure 14 - Stribeck Curve Relating Friction Factor to Viscosity, Speed, and Load

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 service 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.



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

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

Or in SI units:

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

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

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

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.

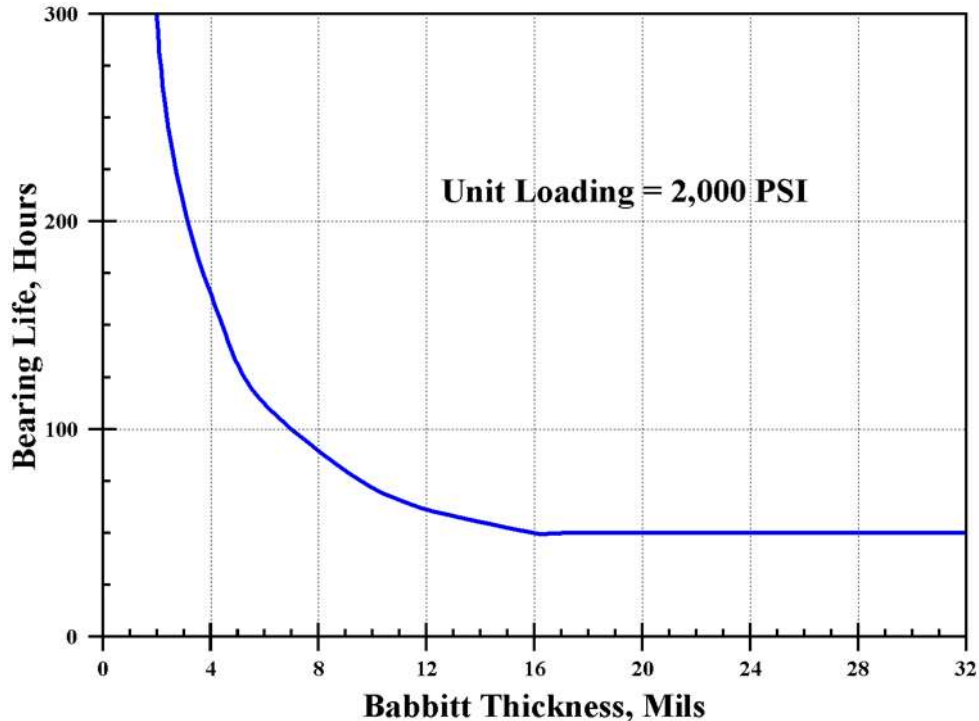


Figure 16 - Bearing Fatigue Life as Babbitt Thickness Varies

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.

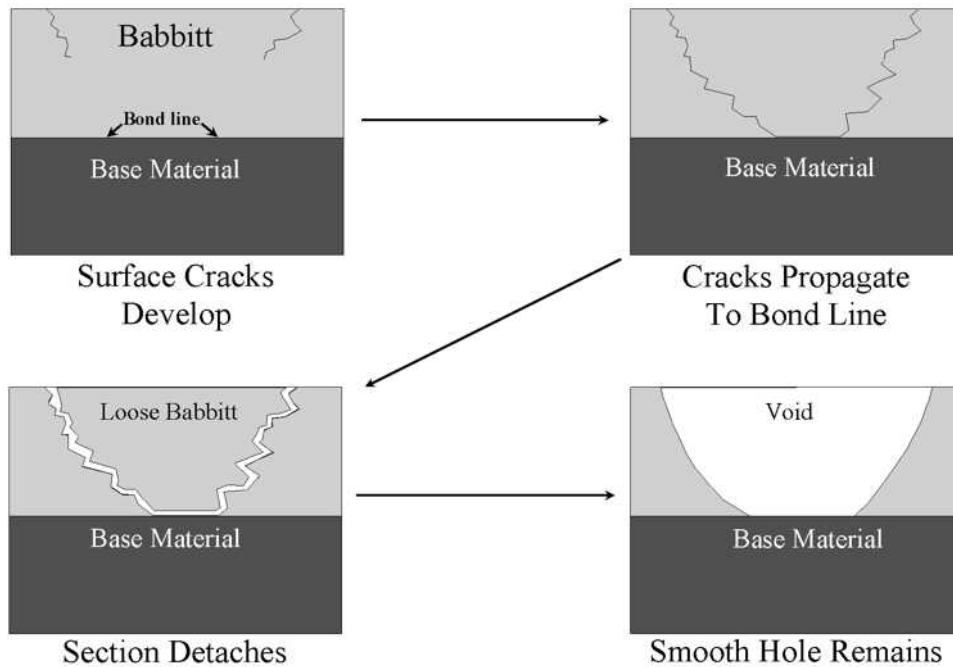


Figure 17 - Babbitt Failure due to Fatigue Pounding Damage

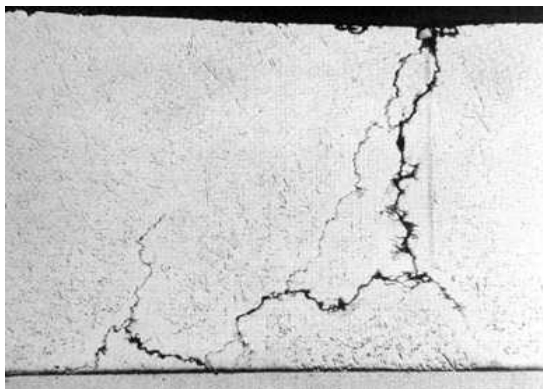
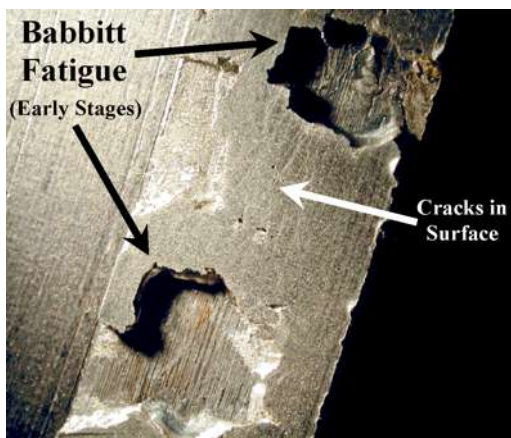


Figure 18 and 19 - Fatigue Cracks in Babbitt



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
 $\pi = 3.14159...$
 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)

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

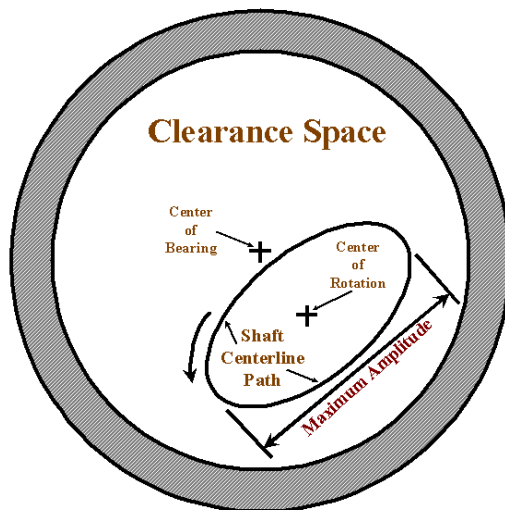
some typical examples are:

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 babbitt 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 babbitt 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 babbitt 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 babbitt or white metal directly into a sleeve containing the actual shaft or a same-size mandrel. Then, the surface of the babbitt 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.

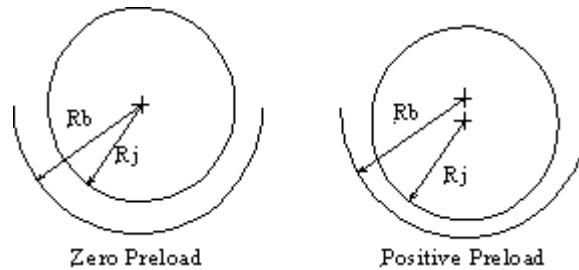


Figure 23 - Pad Preload Definition

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.

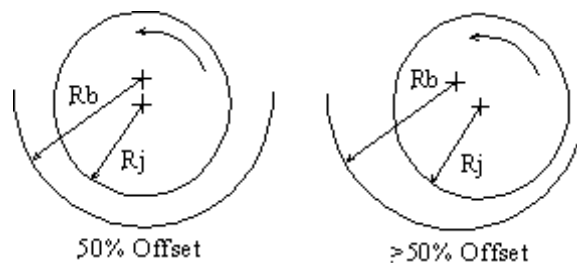


Figure 24 - Pad Offset Definition

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.

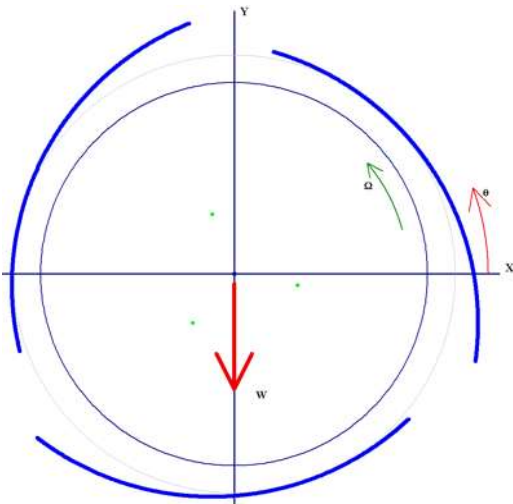


Figure 25 - Multi-Lobe Bearing

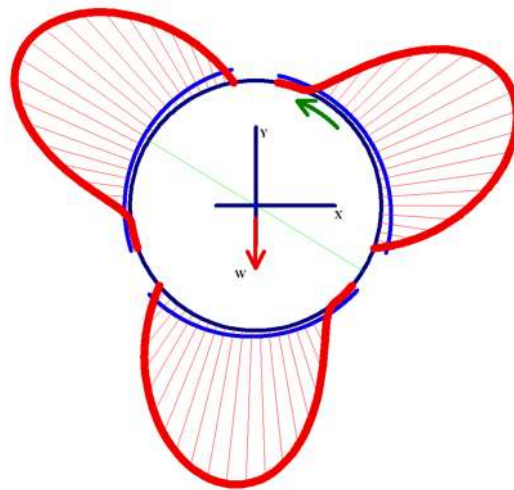


Figure 26 - 3-Lobe Bearing Pressure Profile

●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.

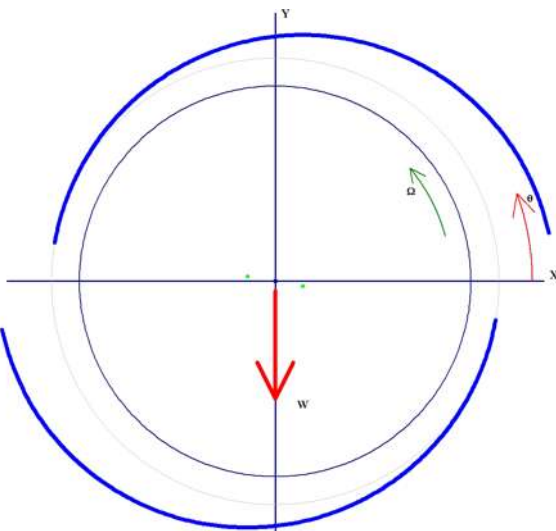


Figure 27 - Offset Half Bearing

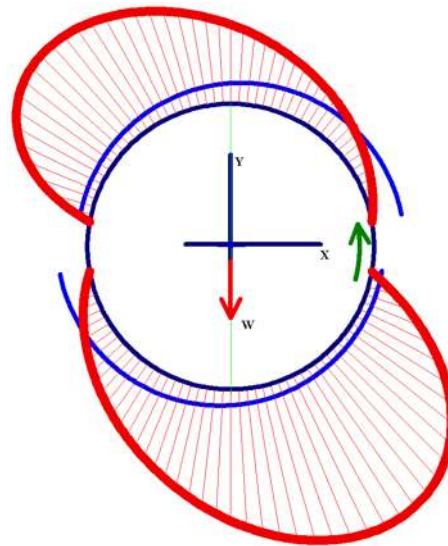


Figure 28 - Offset Half Bearing Hydrodynamic Pressure Profile

● 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.

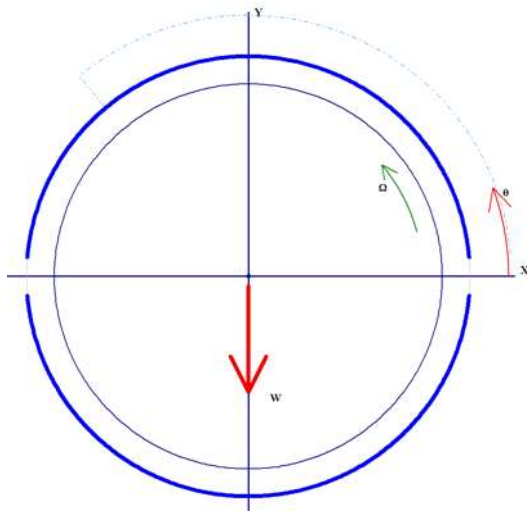


Figure 29 - Pressure Dam Bearing

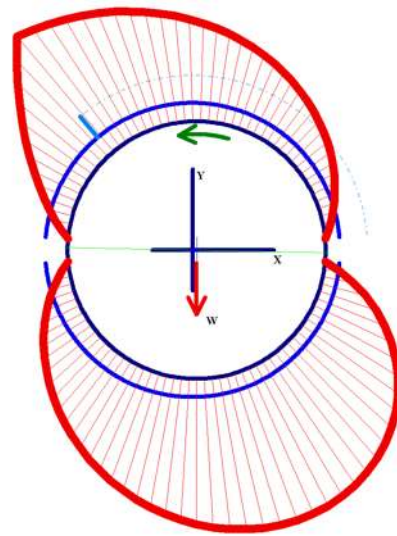


Figure 30 - Pressure Dam Bearing Hydrodynamic Pressure Profile

● 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.

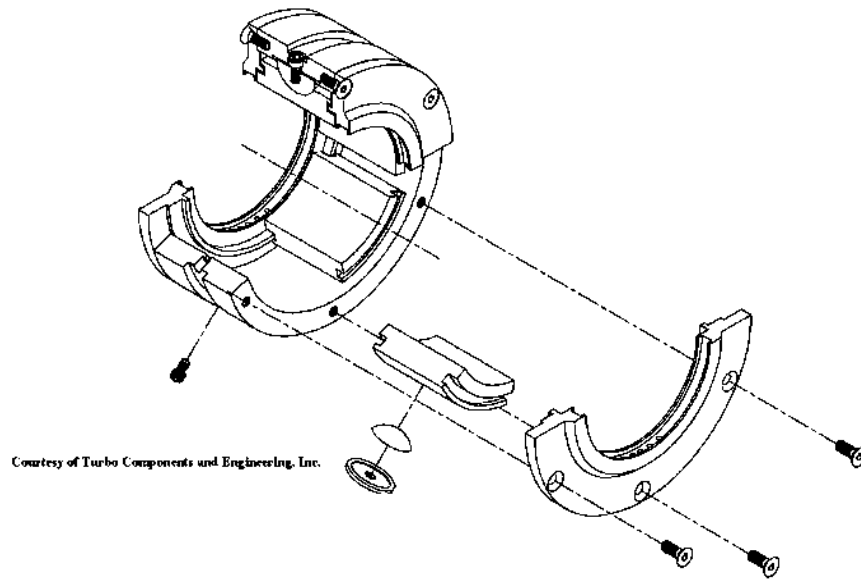


Figure 31 - Exploded View of Tilting Pad Bearing

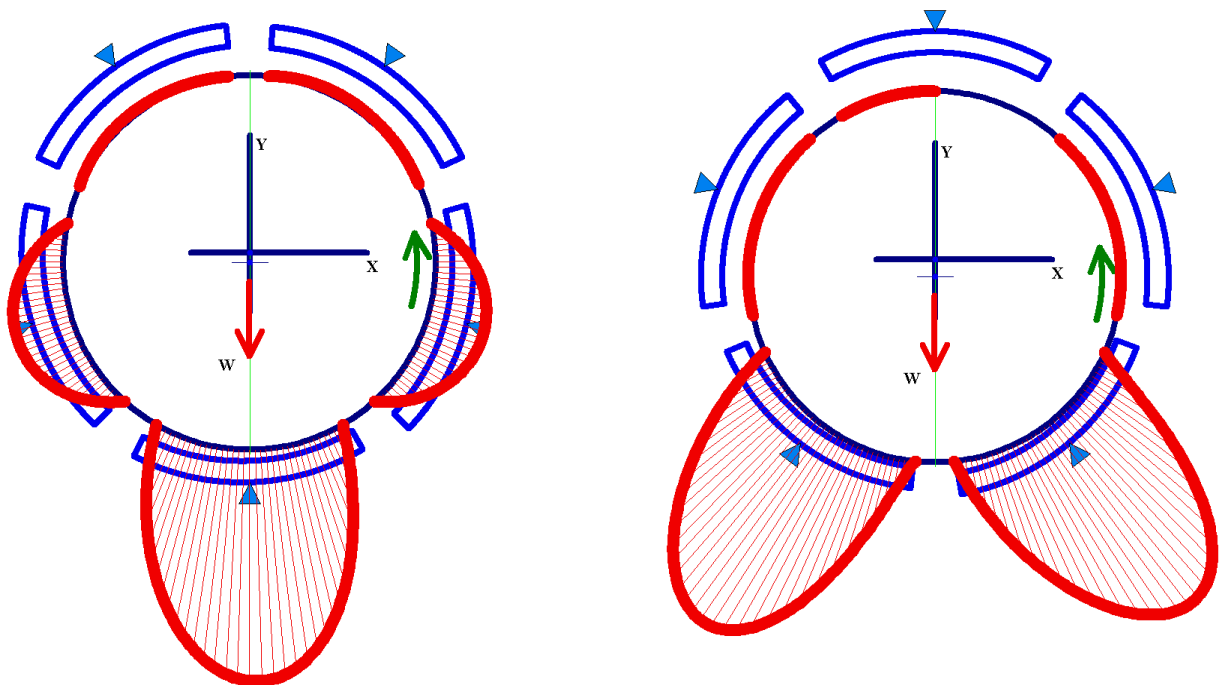


Figure 32 - Hydrodynamic Pressure Profile for Two Tilting Pad Bearings

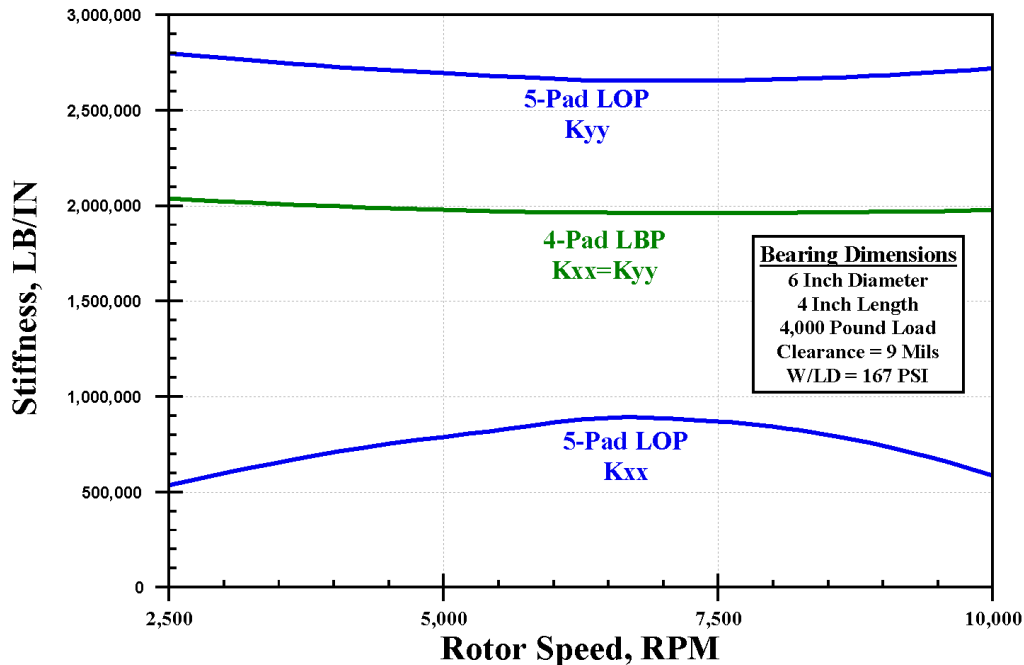


Figure 33 - Using Load Orientation and Number of Pads to Fine-Tune Bearing Stiffness

● 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.

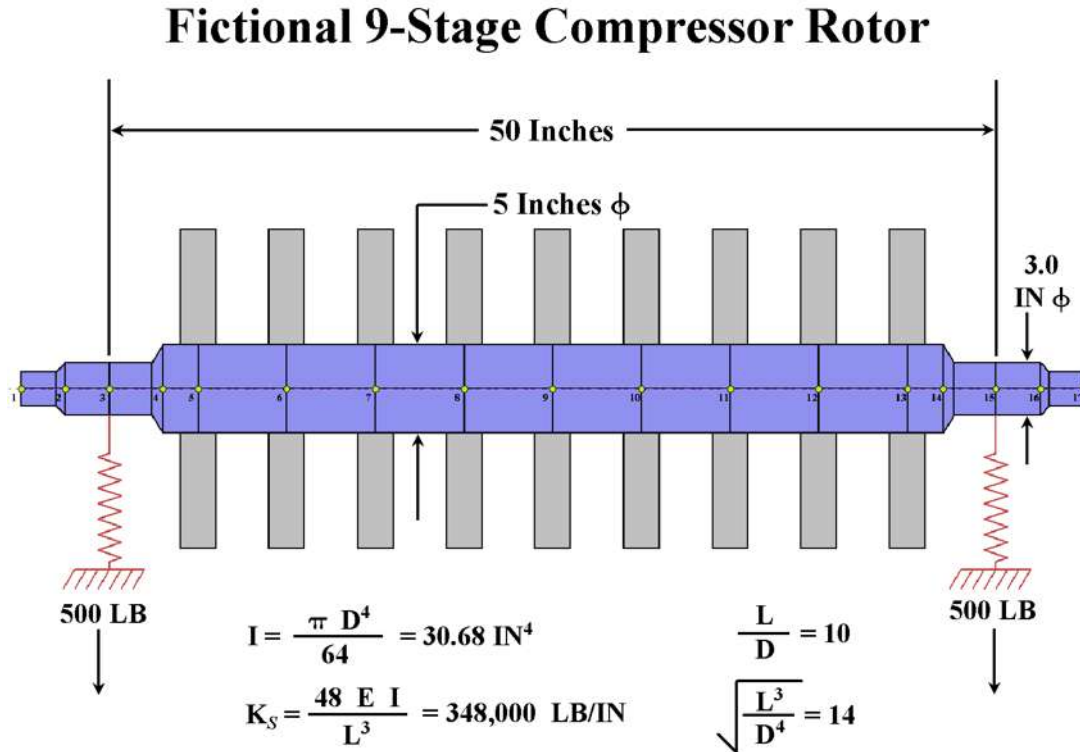


Figure 34 - Compressor Rotor Used for Bearing Comparison

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.

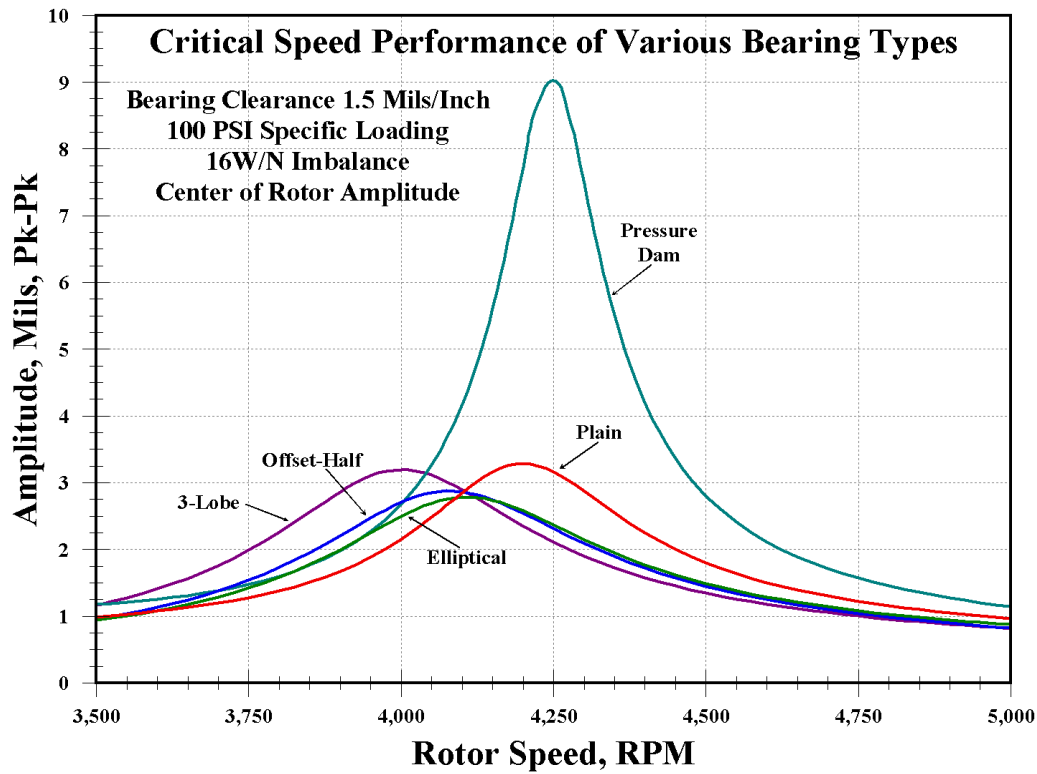


Figure 35 - Critical Speed Performance for Fixed-Profile Bearings

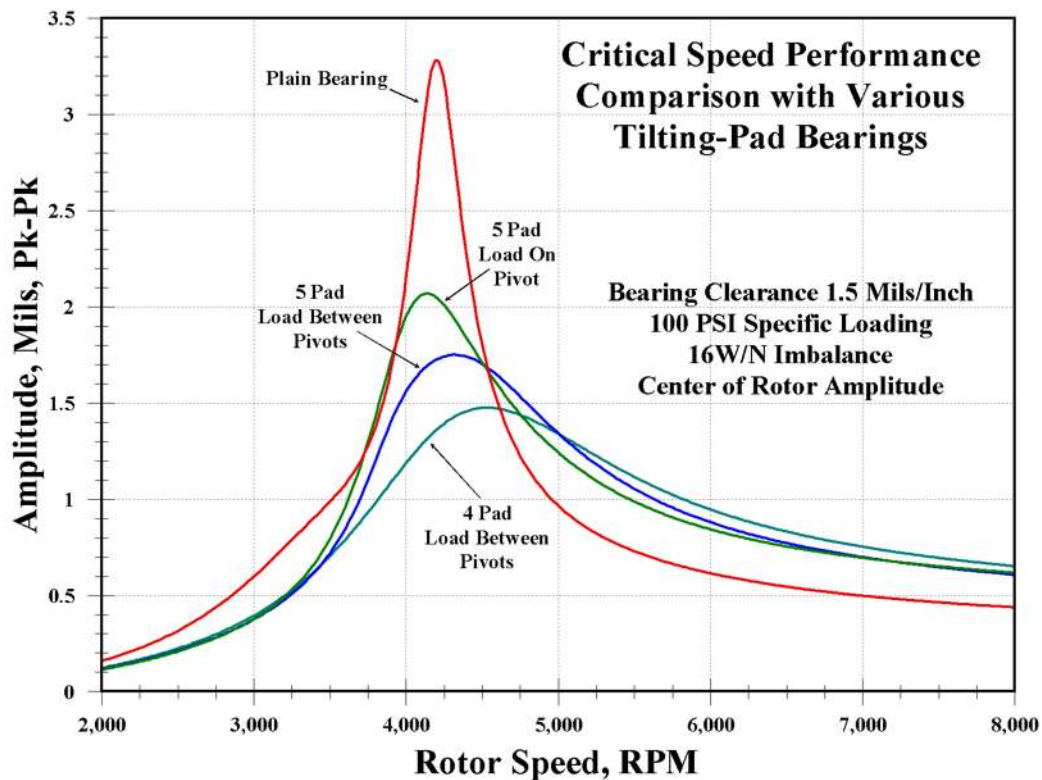


Figure 36 - Critical Speed Performance for Tilting Pad Bearings

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 prelube 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.

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13.1 Introduction

Any bearing can be simply defined as a support or guide which at the same time allows relative movement to take place between two bodies. In a diesel engine, the principal bearings are those which allow rotation of the crankshaft about its own longitudinal axis in the main engine entablature, and those between the connecting rod and crankshaft. In modern engine terminology a bearing has come to mean the component fitted between the journal and either the main bearing housing or the connecting rod, the shaft being supported in the bearing via an oil film. Figure 13.1 indicates the interaction between the various components in any engine bearing assembly, and the environment within which the bearing system must operate¹.

The co-operating surface, the lubricant and the environment all place constraints on the bearing, the resulting design usually being a compromise between various conflicting requirements. In relation to the bearing itself, the choice is usually restricted to material and geometry. However, environmental factors have to be taken into serious consideration at the design stage if a reliable system is to be achieved.

13.2 Bearing design

13.2.1 Wall thickness

To achieve optimum utilization of space, reliability and repeatability of the bearing assembly, thin shell bearings have been almost universally adopted in modern diesel engines, and fitted retrospectively into many older engines. Such a bearing allows the use of a wider range of surface lining materials than would otherwise be possible.

The thin shell bearing is a precision made pre-finished component consisting of a steel backing with a thin coating of the most appropriate lining material necessary to provide sufficient strength for the applied loading and compatibility with the general environment. The overall thickness/diameter ratio is not critical, but typically varies from 0.05 at 40 mm diameter to 0.02 at 400 mm as shown by Figure 13.2.

