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Sleeve Bearing Repair Tips

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SLEEVE BEARING REPAIR TIPS

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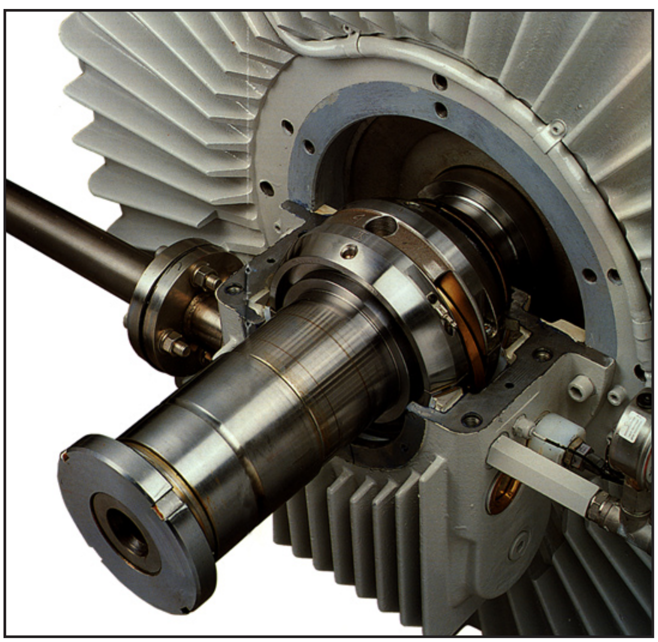
Sleeve bearings, also known as babbitt bearings, plain bearings or white metal bearings, have been used in almost all sizes of electric motors since motors were invented. Although most motors through a 400-frame size have ball bearings for economic reasons, sleeve bearings are still used for large motors where the desired bearing life cannot be achieved with rolling-element bearings. The limiting factors in larger motors are the diameter of the bearing and the speed of rotation.

The purpose of this paper is to bring together information from several resources to explain how a sleeve bearing works, identify areas of concern for repairers, and to offer a more comprehensive explanation of sleeve bearing clearances for horizontal and vertical machines.

This paper will explain the factors that govern bearing clearance for horizontal and vertical machines, explain why they are different, and the problems that can arise when a sleeve bearing has either too little, or too much, clearance. It also will cover fitting procedures for sleeve bearings and some tips for avoiding sleeve bearing failures during the final test run.

FIGURE 1: HORIZONTAL SLEEVE BEARING



The cylindrical overshot bearing is lubricated by oil rings and/or a forced lubrication system.

PRINCIPLES OF SLEEVE BEARINGS

Perhaps the first thing we need to recognize is that most sleeve bearing-equipped horizontal electrical rotating machinery uses what is called a cylindrical overshot bearing design (Figure 1); the term describes how it is lubricated: by oil supplied by oil rings.

Sleeve bearings are deceptively simple in appearance. Made of tin- or lead-based babbitt, they are machined slightly larger than the shaft. A film of oil continuously lubricates the bearing and shaft, minimizing surface wear and cooling the parts. Foreign matter that finds its way between the bearing and the shaft becomes embedded in the soft babbitt material, thus protecting the harder (and more costly) shaft.

CONSIDERATIONS IN SLEEVE BEARING DESIGN

The major factors, in order of importance, that influence sleeve bearing design include:

- Weight to be supported
- Peripheral speed of shaft journal
- Viscosity of lubricant
- Operating temperature

Designers of electrical rotating equipment generally keep sleeve bearing load pressure around 145 psi (1000 PA) compared to 580-725 psi (4000-5000 PA) for internal combustion engines. Some older motors used even lower bearing load pressure, so vintage machines sometimes have a larger bearing than a modern motor with similar characteristics.

Excessive weight can distort soft babbitt, so bearing cross-sectional area increases proportional to the load. Higher hp/kW ratings mean more torque, hence a larger shaft diameter. Peripheral speed increases with the rpm, or when the shaft diameter increases. For hydrostatic, cylindrical-overshot bearings (the sleeve bearings most common to electrical machinery) the upper peripheral speed limit is approximately 6,000 feet (1,830 meters) per minute. For example, a 6” diameter journal rotating 3600 rpm:

$$(6 \times \pi \times 3600) / 12 = 5{,}655 \text{ ft/min.}$$

Higher speeds require special designs, starting with 2-lobe, then 4-lobe bearings, as well as special lubrication methods (Figure 2).