13.2.2 Interference fit²

The assembled form of a thin shell bearing is dictated by the

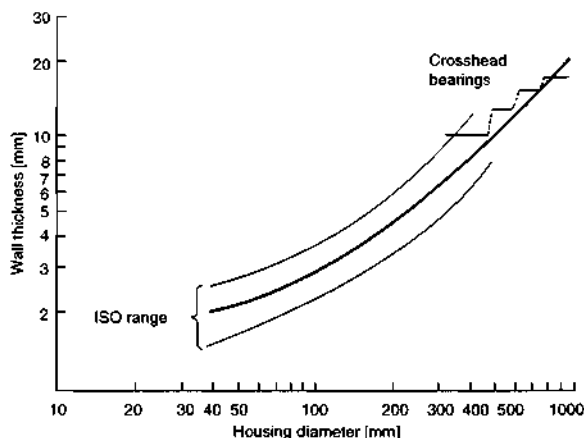


Figure 13.2 Wall thickness/diameter for thin wall bearings

tolerance to which it can be manufactured, and by the housing into which it fits. To ensure conformity of the bearing shell to its housing, an accurate interference fit has to be provided. This is obtained by means of an excess peripheral length in each half-bearing which has to be closely prescribed to allow interchangeability, requiring precise quality control of every half-bearing in a checking fixture (Figure 13.3). On assembly the excess peripheral length creates a circumferential stress around the bearing, and a radial contact pressure between the bearing back and the housing bore. This contact pressure resists relative movement, thus preventing fretting. Unfortunately, there is no theoretically correct level; housings with high flexibility require more contact pressure than stiffer ones. On early engines, having thin shell bearings, a contact pressure as low as 2 MPa was usually sufficient to resist fretting, but as engine ratings increased, and housing stress analysis became more sophisticated, higher pressures became necessary, often reaching 8–10 MPa today. In these very high interference fit assemblies, particular care has to be taken to ensure that the joint face clamping bolts have sufficient capacity to assemble the bearing, and to resist dynamic separating forces generated by engine operation.

As the contact pressure is increased for any given bearing size, the circumferential stress increases to the point where the

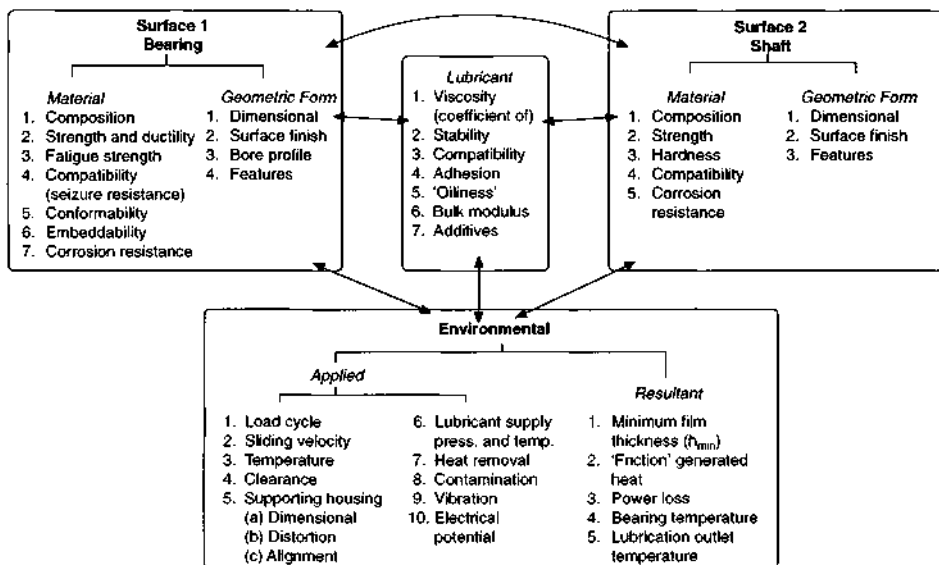


Figure 13.1 The bearing system-interaction between components and environment



Figure 13.3 Checking peripheral length of a thin shell bearing

steel backing begins to yield adjacent to the joint faces. Knowing the combined effect of bearing steel yield strength and the friction force for bearing assembly, a wall thickness can be determined which will avoid yield, although in some automotive-type engines this may not be feasible.

Increased contact pressure requires a greater bolt tension for fitting bearing caps to their opposite half-housings, and as higher values are incorporated, measurement checks should be carried out on prototype assemblies to ensure that satisfactory closure of the housing joints occurs. Figure 13.4 shows a convenient method of checking this closure, the micrometer readings being taken as the bolt load is gradually increased. Insufficient bolt pre-load would result in fretting between the bearing and housing, and could allow the housing joints to separate, giving rise to high dynamic loading of the bolts, with possible fatigue fracture.

To reduce the tendency to fretting, it is recommended that the centre line average (CLA) of the housing bore surface finish should not exceed $1.6 \mu\text{m}$. Bearing backs are typically $0.8 \mu\text{m}$ CLA surface finish and, in highly loaded zones, should not be unsupported. Cyclic variation of hydrodynamic oil pressure on

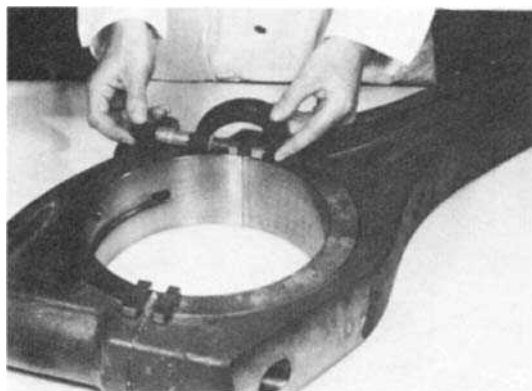


Figure 13.4 Method of checking housing joint face closure

the bearing surface will cause the bearing to flex in these unsupported regions. If, for example, there are grooves or oil holes behind the bearing surface, fretting will almost certainly occur. If flexure is sufficiently great, the lining material on the bearing surface can also suffer fatigue damage. Figure 13.5 shows fretting damage on the back caused by a combination of insufficient interference fit and lack of support. The resulting flexure has also caused the lining material on the bore surface to fatigue.

Provided a bearing is manufactured to the defined length and wall thickness limits, and the housing and bolting arrangements are satisfactory, correct assembly should be achieved every time, with bearing shells being interchangeable. However, locating tangs and free spread greatly assist correct assembly of thin shell bearings, particularly replacement *in situ*.

13.2.3 Locating tangs

Locating tangs allow correct axial positioning of bearing shells thereby ensuring alignment of oil transfer passages and clearance between the ends of the bearing and crankshaft fillets. The tang is located in a corresponding recess in the housing and must be a clearance fit, otherwise the bearing surface may distort. It is also advantageous to relieve the tang below the level of the bearing joint face, thus reducing localized pressure on the tang during assembly.

Tangs are not intended to resist rotation of the bearing shell; that is achieved by the contact pressure. If, under seizure conditions, this contact pressure cannot prevent rotation, the tangs are either sheared or flattened.

13.2.4 Free spread

Free spread is the term given to the excess dimension across the bearing joints with the bearing in its free state, compared with the housing diameter. This allows the bearing to be positively located radially and should generate sufficient friction to hold the shell in its housing even when inverted.

Following operation in an engine, bearings may show a loss of free spread on removal for the following reasons:

- (i) The lining of a steel-backed bearing, because of its higher thermal expansion, can yield in compression at operating temperature producing a residual tensile stress when cooled to room temperature. This effect becomes greater with higher lining to steel thickness ratio.
- (ii) A high radial temperature gradient across the steel backing, due to malfunction, can cause differential yield and therefore additional free spread loss.
- (iii) Relief of strains induced during fitting or by the manufacturing and machining processes

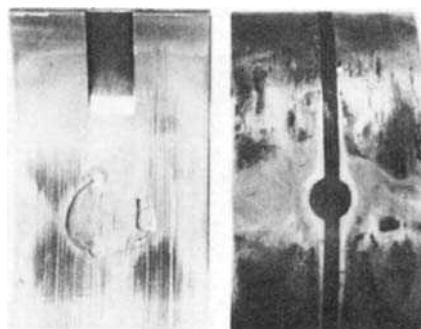


Figure 13.5 Fatigue and fretting damage due to insufficient interference fit and unsupported region of the bearing back

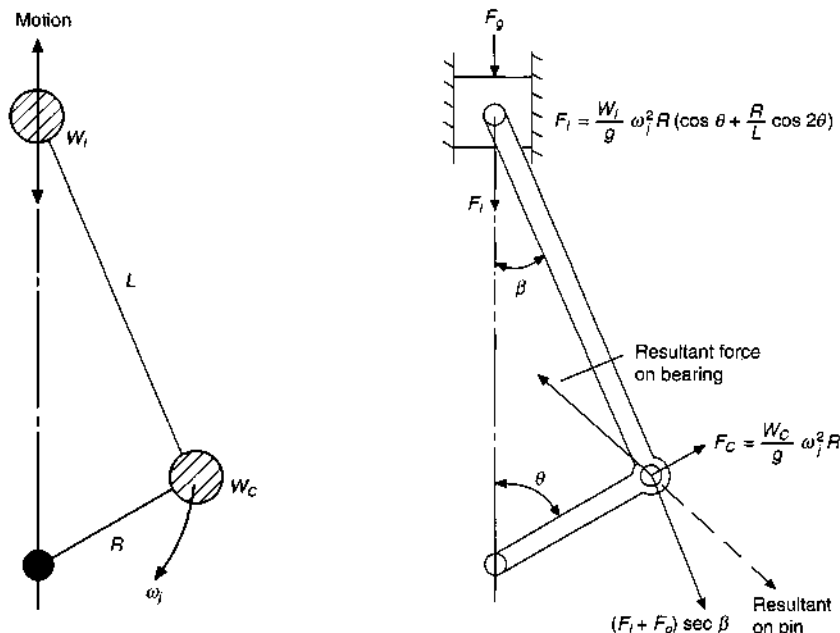


Figure 13.6 Components of load acting on large end bearing

The forces which result in free spread loss after releasing the bearing from its housing are of relatively small magnitude and will not cause malfunction of the bearing whilst in operation. Problems only arise upon reassembly, particularly if one half has negative and the other positive free spread. The half with the negative free spread may become trapped against the shaft and lead to overheating or seizure. Therefore bearings with negative free spread should not be refitted.

13.2.5 Loading on crankpin and main bearings³

The load on both crankpin and main bearings varies in magnitude and direction with time. When studying the performance of such bearings, it is necessary to determine these loads, over each complete loading cycle. Such data are usually presented as polar load diagrams.

The loads on the connecting rod big end bearing can be attributed to three components; reciprocating inertia forces, rotating inertia forces and gas forces, as shown by the two diagrams of Figure 13.6. Vector addition of these forces at regular intervals of crank angle results in the typical polar load diagram of bearing forces for one loading cycle shown in Figure 13.7.

The loads on main bearings are partly due to force reactions from the big end bearings and partly due to out of balance of the crankshaft. The out of balance of the crankshaft is usually reduced by the use of balance weights. When considering the forces from the big end bearing, it is necessary to orientate them to the same non-rotating datum as the main bearing (i.e. relative to a cylinder axis).

In a multi-cylinder engine, which generally in the modern engine has a main bearing between each crank pin, the force on that main bearing is usually assumed to consist only of forces from components in the two crank bays immediately adjacent to that bearing. The resultant force is obtained by treating the crankshaft as a series of simply supported beams resting on supports at the main bearings (statically determinate method). By this means, component forces closest to any one main bearing have the largest effect on that bearing. Then, in much the same way as the polar load diagram of the big end bearing is built up over a complete loading cycle, the overall dynamic load pattern

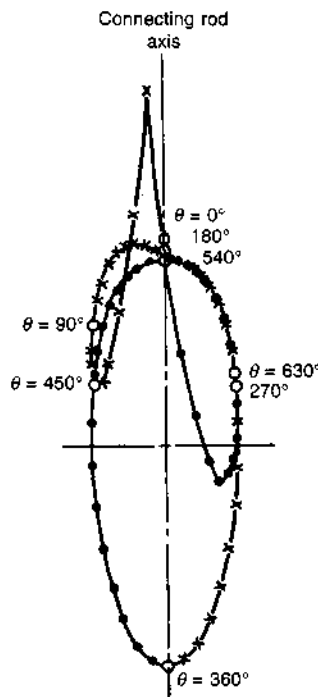


Figure 13.7 Typical large end polar load diagram

on each main bearing is obtained by vector addition of all the forces at each interval of crank angle.

13.2.6 Prediction of oil film thickness

Once the dynamic load pattern as a function of time has been established for any bearings in the engine, it is possible to predict the minimum oil film thickness, and its position on the bearing surface, at each position of the load diagram. Several different procedures exist for this prediction, varying in complexity and

hence in validity with the real case. Each method makes some attempt to solve Reynolds equation governing the performance of hydrodynamic bearings. Some of the methods available are:

- Hand calculation, using an equivalent speed concept^{4,5,6}
- Graphical procedures for the complete locus, using mobility methods⁷.
- Approximate method to predict the minimum film thickness in the cycle where 'squeeze intervals' predominate⁸.
- Computer methods which predict the complete journal locus^{7,9,10}.
- Inertia load studies¹¹.
- Experimental results⁸.

All these methods are thoroughly reviewed by Campbell *et al.*⁸, but whilst each technique has specific advantages in particular circumstances, those most generally employed involve the use of computer methods to predict the complete journal locus over the whole of each loading cycle. The advantage of rapid, low-cost computer programs is that predicted results can be achieved and compared for a large number of different types and sizes of engines at various operating conditions. With the greater flexibility afforded by the use of computers, bearing analysis will continue to advance, but whilst it is already possible to give some consideration to housing, crankshaft and bedplate distortion, the most common computer techniques still treat the crankshaft as a series of simply supported beams. For medium- and slow-speed engines, this is probably realistic due to the high crankshaft stiffnesses demanded by stress limits, and the high standard of bearing alignment enabled by current machine tools and production methods.

Other basic assumptions usually include: perfect alignment of shaft and bearing through the loading cycle; truly circular shaft and bearing; infinitely rigid bearing housing and journal; viscosity unaffected by variations of temperature and pressure around the bearing and through the cycle; and that negative oil film pressures can be neglected.

From the numerous bearing calculations carried out by the Glacier Metal Co., with use of computer calculation procedures incorporating the above assumptions, it has been possible to prepare a graph of computed minimum film thicknesses for different shaft diameters (Figure 13.8). This facilitates comparison of calculated values for any other engine against the norm, but does not allow absolute limits to be set. Deduced levels of 0.000 008 per unit diameter for large ends, and 0.000 01 per unit of diameter for main bearings have been superimposed. Further experience, improved production methods, filtration, etc., will presumably allow these levels to be reduced in the future.

A similar exercise was carried out to compare the calculated maximum specific bearing loads (Figure 13.9) for the same engines as with the minimum film thicknesses of Figure 13.8. The wide scatter band shown by these results is to be expected, but as with the film thickness values, they allow direct comparison with existing practice for any engine design, particularly at the design stage.

13.2.7 Grooving configuration

The simplest grooving arrangement is to have a central circumferential groove of 360° extent in all bearings. This allows unimpeded oil transfer from one area to another. Hydrodynamically, the absence of all grooving is the optimum, and so a compromise between the hydrodynamic requirement and the physical requirement of supplying oil has to be made. This often results in the so-called partially grooved bearing in which there is a central circumferential groove of less than 360° extent. Such a partially grooved bearing usually allows an increase of

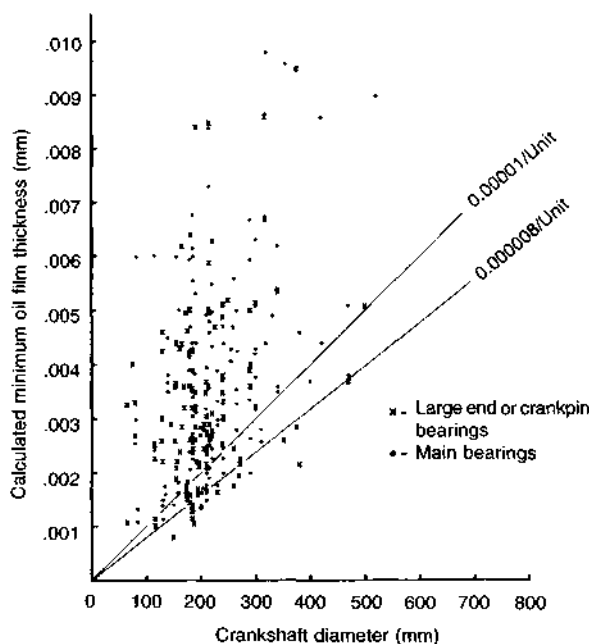


Figure 13.8 Predicted minimum oil film thickness as a function of diameter

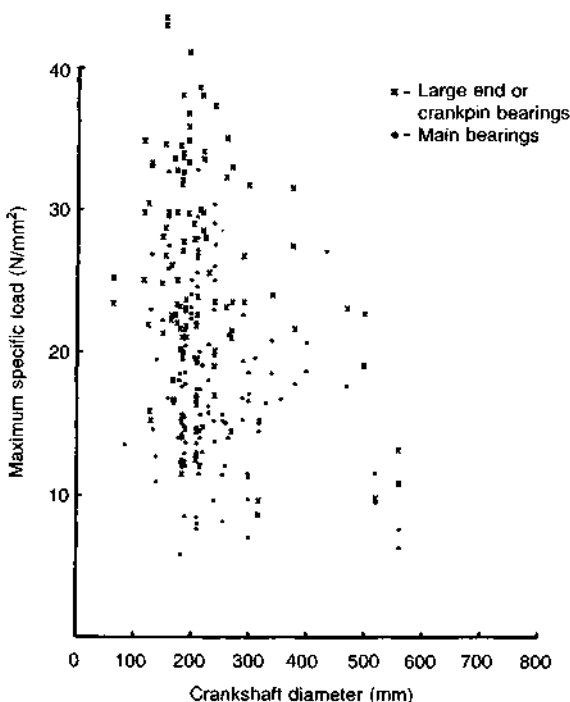


Figure 13.9 Predicted maximum specific load as a function of diameter

the order of 100–200 per cent in the oil film thickness in the original thinnest film region.

When partially grooved bearings are adopted, a system of interconnected cross drillings in the crankshaft is necessary to allow a constant supply of oil to each bearing, or to the piston.

Positioning of, and oil supply passages to, partial grooves can be critical. Predicted oil film thicknesses are based on the

assumption that oil is available at all times within the clearance space to allow a film to be generated. Thus any arrangement of partial grooving must allow sufficient oil to be supplied to the required region of the bearing surface.

Two further distinct disadvantages of the partial groove design of bearing are described in the section on bearing damage. These are differential wear of the crank journal, and cavitation erosion of the bearing surface¹².

Thus one should not automatically design partially grooved bearings purely on the basis that increased oil film thickness will be achieved. Alternative methods of improving the film thickness should also be investigated if improvement is considered necessary. This includes increase of bearing dimensions, adjustment of engine masses and modification of firing orders.

13.2.8 Clearance¹³

There are no precise limits on what clearance there should be between bearing bore and journal surface, and it is often determined by experience. As clearance is increased, the theoretical load-carrying capacity is reduced, but this assumes that bearing temperature and oil viscosity are constant. However, an increased quantity of oil can be pumped through the bearing, thus keeping its temperature lower. Too high a value of clearance in any plane around the bearing may also result in cavitation erosion damage¹². Thus there is a range of clearance over which optimum bearing performance can be expected. Figure 13.10 presents curves of a semi-empirical nature based on a theoretical heat balance for steadily loaded bearings. These curves can be considerably simplified, by the superposition of a practical band of shaft sizes over a given speed range. Comparison of these two sets of curves results in a generalized value for minimum diametral clearance of $0.00075 \times$ bearing diameter. Manufacturing tolerances of shaft, housing diameter and bearing wall thickness then result in a maximum design clearance somewhat greater than $0.001 \times$ bearing diameter.

When oil film thickness computations are carried out on a comparative basis for various values of clearance, allowing for the lower temperature and higher viscosity corresponding to the larger clearances, it is often found for the medium-speed range of engines that optimum oil film conditions occur at a clearance of approximately $0.001 \times$ bearing diameter, but over a range

$0.00075-0.0015 \times$ bearing diameter there is usually only a small change in the predicted conditions. This upper limit of $0.0015 \times$ bearing diameter can be used as an indicator for limiting wear during engine operation, although absolute limits for any particular engine type can be determined only by experience with that engine type. An additional practical limit can also be imposed by oil pump capacity.

13.3 Bearing damage

The main difficulty of establishing separate categories of damage is that they are often related, or that damage initiated by one cause can result in another effect. However, the commonest types of bearing damage likely to be experienced by diesel engine operators can be approximately classified under the following 10 categories (Sections 13.3.1 to 13.3.10).

13.3.1 Abrasion

This is still probably the most common form of bearing damage, even though filtration standards are generally very high. Long term operation with very fine debris in the oil, or short term operation with coarser contaminant can result in abrasion and scoring of the surface of a bearing (Figures 13.11 and 13.12) roughening it to such an extent that overheating can occur due to the surface roughness penetrating the thin oil film. Alternatively, ferrous particles can become embedded in the bearing lining or soft overlay and become work hardened to a level higher than that of the crankshaft, resulting in wear and scoring of the shaft. The presence and degree of ferrous contamination in bearing surfaces can be detected by the technique known as iron printing. In essence, this consists of obtaining a chemical reaction between the embedded particles and a weak solution of potassium ferrocyanide held in a filter paper placed on the bearing surface. Generally speaking, a worn bearing should be renewed, when discovered, if approximately one-third of the projected area of copper-lead or nickel interlayer is exposed. Wear through to expose underlying tin-aluminium is not necessarily a cause for renewal, unless the bearing shows signs of overheating, scuffing, or entrapment of particles proud of the bore surface.

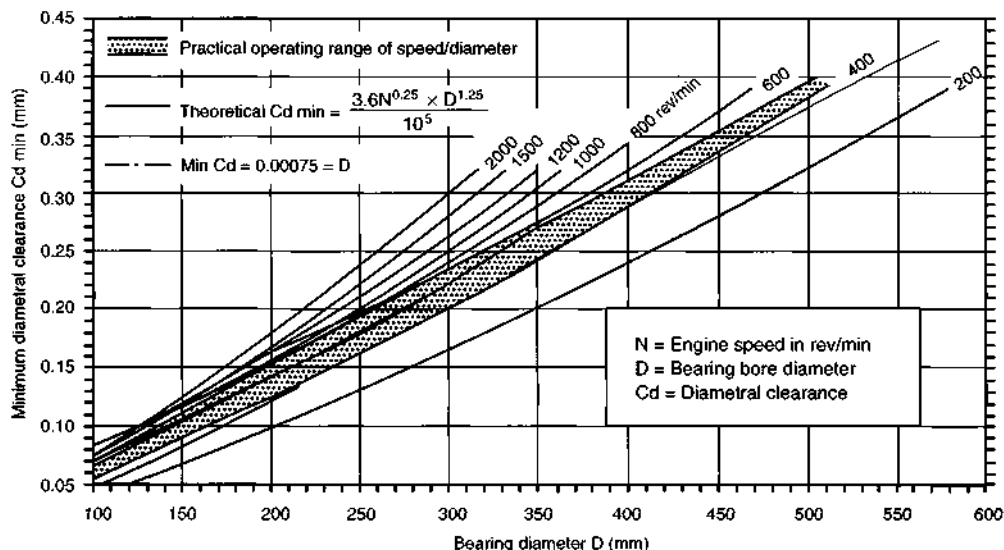


Figure 13.10 Minimum diametral clearance

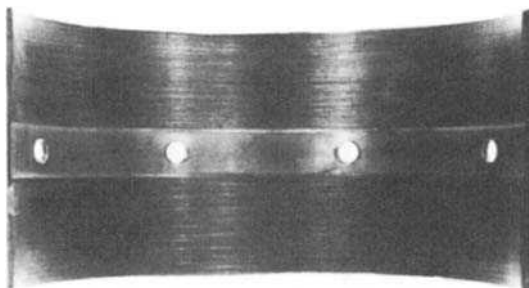


Figure 13.11 Abrasion by small contaminant over a long period

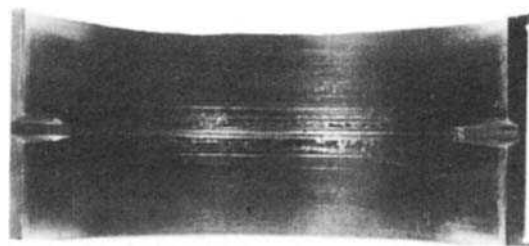


Figure 13.12 Severe scoring by a few large particles

13.3.2 Fatigue

With the general introduction of stronger bearing linings, and improved calculation procedures, fatigue due to design overload is uncommon. Typical fatigue damage of white metal is shown in *Figure 13.13* due to a combination of load, high temperature, and the inconsistency of white metal structure resulting from the direct lining of a large variable-cross-section housing. Fatigue damage of the stronger lining of the modern diesel engine is usually a consequence of some other problem, as was shown in *Figure 13.5*.

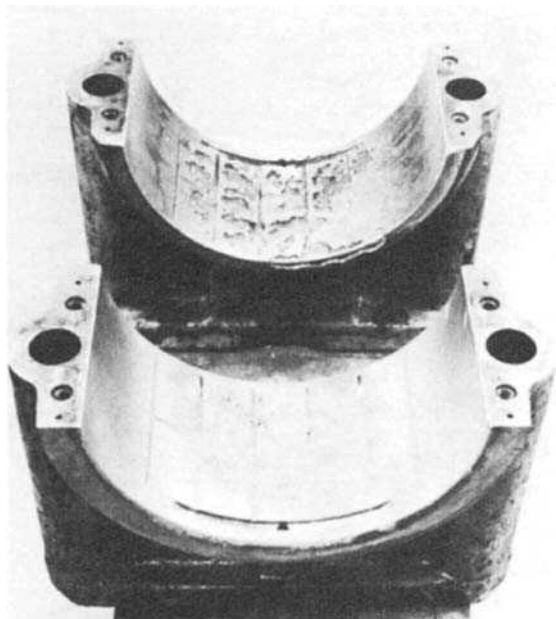


Figure 13.13 Fatigue damage in direct-lined white metal crosshead bearings

Fatigue of the galvanically applied overlay plate is shown in *Figure 13.14*. This layer is applied to the majority of medium-speed engine crankshaft bearings to accommodate oil-borne debris, misalignment and operating deflections.

13.3.3 Corrosion

The different bearing alloys suffer corrosion under different conditions.

Tin-base white metals, particularly in the slow-speed diesel engine, can form a hard, dark (almost black) brittle surface deposit when water is present in the lubricating oil. This has been found to be tin oxide, formed by conversion of the tin matrix of the lining by an electrochemical process¹⁴, and becomes thicker with time. Layers as thick as 0.25 mm have been found. Under these conditions, the bearing becomes prone to overheating and seizure. Alternatively, due to the brittle nature of the layer, it can flake off the surface, leaving a generally pitted appearance.

Copper-lead lining, if exposed to a lubricating oil which has thermally degraded to form organic acids and peroxides, or become contaminated by sulphur containing fuel oils, blow-by of products of combustion of such fuels, or cooling water with antifreeze additives, is prone to corrosion of the lead phase of the lining, be this either of the cast or sinter type. Such attack results in a porous, extremely weak, copper matrix which is easily fatigued by the dynamic loads applied to the surface (*Figure 13.15*).

Aluminium-based linings are completely resistant to engine oils, and to their high-temperature degradation products. However, in direct contact with water, a film of aluminium oxide can form in the bore, together with corrosion of the steel backing and

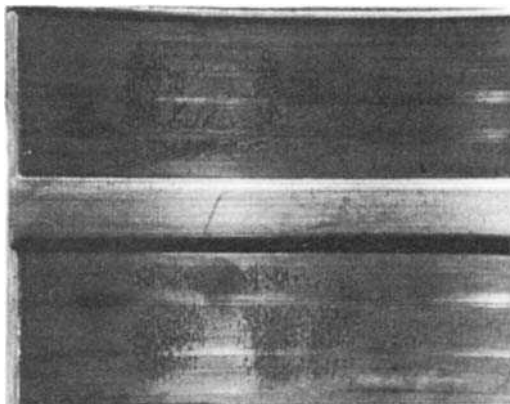


Figure 13.14 Fatigue of an overlay-plated surface

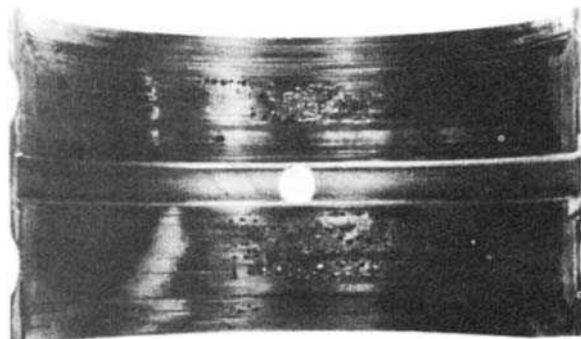


Figure 13.15 Loss of surface material caused by corrosion of the lead phase

possible lifting of the lining material, although in an oil environment in an engine, this is virtually impossible.

13.3.4 Wiping

This type of damage (*Figure 13.16*) can occur with any lining material, and is caused by insufficient lubricating and cooling oil on the bearing surface. This results in overheating and eventually melting of the lowest melting-point phase of the lining alloy. Potential causes of insufficient oil on the bearing surface include inadequate generated oil film thickness, insufficient clearance, housing distortion, restriction in oil supply system, excessively worn bearings in other locations and inadequate oil pump capacity.

13.3.5 Cavitation

Cavitation erosion damage to bearing surfaces is a form of micro fatigue cracking, initiated by the collapse of vapour cavities, and is thoroughly discussed in Reference 12. Typical damage to a large end bearing is shown in *Figure 13.17* and to a main bearing in *Figure 13.18*.



Figure 13.16 Wiping damage

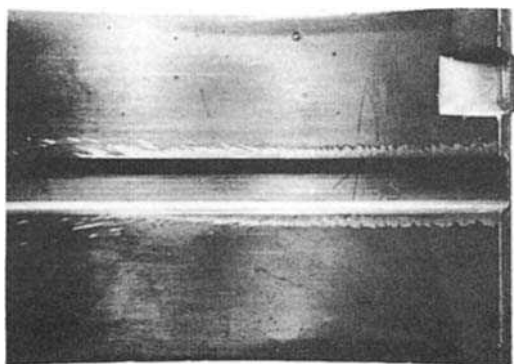


Figure 13.17 Cavitation erosion of a large end bearing

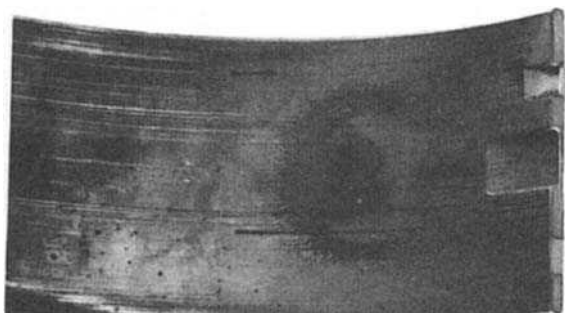


Figure 13.18 Cavitation erosion of a main bearing

13.3.6 Fretting

Figure 13.19 shows a severe case of fretting of a bearing due to insufficient contact pressure, local welding and tearing having taken place between the bearing back and housing bore, resulting in transfer of metal from one to the other. Where fretting of the joint faces of the bearing occurs, as shown in *Figure 13.20*, this usually implies inadequate joint face clamping force. Fretting of the housing joints can also be expected under these conditions.

Before fitting new bearings where fretting has occurred, all trace of fretting build-up in the housing must be removed to avoid premature catastrophic damage of the new bearing.

13.3.7 Design faults

Major design faults will not generally be evident in production engines because they would have been found out very quickly during pre-production development testing. Some faults however, do not necessarily cause problems, but require certain other conditions to exist also. An example of this would be blind location features, such as a dowel fitting in a hole in the bearing back. If positioned away from the heaviest loaded region of the bearing, this would not cause problems, unless during assembly or re-fitting the dowel was not located correctly in the hole.

A bearing load diagram which results in an almost pure rotating load pattern, i.e. where the load vector on the bearing surface travels at approximately shaft speed, can result in wiping and overheating even though the theoretical oil film thickness is

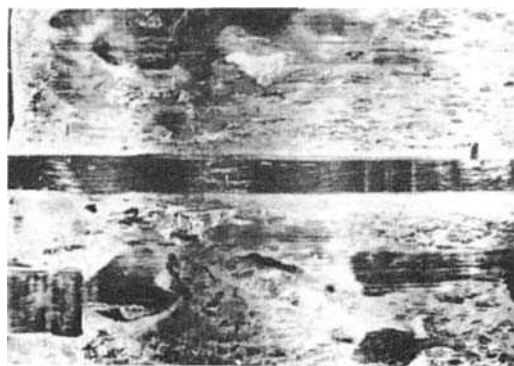


Figure 13.19 Fretting of a bearing back due to insufficient contact pressure



Figure 13.20 Fretting of a bearing joint face due to inadequate bolt load

greater on that bearing than on others in the same engine. Such a condition is more likely to exist in an engine running on very light load and full speed. The solution is to fit larger balance weights to counteract the inertia forces of the crankshaft and connecting rods. Adverse load patterns can be further aggravated by incorrect positioning of oil holes in journal surfaces, and damage due to this cause is typified by *Figure 13.21*.

13.3.8 Incorrect assembly

The commonest causes of incorrect assembly are associated with locating devices. Incorrect positioning will mean that oil feed connections are misplaced and can block off the oil feed. Incorrect location of a tang into its recess means that the bearing back will not be in contact with the housing, and the clearance between the bearing bore and the crankshaft can be lost, resulting in local overheating of the bearing surface, and possibly seizure.

Having located the bearing correctly into its housing, care must be taken to ensure that the housing bolts are correctly tensioned. Insufficient bolt load due to incorrect tightening can result in the damage shown in *Figure 13.20* and also excessive dynamic stressing of the bolts and ultimate fatigue fracture. The bolts should also be tensioned in the prescribed sequence given in the engine manual. Failure to follow the same sequence may result in a modified bore shape and unsatisfactory bearing performance.

13.3.9 Environmental factors

13.3.9.1 Electrical discharge

On generator installations, with inadequate earthing, a discharge of current can occur through the oil film between the journal surface and the bearing bore, resulting in fine, cleanly defined pitting of the bearing surface (*Figure 13.22*). Damage such as this has been found to occur with an electrical potential in excess of only 50 mV.

13.3.9.2 Journal wear ridge

Wear of the journal surface can occur due to embedded hard contaminant in the bearing surface. Less wear occurs in the region associated with grooves, gradually producing a ridge on the journal surface, which in turn causes wear of the bearing surface (*Figure 13.23*) of partially grooved bearings.

13.3.9.3 Static fretting

The damage shown in *Figure 13.24* is caused by vibration of a

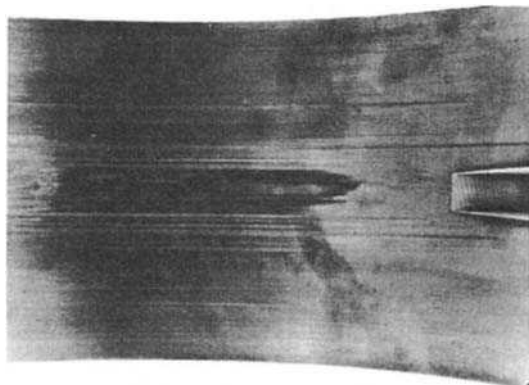


Figure 13.21 Surface damage due to incorrectly positioned oil hole



Figure 13.22 Pitted surface caused by electrical discharge

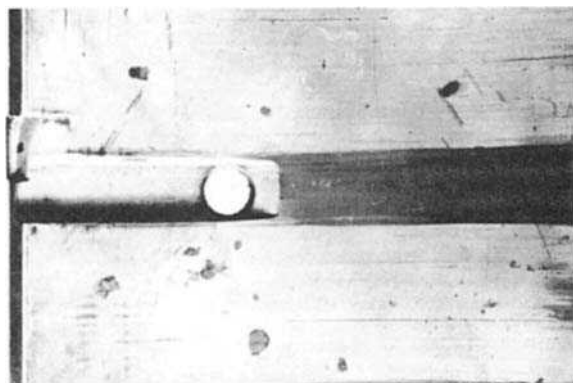


Figure 13.23 Wiping and overloading of a local region of the bore surface

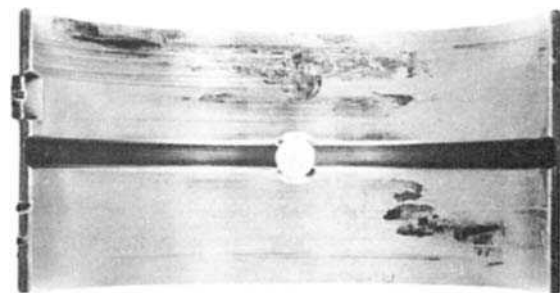


Figure 13.24 Damage to bore surface caused by crankshaft vibration

crankshaft within the bearing clearance. The term 'static' comes from the fact that such damage only occurs when the engine is not running. The damage can occur during transport of assembled engines to site. In marine installations, generating sets mounted on flexible tank tops are prone to similar damage. In such cases, the simplest palliative is to ensure that oil is pumped through the bearing clearance every few hours while the engine is inoperative, although in very severe instances it may also be necessary to redesign the mountings.

13.3.9.4 Nitride damage

The specific form of damage shown in *Figure 13.25* characterized by the vee chevrons is caused by iron nitride particles from a

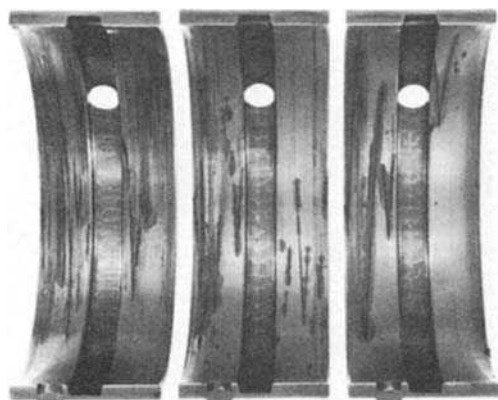


Figure 13.25 Damage caused by iron nitride particles from a nitrided journal surface

nitride-hardened shaft which has not had the thin friable layer ground off from the surface. To be sure of avoiding such damage, up to 0.02 mm needs to be removed.

13.3.9.5 Misalignment

Misalignment through whatever cause is typified by a 'D' pattern of wear, with the longest side on one end of the bearing, as *Figure 13.26*. Causes can be taper in the bearing housing or journal surface, debris trapped between shell and housing, bruising of the steel backing of the bearing shell or housing bore during assembly, taper through the bore of the bearing, or a variation in alignment of a series of main bearing bores along the engine.

13.3.10 Geometric factors

Severe wear of localized regions of the bearing surface are generally caused by specific geometric inaccuracies. Heavy wear in the axial centre region is usually caused by a 'barrelled' journal, whereas heavy wear at both ends of the bearing results from an 'hour glass' profile.

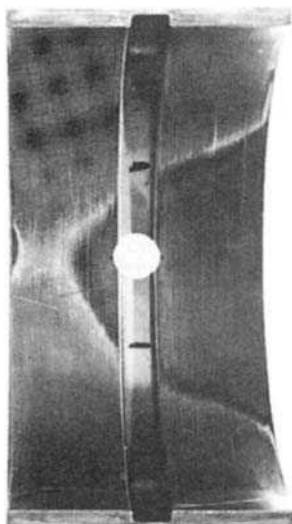


Figure 13.26 Surface wear associated with shaft misalignment

13.4 Slow-speed engine crosshead bearings

Thin shell bearings for probably the most arduously loaded of all engine bearings, the two-stroke crosshead assembly, were tested by the major engine builders in the early 1960s. Following such trials, they were gradually introduced into general production, and as loadings and conditions became even more arduous, the thin shell pre-finished bearing has become almost universally adopted in crossheads, and in several instances, in the bottom end and main bearings also.

The basic problem encountered in direct-lined white metal bearings was lining fatigue (as shown in *Figure 13.13*). Adoption of thin shell bearings permitted an improvement in metallurgical structure and strength, particularly at the steel/white-metal bond line by the use of centrifugal casting and improved uniform cooling of the constant cross-section.

Moreover, once the pre-finished thin shell is adopted, it allows advantage to be taken of the better fatigue properties of the stronger lining materials. One engine range has had 40 per cent tin-aluminium-lined bearings incorporated from inception¹⁵. The thin shell bearing has other inherent advantages. Carrying of onboard spares is much more convenient, as is the re-fitting of replacements. Spares, when required urgently, are much more easily freighted and ship turn-round time is greatly reduced compared with that for the assembly shown in *Figure 13.13*.

The bearing geometry can be much more closely controlled as a result of specialist manufacture and absence of hand fitting. Optimum performance of the crosshead bearing has been obtained with the so-called bedded-arc profile. This is a machinable form, repeatable within fine tolerance, that replaces the hand-scraped bedded-in form. The general configuration is shown in *Figure 13.27*. Over an arc of approximately 120°, symmetrical about the vertical centre line, the radial clearance between pin and bearing is almost zero. This is optimum because the crosshead bearing relies basically upon a squeeze film for lubrication. Axial grooves pitched at the swing of the crosshead pin allow oil to be swept onto the bearing surface. Small bleed grooves at

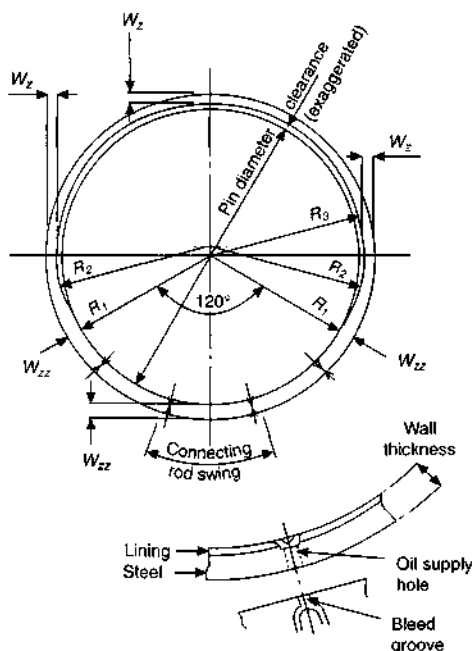


Figure 13.27 General configuration of a 'bedded arc' crosshead bearing

the ends of the main axial grooves induce a flow of oil for added cooling, and also allow lubrication of thrust faces.

Manufacturing tolerances of bearing, pin and housing create a positive clearance between pin and bearing surface, but the application of a precision electroplated overlay, usually 0.02–0.03 mm thick, of a soft lead-tin on the precision-formed bearing bore profile, allows very rapid bedding-in without localized overheating and consequential fatigue cracking, and as such is extremely beneficial, even on white metal.

A different approach has been taken by one engine builder¹⁶, using an eccentric arrangement of bearings and pins where load is transferred from one bearing land to another during the total swing of the rod and pin. The adoption of separate shell bearings for each land, with eccentricity in the wall thickness, simplifies the achievement of the required assembly.

13.5 Bearing metals

The purpose of the remaining sections of this chapter is to describe the materials used as linings for thin shell bearings, with particular emphasis again on crankshaft bearings. The most commonly used alloys are listed in Table 13.1.

To begin with, an outline is given of the properties of materials most relevant to bearing performance.

13.5.1 Fatigue strength

Lining fatigue occurs because of the cyclic nature of the loads applied to the bearings. Fatigue ratings for a range of bearing alloys are given in Figure 13.28. These results were obtained on a test rig in which a dynamic load is applied to a bearing via hydraulic resistance against an eccentric test shaft¹⁴. Results from test rigs cannot be used directly for design purposes. However, it is possible to rank materials in an order that correlates with their performance in engines, as illustrated by comparison of Figure 13.28 with Table 13.2, which contains fatigue limits of bearing materials derived from engine experience.

13.5.2 Scuff resistance

Scuffing, also known as seizure or scoring, is a result of local solid-phase welding ('pick-up') between the shaft and bearing surfaces. In extreme cases, scuffing leads to wiping and possibly to complete bearing seizure.

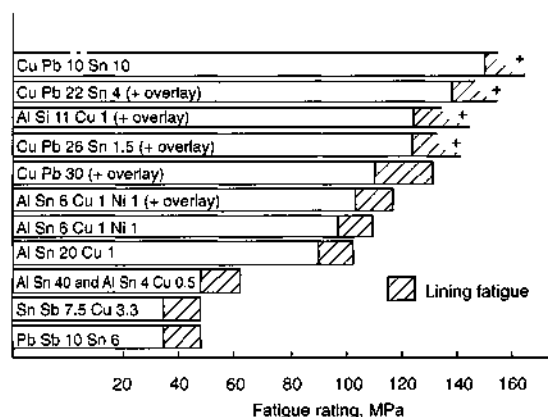


Figure 13.28 Fatigue ratings for a range of commercial bearing alloys.

Table 13.2 Recommended maximum loadings (fatigue limit) for bearing materials in slow- and medium-speed diesel engines

Material	Maximum loading (MPa)
AlSn6Cu1Ni1	38
CuPb26Sn1.5	38
AlSn20Cu1	35
AlSn40Cu0.5	20
SnSb8.5Cu3.5Cd1Cr0.1	14
SnSb7.5Cu3.3	12
PbSb10Sn6	12

The ability of a bearing material to resist scuffing depends on three factors—compatibility, conformability and embeddability. *Compatibility* is an inherent tendency of a material to resist solid-phase welding. A *conformable material*, is able to deform under conditions of shaft misalignment, thereby reducing local load concentrations and maintaining an adequate oil film thickness. A material with good *embeddability* allows hard particles to embed in the surface of the bearing, thus reducing any abrasive damage which such particles can cause both to the bearing and the shaft.

With metallic bearing materials, both conformability and embeddability are related inversely to hardness. In Table 13.1

Table 13.1 Composition and hardness of some commercial bearing alloys

Nominal composition (% by weight)	Hardness* (HV 2.5)
White-metal alloys:	
Tin/7–8% antimony/3–3.5% copper	SnSb7.5Cu3.3 27–29
Tin/8–9% antimony/3–4% copper/0.5–1.0% cadmium/0.05–0.15% chromium	SnSb8.5Cu3.5Cd1Cr 28–33
Lead/9–11% antimony/5–7% tin	PbSb10Sn6 16–20
Lead-bronze alloys:	
Copper/28–32% lead	CuPb30 46–51
Copper/24.5–27.5% lead (–2% tin)	CuPb26Sn1.5 78–82
Copper/20–24% lead/3.5–5% tin	CuPb22Sn4 98–105
Copper/9–11% lead/9–11% tin	CuPb10Sn10 124–140
Aluminium alloys:	
Aluminium/37–43% tin	AlSn40 23–25
Aluminium/37–43% tin/0.3–0.5% copper	AlSn40Cu0.5 29–31
Aluminium/18.5–24% tin/0.8–1.2% copper	AlSn20Cu1 34–41
Aluminium/5.5–7.0% tin/0.7–1.3% copper/0.7–1.3% nickel	AlSn6Cu1Ni1 38–43
Aluminium/10–11% silicon/0.8–1.2% copper	AlSi11Cu1 59–62

This is the hardness of a 0.25–0.50 mm thick lining on steel, in the form of a 50 mm diameter half-bearing. Flat strip or larger bearings have lower hardness.

typical hardness values are given for the range of bearing alloys. Compatibility is less easy to quantify; in a test used in the Glacier Metal Co., a shaft runs in a bush in a stop-start cycle with the load maintained during the stop part of the cycle. Lubrication is limited to approximately one drop of oil per minute. The load is increased until scuffing occurs, indicated by an increase in running temperature. Alignment between shaft and bearing surface is good so that resistance to scuffing depends primarily on the compatibility of the bearing material. The relative performance of bearing alloys on this test rig is shown in Figure 13.29.

Comparison of Figures 13.28 and 13.29 and Table 13.1 shows that, in general, bearing alloys with high fatigue strength have high values of hardness as well as relatively poor compatibility. Thus, selection of a bearing material is always a compromise between fatigue strength and scuff resistance. As a general guide, the least hard material is chosen which has sufficient fatigue strength for the application.

13.5.3 Wear resistance

Many wear mechanisms have been proposed for different materials¹⁷, but for engine bearings only two need be considered:

- 'severe abrasion' of the bearing and shaft by hard particles in the oil.
- 'mild abrasion' of the bearing by asperities on the shaft.

Wear rates in the two regimes may differ by a factor of up to 10^6 . Mild wear without scuffing can be beneficial if it improves conformability between shaft and bearing.

The wear rate of a material is a highly system-dependent property, and test rigs provide an unreliable guide to performance in an application. In general, a harder material has greater resistance to 'severe abrasion' but a correlation between material properties and 'mild abrasion' resistance has not been established.

13.5.4 Cavitation erosion resistance

A simple test rig for producing cavitation erosion of bearing linings has been described¹², in which cavitation is produced close to a bearing surface by an ultrasonically vibrating probe. Some results from this rig are shown in Figure 13.30 and suggest that cavitation erosion resistance depends on material type rather than mechanical properties. For each type of alloy, however, e.g. lead-bronze, increased hardness and strength produce increased cavitation erosion resistance. It should be noted that

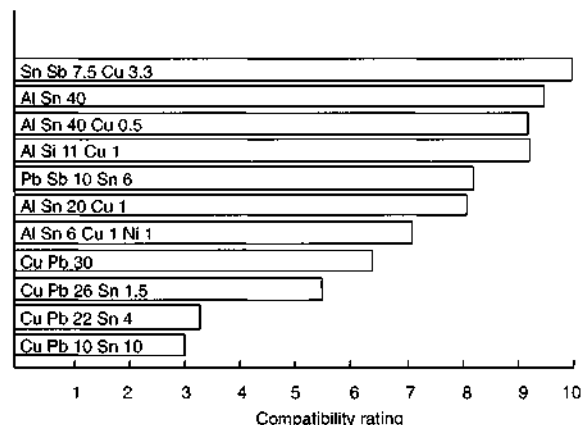


Figure 13.29 Relative compatibility ratings (Sn-based white metal=10) of bearing alloys

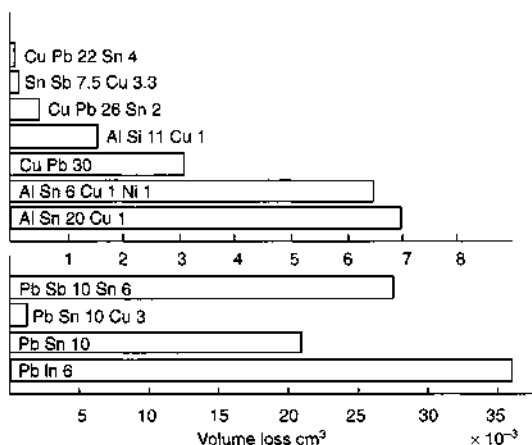


Figure 13.30 Cavitation erosion resistance of bearing alloys and overlays. Volume loss measured after 10 min. Note different scale for lead-based alloys

the results in Figure 13.30 for the overlay compositions (PbSn10Cu3, PbSn10 and PbIn6) were obtained under less severe conditions than for the other alloys. The two sets of results are not, therefore, directly comparable.

13.5.5 Overlays

Overlays are thin coatings (0.02–0.05 mm) of a soft alloy, usually lead-based. Commonly used compositions are given in Table 13.3. Their original purpose was to improve the corrosion resistance and compatibility of lead-bronze bearings although, being soft, they also have good dirt embeddability and conformability. With cast lead-bronze linings a nickel interlayer, 0.002 mm thick, is normally applied to ensure the long-term integrity of the bond between overlay and substrate. Overlays can also be applied to the full range of aluminium alloy bearings. A nickel interlayer is again used, to protect the aluminium from chemical attack during deposition of the overlay.

Overlays are electrodeposited onto the bearing surface, normally after final boring. Lead/tin and lead/tin/copper are co-deposited directly, whereas lead and indium are deposited consecutively and diffused together by a heat treatment. Typical hardness figures are given in Table 13.3. In bulk these overlay compositions are weak, but are significantly stronger in the form of thin coatings. Figure 13.31 shows how fatigue strength increases as thickness is reduced. For a given thickness, lead/indium overlays give slightly higher fatigue ratings than lead/tin or lead/tin/copper.

Overlay fatigue usually occurs at a lower load than for the substrate material, as shown in Figure 13.28. However, even a fatigued overlay provides some protection for the substrate and the fatigue strength of a bearing is always higher when overlay plated, as illustrated in Figure 13.28 for aluminium/6% tin. This does not imply, though, that overlay fatigue can be ignored. With lead-bronze bearings for example, fatigue cracks in the overlay expose lead in the substrate to corrosion.

Table 13.3 Composition and hardness of overlays

Nominal composition		Hardness (Hv 0.025)
Lead/10% tin	PbSn10	9–10
Lead/10% tin/3% copper	PbSn10Cu3	12–15
Lead/6% indium	PbIn6	8–9

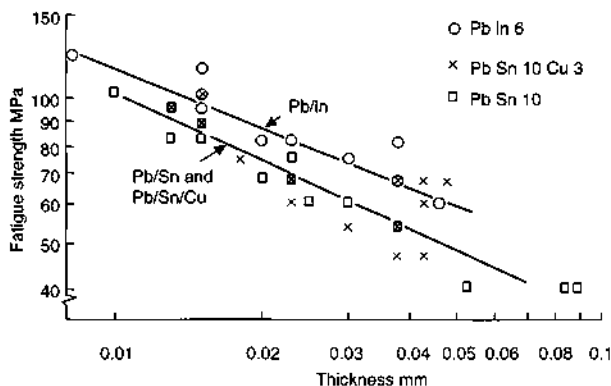


Figure 13.31 Fatigue-strength/thickness relationship for overlays

Relative wear resistance is demonstrated by measurements obtained using the dynamically loaded bearing test rig, and results are shown in Figure 13.32. Lead/tin/copper has a higher wear resistance than lead/tin, which is in turn superior to lead/indium. These results concur with those reported by Schaefer¹⁹. The relatively poor wear resistance of lead/indium negates to some extent its high fatigue rating, since for a given wear life a thicker overlay is required which leads to a lower fatigue strength.

Relative cavitation erosion rates are given in Figure 13.30. Lead/tin/copper has the highest resistance to cavitation, followed by lead/tin.

Tin or indium is necessary to improve corrosion resistance of the lead. The minimum levels required for adequate corrosion resistance are 3 per cent tin and 5 per cent indium²⁰. Although these levels are below those in the as-plated overlays, in service, tin and indium are lost by diffusion into the substrate¹⁸. The result is formation of intermetallic compounds with copper or nickel in the substrate or interlayer²¹, and corrosion of the overlay if the tin or indium level falls below the safety limit. Loss of tin is demonstrated by results shown in Figure 13.33, which were obtained by heating sintered copper-lead bearings plated with lead/10% tin. It is sometimes claimed that a nickel interlayer eliminates tin loss. However, this is not so, as shown in Figure 13.33, although the rate of loss of tin is significantly reduced by the presence of a nickel interlayer. Increasing the tin content of the overlay is not a complete solution to the tin diffusion problem, since the rate of diffusion is correspondingly increased.

Lead/tin/copper overlays are more corrosion resistant than lead/tin, possibly because the rate of tin loss is lower with lead/tin/copper as a result of the formation of copper-tin compounds

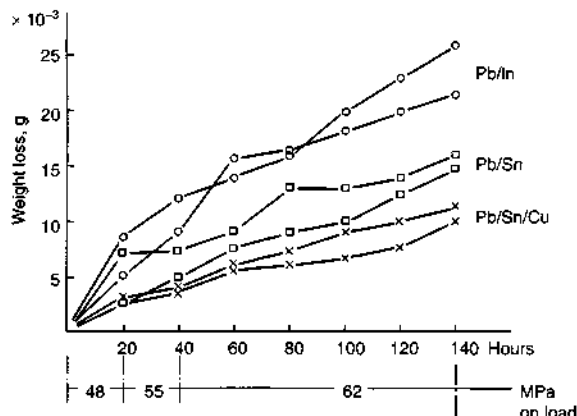


Figure 13.32 Comparative wear resistance for overlays

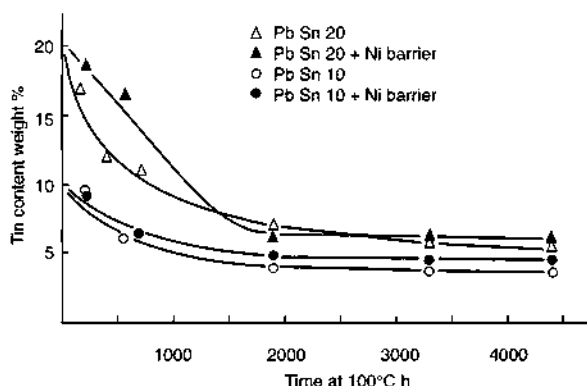


Figure 13.33 Diffusion of Sn from Pb/Sn overlays on Cu/Pb with and without Ni barrier

within the overlay. Thus, the combination of high wear resistance, cavitation erosion resistance and corrosion resistance makes lead/tin/copper the preferred overlay for the modern highly rated medium-speed diesel engine, particularly when these operate on residual fuels.

13.5.6 White metals

Tin-based white metals are alloys of tin, antimony and copper, the most commonly used composition being tin/7.5% antimony/3.3% copper. The metallurgical structure (Figure 13.34) consists of a tin-antimony matrix which is strengthened by needles of copper-tin compound.

The effects of composition and method of manufacture on the mechanical and bearing properties of white metals have been described in detail by Pratt¹⁴. The effect of cooling conditions during production of bearing linings, referred to earlier, is demonstrated by the results in Figure 13.35. Rapidly quenched material, obtained during production of thin shell bearings, has a finer copper-tin compound distribution and a higher fatigue strength than slow-cooled linings obtained when white metal is cast directly into large bearing structures.

The fatigue strength of the basic white metal composition

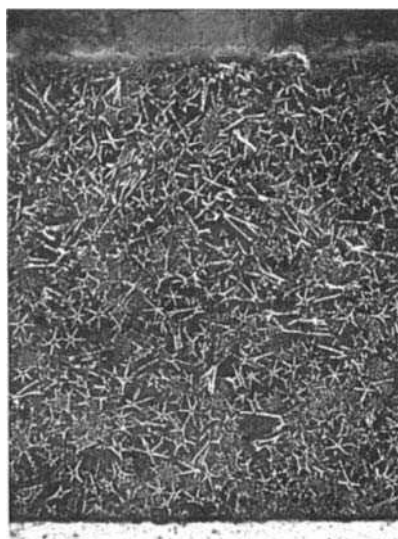


Figure 13.34 Typical structure of Sn7.5Cu3.3, showing star-shaped Cu-Sn needles in Sn-Sb matrix

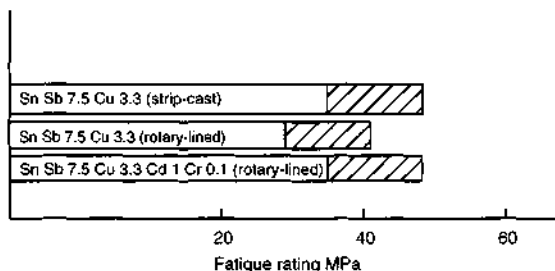


Figure 13.35 Effect of casting method and composition on fatigue strength of Sn-based white metals

can be increased by adding cadmium, which goes into solution in the matrix, and by adding trace quantities (0.1 per cent) of chromium, which refines the distribution of the copper-tin compound. By this means, it is possible to produce commercially a slow-cooled white-metal-lined bearing with a fatigue strength as high as a rapidly cooled or quenched lining (*Figure 13.35*).

Conformability and embeddability of tin-based white metals are very good, as would be expected for a material of low hardness. Corrosion resistance is good, except when the lubricating oil is contaminated by water as described earlier. However, the outstanding characteristic of tin-based white metal is its compatibility, scuffing in service being virtually unknown. In conditions of oil starvation linings suffer wiping rather than scuffing and the wiped surfaces are capable of continued operation, which is not the case with most other bearing alloys.

Lead-based white metals are basically alloys of lead, tin and antimony, containing 6–12 per cent tin and 10–15 per cent antimony. The structure of the 10 per cent antimony/6 per cent tin composition consists of a single-phase lead-tin-antimony matrix. At higher antimony contents, tin-antimony cuboids are formed. Another version contains 15 per cent antimony and 1 per cent arsenic, the structure in this case being a fine antimony-arsenic precipitate, in a lead-antimony matrix.

Cooling conditions during bearing manufacture affect mechanical properties similarly to tin-based white metal, although rapidly cooled lead-based linings must be annealed to provide adequate ductility. Likewise, minor additions of certain metals—arsenic, copper, cadmium or nickel—can increase the fatigue strength of a slow-cooled lining by refining the alloy structure.

In comparison with tin-based white metals, the lead-based alloys are softer (*Table 13.1*) and hence have even better conformability and dirt embeddability. Fatigue strength is similar (*Figure 13.28*). Tin and antimony contents are generally high enough to avoid corrosion problems in most applications. Compatibility, as assessed on the stop-start test rig described above, is not as good as tin-based white metal (*Figure 13.29*) although in service, scuffing of lead-based white metal is generally not a problem. Wear and cavitation erosion resistance of the lead-based alloys are, however, inferior.

White metals were the first alloys developed specifically for plain bearings. Although their fatigue strength is relatively low, they are still widely used on account of their surface properties. Because of the rather better corrosion and scuff resistance of tin-based white metal, it is preferred to lead-based alloys in certain critical applications, e.g. crosshead bearings. For general applications, lead-based white metal bearings are significantly cheaper than their tin-based equivalents, but even so, the tin-based alloys are, by tradition, used more widely.

13.5.7 Copper-lead and lead-bronze alloys

Historically, lead-bronze bearing linings were introduced when fatigue of white metal became a serious problem, as a result of

increased engine ratings. Modern thin shell bearings are made either by casting on to steel strip or by a sinter route, in which pre-alloyed powder is applied to steel and consolidated by sintering and rolling. All of the lead-bronze alloys in *Table 13.1* can be made into bearings by either method; copper-lead linings with 30 per cent or more lead are made only by the sinter route.

The most widely used compositions lie within the range 22–26% lead/1–2% tin. The tin goes almost exclusively into the copper, and so the structure consists of a bronze matrix containing lead islands. However, the detailed structure depends on the method of manufacture, as illustrated in *Figure 13.36* (a) and (b). Sintered linings tend to have a finer and more uniform lead distribution than cast alloys. In general, strip-cast linings are harder than sintered ones of the same composition.

The 22–26% lead/1–2% tin alloy has a high fatigue strength and is suitable for most medium- and high-speed diesel engines. It is always overlay-plated, primarily to protect the lead phase from corrosion but also because the alloy is relatively hard and incompatible (*Figure 13.29*). For medium-speed applications the overlay thickness may be as high as 0.05 mm, but where higher load-carrying capacity is required a thickness of 0.025 mm is used (see *Figure 13.31*). Hardened shafts, of minimum hardness 300 HV, are recommended.

An improvement in surface properties and dirt embeddability, at the expense of fatigue strength, is obtained with copper/30% lead. Even so, this alloy is overlay-plated for corrosion protection. Stronger linings can be made from copper/22% lead/4% tin. Such alloys can be used in engines which are too highly rated

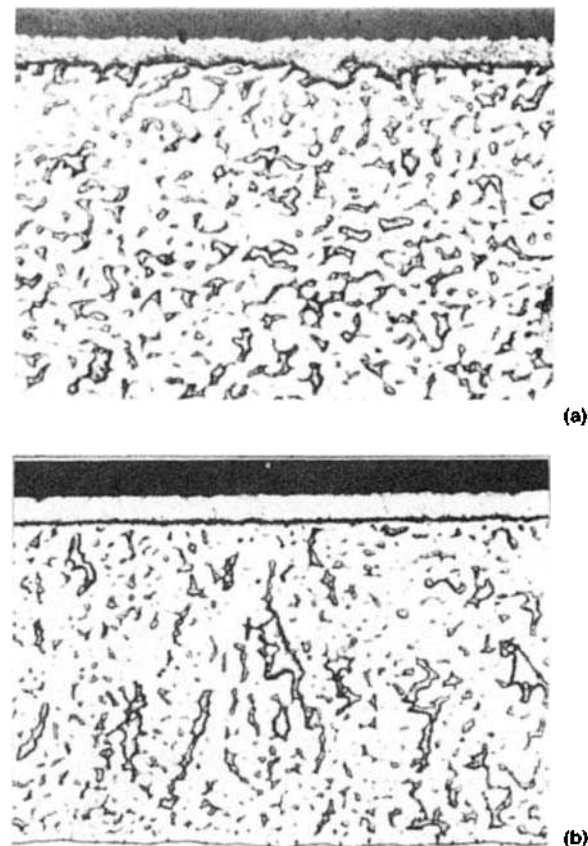


Figure 13.36 Typical structures of CuPb26Sn1.5; (a) sintered; (b) cast. The different lead distributions and matrix structures are shown

for the 26% lead/1.5% tin alloy; a thin overlay and a hardened shaft are required.

The copper/10% lead/10% tin alloy is very strong, but is too hard and prone to scuffing to be used as a crankshaft bearing. However, it is widely used for small end bushes, which are very highly loaded. In order to avoid scuffing, a hardened gudgeon pin (700 HV) is used, with a very fine surface finish.

13.5.8 Aluminium-tin alloys

The major impetus for the development of aluminium-tin bearings was their superior corrosion resistance relative to lead-bronze. Four aluminium-tin alloys are used in diesel engines, and their nominal compositions are given in Table 13.1. Bearings are made by roll bonding the alloy to steel strip via an interlayer (for aluminium-tin alloys) which is usually aluminium.

Aluminium/6% tin is a long-established bearing material²², developed originally for cast bearings. Copper and nickel are added to increase strength; copper is a solution hardener and the nickel forms a fine dispersion of nickel-aluminium compounds, as shown in Figure 13.37 (a).

Fatigue strength is slightly lower than that of the lead-bronzes (Figure 13.28), but is more than adequate for most engine applications. Compatibility is better (Figure 13.29), but the alloy is relatively hard and so is normally overlay-plated. Overlay thickness is normally in the range 0.02–0.05 mm, and shafts hardened to 250–300 HV are recommended.

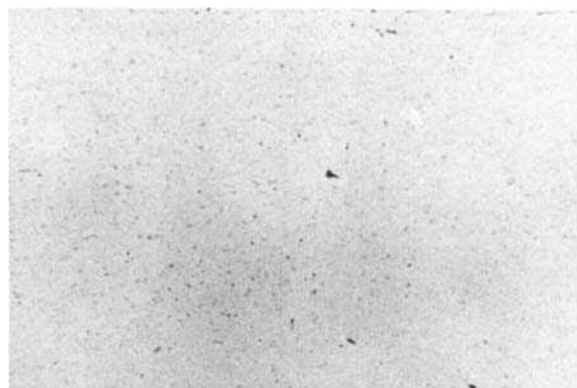
Aluminium/20% tin with a 'reticular' tin structure was developed as a more compatible version of the 6% tin alloy. The

structure (Figure 13.37 (b)) consists of an aluminium/1% copper matrix containing an interconnecting network of tin. This structure is essential if the alloy is to have adequate high-temperature strength, and is obtained by controlled cold working and heat treatment of the as-cast alloy. According to results from the stop/start test rig (Figure 13.29), the compatibility of aluminium/20% tin is superior to the 6% tin alloy, and in practice it has proved to be sufficiently scuff resistant to operate successfully without an overlay. Although its fatigue strength is marginally lower than aluminium/6% tin it is high enough to use without an overlay in the majority of high-speed engines; for medium-speed diesel engines it is usually overlay-plated and run against a shaft of minimum hardness 250 HV.

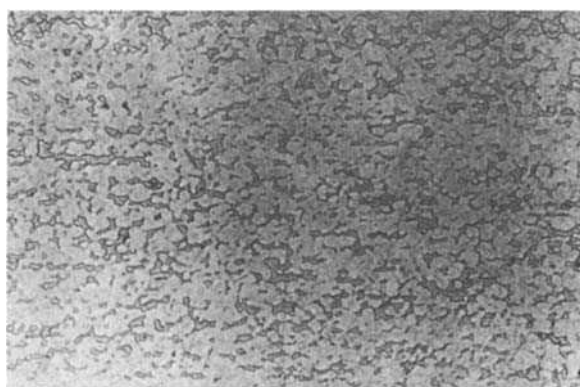
High-tin aluminium alloys, containing 40 per cent or more tin, were developed as a higher strength replacement for white metal in slow-speed diesel engines. Their use in such applications was not possible until thin shell bearings were adopted, because cast-in linings cannot be produced with aluminium alloys. Two forms are available, aluminium/40% tin which is used unplated, and, for crosshead applications, overlay-plated aluminium/40% tin/0.5% copper.

The 'reticular' tin structure and continuous aluminium matrix are also present in these high-tin alloys (Figure 13.37 (c)). As a result, aluminium/40% tin maintains its fatigue strength at elevated temperature, whereas white metal becomes significantly weaker, as shown in Figure 13.38. At typical engine operating temperatures aluminium/40% tin is approximately 30% stronger than white metal.

The compatibility of aluminium/40% tin is not as good as



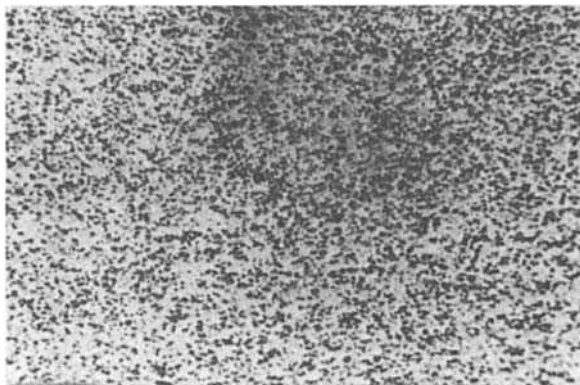
(a)



(c)



(b)



(d)

Figure 13.37 Typical structures of (a) AlSn6Cu1Ni1; (b) AlSn20Cu1; (c) AlSn40; (d) AlSi11 Cu1

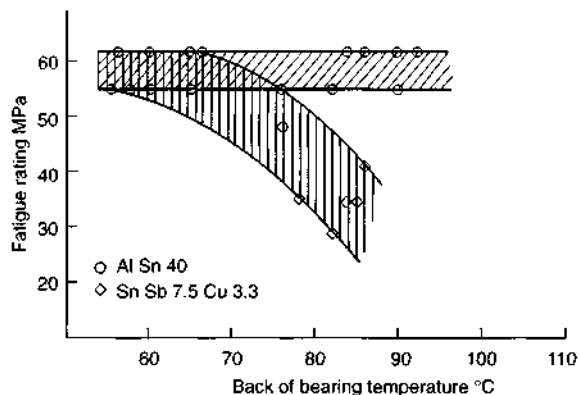


Figure 13.38 Fatigue-strength/temperature relationships for AlSn40 and SnSb 7.5Cu3.3

tin-based white metal, as assessed on the stop/start test rig. Nevertheless, the copper-free alloy does have the ability to 'wipe' under incipient scuffing conditions in the same way as white metal, and has been tested successfully without an overlay in crosshead bearings.

Aluminium containing up to 70 per cent tin has been produced experimentally, in an attempt to further improve compatibility and reduce hardness. However, at the 60 per cent level and above, tin tends to form the matrix in the finished lining, with a consequent reduction in high-temperature fatigue strength.

13.5.9 Aluminium-silicon alloys

Two aluminium-silicon compositions are used as high-strength bearing materials. The first, aluminium/4% silicon/1% cadmium, consists of a dispersion of silicon particles in an aluminium matrix; cadmium provides a soft phase to improve scuff resistance. Copper and magnesium are added, approximately 0.1 per cent of each. The magnesium forms magnesium-silicon compound during heat treatment of the alloy and a heat-treated high-strength version is available.

The second composition is aluminium/11% silicon/1% copper. The structure of this alloy is shown in Figure 13.37 (d). For maximum strength and compatibility the silicon must be finely dispersed throughout the aluminium/1% copper matrix, and this is achieved by cold working the alloy during manufacture. Bimetal is produced by roll bonding directly to steel strip. The material was developed specifically for the turbocharged high-speed diesel and its strength equates to that of the strong lead-bronze alloys also used in this application. However, the superior corrosion resistance of the aluminium-based alloy offers a major advantage over lead-bronze. In very highly loaded high-speed engines the operating temperatures of the oil may be up to 170°C, and under these conditions breakdown of oil can occur very rapidly, especially when catalysed by a copper-based bearing alloy.

In general, the compatibility of aluminium-silicon is very good considering that there is no major soft phase, and on the stop/start test rig aluminium/11% silicon/1% copper is more

compatible than lead-based white metal (Figure 13.29). This good compatibility is attributed to the non-metallic nature of the silicon which does not exhibit solid-phase welding to the shaft under scuffing conditions²³. However, the alloy is relatively hard, and must be overlay-plated. Overlay thickness is normally restricted to 0.02–0.03 mm because of the high operating loads. Hard shafts, of minimum hardness 300 Hv, are also required.

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Basics of sliding metallic-bearing materials: Part 1

This is the first of a series on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses.

George R. Kingsbury | *Motion System Design* Jun 1, 2000

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A sliding bearing is a machine element that transmits reaction forces to a shaft. A journal bearing is cylindrical (either continuous or segmented) and serves when the force on it is essentially radial. A thrust bearing is washer-shaped (either continuous or segmented) and serves where the force is axial relative to the shaft. Sliding bearings are also called plain bearings. (Some sources prefer to call them "plane" bearings.) This article deals mostly with metallic sliding bearings, though parts apply also to the growing class of nonmetallic bearings such as nylon, PTFE, and polymer composites.

A flange bearing can accommodate both radial and axial force. It is a journal bearing constructed with one or two integral thrust-bearing surfaces. Sliding of the shaft or thrust-collar surface relative to the bearing surface is characteristic of all plain bearings. In many applications, plain bearings offer advantages over rolling-element bearings, such as lower cost, smaller space needs, ability to run with marginal lubricants, corrosion resistance, and capacity for high specific loads.

The first use of a metal alloy for its special properties was probably Isaac Babbitt's adaptation of a pewter composition in 1839. Since the 1930s, developments have proliferated on all counts for metallic plain bearings. Single-metal, cast-in-place bearings have been mostly supplanted by replaceable-insert bearings of multilayer laminated metals. Bearing mechanical design is now a much more refined engineering specialty. And manufacturing technology has evolved into a complex matrix of metallurgical, chemical, and mechanical processes.

Size, configuration, manufacturing method

Sliding bearings are often classified as thin-wall (less than about 0.2-in. wall thickness) or heavy-wall. In general, bearings of diameter greater than about 6 in. are in the heavy-wall class.

Configurations are also described as half round, full round, flanged, or washer. Society of Automotive Engineers (SAE) standards classify thin-wall bearings as sleeve-type half bearings, split-type bushings, and thrust washers. Most such bearings are made by high-speed forming and machining processes from flat strip. You may sometimes hear them called "striptype" or "sheet-metal" bearings.

Heavy-wall bearings are produced in small lots, say, no more than a few thousand pieces, by more conventional machine-shop processes. Starting materials may be in flat slab or tubular form. They are sometimes called "slab" or "shellcast" bearings. Structural characteristics Sliding bearings are also often classified according to material construction. They can be solid (single-metal), bimetal (two-layer), or trimetal (three-layer) bearings. The terms refer to the number of principal layers used. Each construction is in wide use. Separate layers allow combinations of properties unobtainable with single metals.

Environments & material requirements

In bearing materials design and selection, at least some of these wear damage mechanisms must be considered:

- Surface fatigue wear.

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- Abrasive wear.
- Adhesive wear.
- Erosive wear.
- Corrosion.

Figure 1 shows examples of damage these processes cause.

Wear-damage potentials are linked intimately to interactions among system characteristics that include:

- Magnitude of bearing loads.
- Nature of loads (cyclic or steady, unidirectional or reversing).
- Speeds.
- Lubricants and lubrication system characteristics.
- Operating temperature.
- Counterface (shaft) material and finish.
- Alignment and rigidity.
- Life expectancy.

Continue on page 2

Bearing-material properties

Surface and bulk properties. The conditions at which plain bearings must operate and the wide ranges over which these conditions can vary bring up material- properties concerns of two kinds:

- Those of the bearing surface and immediate subsurface.
- Those of bulk properties.

Here are some of the more important relationships between bearing material properties and wear-damage mechanisms (material properties in *italic type*; damage mechanisms, normal type):

Compatibility: Adhesive wear.

Conformability: Adhesive wear; surface fatigue wear.

Embeddability: Abrasive wear; adhesive wear.

Fatigue strength: Surface fatigue wear.

Hardness: Extrusion; erosive wear.

Corrosion resistance: Corrosive wear.

You can think of compatibility as a purely surface characteristic. Conformability and embeddability involve the surface and immediate subsurface, and relate strongly to the bulk properties of strength and hardness. The other characteristics relate mostly to bulk properties.

Of those six characteristics, only hardness can be measured satisfactorily by standard laboratory methods. Many dynamic test rigs and methods have been developed in the plain-bearing industry to evaluate the other characteristics and their interactions. Most are designed to subject specimen bearings to conditions qualitatively similar to those of intended service, but quantitatively more severe with respect to one or more key parameters, such as:

- Load magnitude.
- Surface speed.
- Running temperature.
- Lubricant supply.
- Lubricant cleanliness.

Such tests are costly and time-consuming and should be preceded by consideration of what is really needed. Then, an appreciation of relative importance of various material characteristics in the given application and an understanding of possible trade-offs will limit practical choices to a manageable number. Fullscale tests then, can do what they should do: confirm the validity of design and material choices.

¹Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, *ASM HANDBOOK, Friction, Lubrication and Wear Technology*, ASM International, Materials Park, Ohio, 1992, pages 741-757. For ordering information about the entire book, contact ASM International, Materials Park, OH 44073-0002, ph. (216)-338-4634.

George R. Kingsbury, P.E., recently retired Senior Engineer from Glacier Vandervell Inc., a major producer of metal plain bearings, is principal of his own metallurgical engineering consulting practice in Lyndhurst (Cleveland), Ohio. He is well known in the bearing materials field as an author, lecturer, inventor, and consultant.

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Basics of sliding metallic-bearing materials: Part 2

This is the second of a series on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses. See [June 1994](#) for Part 1.

George R. Kingsbury | *Motion System Design* | Jul 1, 2000

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We continue the discussion of bearing- material properties, covering some of the more important relationships among bearing-material properties and wear/damage mechanisms.

Compatibility. This is an antiseizure and antiscore characteristic. It relates mostly to the ease with which the bearing material surface can adhere or weld to a steel or iron journal surface with pressure and heat and no lubricant or other interfering surface film. Potential for scoring and seizure exists in all boundary and thin-film lubrication conditions. There are only seven commercially significant elemental metals of good compatibility with ferrous journal surfaces: silver (Ag), cadmium (Cd), indium (In), tin (Sn), antimony (Sb), lead (Pb), and bismuth (Bi). Tin and lead show the most attractive combinations of cost, availability, and engineering properties. Both are used widely in bearing alloys because of their contributions to compatibility, either as the alloy base as in lead and tin babbitts, or as major alloy constituents as in leaded bronzes and aluminum-tin (Al-Sn) alloys.

Many nonmetals also show useful compatibility with steel. They include synthetic resins, carbon, cemented carbides, intermetallic compounds, and ceramics.

Conformability & embeddability vs. hardness & fatigue strength. Conformability and embeddability depend on yielding and plastic flow of the loaded bearing material. Therefore, soft, weak, low-modulus metals such as tin and lead have the best of these characteristics. However, harder, stronger, higher-modulus metals such as copper (Cu) and aluminum have high fatigue strength. You can get useful compromises among these sets of opposed properties by alloying to produce polyphase structures with intermediate properties. Also, you can get such compromises with layered constructions where at least one harder, stronger backing layer reinforces softer and weaker surface layers.

Bearing material microstructures. All commercially significant bearing metals except silver are polyphase alloys. Three basic microstructural types are:

- Type I — Soft Matrix with Discrete Hard Particles, [Figures 2 and 3](#). Lead and tin babbitts are of this type. These alloys have lower compatibility, conformability, and embeddability than unalloyed lead or tin — hard intermetallic and metalloid particles are present, effectively increasing bulk-strength properties.
- Type II — Interlocked Soft and Hard Continuous Phases, [Figures 2 and 3](#). Many copper-lead and leaded bronze alloys are of this type. Structures consist of continuous, mutually supporting copper and lead sponges. A large volume of lead helps compatibility. Conformability, embeddability, hardness, and strength are intermediate between those of lead and those of copper.
- Type III — Strong Matrix with Discrete Soft-Phase Pockets, [Figures 2 and 3](#). Low-lead bronzes and some aluminum alloys are of this type. Structures consist of a continuous copper-base or aluminum-base metallic matrix that contains discrete pools or pockets of lead or tin. The strength of the matrix phase dictates conformability,

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embeddability, strength, and hardness. The soft metal phase exposed at the bearing surface enhances compatibility.

Alloys containing silicon (Si), such as aluminum-silicon and aluminum-silicon-lead alloys are examples of mixed microstructures that combine Type I and Type III characteristics. Here, the aluminum matrix contains dispersed hard particles (silicon) and soft-phase (lead or tin) particles.

Bearing material design gained much with the recognition that effective load capacities and fatigue strengths of lead and tin alloys increase sharply when the alloys serve as thin layers intimately bonded to strong bearing backs of bronze or steel. Figure 4 shows the value of this principle in two-layer constructions like that of Figure 5 using a surface layer of lead or tin alloy, usually no more than 0.005 in. thick. Such a layer provides unimpaired compatibility, together with good conformability and embeddability. You can get other useful compromises between surface and bulk properties with an intermediate copper or aluminum alloy layer between the surface alloy layer and a steel back. In these three-layer constructions, surface layer thickness as low as 0.0005 in. offers even more favorable surface-and-bulk property compromises than you can get with two-layer construction.

Corrosion resistance. Bearing failure due to corrosion alone is rare. Corrosion usually interacts with mechanical and thermal factors to cause failure by fatigue or seizure in conditions the bearing could normally tolerate. You can avoid most bearing corrosion with oxidation inhibitors in commercial lubricating oils, and by periodic oil change. However, in some situations neither of these is dependable. There, use materials with inherently high corrosion resistance.

Continue on page 2

Commercially pure lead is susceptible to corrosion by fatty acids. Lead-base and copper-lead bearing alloys can corrode severely in acidic lubricating oils. Tin additions to lead above about 5% protect effectively against this kind of corrosion. Thus, tin is used extensively in lead-base bearing alloys. Acidic oils that contain sulfur attack copper and lead. This is of special concern with copper-lead and leaded bronze bearing alloys. A surface layer of a lead alloy containing tin, or a tin alloy, can provide effective protection. As long as the corrosion-resistant surface layer is intact, corrosion can't damage the underlying copper-lead alloy.

Tin and aluminum bearing alloys are substantially impervious to corrosion by oil-oxidation products. They are used extensively where the potential for lubricating-oil corrosion is high. Although lubricating-oil oxidation and contamination are the most common causes of bearing corrosion, there are others. They include seawater, animal and vegetable oils, and corrosive gases. When you select and specify a bearing material for a given application, consider the anticipated service conditions and the corrosion potentials these conditions may involve.

Thermal effects. When selecting a bearing material, it is important to consider the reduced mechanical strength of bearing liner materials at high temperature. Fatigue strength, compressive yield strength, and hardness decrease significantly with increasing temperature. The softening curves in Figure 6 show lead and tin-base bearing alloys are most severely limited; copper alloys, least.

Load capacity. For a bearing material, load capacity is the maximum unit pressure at which the material can operate without excessive friction or wear damage. Capacity ratings published for designer guidance are usually upper limits, which may be safely used only with very good lubricant-film integrity, counterface finish, mechanical alignment, and temperature control.

In cyclic load service (such as in crankshaft bearings), fatigue strength is the primary limit to load capacity. For steady loads, capacity depends more on compressive yield strength, reflected in indentation hardness. In all situations, material strength at operating temperature governs. Thus, successful operation of sliding bearings depends on temperature and its control.

You may find load-capacity ratings useful as references, but you must recognize them as imprecise and somewhat judgmental approximations. They are not guaranteed or directly measurable.

*Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, ASM HANDBOOK, Friction, Lubrication and Wear Technology, ASM International, Materials Park, Ohio, 1992, pages 741-757. For ordering information about the entire book, contact ASM International, Materials

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Basics of sliding metallic-bearing materials: Part 3

This is the third of a series on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses. Part 1 appeared in June 1994 and Part 2 in July.

George R. Kingsbury | *Motion System Design* Aug 1, 2000

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Part 2 concluded a discussion of bearing-material properties in general, including both surface and bulk properties. Here, we begin coverage of bearing-material systems, in this part covering only single-metal systems. Discussion of multilayer systems will begin in Part 4.

Bearing-material systems

Because of widely varying operating conditions for bearings, commercial bearing materials evolved as specialized engineering-material systems rather than commodity products. Total production is in small tonnages produced by just a few manufacturers. Alloy formulation and processing methods involve much proprietary technology. Successful bearing material selection often requires close user-producer cooperation.

Single-metal systems. Most singlemetal sliding bearings are made of either copper alloy or aluminum alloy. Cast zinc-base (Zn) alloys sometimes serve as lower-cost substitutes for solid bronze. Table 1 lists commercially significant alloys used as single-metal bearings.

Wide ranges of compositions and properties are available in the older copper group, some dating to the 19th Century. World War II metal shortages stimulated interest in aluminum alloys, which accelerated with the commercial introduction of aluminum-tin bearing alloys in 1946. Since then, metal economies have encouraged aluminum alloy bearings, but many designers of heavy and special-purpose machinery continue to prefer brasses and bronzes.

Single-metal systems have no outstanding surface properties, and tolerance of boundary and thin-film lubrication conditions is limited. Therefore, the load-capacity rating for a single-metal bearing is usually low relative to the fatigue strength of the material from which it was made. Because of metallurgical simplicity, single metals suit small-lot manufacturing from cast tubes or bars, by conventional machining.

- **Copper alloys.** Except for commercial bronze and low-lead tin bronze, copper alloys in single-metal systems are almost always in cast form. This provides thick bearing walls (0.125 in. and greater), strong enough so that the bearing is retained when press-fitted into the housing.

Commercial bronze and medium-lead tin bronze alloys C83420 and C83520 serve extensively in wrought strip for thin-wall bushings. They are made in large volumes by high-speed press forming. Poor compatibility of these alloys can be improved by embedding a graphiteresin paste in rolled or pressed-in indentations, so that the running surface of the bushing consists of interspersed areas of graphite and bronze. Such bushings serve in automotive-engine starting motors.

The lead in leaded tin bronze is free lead, dispersed throughout a copper-tin matrix so the bearing surface consists of interspersed areas of lead and bronze. In general, the best selection from this group of materials for a given application is the highest-lead composition you can use without risking excessive wear, plastic deformation, or fatigue damage.

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- **Aluminum alloys.** Most solid aluminum bearings in the United States are of alloys containing 5.5 to 7% tin, plus smaller amounts of copper, nickel, silicon, and magnesium. Starting forms for fabrication include cast tubes as well as rolled plate and strip, which can be press-formed into half-round shapes. As with solid bronze bearings, bearing walls are thick.

Tin in these alloys is free tin, dispersed throughout an aluminum matrix so that the bearing surface consists of interspersed areas of aluminum and tin. Free tin enhances surface properties much like free lead improves those of bronze.

Aluminum's high thermal expansion coefficient poses special problems in maintaining press-fit and running clearances. Various methods are used to increase yield strength, through heat treatment and cold work, to overcome plastic flow and permanent deformation at service temperatures and loads.

- **Zinc alloys.** Over the past two decades, zinc-aluminum-copper casting alloys replaced cast bronze alloys in some low-speed machinery bearings. The practice has advanced most in Europe.

These alloys contain no soft microconstituents that correspond to lead used in bearing bronzes and to tin in cast aluminum bearing alloys. Compatibility of the zinc-base alloys seems to derive mostly from their chemical behavior with hydrocarbon lubricants. Formation of a stable, low-shear-strength film of zinc-base soap seems important.

- **Porous metal bushings.** You can also consider oil-impregnated porous metal bushings as single-metal systems. Compositions include unleaded and leaded tin-bronze, bronze-graphite, iron-carbon, iron-copper, and iron-bronze-graphite. Oil content is 8 to 30% of total volume.

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Basics of sliding metallic-bearing materials: Part 4

This is the fourth of a series on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses. See August 1994 PTD for Part 3.

George R. Kingsbury | *Motion System Design* Nov 7, 2000

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In Part 3, we discussed single-metal bearing-material systems. Now, we cover multilayer systems.

Bimetal systems. Every bimetal system uses a strong bearing back to which a softer, weaker, thin layer of a bearing alloy is metallurgically bonded. Low-carbon steel is by far the most common bearing-back material. Alloy steels, bronzes, brasses, and (to lesser extent) aluminum alloys are also used. Table 2 classifies bimetal bearing material systems in significant commercial use.

Classes 3 and 4 in Table 2 show the strengthening effect of a steel bearing back. Compare that with the aluminum and copper alloy single-metal systems in Table 1. With steel bearing backs, load-capacity ratings for copper and aluminum alloys increase sharply over those of the corresponding single metals, without degrading any other property. Similarly, in classes 1, 2, 5, 6, and 7, strong bearingback materials permit use of lead and tin alloys that have extremely good surface properties but such low strength that they can serve as single-metal bodies only at very light load.

Classes 1 and 2 show the strengthening effect of thin-layer construction on lead and tin alloys, where a 25% load-capacity increase is gained by reducing babbitt layer thickness. Although similar behavior has been seen with aluminum and copper alloys, the thin liner effects are less pronounced. Liner thicknesses for these stronger alloys are established by metal economics and manufacturing process considerations rather than by strength-and-thickness relationships.

In Table 2, you can see deterioration in surface properties with increasing liner alloy fatigue strength by comparing classes 1 and 2 with classes 3 and 4, and by comparison within classes 3 and 4. In practice, only systems with surface property ratings of D or better succeed in boundary and thin-film lubrication.

Bronze-back bearings of classes 5, 6, and 7, Table 2, do not show performance-characteristic combinations much different from those of steel-back bearings. The practical advantages of bronze as a bearing-back material lie partly in the economics of small-lot manufacturing and partly in the ease with which rebarbbiting and remachining can salvage worn bronze-back bearings. Regarding performance, bronze beats steel as a bearing-back material in the protection it offers against catastrophic bearing seizure with severe liner wear or fatigue. The aluminum alloy bearing back in class 8 provides similar protection.

The surface properties of bronze bearing-back materials are not impressive, but they exceed those of steel, and these "reserve" properties can be important in some large, expensive machines.

Trimetal systems. Nearly every trimetal system uses a steel bearing back, a high-strength intermediate layer, and a tin alloy or lead alloy surface layer. Table 3 lists systems in commercial use. Most are derived from the bimetal systems of Table 2, classes 3 and 4, by adding a lead or tin-base surface layer.

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Strengthening effects of thin-layer construction are notable in systems that have electroplated lead alloy surface layers no more than about 0.001 in. thick, Table 3, classes 4 through 11. In comparing fatigue-strength and load-capacity ratings of these systems with those of corresponding bimetal systems, Table 2, you see that the thin lead alloy surface layer upgrades not only surface properties, but also fatigue strength. The gain in fatigue strength is at least partly due to elimination of stress raisers from which fatigue cracks can propagate.

Class 1 and class 2 trimetal systems comprise leaded bronze intermediate layers with relatively thick tin alloy surface layers. They are an evolution from bronze-back babbitt construction wherein steel has replaced most of the bronze. This produces the expected economy and bearing-back yield strength, but keeps the desirable "reserve" bearing properties of bronze-back construction.

Class 11 trimetal systems, which have silver intermediate layers, are too costly for most commercial uses. However, they offer an unequalled combination of high load capacity and corrosion resistance. They are still in limited use in aircraft radial-piston engines.

Trimetal systems with electroplated lead-base surface layers and copper or aluminum alloy intermediate layers offer the best combinations of cost, fatigue strength, and surface properties. They tolerate boundary and thin-film lubrication well, and thus can be used at higher loads than any bimetal system. Although more costly than corresponding steelback bimetal systems, they serve in some high-volume automotive uses as well as larger mobile and stationary engines. A highly developed body of mechanical, metallurgical, and chemical manufacturing technology is established in plain bearings, and it permits mass production of precision trimetal bearings without severe cost penalty.

*Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, *ASM HANDBOOK, Friction, Lubrication and Wear Technology*, ASM International, Materials Park, Ohio, 1992, pages 741-757. For ordering information about the entire book, contact ASM International, Materials Park, OH 44073-0002, ph. (216)-338-4634.

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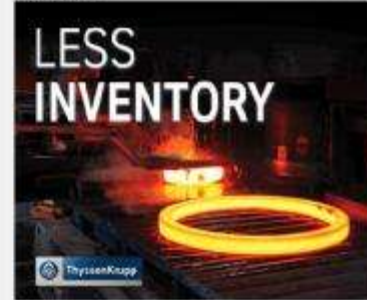
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Basics of sliding metallic-bearing materials - Part 5

This is the fifth of a series on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses. Part 1 appeared in June 1994, 2 in July 1994, 3 in Aug. 1994, and 4 in Nov. 1994.

GEORGE R. KINGSBURY | Motion System Design

Aug 7, 2009

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In Part 4, we discussed multilayer bearing-material systems. Now, we cover casting, powder-metallurgy, and roll bonding processes for manufacturing bearing-material systems in single and multilayer types.

Casting processes

Single-metal systems. Except for porous-metal, oil-impregnated bushings, all single-metal systems in Table 1 are commercially produced by casting, with or without subsequent mechanical working. Plate, strip, and sheet forms of commercial bronze, of low-lead and lead-free tin bronzes, and of aluminum-tin alloys are initially cast as ingots, slabs, or bars. Methods are static and continuous casting similar to those for other brass and aluminum mill products. Subsequent rolling and annealing operations are also similar to those of conventional mill products. Because of extreme hot shortness of leaded tin bronzes and aluminumtin alloys, they must be rolled either cold or at only slightly elevated temperature, with frequent intermediate annealing.

The recrystallized wrought structures of bronze and aluminum-tin bearing alloys differ substantially from the initial cast structures, with respect to the configurations of copper and aluminum phases and of free-lead and free-tin phases. Improvements in ductility and forming characteristics that come from these structural changes are important in subsequent bearing manufacture. The changes have no strong effect on performance properties. As-cast and wrought forms of these alloys are in commercial use and equally acceptable in bearings. Tubular and cylindrical bronze, zinc, and aluminum-tin alloy shapes are produced by static, centrifugal, and continuous casting methods, and subsequently machined into bearings. High-lead bronzes are used only as-cast, because of their low ductility and extreme hot shortness, which preclude any substantial plastic deformation of cast shapes. Cast aluminum-tin alloy tubes can withstand limited cold work, however, and in some instances cold compression of 4 to 5% increases yield strength and improves press-fit retention in finished bearings.

Bimetal systems. Specialized casting methods are widely used to produce bimetal materials in tubes and flat strips. Except for aluminum alloy systems, Table 2, classes 3 and 8, all commercial bimetal systems, at least in principle, can be produced by casting. Systems with tin and lead babbitt liners more than about 0.004 in. thick are universally cast.

- **Babbitt centrifugal casting.** Short tubular steel and bronze shapes (bearing shells) are commonly lined with tin or lead alloys by various forms of centrifugal casting. Here, a machined steel or bronze shell is preheated and coated by immersion in molten tin or tin alloy. The prepared shell is then placed in a lathe-like "spinner" and turned about its axis at controlled speed. Molten babbitt is admitted through one end and uniformly distributed around the inside wall of the shell by centrifugal action. The layer then is cooled and solidified by spraying water against the outside of the rotating shell. These processes can make fine-grain, uniformly thick liner layers, fully bonded to the steel or bronze bearing-back material. Centrifugal casting is especially suited to large-diameter, thick-wall bearings which are made in small quantities, and to full-round seamless bearings that cannot be made from flat strip.

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- **Bronze centrifugal casting.** Leaded tin bronzes also can be applied to the inner walls of steel shells by centrifugal casting. There are various methods of shell preparation, including both molten salt and controlled-atmosphere preheating. Total absence of oxidation of the steel shell's inner wall is a fundamental requirement for complete metallurgical bonding. Centrifugal casting of bronzes succeeds best with alloys containing more than about 3% tin and not more than about 20% lead. Outside this range, leaded tin bronze and copper-lead alloys are sensitive to lead segregation and consequent nonuniform "centrifuged" microstructures. But inside this range and with well-controlled process conditions, mechanically sound, well-bonded bronze layers with uniform microstructures can be produced.

- **Bronze gravity casting.** All copper-lead alloys and leaded bronzes containing up to about 35% lead can be cast in and bonded to steel shells by gravity casting, in which there is no centrifugal force. In such processes, a core usually forms an annular space inside the shell, into which molten bronze or copper-lead alloy is poured. Many such processes are in commercial use, using a variety of preheating methods, core materials, pouring methods, and quenching procedures.

As in centrifugal casting, absence of oxides on the inner shell wall is necessary for complete bond of alloy layer and steel back. Liner microstructures made by gravity shell casting generally are more uniform than those by centrifugal casting. For low-tin or high-lead compositions, or both, gravity casting is preferred; there is no centrifuging effect on the solidifying alloy.

- **Babbitt strip casting.** Steel-backed tin alloy and lead alloy bearing-strip materials are commonly produced by continuous casting. It is done on process lines in which separate cleaning, etching, hot tinning, liner-alloy casting, and quenching are carried out continuously on a moving steel bearing-back strip. In-line machining may be included so the strip emerges with closely controlled thickness, suitable for bearing fabrication.

- **Bronze strip and slab casting.** The oldest commercial processes for steelback copper-lead and leaded-bronze bearing strip also use continuous casting on a moving steel strip. Steel preheating, alloy casting, and quenching are done in a strongly reducing atmosphere to ensure freedom from oxidation. Some in-line machining also can be done, but the cast strip usually is machined in a separate line for close thickness control. There can be additional cold rolling and annealing, especially with the low and medium lead-tin bronze alloys, in which recrystallized structures are frequently preferred for their superior fabrication properties.

Strip casting of copper alloys is difficult, requiring close process control, high operator skill, and expensive special equipment. Only a few bearing manufacturers use it, but with much success. It is used not only for thin gage coiled materials, but also for heavy-gage slabs with steel thicknesses to 0.60 in.

Trimetal systems. Trimetal materials with thick surface layers, Table 3, classes 1, 2, and 3, are used mostly in large bearings. These are produced in low volumes from steel shells initially lined by casting with copper-lead alloys or bronze. After intermediate machining to remove excess liner alloy, such shells are commonly relined with tin or lead babbitt by centrifugal casting. Methods used are essentially the same as those for casting in bare steel or solid bronze shells.

Powder metallurgy processes

Single-metal systems. The only commercial use of powder metallurgy (P/M) methods for making single-metal bearing materials is in fabrication of copper-base and iron-base porous metal bushings, which are subsequently impregnated with oil. The methods are similar to those for making structural P/M shapes — that is, pressing in a closed die and sintering in a reducing atmosphere. Bars, tubes, and finished parts are made in this way. Often, post-sinter coining and re-pressing control final dimensions.

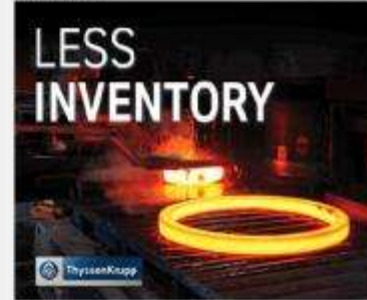
See the "Powder Metallurgy" section of the *ASM Metals Handbook*, ASM International, Materials Park, Ohio, for detailed information on this technology.

Bimetal and trimetal systems. No powder metallurgy process is in commercial use with lead-base or tin-base bearing alloys, nor is there any commercial process for lining bearing shells by powder metallurgy methods. In manufacture of steel-back copper-lead alloy and leaded bronze strip, however, powder metallurgy methods are used more extensively than any other.

Continuous sintering on a steel backing strip can produce a variety of steelback copper alloy materials, including counterparts of all cast copper-lead and leaded bronze bearing alloys, Table 2, class 4. Here, prealloyed powder particles are spread uniformly

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onto a mazing steel strip. As the strip passes through a furnace in a reducing atmosphere, the particles sinter together, forming an open grid bonded to the strip. After cooling, this bimetal is rolled to densify the liner alloy, then resintered to develop complete interparticle and alloy-to-steel bonds. After resintering, the strip may receive further rolling, to attain finish steel size and, sometimes, to strain-harden the alloy liner for higher strength.

Strip sintering permits production of steel-core "sandwich" material, which is especially suitable for applications requiring two bearing surfaces, such as in some thrust washers. Here, powder spreading, sintering, rolling, and rolling are repeated on the opposite side, after which the strip is finally resintered. Sintered strip for most automotive and truck bearings is processed in coils up to about 0.2 in. thick. Thick-wall materials with steel layers up to about 7/8 in. thick also can be processed in flat slab lengths.

Both bimetal and trimetal bearing materials also can be made by impregnation or infiltration of a sinter-melting lead alloy into a layer of sintered copper alloy powder. In impregnation, a bilayer strip of predeformed copper-lead alloy or leaded bronze powder is immersed in a molten lead-tin alloy bath heated above the melting point of lead. During immersion, the lead-tin alloy seeps some lead at the strip surface. In infiltration, the copper alloy powder layer is first sintered and not compacted after sintering. The open-grid sintered layer is then infiltrated with material having a lower melting temperature than that of the grid alloy.

The infiltrant is usually molten lead or a lead alloy, but it may be a nonmetallic material such as PTFE, which can be introduced in paste or slurry form. A useful class of self-lubricating trilayer structures is made commercially this way — where a PTFE-based infiltrant also forms a thin low-shear-strength surface film.

Powder rolling. A useful application of direct powder rolling developed commercially for plain bearings is production of an aluminum-lead alloy strip for subsequent bonding to a steel back. (See the third item under class 3, Table 2.) Here, prealloyed lead-aluminum powder and unalloyed aluminum powder are fed separately to a powder rolling mill and continuously compacted into a bilayer aluminum strip. After sintering, this strip is roll-bonded to low-carbon steel, with the unalloyed aluminum bonding layer next to the steel. This steel-back strip serves as a trimetal material for bearings, where unit load exceeds the capacity of tin or lead babbitt bimetal.

Roll bonding processes

Most commercial manufacture of trimetal aluminum alloy bearing strip materials (Table 2, class 3) is done by roll bonding the liner alloy to a steel backing strip. Both batch and continuous processes are used. Continuous processes are favored for economical, high-volume processing of lighter gages.

In all roll bonding — batch or continuous — a rolling mill forces clean aluminum and steel surfaces together at intense pressure, so solid-phase bonding (cold welding) can occur between the two metals at many interface sites. Heat may be applied simultaneously with pressure in hot rolling or subsequently in postroll annealing. It helps develop complete diffusion bonding from initial weld sites and recrystallize the aluminum alloys. Thus, the final bimetallic strip has useful liner alloy ductility and full bonding. Because of undesirable interactions between the fine tin constituent and the steel backing, tin-aluminum alloys usually are not bonded directly to steel. A layer of electrolytic nickel plating on the steel surface is commonly used to alleviate these effects with both low-tin and high-tin alloy compositions.

Another common method with tin-aluminum alloys uses a tin-free aluminum interlayer. This is done by using alclad tin-aluminum alloy strip. The tin-free cladding layer serves as the bonding surface and is present as a distinct bond interlayer in the finished bimetal strip.

Direct roll bonding to steel is used most commonly with tin-free aluminum alloys (fifth item under class 3, Table 2), and with lead-aluminum strip materials.

*Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, *ASM HANDBOOK, Friction, Lubrication and Wear Technology*, ASM International, Materials Park, Ohio, 1992, pages 711-737. For ordering information about the entire book, contact ASM International, Materials Park, OH 44073-0002, p. 62161-328-4634.

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Plated silver intermediate layers. Pure silver and silver-lead alloy bearing liners are applied to steel shells by electrodeposition from cyanide plating baths. Usually, final machining comes after plating, leaving a substantially thick layer, (typically 0.010 to 0.015 in.), of bonded silver liner material. Although as-plated thickness tolerances are not critical, special racking and masking techniques restrict plating to surfaces where it is required and to eliminate local high current densities. If the plated layer structure is to be uniform and the bond strength of the liner uniformly high, the steel basis metal must be prepared carefully, and plating-bath compositions and cleanliness must be properly controlled. The principles of silver plating of bearing liners are the same as for decorative silver plating. However, unusually thick deposits (normally more than 0.020 in.) and extremely high quality requirements for bond and plated-metal soundness have spawned several unique operating and control practices.



Tin alloys

Tin-base bearing materials (babbitts) are alloys of tin, antimony, and copper that contain limited amounts of zinc, aluminum, arsenic, bismuth, and iron. Zinc in these bearing metals generally is not favored. Arsenic increases resistance to deformation at all temperatures; zinc has a similar effect at 100 F, but causes little or no change at room temperature. Zinc has a marked effect on microstructures of some of these alloys. Small quantities of aluminum — even less than 1% — modify microstructures. Bismuth is objectionable because, in combination with tin, it forms a eutectic that melts at 279 F. Above this eutectic temperature, alloy strength decreases appreciably.

Bulk mechanical properties of ASTM grades 1 to 3 are available in "Friction and Wear of Sliding Bearing Materials," *ASM HANDBOOK: Friction, Lubrication and Wear Technology*. These properties have some value for initial materials screening comparisons among alloys. However, they are not reliable predictors of performance of thin layers bonded to strong backings, which is how most tinbase babbitts are used in modern bearing practice. Layer thickness effects in Figure 4 and temperature effects in Figure 6 are more important practical considerations than mechanical-property differences among various alloy compositions.

Compared with most other bearing materials, tin alloys have low fatigue resistance but strength is enough to warrant use at low loads. They are easy to bond and handle and have excellent antiscizure qualities.

The alloys vary in microstructure in accordance with composition. Alloys containing about 0.5 to 8% Cu and less than about 8% Sb are characterized by a solid-solution matrix in which are distributed needles of a copper-rich constituent and fine, rounded particles of precipitated SbSn. The proportion of the copper-rich constituent increases with copper content. Alloys containing about 0.5 to 8% Cu and more than about 8% Sb exhibit primary cuboids of SbSn, and needles of the copper-rich constituent in the solid-solution matrix. In alloys with about 8% Sb and 5 to 8% Cu, rapid cooling suppresses cuboid formation. This is especially so of alloys with lower copper percentages.

Lead alloys

Lead-base bearing materials (lead babbitts) are alloys of lead, tin, antimony, and in many cases arsenic. Many such alloys have been used for centuries as type metals, and were probably first used as bearing materials because of properties they were known to possess in other uses. The advantage of arsenic was recognized about 1938.

Comments in the previous section about significance of bulk mechanical properties of tin-base babbitt alloys apply equally to those of lead-base alloys.

For many years, lead-base bearing alloys were considered just low-cost substitutes for tin alloys. However, the two groups of alloys do not differ greatly in antiscizure characteristics, and when lead-base alloys are used with steel backs and in thicknesses below 0.03 in., fatigue resistance is equal to, if not better than, that of tin alloys. Bearings of any of these alloys are serviceable longest when they are less than 0.005 in. thick, Figure 4.

Without arsenic, microstructures of these alloys comprise cuboid primary crystals of SbSn, or of antimony embedded in a ternary mixture of Pb-Sb-SbSn in which lead forms the matrix. The number of these cuboids per unit volume of alloy increases with increasing antimony content. If antimony content exceeds about 15%, the total amount of hard constituents increases so much that the alloys become too brittle to use as bearing materials.

Arsenic added to lead babbitts improves mechanical properties, particularly at high temperature. All lead babbitts are subject to softening or loss of strength during prolonged exposure to temperatures at which they serve as bearings in internal-combustion engines, 200 to 300 F. Arsenic minimizes such softening. With suitable casting conditions, arsenical lead babbitts develop remarkably fine, uniform structures. They also have better fatigue strength than arsenic-free alloys.

Pouring temperature and cooling rate markedly influence lead-alloy microstructures and properties, especially when used as heavy liners for railway journals. High pouring temperatures and low cooling rates, such as those from overly hot mandrels, promote segregation and formation of a coarse structure. A coarse structure may cause brittleness along with low compressive strength and low hardness.

Overlays

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babbitt-layer thickness. Economically as well as mechanically it is hard to get consistent, thin, uniform babbitt layers bonded to bimetal shells by casting. Therefore, electroplating a thin precision babbitt layer on an accurately machined bimetal shell was developed. Special plating racks let the plated babbitt layer thickness be regulated so accurately that machining usually is not needed.

Electroplated tin alloys were found to be generally inferior to lead alloys, and only lead alloys are in commercial use as electroplated bearing overlays. The four most common compositions are SAE 191, 192, 193, and 194. Tin in Alloys 191, 192, and 193, and indium in Alloy 194 confer corrosion resistance. Tin also increases wear resistance. Copper and indium both enhance fatigue resistance.

When a tin-containing overlay is plated directly onto a copper-lead or bronze surface, the tin tends to migrate to the copper interface, forming a brittle copper-tin intermetallic compound. This decreases the corrosion resistance of the overlay and causes embrittlement along the bond line. To avoid this, a continuous barrier layer, preferably nickel about 0.0005 in. thick, is plated onto the copper alloy surface just before overlay plating. Besides providing better surface behavior, overlays improve fatigue performance of some intermediate layers by preventing cracking in this layer. Plated overlays generally range from 0.0005 to 0.002 in. thick, with fatigue life increasing markedly with decreasing overlay thickness. To take full advantage of improved fatigue life gains with thin overlays, you must minimize assembly imperfections such as misalignment, and maintain close tolerances on machined shafts and bearing bones. In adverse wear conditions, premature removal of the overlay will not necessarily impair bearing operation, because the exposed intermediate bearing alloy layer should keep working satisfactorily.

Copper alloys

Copper-base bearing alloys comprise a large family of materials with a wide range of properties. They include commercial bronze, copper-lead alloys, and leaded and unleaded tin bronzes. They are used alone in single-metal bearings, as bearing backs with babbitt surface layers, as bimetal layers bonded to steel bushings, and as intermediate layers in steel-bar and trimetal bearings, Tables 1, 2, and 3.

Pure copper is a soft, weak metal. The principal alloying element used to harden and strengthen it is tin, with which it forms a solid solution. Lead is present in cast copper-base bearing alloys as a nearly pure, discrete phase, because it has nearly no solid solubility in the matrix. The lead phase, which is exposed on the running surface of a bearing, constitutes a site vulnerable to corrosion in some operating conditions.

The anti-friction behavior of copper-base bearing alloys improves with increasing lead content, though at the same time strength is degraded because of more interruption of the continuity of the copper alloy matrix by the soft, weak lead. Thus, by judicious control of tin content, lead content, and microstructure, a large family of alloys has evolved to suit a variety of bearing applications.

Commercial bronze. Pure anti-friction properties but fairly good load capacity characterizes lead-free copper alloys. You can readily press-form wrought commercial bronze strip (SAE 705) with 0.0025 in. into cylindrical bushings and thrust washers. Cold working can increase strength of this inexpensive material.

Unleaded tin bronze. Unleaded copper-tin alloys are called phosphor bronzes because they are deoxidized with phosphorus. They serve principally in cast form as shapes for specific applications, or as rods or tubes from which solid bearings are machined. They have excellent strength and wear resistance, both of which improve with increasing tin content, but poor surface properties. They are used on bridge turntables and trunnions in contact with high-strength steel, and in other slow-moving applications.

Low-lead tin bronzes. Small amounts of lead can improve the inherently poor machinability of tin bronzes. Such additions do not significantly improve surface properties, however, and applications for these alloys are essentially the same as those for unleaded tin bronzes.

Medium-lead tin bronzes. The only wrought strip material in this alloy group is SAE 704, which is press-formed into solid bushings and thrust washers. C86800 is used in cast form as bearing backs in bimetal bearings. SAE 703 is a low-tin, medium-lead alloy that is cast or sintered on a steel back and used as a surface layer for medium-load bimetal bushings. SAE 702 is higher in tin and slightly higher in lead, cast or sintered on a steel back, and used for heavy-duty applications such as wrist-pin bushings and heavy-duty thrust surfaces.

High-lead tin bronzes. These contain medium to high amounts of tin, and high lead contents to markedly improve antifriction characteristics. SAE 794, widely used in bushings for rotating loads, has the same bronze matrix composition as SAE 793 (4.5% Sn), but three times as much free lead. It is cast or sintered on a steel back and used for somewhat higher speeds and lower loads than Alloy 793. The bronze matrix of SAE 794 is much stronger than that of a plain 75-25 copper-lead alloy. Alloy 794 can serve as the intermediate layer with a plated overlay, in heavy-duty trimetal applications such as main and connecting-rod bearings in diesel truck engines. This construction provides the highest load capacity available in copper alloy trimetals.

Copper-lead alloys. These are used extensively in automotive, aircraft, and general engineering applications. They are cast or sintered to a steel backing strip from which parts are blanked and formed into full or half-round shapes.

High-lead alloy SAE 48 can be used bare on steel or cast iron journals. Its tin content is restricted to maintain a soft copper matrix, which together with the high lead content improves the alloy's antifriction and antiseizure properties. Bare bimetal copper-lead bearings are used infrequently today because the lead phase, present as nearly pure lead, is susceptible to attack by corrosive products that can form in the crankcase lubricant between extended oil-change intervals. Therefore, most copper-base alloys with lead contents above 20%, including both SAE 48 and 49, are now used with plated overlays in trimetal bearings for automotive and truck engines.

SAE 485 is a special sintered and infiltrated composite material, produced by methods described in the preceding section, "Powder Metallurgy Processes." By these methods you can combine a strong, continuous copper alloy matrix with a high lead content; and alloy the lead-rich constituent with enough tin to resist corrosion. SAE 485 is used mostly in bushing and bearing applications involving problems in alignment, shaft surface finish, or unusual dirt contamination.

Mechanical properties of copperbase bearing alloys. Table 4 shows the ranges of mechanical strength properties of copper-base bearing alloys, according to alloy families just discussed. Indentation hardness tests provide the best indications of behavior with compressive loads, and are the only standard strength tests applicable to all alloy forms. Conventional tensile and compression testing can be done only on solid alloy bodies, which represent a small part of total copper-base bearing alloy use.

Test information of this kind helps in materials selection, as a supplement to information generated in dynamic rig tests and in real service. Except in certain solid-alloy bearings and bushing applications, alloy strength and hardness values are rarely stated as absolute specification requirements.

*Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, *ASM HANDBOOK, Friction, Lubrication and Wear Technology*, ASM International, Materials Park, Ohio, 1992, pages 741-757. For ordering information about the entire book, contact ASM International, Materials Park, OH 44073-0002, ph. (216)-338-4634.

George R. Kingsbury, P.E., recently retired as Senior Engineer from Glacier Vanderwell Inc., a major producer of metal plain bearings, is principal of his own metallurgical engineering consulting practice in Lyndhurst (Cleveland), Ohio. He is well known in the bearing materials field as an author, lecturer, inventor, and consultant.

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Basics of sliding metallic-bearing materials — Part 7

This is the seventh and last of a series* on principles and characteristics of metallic plain bearing materials and how to optimize them in given uses. Part 1 appeared in June 1994, p. 36; Part 2, July 1994, p. 32; Part 3, Aug. 1994, p. 90; Part 4, Nov. 1994, p. 56; Part 5, Aug. 1995, p. 62; Part 6, Apr. 1996, p. 51

GEORGE R. KINGSBURY | *Motion System Design* Feb 1, 2000

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In Part 6, we discussed electroplating processes along with tin alloys, lead alloys, overlays, and copper alloys. Here in this last part, we cover aluminum alloys, silver alloys, zinc alloys, other metallic bearing materials, nonmetallic bearing materials, and bearing-material selection.

Aluminum alloys

Successful commercial use of aluminum alloys in plain bearings dates to about 1940, when low-tin aluminum alloy castings began replacing solid bronze bearings for heavy machinery. Production of steel-backed strip materials by roll bonding became commercially successful about 1950, allowing development of practical bimetal and trimetal systems using aluminum alloys in place of babbits and copper alloys.

Ready availability of aluminum and its stable cost have been incentives for broadening its use in plain bearings. Aluminum single-metal, bimetal, and trimetal systems now can be used in the same load ranges as babbits, copperlead alloys, and high-lead tin bronzes. Moreover, aluminum's outstanding corrosion resistance has gained importance and led to widespread preference of aluminum alloy materials in auto engine bearings over copper-lead alloys and leaded bronzes.

In these alloys, additions of silicon, copper, nickel, magnesium, and zinc strengthen the aluminum through solid-solution and precipitation mechanisms. These elements largely control fatigue resistance and opposing properties of conformability and embeddability.

Tin and lead help upgrade the inherently poor compatibility of aluminum. Cadmium also serves as an alloy addition for this reason. Silicon helps compatibility besides its moderate strengthening effect. Although not well understood, this compatibility-enhancing mechanism is of much practical value. Silicon serves effectively in many alloys for this reason, usually along with tin, lead, or cadmium.

Conventional mechanical properties are more valuable in understanding fabrication behavior of aluminum-base bearing alloys than in predicting bearing performance. Except for solid aluminum alloy bearings — in which there is no steel back and where press-fit retention depends entirely on aluminum-alloy strength — mechanical properties of finished bearings are rarely specified, and then usually for control purposes only.

Most current commercial applications of aluminum-base bearing alloys involve steel-backed bimetal or trimetal bearings. To determine the most cost-effective aluminum material for any given application, consider the economic advantages of bimetal vs. trimetal systems. The higher cost of high-tin and high-lead alloys usually is offset by eliminating the cost of the lead alloy overlay plate. Here is a clear demonstration of the cost-effectiveness of aluminum bimetal materials: About 75% of U.S.-built passenger car engines use high-lead aluminum alloy bimetals for main and connecting rod bearings. In Europe and Japan, intermediate and high-tin aluminum bimetals are

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likewise preferred.

If you need the higher load capacity of a trimetal material, then select an aluminum liner alloy that provides adequate — but not excessive — strength. That way, you don't sacrifice conformability and embeddability unnecessarily. The tin-free alloy group offers a wide range of strength properties, and the most economical choice usually is in this group.

Silver alloys

Use of silver in bearings is largely confined to unalloyed silver electroplated on steel shells, which then are machined to close tolerances and finally precisionplated to size with a thin soft-metal overlay. The overlay may be lead-tin, lead-tincopper, or lead-indium. As a bearing material, plated silver is invariably used with an overlay. Silver on steel with an overlay is the ultimate fatigue-resistant bearing material.

Silver was widely used during and after World War II in aircraft, where its high cost was justified. As piston engines phased out, however, the use of silver in bearings greatly declined. Current applications are special, chiefly in the aircraft and locomotive industries. In view of the high cost of silver, any increase in demand for it would stimulate a search for a comparable, less costly substitute.

Zinc alloys

The zinc-base alloys that have served successfully for machinery bearings are standard zinc foundry alloys of the Zn-Al-Cu-Mg high-performance type. Tubular shapes made by conventional sand, permanent-mold, and pressure die casting methods are machined into bearings much like solid bronze bearings are. Most applications have been direct substitutions for solid bronze bearings, made mostly to reduce cost.

High compressive strength and hardness of these materials suggest greater load capacities than those of solid bronze and solid aluminum bearing materials. This is not so in practice however, largely because of the high rate at which the zinc alloys soften with increasing temperature. Maximum recommended running temperatures of 200 to 250 F for zinc-base alloys are more than 200 F less than temperature limits for copper and aluminum-base bearing alloys.

Because of low cost, zinc-base alloys will probably continue to replace copper-base alloys in some applications in construction, earthmoving, mining, and mill/machinery markets. However, technical limits with respect to high-temperature strength and corrosion resistance will prevent massive movement away from bronzes to zinc alloys.

Other metallic bearing materials

Gray cast irons. Cast irons are standard materials for some applications involving friction and wear, such as brake drums, piston rings, cylinder liners, and gears. Cast irons do well in such applications, and thus should be considered bearing materials. Gray iron bearings have been successful in refrigeration compressors where bearing pressures seldom exceed 650 psi for main bearings and 800 psi for connecting-rod bearings. Normally, journals in refrigeration compressors are either of steel, carburized and hardened to 55 to 60 Rockwell C (Rc), or of pearlitic malleable or ductile iron, hardened to 44 to 48 Rc and having a surface finish of 12 min. rms or better. Because of occasional dilution of oil with liquid refrigerant and heavy foaming of the oil, lubrication may become marginal for short periods. Fine-grain iron with uniformly distributed graphite flakes usually does well during these periods. Often, the bearings are phosphate-coated to improve seizure resistance. Such coating also creates a sponge-like surface that promotes oil retention.

For good wear resistance, gray cast iron should be pearlitic with randomly distributed graphite flakes. Cast irons have been heat treated to martensitic structures for use as cylinder liners, but benefits of such heat treatment have not been economically justifiable. Hardened cast iron has been successful in machine-tool ways.

Cemented carbides. Extremely hard materials, including cemented tungsten and titanium carbides and combinations, have been successful for various special bearing and seal applications. They have essentially no conformability and embeddability, but rank high in strength, hardness, corrosion resistance, and compatibility. They have been of most interest in high-temperature aerospace uses, but have also been useful in some machinery and machine-tool applications.

Nonmetallic bearing materials

Today, nonmetallic bearing materials are widely used. They have many inherent advantages over metals, including better corrosion resistance, lighter weight, better

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mechanical-shock resistance, and the ability to function with little or no lubricant. The major disadvantages of most nonmetallics are their high coefficients of thermal expansion and low thermal conductivity. For many years, carbon-graphites, wood, rubber, and laminated phenolics dominated nonmetallic bearing materials. In the early 1940s, development of nylon and PTFE (Teflon) gave designers two new nonmetallics with unique characteristics, especially the ability to operate dry.

A variety of polymer composites now serves bearings. Adding fiber reinforcements and fillers such as solid lubricants and metal powders to the resin matrix can significantly improve physical, thermal, and tribological properties.

Bearing-material selection

Bearing-material system selection for a given application and mechanical design for the bearing itself are interrelated processes. Neither is entirely straightforward, neither can be approached independently, and both require a good understanding of other interacting components of the machine system.

This article has considered the principles involved in bearing operation, but it hasn't tried to discuss mechanical design factors in detail. Don't expect to make final decisions on materials for specific applications based on this text alone.

Most plain-bearing manufacturers can help with mechanical design and with material selection. Because of the wide material selection most of these specialized producers offer and their broad experience in practical applications, you should take advantage of the engineering services they can provide.

*Material in this series is condensed from the chapter "Friction and Wear of Sliding Bearing Materials," by George R. Kingsbury, *ASM HANDBOOK, Friction, Lubrication and Wear Technology*, ASM International, Materials Park, Ohio, 1992, pages 741-757. For ordering information about the entire book, contact ASM International, Materials Park, OH 44073-0002, ph. (216)-338-4634.

George R. Kingsbury, P.E., recently retired as Senior Engineer from Glacier Vandervell Inc., a major producer of metal plain bearings, is principal of his own metallurgical engineering consulting practice in Lyndhurst (Cleveland), Ohio. He is well known in the bearing materials field as an author, lecturer, inventor, and consultant.

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Engine Bearing



*A guide to analysis and correction of
premature engine bearing failures*

Recommended Bearing Installation Tolerances

CRANKCASE TOLERANCES

Finish of Main Bores:

- 60-90 micro inches Ra.

Bore Tolerance:

- .001" (.025mm) up to 10.000" (250mm) bore

Out-of-Round:

- .001" (.025mm) maximum if horizontal is larger than vertical

Alignment:

- .002" (.050mm) maximum overall misalignment (.001"-.025mm for heavy duty or highly loaded engines)
- .001" (.025mm) maximum misalignment on adjacent bores (.0005"-.013mm for heavy duty or highly loaded engines)

CRANKSHAFT TOLERANCES

MAIN BEARING AND CRANKPIN JOURNALS

Finish of Journals:

- 15 micro inches Ra or better (10 micro inches Ra or better for heavy duty or highly loaded engines)

Diameter Tolerance:

- .0005" (.013mm) up to 1.500" (38mm) journal
- .001" (.025mm) for 1.500" (38mm) to 10.000" (250mm) journal

Out-of-Round:

- .0005" (.013mm) maximum up to 5.000" (125mm) journal (.0002"-.005mm for heavy duty or highly loaded engines)
- Never use a maximum out-of-round journal with a maximum out-of-round bore.

Taper:

- .0002" (.005mm) maximum up to 1.000" (25mm) long journal (.0001"-.003mm for heavy duty or highly loaded engines)
- .0004" (.010mm) maximum for 1.000" (25mm) to 2.000" (50mm) long journal (.0002"-.005mm for heavy duty or highly loaded engines)
- .0005" (.013mm) maximum for 2.000" (50mm) or longer journal (.0003"-.008mm for heavy duty or highly loaded engines)

Alignment:

- .001" (.025mm) maximum misalignment on adjacent journals (.0005"-.013mm for heavy duty or highly loaded engines)
- .002" (.050mm) maximum overall misalignment (.001"-.025mm for heavy duty or highly loaded engines)
- Crankpin and main journals should be parallel within .001" (.025mm) (.0005"-.013mm for heavy duty or highly loaded engines)

Hour-Glass or Barrel Shape Condition: Same as taper

Oil Holes must be well blended into journal surface.

CRANKSHAFT END CLEARANCE

Shaft Diameter

End Clearance

2.000"-2.750" (50mm-70mm)	.003"-.007" (.075mm-.175mm)
2.813"-3.500" (71mm-88mm)	.005"-.009" (.125mm-.225mm)
3.500" or over (89mm or over)	.007"-.011" (.175mm-.275mm)

CONNECTING ROD TOLERANCES

Finish of Rod Bores:

- 60-90 micro inches

Rod Tolerance:

- .0005" (.013mm) up to 3.250" (81mm) diameter
- .001" (.025mm) from 3.250" (81mm) to 10.000" (250mm) diameter

Out-of-Round:

- .001" (.025mm) maximum if horizontal is larger than vertical

Taper:

- .0002" (.005mm) up to 1.000" (25mm) length (.0001"-.003mm for heavy duty or highly loaded engines)
- .0004" (.010mm) for 1.000" (25mm) to 2.000" (50mm) length (.0002"-.005mm for heavy duty or highly loaded engines)
- .0005" (.013mm) for 2.000" (50mm) or longer (.0003"-.008mm for heavy duty or highly loaded engines)

Hour-Glass or Barrel Shape Condition: Same as taper

Parallelism between rod bore and wrist pin hole .001" (.025mm) in 5.000" (125mm)

Twist .001" (.025mm) in 6.000" (150mm)

CONNECTING ROD END CLEARANCE

Fillets at end of crankpin should not bind on ends of rod bearing, .004" (.10mm) to .010" (.25mm) clearance recommended.

BEARING SPREAD

Main bearings: .005" (.13mm) to .020" (.50mm) in excess of crankcase bore diameter

Connecting rod bearings: .020" (.50mm) in excess of rod bore

OIL CLEARANCE - RESIZED BEARINGS

The oil clearance shown in this catalog are for the factory manufactured precision sizes. When installing a resized bearing, adjust the oil clearance shown as follows:

For babbitt and TM-77 copper-lead:

Add .0004" (.010mm) to both low and high limit

For TM-112 copper-lead:

Add .0008" (.020mm) to low limit and .0004" (.010 mm) to high limit

PIN BUSHINGS

Resizing:

Light Ream: .007"/.015"

Bore: .015"/.030"

As you know, every automotive engine part will eventually wear out. And if every part always performed for the full length of its expected life, your job would be fairly simple. . . to replace parts that have worn. Unfortunately, we cannot always count on an engine part failing only because its normal lifespan is exceeded. A mechanic must not only be a “replacer of parts” but, like a doctor, he must be capable of diagnosing his “patient” to determine why a part failed prematurely. The table below lists the eight major causes of premature engine bearing failure, along with percentage figures which indicate how often each has been found to be the prime contributor to a bearing’s destruction. However, it is important to note that in many cases a premature bearing failure is due to a combination of several of these causes.

MAJOR CAUSES OF PREMATURE BEARING FAILURE

Dirt	45.4%
Misassembly	12.8%
Misalignment	12.6%
Insufficient Lubrication	11.4%
Overloading	8.1%
Corrosion	3.7%
Improper Journal Finish	3.2%
Other	2.8%

Thus we can reason that if a mechanic merely replaces a damaged bearing in an engine, without determining the cause of its failure, more than 99% of the time he will be subjecting the replacement bearing to the same cause that was responsible for the original failure. What this all means is that just as a doctor cannot cure a patient until he has determined what ails him, so, too, a mechanic cannot correct the cause of premature bearing failure until he first determines what causes the failure.

The pages of this manual are organized, for your convenience, into four major subjects:

1. **Appearance** – an illustration and brief description of a bearing that has failed due to a specific cause.
2. **Damaging Action** – what actually damaged the bearing under the conditions which were present.
3. **Possible Causes** – a listing of those factors capable of creating the particular damaging action.
4. **Corrective Action** – the action that should be taken to correct the cause of failure.

We believe you will find this reference manual easy to read and use, and that it will be very helpful to you in properly determining the cause of premature bearing failure.



NORMAL WEAR PAGE 1



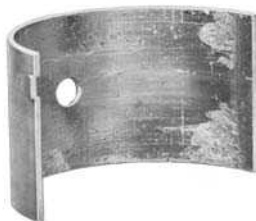
FOREIGN PARTICLES IN LINING PAGE 3



**FOREIGN PARTICLES
ON BEARING BACK PAGE 4**



SHIFTED BEARING CAP PAGE 6



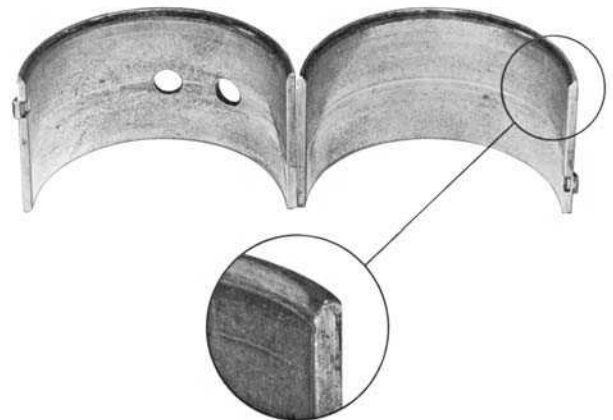
EXCESSIVE CRUSH PAGE 7



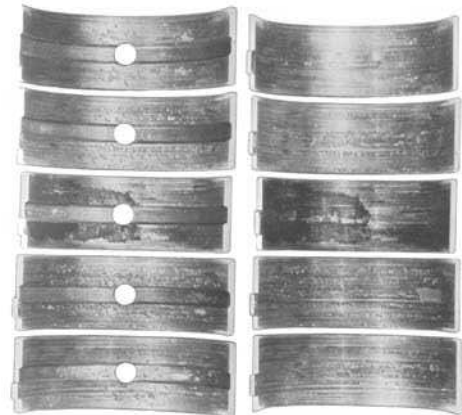
INSUFFICIENT CRUSH PAGE 8



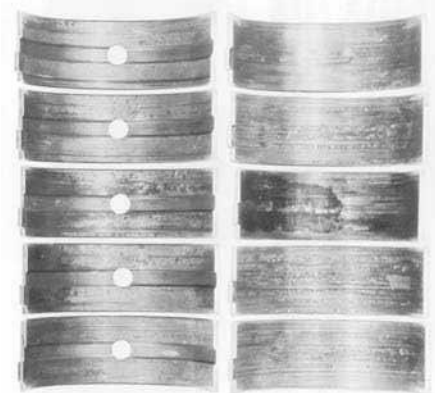
**BENT OR TWISTED
CONNECTING ROD PAGE 9**



FILLET RIDE PAGE 10



DISTORTED CRANKCASE PAGE 11



BENT CRANKSHAFT PAGE 12



OUT-OF-ROUND BOREPAGE 13



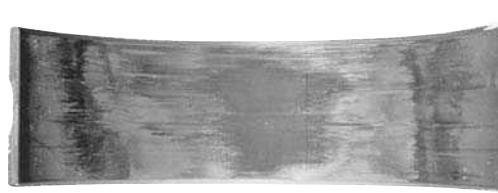
HOT SHORTPAGE 16



OUT-OF-SHAPE JOURNALPAGE 14



SURFACE FATIGUEPAGE 17



CORROSIONPAGE 18



OIL STARVATIONPAGE 15

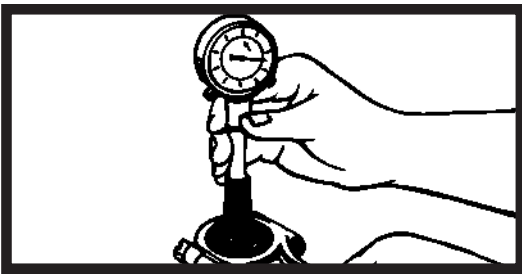


ACCELERATED WEARPAGE 19

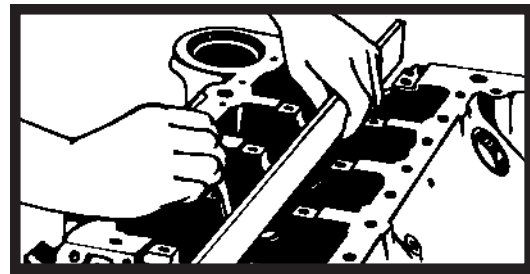


Uniform wear pattern over approximately 2/3 of the bearing's surface. Wear should diminish near the parting line ends of the bearing, and the wear pattern should extend uniformly across the bearing in the axial direction.

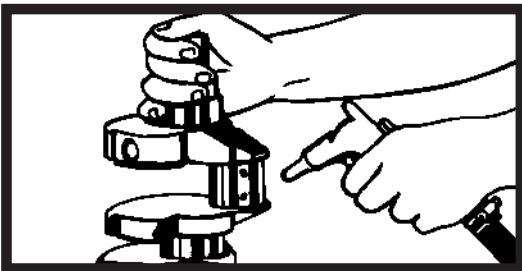
CRANKSHAFT & ENGINE BEARING INSTALLATION GUIDELINES



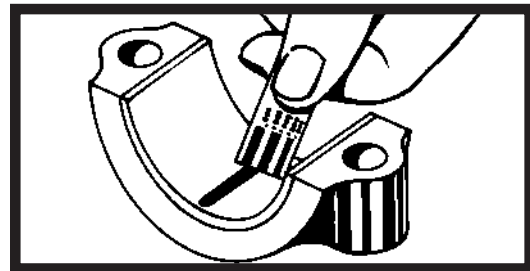
1. Check all connecting rod housing bores for taper, roundness and size, using a bore gauge or inside micrometer. Check for parallelism between the large and small ends of rod. Check condition of bolts and threads.



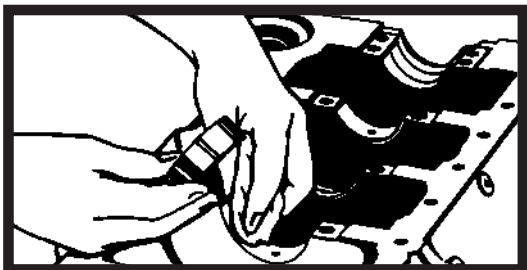
2. Check main bearing bores for alignment, taper, roundness and size. Check condition of bolts and threads.



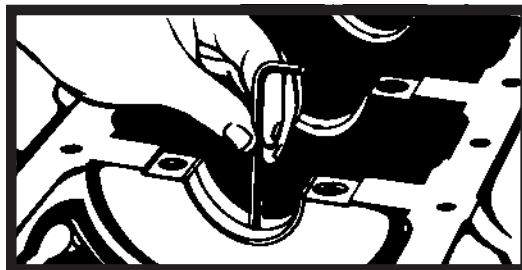
3. Make sure that engine block and crankshaft are clean, checking oil passageways for hidden dirt and debris. Hot water and detergent work best for cleaning.



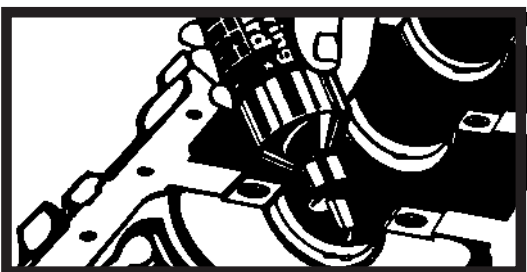
4. While installing both connecting rod bearings and main bearings, check for proper oil clearance, using Clevite 77[®] Plastigage[®].*



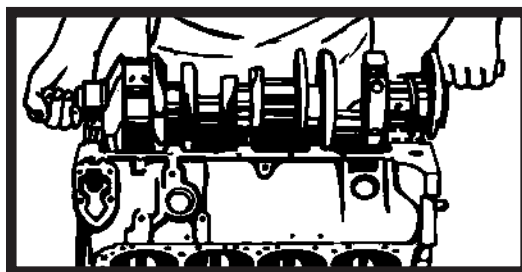
5. With engine block upside down, install the upper main halves, making sure bearing backs and bores are clean and dry.



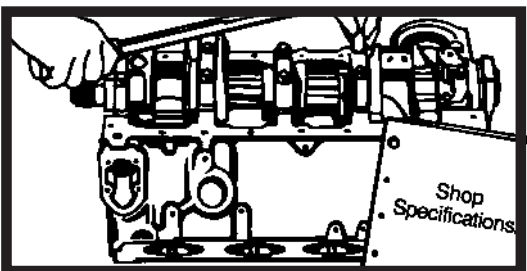
6. Check oil hole alignment to insure proper oil flow.



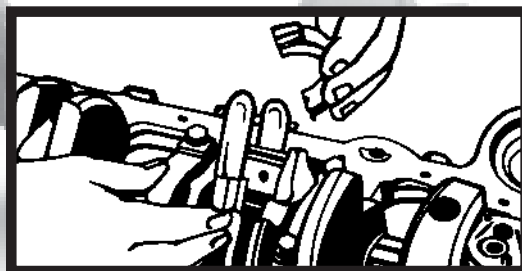
7. Lubricate all bearings and seal lip surfaces using Clevite 77® Bearing Guard**. Bearing Guard can be used on any internal engine component as an assembly lube.



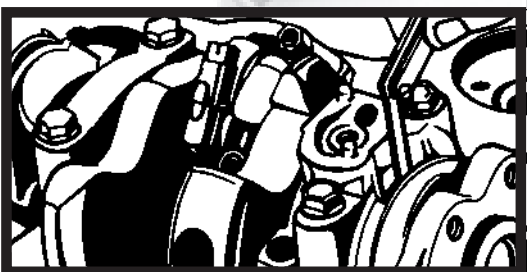
8. Place the crankshaft squarely into the main bearings and assemble lower main bearings and caps. Check to insure proper position.



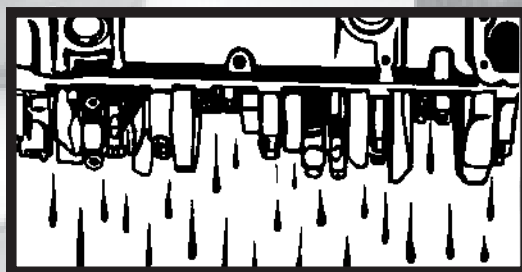
9. Pry crankshaft squarely into the main bearings and assemble lower main bearings and caps. Check to insure proper position.



10. Assemble the connecting rod and piston assembly to the crankshaft, using Clevite 77® Bolt Boots[†] to protect crankshaft journals and cylinder walls. Make sure caps match correct rods, and torque bolts to proper OE specifications.



11. With crankshaft forced all the way forward in block, check end clearance between the crankshaft thrust face and the bearing flange to OE specification, using a feeler gauge of proper thickness.



12. Prime engine oiling system before starting engine to assure operation of oil pump and lubrication system immediately upon start-up.

* Part #MPG-1,.001"-.003"; #MPR-1,.002"-.006"; #MPB-1,.004"-.009"; #MPY-1,.009"-.020" • ** Part #2800B2 - 8 oz. bottle; #2800B4 - 1 gal. jug; #2800B5 - 1 1/3 oz. pkg. • † Part # 2800B1

Foreign Particles in Lining



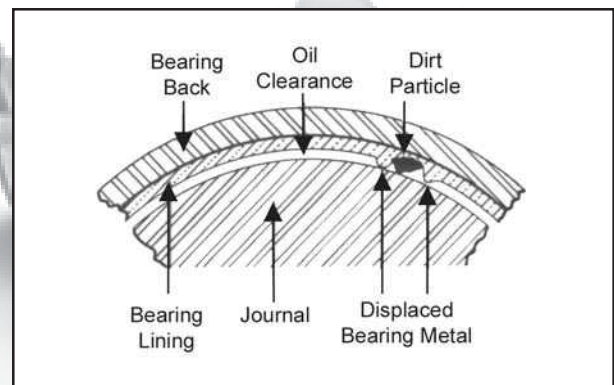
APPEARANCE

Foreign particles are embedded in the lining of the bearing. Scratch marks may also be visible on the bearing surface.

DAMAGING ACTION

Dust, dirt, abrasives and/or metallic particles present in the oil supply embed in the soft babbitt bearing lining, displacing metal and creating a high-spot.

The high-spot may be large enough to make contact with the journal causing a rubbing action that can lead to the eventual breakdown and rupture of the bearing lining. Foreign particles may embed only partially and the protruding portion may come in contact with the journal and cause a grinding wheel action.



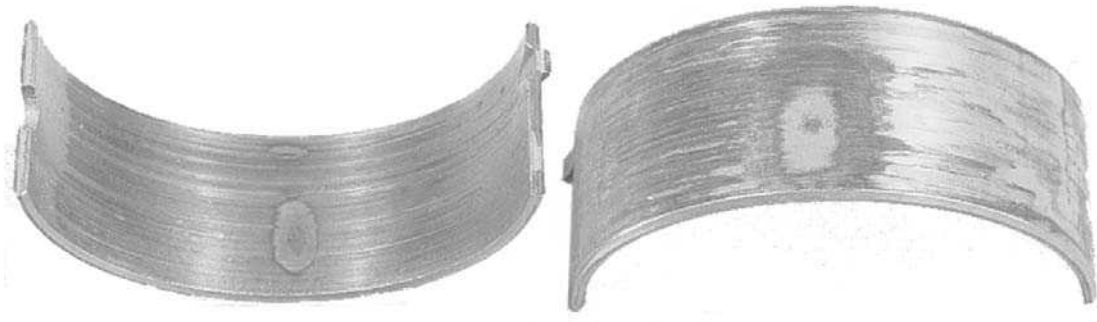
POSSIBLE CAUSES

1. Improper cleaning of the engine and parts prior to assembly.
2. Road dirt and sand entering the engine through the air-intake manifold or faulty air filtration.
3. Wear of other engine parts, resulting in small fragments of these parts entering the engine's oil supply.
4. Neglected oil filter and/or air filter replacement.

CORRECTIVE ACTION

1. Install new bearings, being careful to follow proper cleaning procedures.
2. Grind journal surfaces if necessary.
3. Recommend that the operator have the oil, air filter, oil filter and crankcase breather-filter replaced as recommended by the manufacturer.

Foreign Particles on Bearing Back



APPEARANCE

A localized area of wear can be seen on the bearing surface. Also, evidence of foreign particle(s) may be visible on the bearing back or bearing housing directly behind the area of surface wear.

DAMAGING ACTION

Foreign particles between the bearing and its housing prevent the entire area of the bearing back from being in contact with the housing base. As a result, the transfer of heat away from the bearing surface is not uniform causing localized heating of the bearing surface which reduces the life of the bearing.

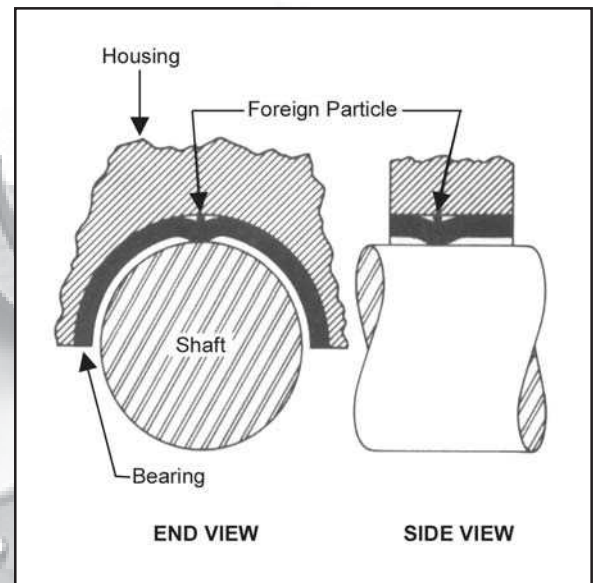
Also, an uneven distribution of the load causes an abnormally high pressure area on the bearing surface, increasing localized wear on this material.

POSSIBLE CAUSES

Dirt, dust abrasives and/or metallic particles either present in the engine at the time of assembly or created by a burr removal operation can become lodged between the bearing back and bearing housing during engine operation.

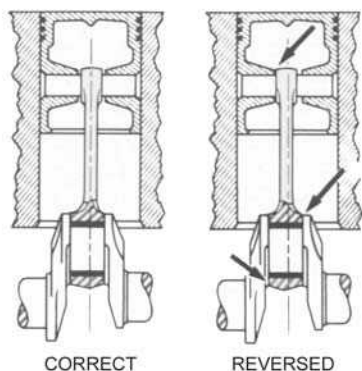
CORRECTIVE ACTION

1. Install new bearings following proper cleaning and burr removal procedures for all surfaces.
2. Check journal surfaces and if excessive wear is discovered, regrind.

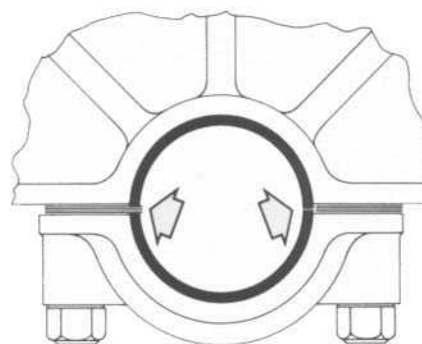


Engine bearings will not function properly if they are installed wrong. In many cases, misassembly will result in premature failure of the bearing.

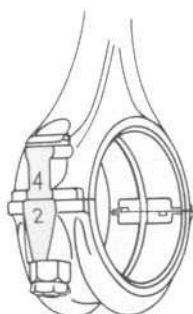
The following are typical assembly errors most often made in the installation of engine bearings.



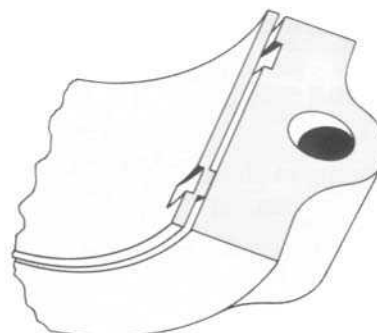
Position of Offset
Connecting Rod Reversed



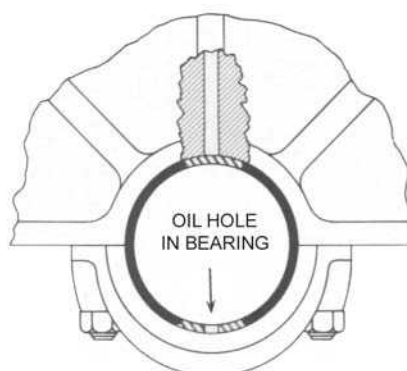
Shims Improperly
Installed



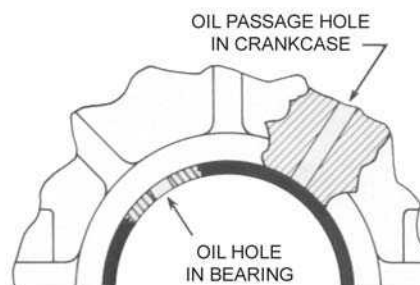
Bearing Caps in Wrong
or Reversed Position



Locating Lugs Not Nested



Bearing Halves Reversed



Bearing Oil Hole
Not Aligned With Oil
Passage Hole

Shifted Bearing Cap



APPEARANCE

Excessive wear areas can be seen near the parting lines on opposite sides of the upper and lower bearing shells.

DAMAGING ACTION

The bearing cap has been shifted, causing one side of each bearing-half to be pushed against the journal at the parting line.

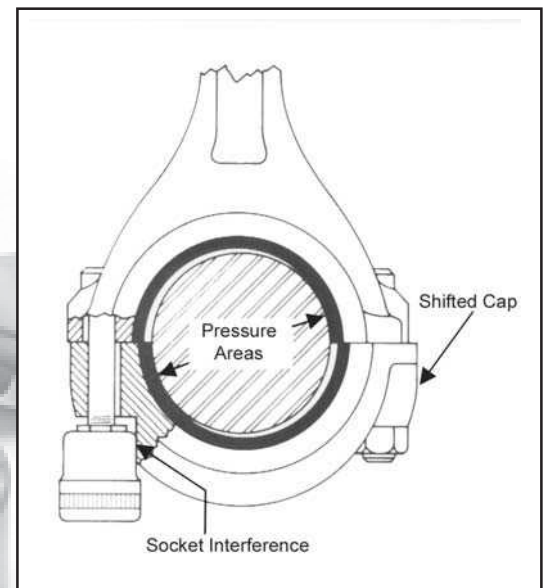
The resulting metal-to-metal contact and excessive pressure cause deterioration of the bearing surface and above normal wear areas.

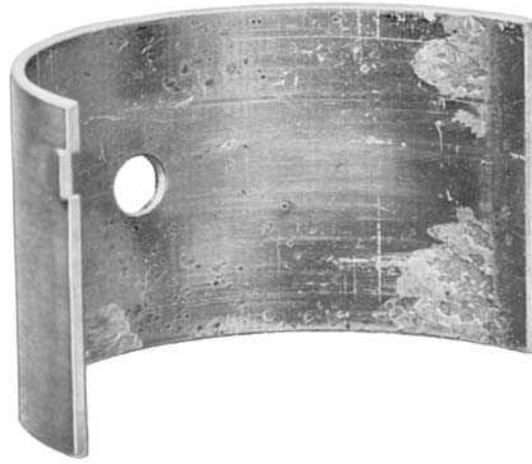
POSSIBLE CAUSES

1. Using too large a socket to tighten the bearing cap. In this case, the socket crowds against the cap causing it to shift.
2. Reversing the position of the bearing cap.
3. Inadequate dowel pins between bearing cap and housing (if used), allowing the cap to break away and shift.
4. Improper torquing of cap bolts resulting in a "loose" cap that can shift positions during engine operation.
5. Enlarged cap bolt holes or stretched cap bolts, permitting greater than normal play in the bolt holes.

CORRECTIVE ACTION

1. Check journal surfaces for excessive wear and regrind if necessary.
2. Install the new bearing being careful to use the correct size socket to tighten the cap and the correct size dowel pins (if required).
3. Alternate torquing from side to side as to assure proper seating of the cap.
4. Check the bearing cap and make sure it's in its proper position.
5. Use new bolts to assure against overplay within the bolt holes.





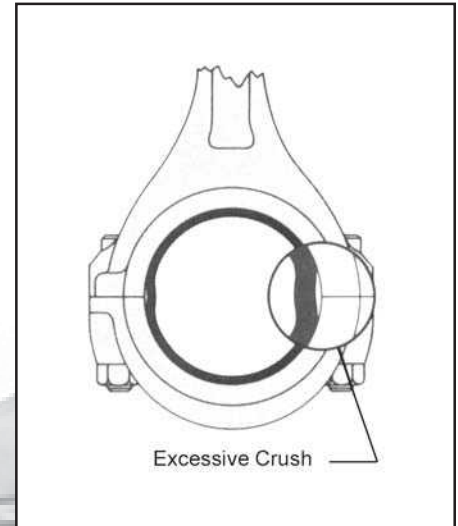
APPEARANCE

Extreme wear areas visible along the bearing surface adjacent to one or both of the parting line.

DAMAGING ACTION

Before the bearing cap is assembled, a small portion of the bearing extends just a little beyond the edge of the bearing housing. Thus when the bearing cap is tightened into place, the bearing is forced against the bearing housing. That portion of the bearing which extends beyond the housing is called “crush”.

When there is too much crush, however, the additional compressive force created by the surplus crush that still remains after the bearing is fully seated causes the bearing to bulge inward at the parting faces. This bearing distortion is called “side pinch.”

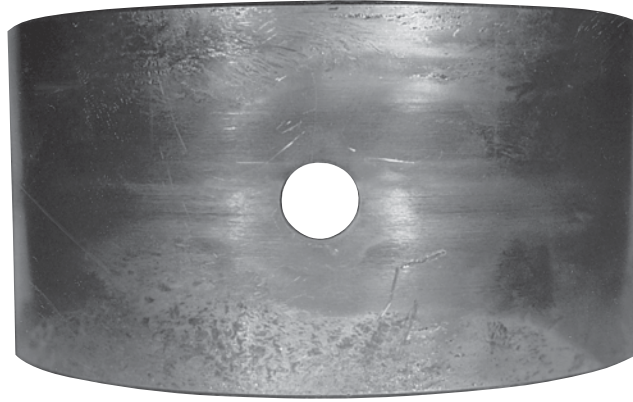


POSSIBLE CAUSES

1. The bearing caps were filed down in an attempt to reduce oil clearance.
2. The bearing caps were assembled too tightly due to excessive torquing.
3. Not enough shims were utilized (if shims were specified).

CORRECTIVE ACTION

1. Rework the bearing housing of the engine block if it has been filed down.
2. Replace the connecting rod if its bearing cap has been filed down.
3. Check journal surfaces and regrind if necessary.
4. Install the new bearing and follow proper installation procedures by never filing down bearing caps and using the recommended torque wrench setting.
5. Correct the shim thickness (if applicable).
6. Check for out-of-roundness of the inside diameter of the assembled bearing by means of an out-of-roundness gauge, inside micrometer, calipers or prussian blue to assure that any out-of-roundness is within safe limits. The maximum assembled bearing I.D. should always be across the split line.



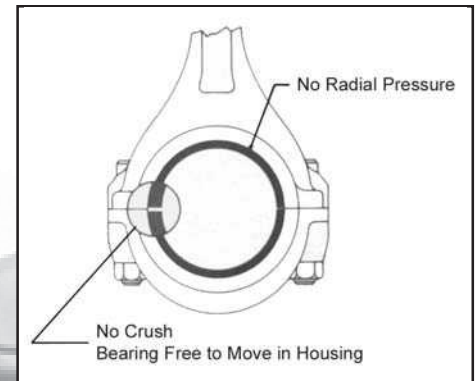
APPEARANCE

Highly polished areas are visible on the bearing back and/or on the edge of the parting line. Areas of pock marks or build-up due to metal transfer between bearing and housing. This is commonly referred to as “fretting”.

DAMAGING ACTION

When a bearing with insufficient crush is assembled in an engine, it is loose and therefore free to work back and forth within its housing.

Because of the loss of radial pressure, there is inadequate contact with the bearing housing, thus impeding heat transfer away from the bearing. As a result, the bearing overheats causing deterioration of the bearing surface.



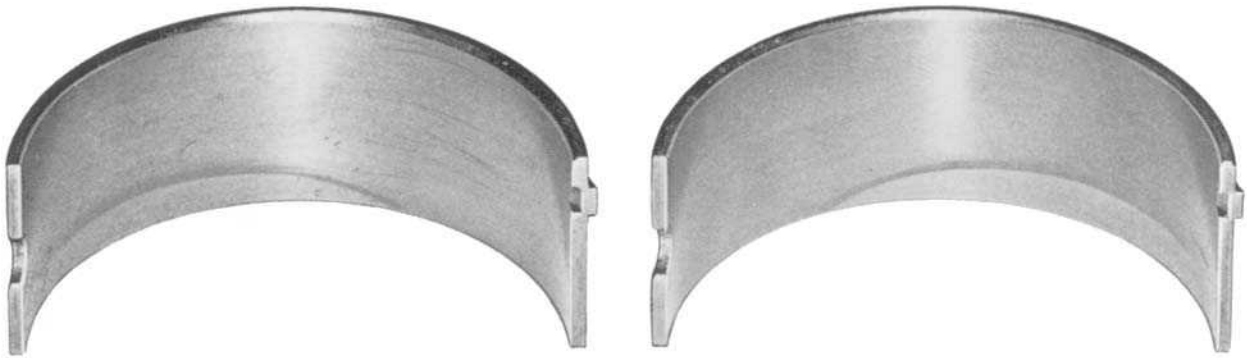
POSSIBLE CAUSES

1. Bearing parting faces were filed down in a mistaken attempt to achieve a better fit, thus removing the crush.
2. Bearing caps were held open by dirt or burrs on the contact surface.
3. Insufficient torquing during installation (be certain bolt doesn't bottom in a blind hole).
4. The housing bore was oversize or the bearing cap was stretched, thus minimizing the crush.
5. Too many shims were utilized (if shims are specified).

CORRECTIVE ACTION

1. Clean mating surfaces of bearing caps and inspect for nicks and burrs prior to assembly.
2. Check journal surfaces for excessive wear and regrind if necessary.
3. Check the size and condition of the housing bore and recondition if necessary.
4. Correct shim thickness (if applicable).
5. Install new bearings using correct installation procedures (never file bearing parting faces).

Bent or Twisted Connecting Rod



APPEARANCE

Excessive wear areas can be seen on opposite ends of the upper and lower connecting rod bearing shells. The wear is localized on one portion of the bearing surface with little or no wear on the remainder.

DAMAGING ACTION

A bent or twisted connecting rod results in misalignment of the bore, causing the bearing to be cocked so the bearing edge makes metal-to-metal contact with the journal. These metal-to-metal contact areas cause excessive wear on the bearing surface.

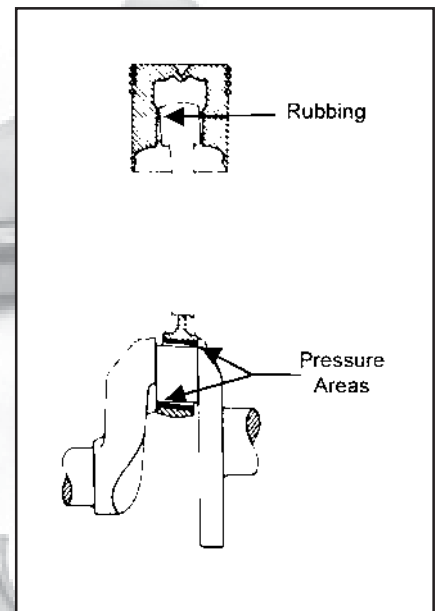
POSSIBLE CAUSES

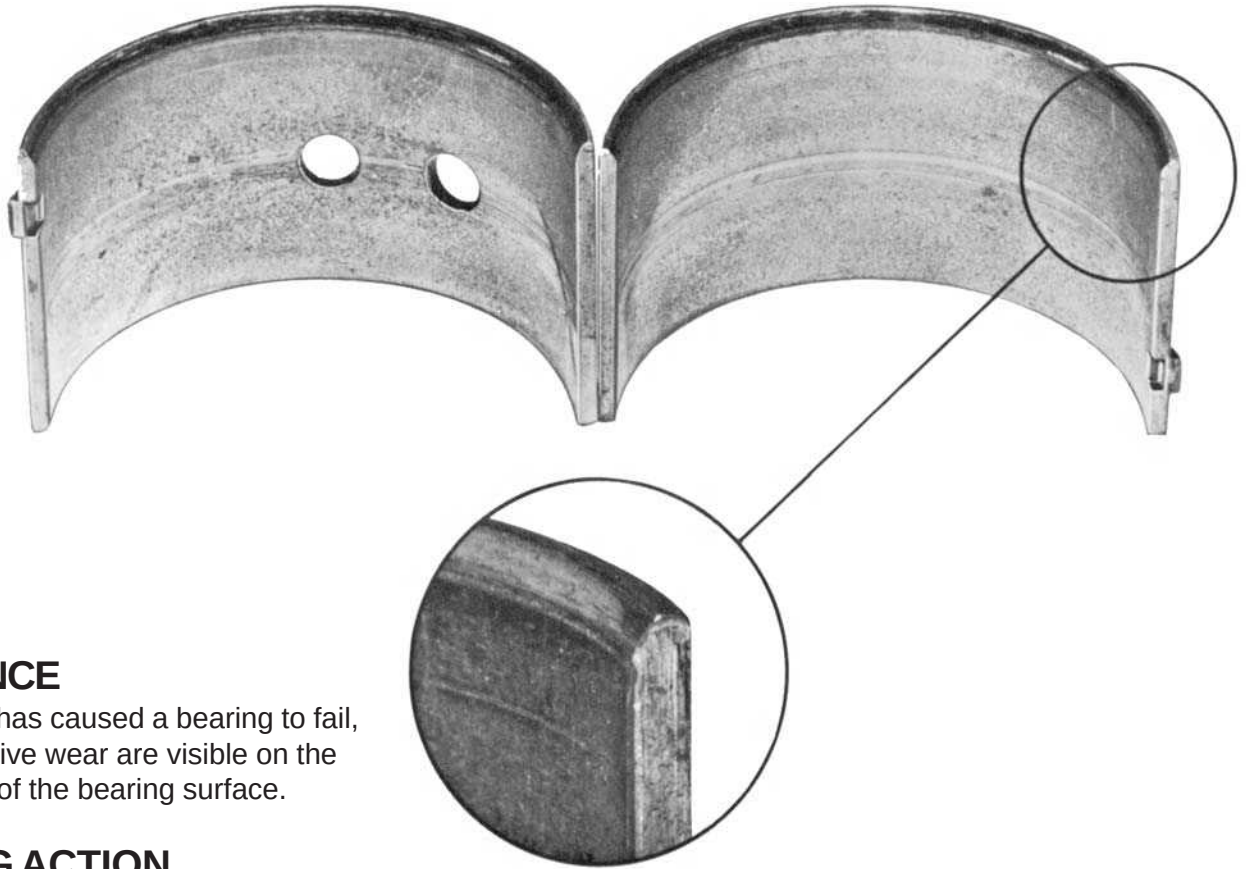
Three factors can contribute to connecting rod distortion:

1. Extreme operating conditions such as “hot rodding” and “lugging.”
2. Improper reconditioning. See inside front cover.
3. Dropping or abusing the connecting rod prior to assembly.

CORRECTIVE ACTION

1. Inspect connecting rod and recondition or replace if bent or twisted.
2. Check journal surfaces for excessive wear and regrind if necessary.
3. Install bearing.
4. Avoid dropping or abusing the connecting rod prior to assembly.
5. Use proper installation techniques.
6. Check related upper cylinder parts and replace if necessary.





APPEARANCE

When fillet ride has caused a bearing to fail, areas of excessive wear are visible on the extreme edges of the bearing surface.

DAMAGING ACTION

If the radius of the fillet at the corner where the journal blends into the crank is larger than required, it is possible for the edge of the engine bearing to make metal-to-metal contact and ride on this oversize fillet.

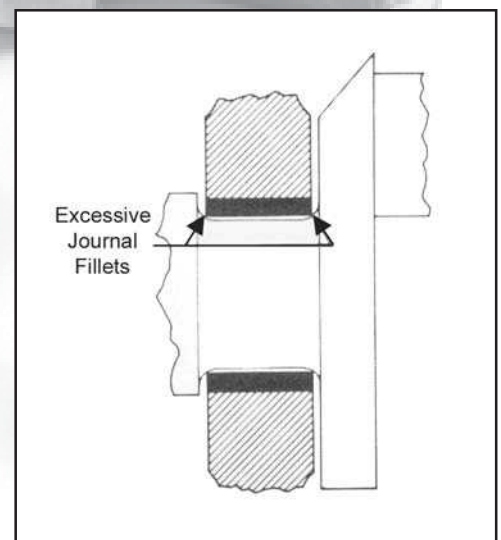
This metal-to-metal contact between the bearing and fillet causes excessive wear, leading to premature bearing fatigue.

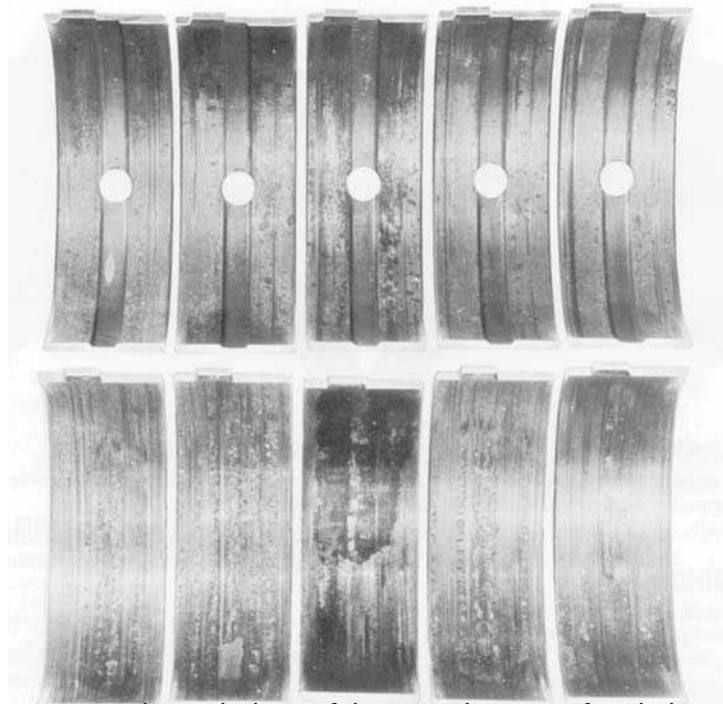
POSSIBLE CAUSES

Fillet ride results if excessive fillets are left at the edges of the journal at the time of crankshaft machining.

CORRECTIVE ACTION

1. Regrind the crankshaft paying particular attention to allowable fillet radii. **NOTE:** Be careful not to reduce fillet radius too much, since this can weaken the crankshaft at its most critical point.
2. Install new bearings with enlarged chamfers that allow proper fillet clearance.





APPEARANCE

A wear pattern is visible on the upper or lower halves of the complete set of main bearings. The degree of wear varies from bearing to bearing depending upon the nature of the distortion. The center bearing usually shows the greatest wear.

DAMAGING ACTION

A distorted crankcase imposes excessive loads on the bearings, with the point of greatest load being at the point of greatest distortion. These excessive bearing loads cause excessive bearing wear. Also, oil clearance is reduced and metal-to-metal contact is possible at the point of greatest distortion.

POSSIBLE CAUSES

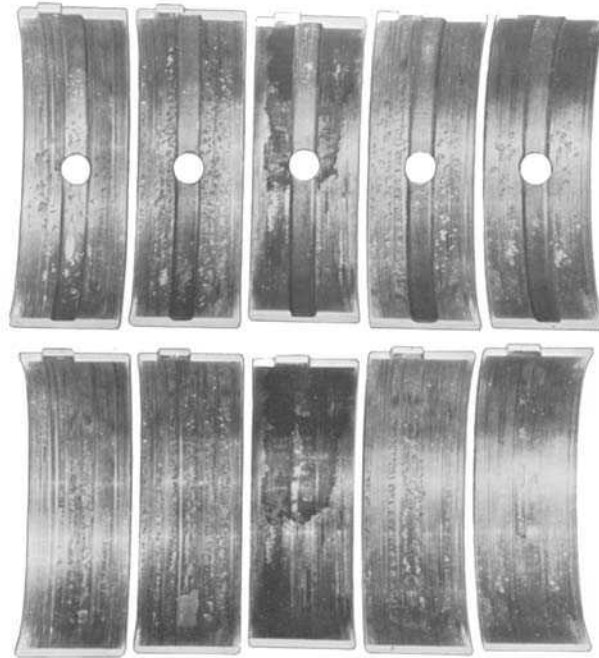
Alternating periods of engine heating and cooling during operation is a prime cause of crankcase distortion. As the engine heats the crankcase expands, and as it cools, the crankcase contracts. This repetitive expanding and contracting causes the crankcase to distort over time in some situations.

Distortion may also be caused by:

- Extreme operating conditions (for example “overheating” and “lugging”)
- Improper torquing procedure for cylinder head bolts, particularly with overhead valve V-8 engines

CORRECTIVE ACTION

1. Determine if distortion exists by use of Prussian blue or visual methods.
2. Align bore the housing (if applicable).
3. Install new bearings.



APPEARANCE

A wear pattern is visible on the upper and lower halves of the complete set of main bearings. The degree of wear varies from bearing to bearing depending upon the nature of the distortion. The center bearing usually shows the greatest wear.

DAMAGING ACTION

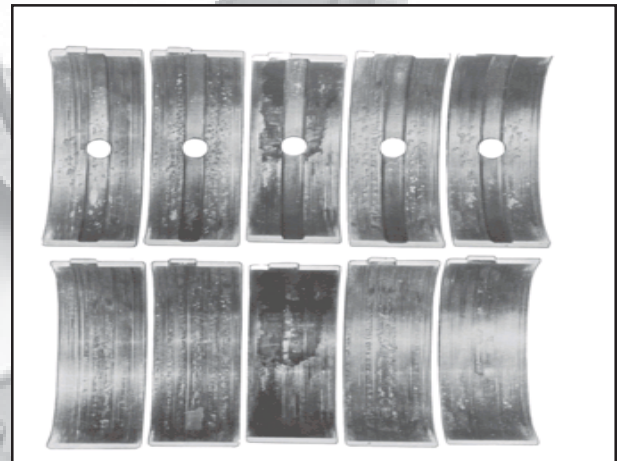
A distorted crankshaft subjects the main bearings to excessive loads, with the greatest load being at the point of greatest distortion. The result is excessive bearing wear. Also, the oil clearance spaces between journals and bearings are reduced, making it possible for metal-to-metal contact to occur at the point of greatest distortion.

POSSIBLE CAUSES

A crankshaft is usually distorted due to extreme operating conditions, such as “over-speeding” and “lugging”. It may also be caused by improper handling prior to installation.

CORRECTIVE ACTION

1. Determine if distortion exists by means of prussian blue or visual methods.
2. Install a new or reconditioned crankshaft.
3. Install new bearings.





APPEARANCE

Localized excessive wear areas are visible near the parting line on both top and bottom shells.

DAMAGING ACTION

Oil clearance near the parting line is decreased to such an extent that metal-to-metal contact between bearing and journal takes place, resulting in areas of above-normal wear.

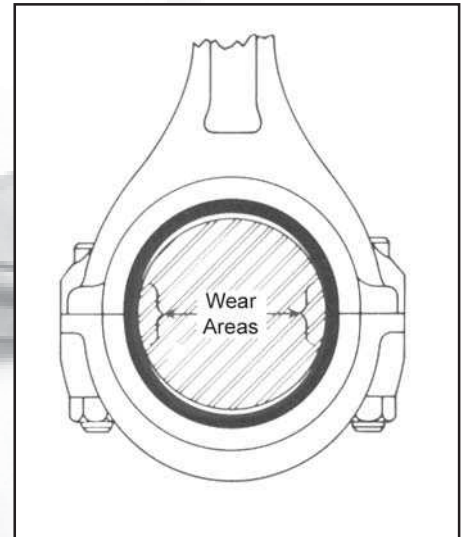
Also, improper seating between the bearing back and the housing bore may be present which hinders proper heat transfer causing localized heating of the bearing surface and thus reducing fatigue endurance.

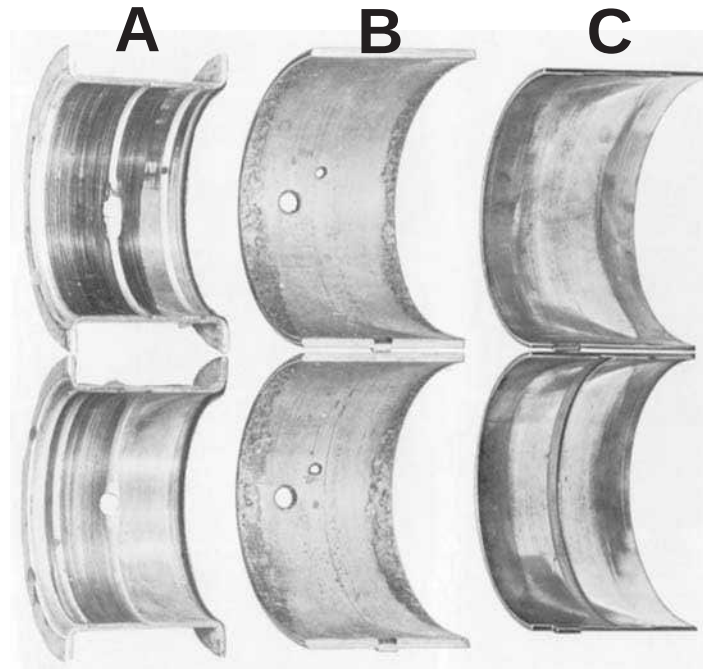
POSSIBLE CAUSES

Alternating loading and flexing of the connecting rod can cause the bearing housing to become elongated. And because replacement bearing shells, when installed, tend to conform to the shape of the bearing housing, this can result in an out-of-round bearing surface.

CORRECTIVE ACTION

1. Check the roundness of bearing housings before installing the new bearings. If they are found to be out-of-round, recondition the bearing housings (or replace connecting rod).
2. Check the journal surfaces for excessive wear and regrind if necessary.
3. Install new bearings.





APPEARANCE

In general, if a bearing has failed because of an out-of-shape journal, an uneven wear pattern is visible on the bearing surface. Specifically, however, these wear areas can be in any one of three patterns: Photo A above shows the wear pattern caused by a **tapered** journal. Photo B shows the wear pattern caused by an **hour-glass** shaped journal. Photo C shows the pattern of a **barrel** shaped journal.

DAMAGING ACTION

An out-of-shape journal imposes an uneven distribution of the load on the bearing surface, increasing heat generated and thus accelerating bearing wear. An out-of-shape journal also affects the bearing's oil clearance, making it insufficient in some areas and excessive in others, thereby upsetting the proper functioning of the lubrication system.

POSSIBLE CAUSES

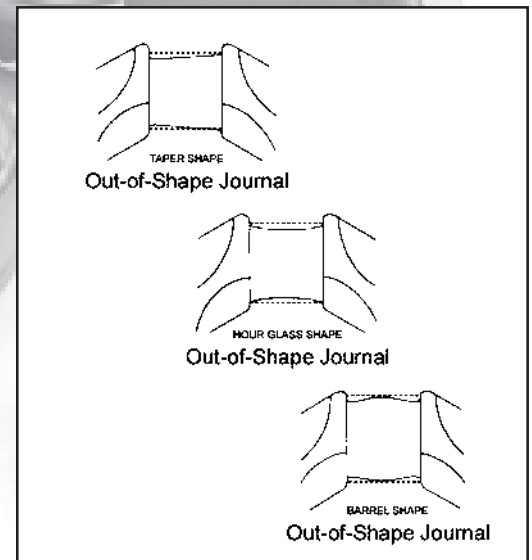
If the journal is tapered there are two possible causes:

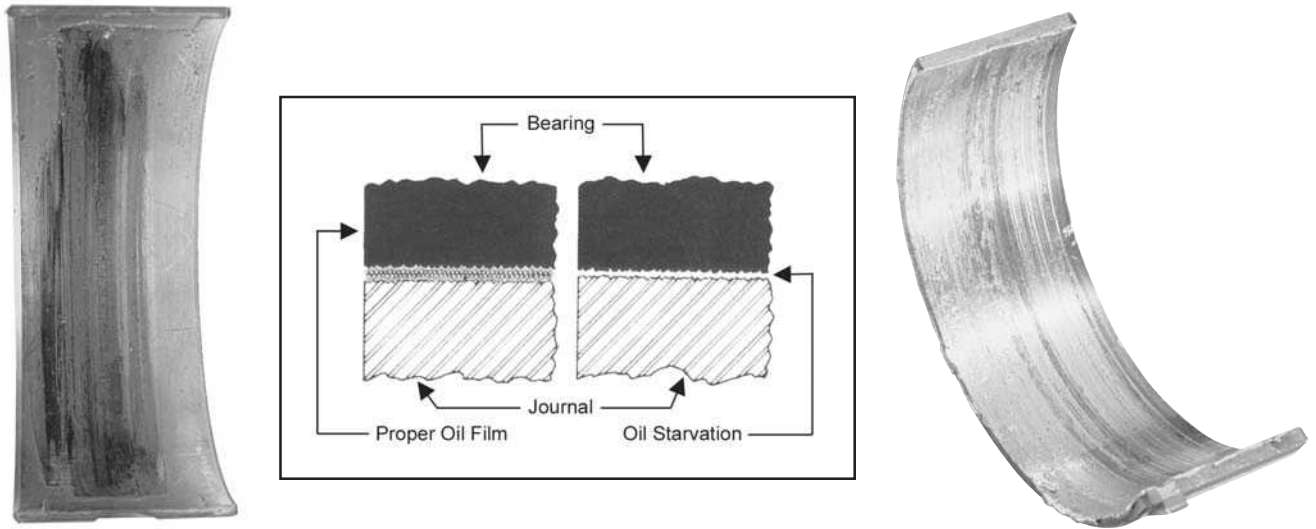
1. Uneven wear of the journal during operation (misaligned rod).
2. Improper machining of the journal at some previous time.

If the journal is **hour-glass** or **barrel** shaped, this is usually the result of improper machining or polishing.

CORRECTIVE ACTION

Regrinding the crankshaft can best remedy out-of-shape journal problems. Then install new bearings in accordance with proper installation procedures.





APPEARANCE

When a bearing has failed due to oil starvation, its surface is usually very shiny. In addition, there may be excessive wear of the bearing surface due to the wiping action of the journal. In the most severe cases the bearing surface will be smeared or scratched and torn. The bearing will begin to turn dark blue to black in color.

DAMAGING ACTION

The absence of a sufficient oil film between the bearing and the journal permits metal-to-metal contact. The resulting wiping action causes premature bearing fatigue.

POSSIBLE CAUSES

1. Insufficient oil clearance – usually the result of utilizing a replacement bearing that has too great a wall thickness. In some cases, the journal may be oversize.
2. Broken or plugged oil passages, prohibiting proper oil flow.
3. A blocked oil suction screen or oil filter.
4. A malfunctioning oil pump or pressure relief valve.
5. Misassembling main bearings metering off an oil hole.
6. Lubrication system not primed before start up.

CORRECTIVE ACTION

1. Double-check all measurements taken during the bearing selection procedure to catch any errors in calculation. (During assembly check oil clearance with Plastigage[®].)
2. Check to be sure that the replacement bearing you are about to install is the correct one for the application.
3. Check the journals for damage and regrind if necessary.
4. Check engine for possible blockage of oil passages, oil suction screen and oil filter.
5. Check the operation of the oil pump and pressure relief valve.
6. Be sure that the oil holes are properly indexed when installing the replacement bearings.
7. Advise the operator about the results of engine lugging.
8. Always prime the lubrication system before the engine is started the first time.



APPEARANCE

Bearing surface wiped and torn, blackened from heat, with patches of lining material torn cleanly from steel backing.

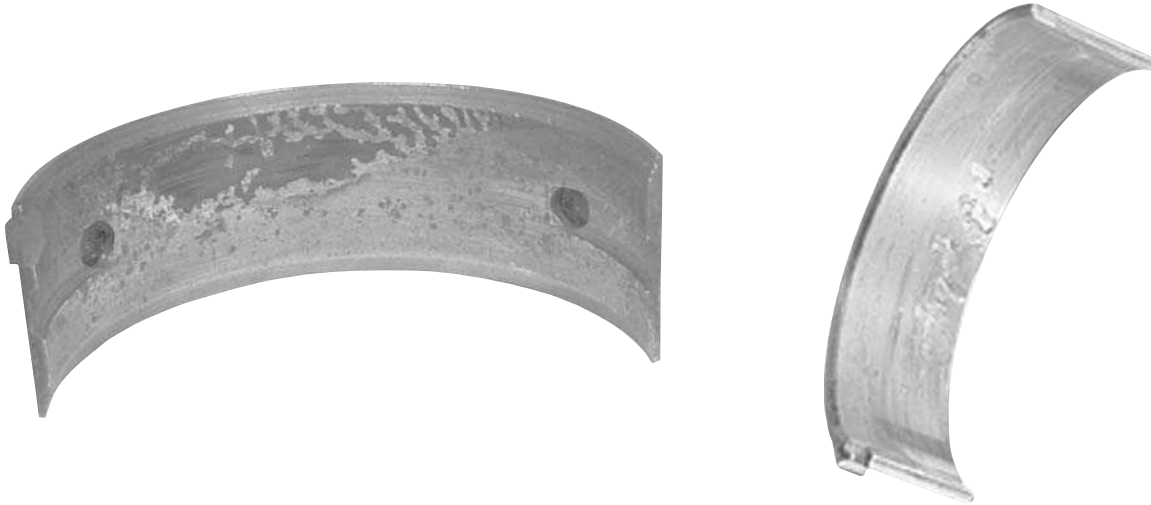
POSSIBLE CAUSES

1. Breakdown of lubrication and resulting high friction elevates operating temperature.
2. Lead in bearing material melts and allows shaft to tear away patches of bearing lining.
3. Lack of lubrication.
4. Wiping.
5. Dirt contamination.
6. Concentrated loading (misalignment, etc.).

CORRECTIVE ACTION

A hot short is a catastrophic failure that results from one of the conditions already covered in detail. To properly correct this it must first be determined which specific condition lead to the hot short. For further details please see:

1. Foreign Particles in Lining (Page 3)
2. Foreign Particles on Bearing Back (Page 4)
3. Out-of-Round Bore (Page 13)
4. Excessive Crush (Page 7)
5. Bent or Twisted Connecting Rod (Page 9)
6. Shifted Bearing Cap (Page 6)
7. Distorted Crankcase (Page 11)
8. Bent Crankshaft (Page 12)
9. Oil Starvation (Page 15)
10. Misassembly (Page 5)



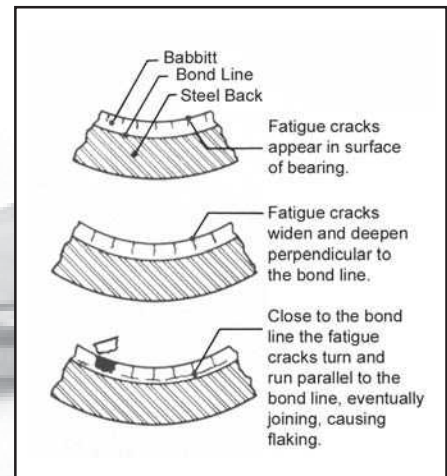
APPEARANCE

Bearing surface cracked, small irregular areas of surface material missing from the bearing lining.

DAMAGING ACTION

Heavy pulsating loads imposed upon the bearing by reciprocating motion within the engine cause the bearing surface to crack due to metal fatigue.

Fatigue cracks widen and deepen perpendicular to the bond line. Close to the bond line, fatigue cracks turn and run parallel to the bond line, eventually joining and causing pieces of the surface to flake out.

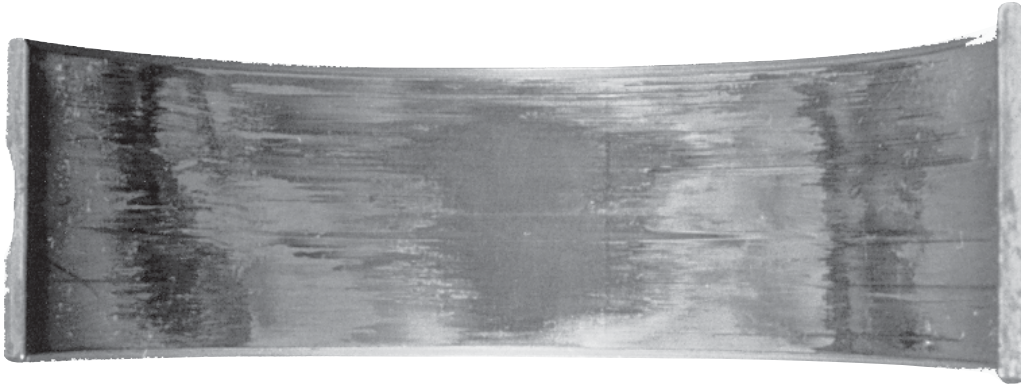


POSSIBLE CAUSES

1. Overloading (lugging, detonation, or overfueling).
2. Uneven loading (see sections on misalignment).
3. Bearing material of inadequate fatigue strength for application.
4. Bearing failure due to surface fatigue can be the result of the normal life span of the bearing being exceeded.

CORRECTIVE ACTION

1. If the service life for the old bearing was adequate, replace with the same type of bearing to obtain a similar service life.
2. If the service life of the old bearing was too short, replace with a heavier duty bearing to obtain a longer life.
3. Replace all other bearings (main connecting rod and camshaft) as their remaining service life may be short.
4. Recommend that the operator avoid "hot rodding" and "lugging" as these tend to shorten bearing life.
5. Check for misassembly. Use proper installation techniques.



APPEARANCE

Bearing surface darkened, spongy, etched by chemical attack.

DAMAGING ACTION

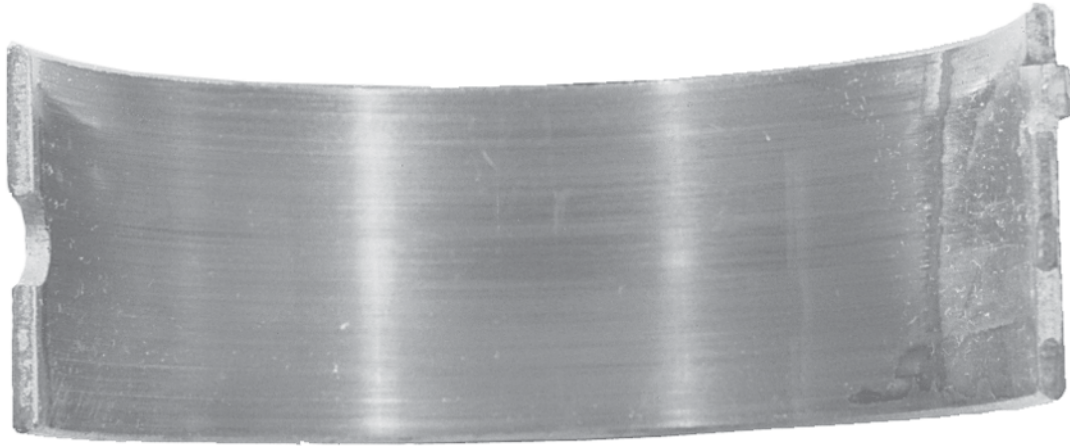
This is usually the result of contamination of the oil from either the fuel system or internal engine leaks. This condition is further pronounced when there is poor routine maintenance.

POSSIBLE CAUSES

1. Acids in oil.
2. Excessive operating temperature.
3. Excessive blow-by.
4. Coolant contamination of oil.
5. Use of high sulfur fuel.
6. Excessive oil change interval.

CORRECTIVE ACTION

1. Identify and correct source of contamination.
2. Install new bearings using correct installation procedures.
3. Use a better quality fuel, if possible.
4. Oil should be changed at recommended intervals using the proper grade and rating. In some applications oil analysis may be needed to determine the optimum oil change intervals.



APPEARANCE

Wall thickness reduced from original dimension. Bearing surface worn and polished but not smeared, torn or scored. No evidence of heat, no embedded foreign particles.

DAMAGING ACTION

Grinding and polishing the crankshaft journals produce burrs that are so small that we can't see or feel them. Not only is it important that the surface finish meet recommended average Ra or better, but it is also important to always grind opposite to rotation and polish in the direction of rotation. Otherwise it is possible for these microscopic burrs to disrupt the oil film and abrade away the bearing surface.

POSSIBLE CAUSES

1. Poor journal surface finish.
2. Wear in the presence of adequate lubrication to prevent heat build-up and wiping is caused by peaks in the journal surface finish profile which penetrate the oil film and abrade the bearing.

CORRECTIVE ACTION

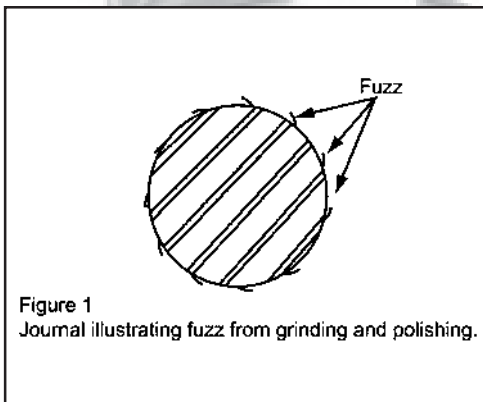
1. Check journal surface finish for proper average Ra and regrind as needed.
2. If surface finish is acceptable lightly re-polish in the direction of rotation.
3. Install new bearings.

Crankshaft Grinding And Polishing

Crankshaft journal surfaces should be ground and polished to a surface finish of 15 micro inches roughness average Ra or better. Journals on highly loaded crankshafts such as diesel engines or high performance racing engines require a finish of 10 micro inches Ra or better.

The above is a simple straight forward specification which can be measured with special equipment. However, there is more to generating a ground and polished surface than just meeting the roughness specification. To prevent rapid, premature wear of the crankshaft bearings and to aid in the formation of an oil film, journal surfaces must be ground opposite to engine rotation and polished in the direction of rotation. This recommendation can cause a great deal of confusion in actual execution. Understanding the reasons behind the recommendation and examination of the following illustrations will help make the recommendation more clear.

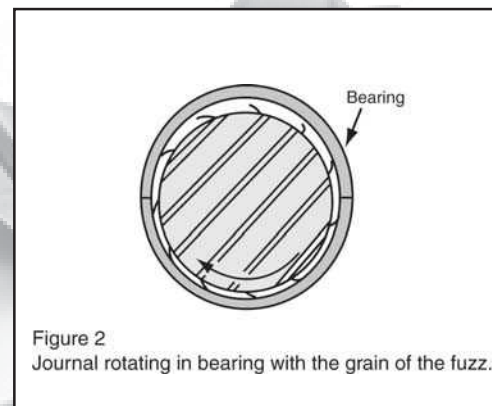
Metal removal tends to raise burrs. This is true of nearly all metal removal processes. Different processes create different types of burrs. Grinding and polishing produces burrs that are so small that we can't see or feel them but they are there and can damage bearings if the shaft surface is not generated in the proper way. Rather than "burrs," let's call what results from grinding and polishing "microscopic fuzz." This better describes what is left by these processes. This microscopic fuzz has a grain or lay to it like the hair on a dog's back. Figure 1 is an illustration depicting the lay of this fuzz on a journal. (Note: All figures



are viewed from nose end of crankshaft.) The direction in which a grinding wheel or polishing belt passes over the journal surface will determine the lay of the micro fuzz.

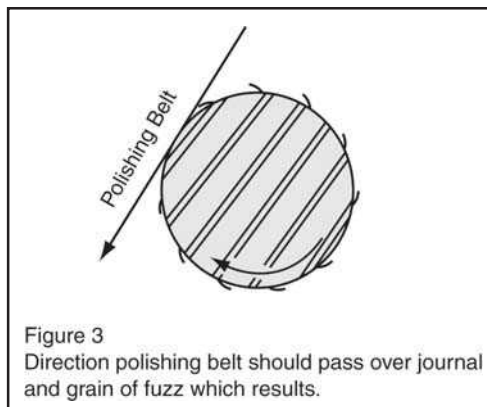
In order to remove this fuzz from the surface, each successive operation should pass over the journal in the opposite direction so that the fuzz will be bent over backward and removed. Polishing in the same direction as grinding would not effectively remove this fuzz because it would merely lay down and then spring up again. Polishing must, therefore, be done opposite to grinding in order to improve the surface.

In order to arrive at how a shaft should be ground and polished, we must first determine the desired end result and then work backwards to establish how to achieve it. Figure 2 depicts a shaft turning in a bearing viewed from the front of a normal clockwise rotating engine. The desired condition is a journal with any fuzz left by the polishing operation oriented so it will lay down as the shaft passes over the bearing (Figure 2).

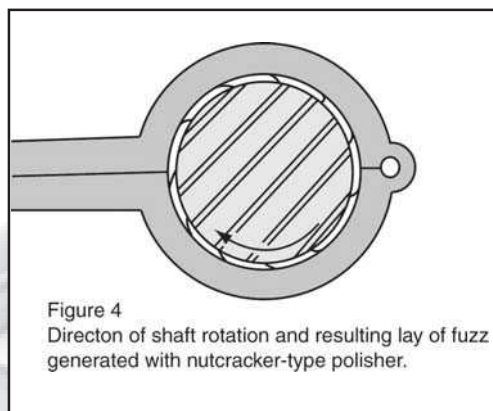


The analogy to the shaft passing over the bearing is like petting a dog from head to tail. A shaft polished in the opposite direction produces abrasion to the bearing which would be like petting a dog from tail to head. To generate a surface lay like that shown in Figure 2, the polishing belt must pass over the shaft surface as shown in Figure 3.

The direction of shaft rotation during polishing is not critical if a motorized belt type polisher is used be-



cause the belt runs much faster than the shaft. If a nutcracker-type polisher is used, then proper shaft rotation must be observed (Figure 4). Stock removal during polishing must not exceed .0002" on the diameter.

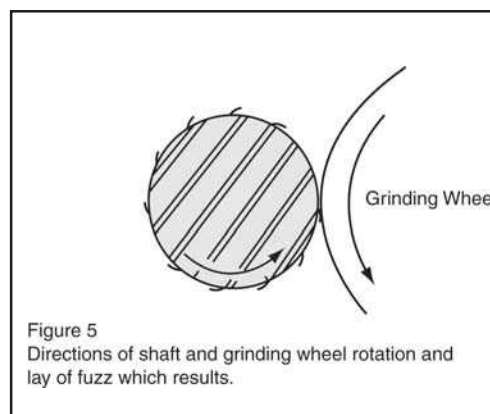


Having determined the desired surface lay from polishing, we must next establish the proper direction for grinding to produce a surface lay opposite to that resulting from polishing. Figure 5 shows the grinding wheel and shaft directions of rotation and surface lay for grinding when viewed from the front or nose end of the crankshaft. This orientation will be achieved by chucking the flywheel flange at the left side of the grinder (in the headstock). Achieving the best possible surface finish during grinding will reduce the stock removal necessary during polishing.

The surface lay generated by grinding would cause abrasion to the bearing surfaces if left unpolished. By polishing in the direction shown in either Figure 3 or 4,

the surface lay is reversed by the polishing operation removing fuzz created by grinding and leaving a surface lay which will not abrade the bearing surface.

Nodular cast iron shafts are particularly difficult to grind and polish because of the structure of the iron.



Nodular iron gets its name from the nodular form of the graphite in this material. Grinding opens graphite nodules located at the surface of the journal leaving ragged edges which will damage a bearing. Polishing in the proper direction will remove the ragged edges from these open nodules.

All of the above is based on normal clockwise engine rotation when viewed from the front of the engine. For crankshafts which rotate counterclockwise, such as some marine engines, the crankshaft should be chucked at its opposite end during grinding and polishing. This is the same as viewing the crank from the flanged end rather than the nose end in the accompanying figures.

Premature Thrust Bearing Failure

BACKGROUND

Although thrust bearings run on a thin film of oil, just like radial journal (connecting rod and main) bearings, they cannot support nearly as much load. While radial bearings can carry loads measured in thousands of pounds per square inch of projected bearing area, thrust bearings can only support loads of a few hundred psi. Radial journal bearings develop their higher load capacity from the way the curved surfaces of the bearing and journal meet to form a wedge. Shaft rotation pulls oil into this wedge shaped area of the clearance space to create an oil film, which actually supports the shaft. Thrust bearings typically consist of two flat mating surfaces with no natural wedge shape in the clearance space to promote the formation of an oil film to support the load.

Conventional thrust bearings are made by incorporating flanges at the ends of a radial journal bearing. This provides ease in assembly and this design has been used successfully for many years. Either tear-drop or through grooves on the flange faces and wedge shaped ramps at each parting line allow oil to enter between the shaft and bearing surfaces. However, the vast majority of the bearing surfaces and the entire shaft surface are flat making it much harder to create and maintain an oil film. If you have ever taken two gauge blocks and wiped them perfectly clean and pressed them together with a twisting action you know that they will stick together. This is very much like what happens as a thrust load applied to the end of a crankshaft squeezes the oil out from between the shaft and bearing surfaces. If the load is too great, the oil film collapses and the surfaces want to stick together, resulting in a wiping failure.

For many years some heavy-duty diesel engines have used separate thrust washers with a profiled face to enable them to support higher thrust loads. These thrust washers either have multiple tapered ramps and relatively small flat pads or have curved surfaces that follow a sine wave contour around their circumference.

CAUSES OF FAILURE

Aside from the obvious causes, such as dirt contamination and misassembly, there are only three common factors that generally cause thrust bearing failures. These are:

- Poor crankshaft surface finish
- Misalignment
- External overloading

SURFACE FINISH

Crankshaft thrust faces are difficult to grind because they are done using the side of the grinding wheel. Grinding marks left on the crankshaft face produce a visual swirl or sunburst pattern with scratches sometimes criss-crossing one another in a crosshatch pattern similar to hone marks on a cylinder wall. If these grinding marks are not completely removed by polishing, they will remove the oil film from the surface of the thrust bearing much like multiple windshield wiper blades.

A properly finished crankshaft thrust face should only have very fine polishing marks that go around the thrust surface in a circumferential pattern. A surface finish of 15 Ra or less is recommended (10 Ra or less for heavy duty or highly loaded engines).

ALIGNMENT

The grinding wheel side face must be dressed periodically to provide a clean, sharp cutting surface. A grinding wheel that does not cut cleanly may create hot spots on the work piece leading to a wavy, out-of-flat surface. The side of the wheel must also be dressed at exactly 90° to its OD to produce a thrust face that is square to the axis of the main bearing journal.

When assembling thrust bearings:

1. Tighten main cap bolts to approximately 10 to 15 ft. lbs. to seat bearings then loosen.
2. Tap main cap toward rear of engine with a soft-faced hammer.
3. Tighten main cap bolts, finger tight.
4. Using a bar, force the crankshaft as far forward in the block as possible to align bearing rear thrust faces.
5. While holding shaft in forward position, tighten main cap bolts to 10 to 15 ft lbs.
6. Complete tightening main cap bolts to specifications in 2 or 3 equal steps.

The above procedure should align the bearing thrust faces with the crankshaft to maximize the amount of bearing area in contact for load carrying.

LOADING

A number of factors may contribute to wear and overloading of a thrust bearing, such as (in no specific order):

- Excessive pump pressure
- Torque converter expansion
- Torque converter internal wear
- Pump drive gear installed backwards
- Wrong torque converter
- Wrong flex plate
- Wrong flywheel bolts
- Misalignment of the engine and the major transmission components
- Improper throw out bearing adjustment
- Riding the clutch pedal

DIAGNOSING PROBLEMS

By the time a thrust bearing failure becomes evident, the parts have usually been so severely damaged that there is little if any evidence of the cause. The bearing is generally worn into the steel backing which has severely worn the crankshaft thrust face as well. So how do you tell what happened?

Start by looking for the most obvious internal sources:

- Is there evidence of distress anywhere else in the engine that would indicate a lubrication problem or foreign particle contamination?
- Were the proper bearing shells installed correctly?
- If the thrust bearing is in an end position, was the adjacent oil seal correctly installed? An incorrectly installed rope seal can cause sufficient heat to disrupt bearing lubrication.
- Examine the un-failed thrust face on the crankshaft for surface finish and geometry. This may give an indication of the original quality of the failed face.
- Once you are satisfied that all potential internal sources have been eliminated; ask about potential external sources of either overloading or misalignment.
- Did the vehicle have a prior thrust bearing failure?
- What external parts were replaced?
- Was the correct transmission installed?
- Was the correct torque converter installed?

- Was the correct flex plate used? At installation there should be a minimum of 1/16" (1/8" preferred) clearance between the flex plate and converter to allow for converter expansion.
- Were the correct flex plate mounting bolts used?
- Is there evidence of the converter hitting the flex plate mounting bolts?
- Was the transmission properly aligned to the engine?
- Were all dowel pins in place?
- Check condition of pilot bearing.
- If a used torque converter was re-used, is it worn internally?
- If a rebuilt transmission was installed, did the torque converter engage the pump drive spline properly? An improperly installed pump drive gear may prevent full engagement of the converter.
- Was the transmission pump pressure checked and found to be within specification?
- Check external transmission cooling lines and heat exchanger for restrictions that will increase pump pressure.
- If a manual transmission was installed, was the throw out bearing properly adjusted?
- What condition was the throw out bearing found to be in? A properly adjusted throw out bearing that is worn or overheated may indicate the operator was "Riding the Clutch".

HELP FOR THE THRUST BEARING

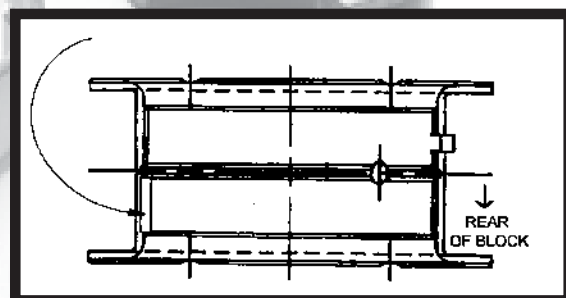
When a problem application is encountered, every effort should be made to find the cause of distress and correct it before completing repairs or you risk a repeat failure.

A simple modification to the upper thrust bearing may help in problem applications. Install the upper thrust bearing in the block to determine which thrust face is toward the rear of the engine.

Using a small, fine tooth, flat file, increase the chamfer on the ID edge of the bearing parting line from the oil groove to the rear thrust face only. (See diagram.) This enlarged ID chamfer will allow pressurized oil from the bearing oil groove to reach the loaded thrust face without passing through the bearings clearance space first. Since there is a load against the rear thrust face, the load should restrict oil flow and there should not be a noticeable loss in oil pressure. Although this modification is not a guaranteed cure-all it should help if all other conditions, such as surface finish, alignment, cleanliness and loading are within reasonable limits.

RECENT DEVELOPMENTS

In the past few years some new automotive engine designs have begun using various methods to enable them to carry higher thrust loads imposed by some of the newer automatic transmissions. The use of thrust washers exclusively, a combination of thrust washers and flange bearings, radial grooving (a series of 3 - 5 straight grooves on the flange face), flange bearings with profiled (ramp and pad design) flange faces, or assembled flanges (separate thrust washers which are mechanically attached to the bearing). It is important to keep in mind that most of these developments are application specific and may not be available or even feasible for older applications.



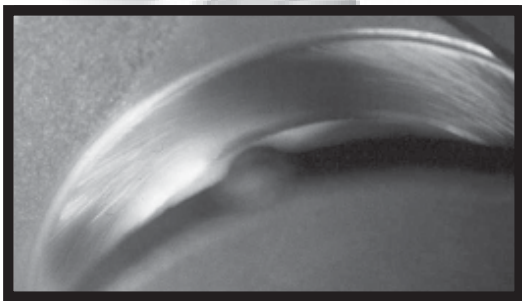
SEVERE USE RECOMMENDATIONS

Crankshaft surface finish and shape are key factors affecting the performance of all bearings. These factors become even more critical for thrust surfaces. As in any bearing, increased loading reduces oil film thickness between shaft and bearing surfaces. This is a much more critical situation in thrust bearings due to their flat faces which make formation of an oil film extremely difficult. Radial bearings (those which carry loads in a radial direction like rod and main bearings) form a natural wedge where shaft and bearing surfaces come together in the clearance space. Shaft rotation pulls a wedge of oil into the loaded area of the bearing and forms an oil film that supports the load.

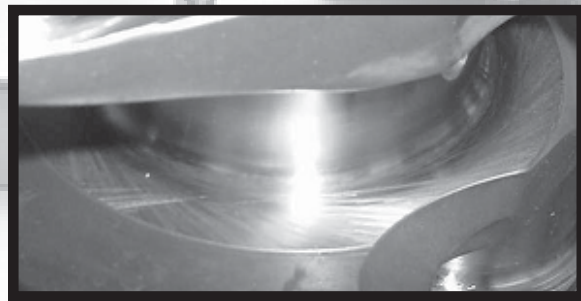
Thrust faces, on the other hand, are made up of two flat surfaces that do not form a natural wedge where they meet. In order to help form an oil film, artificial wedge shaped areas are machined into the bearing surfaces at the ends and sometimes adjacent to the grooves. In spite of all the common design efforts, thrust bearings still run on a much thinner film of oil that makes crankshaft surface finish **critical** in the successful performance of these bearings.

Recent samples of thrust face surface finish on crankshafts from blown fuel “Hemi” engines have confirmed that better finishes resulted in a reduced rate of bearing distress. The study also showed that when no damage occurred, the crankshaft surface finish was improved after running. The surface finishes of 12 crankshafts were measured (7 new and 5 used). The new shafts ranged from a high of 30 Ra to a low of 5 Ra. The used shafts had a very similar range from 31Ra to 4 Ra. Although this represents only a small sampling, it does demonstrate a correlation between surface finish and performance when the condition of mating bearing surfaces was evaluated. Prior to these measurements, race experience had shown no problems on a crankshaft with a thrust-face Ra of 6 and DID show problems on crankshafts when the Ra was over 20!

Obtaining a good finish on the thrust face of a crankshaft is difficult to do because it uses side-wheel grinding. Side grinding causes marks that spiral outward toward the OD of the thrust face and may also cause cross-hatch marks resembling honing patterns. Both patterns are detrimental to the formation of an oil film because they work like wipers as the shaft rotates. **Grinding marks must be removed by polishing. Only a circular pattern should remain. Surface finish should be checked in a tangential direction and must be held to 10 Ra max. The thrust surface should be flat within .0002” max.**



AVOID - SWIRL PATTERN



AVOID - CROSSHATCH PATTERN

Preventing Premature Cam Bearing Failure

For many years, nearly all camshaft bearings were manufactured with a lining of babbitt. Babbitt is a soft slippery material made up primarily of lead and tin and is quite similar to solder. As a bearing surface layer, babbitt possesses the desirable properties necessary to survive under adverse conditions such as foreign particle contamination, misalignment and marginal lubrication on start up.

The trend in modern engines has been toward higher operating temperatures and higher valvetrain loads. Babbitt is limited in its ability to survive under these conditions due to its relatively low strength. When babbitt cam bearings are installed under these demanding conditions, the lining may extrude or fatigue. Fatigue can be identified by craters in the bearing surface where sections of lining material have flaked out.

To meet the demands of higher loads and operating temperatures in modern engines as well as the requirements imposed by high performance, babbitt has been replaced by an alloy of aluminum. This aluminum alloy is much stronger than babbitt and will withstand several times the load. However, this added strength is obtained at the expense of some of the more forgiving properties of babbitt. The aluminum alloy is harder, making it somewhat less compatible with dirt, misalignment and marginal lubrication. This is typical of the compromises or trade offs that are frequently necessary when selecting a bearing material to suit the requirements of a specific application and in this case, higher loading.

Typically, whenever a higher level of loading is encountered, greater precision is required to maintain reliability. Conditions such as cleanliness, alignment, clearances, journal surface finishes and lubrication must all be controlled more closely. Following are some recommendations to help optimize performance when using aluminum alloy camshaft bearings.

Sufficient clearance is necessary in the initial installation. These stronger bearings will not wear in rapidly to make their own clearance like softer babbitt materials. Minimum clearance should be .002" for stock engines and .003" for high performance. Optimum clearance range for high performance applications is .003" to .004". Because of the stack up of tolerances on the block, shaft and bearing it is impossible to control clearance to this range in the manufacture of the bearing alone. Clearances must be measured at installation.

Honing the ID's of cam bearings to increase clearance is not recommended because hone grit may become embedded in bearing surfaces that will cause shaft wear. Bearing ID's may be reamed, but the most practical means is to adjust camshaft journal diameters by grinding the journal. Even if not ground to provide additional clearance, camshaft bearing journals should be polished to the proper surface finish with the camshaft rotating in the same direction it will rotate in the engine.

Like clearance, alignment is also extremely important especially for high performance applications. Any block that has needed to have its main bearing bore alignment corrected due to distortion is likely to have experienced cam bearing bore distortion as well. Adequate clearance can help compensate for minor misalignment of less than .001".

Installation of bearings into the block must be done with care to avoid shaving metal off the backs of the bearings. This galling action may cause a build-up of metal between the bearing OD and the housing bore which will result in a reduction in clearance. To prevent galling, check housing bores for a proper 25 to 30 degree lead-in chamfer before installing cam bearings. On blocks without grooves behind the cam bearings, care must be taken to insure that oil holes line up between the bearings and block. Where the block has a groove behind the bearing, the bearing should be installed with the oil hole at the 2 o'clock position when viewed from the front for normal clockwise camshaft rotation. This will introduce oil into the clearance space outside of the loaded area and allow shaft rotation to build an oil film ahead of the load.



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