Most rotating electrical machinery sleeve bearing lu-

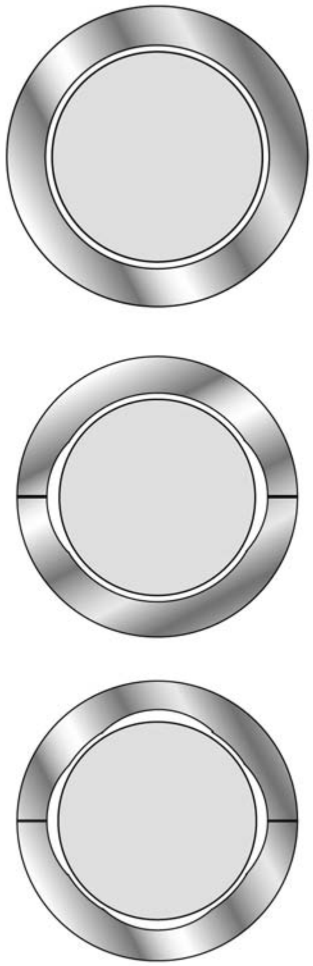
brication is supplied by oil rings, which require a minimum speed be maintained to ensure that the rings deliver sufficient oil to the bearings. That minimum speed is around 25-30 feet (7.5-9 meters) per minute; be very cautious when an application, such as a VFD used for variable speed operation, significantly alters the original design. A force-lubrication system may be required, or even hydraulic jacking to lift the shaft onto an oil film before starting.

When the peripheral speed limit restricts journal diameter, the designer must increase the length of the sleeve bearing. That brings us to the ratio of bearing length to diameter (L/d). For reasons of economy, the preference is for a 1:1 ratio – e.g. a bearing with 3” bore diameter and 3” length. There are drawbacks to a proportionally longer bearing. As the L/d ratio increases:

- Less oil flow exiting the bearing = higher bearing temperature
- Shaft deflection = diagonal contact with ends of bearing
- Longer machine = Higher production costs

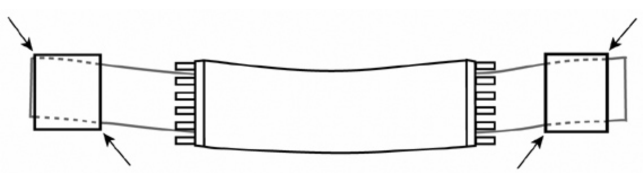
The longer and heavier the rotor, or the more flexible the shaft, the more shaft deflection (Figure 3) should be expected. Shaft deflection may force the designer to increase the clearance between the sleeve bearing and journal. Longer bearings require longer shaft journals, which in turn require longer bearing brackets and larger machines.

FIGURE 2: BEARING BORE SHAPES



Cylindrical overshot, 2-lobe and 4-lobe bearing shapes.

FIGURE 3: ROTOR SAG



A heavy rotor increases shaft sag. The journal may rub the bearings at the places indicated by arrows. To prevent that, the designer increases the clearance.

Viscosity of the oil is less of a factor than the supported weight and peripheral speed, although one the designer must consider. The OEM manual specifies the recommended oils for the machine so, unless bearing design modifications have occurred, we as repairers should stick with the recommended lubricants. Sometimes, a 2-pole machine benefits from a lower viscosity oil but such changes should only be made in consultation with the OEM or customer, and even then with caution.

Clearance

Proper clearance between the shaft and bearing keeps the shaft position stable. Insufficient clearance results in excessive heat due to friction between the shaft and bearing. Excessive clearance can lead to unwanted movement (vibration or loss of concentric orbit).

The optimal shaft clearance for sleeve bearings is governed by several factors:

- Horizontal or vertical shaft orientation
- Weight to be supported
- Peripheral speed of shaft journal
- Ratio of length to diameter
- Oil viscosity and load

One of the first questions to consider, when looking at guidelines for bearing clearance: Is this radial or diametral? We are going to use diametral clearance—total clearance—because we physically measure the shaft and bearing diameters to determine the clearance. Another reason to use diametral clearance: In operation, a horizontal machine rarely has the same radial clearance at the 12:00 and 6:00 positions.

To illustrate just how many guidelines there are for this simple topic, I have combined guidelines from five manufacturers and three other reputable sources into a single graph (Figure 4).

CLEARANCE DEPENDS ON APPLICATION

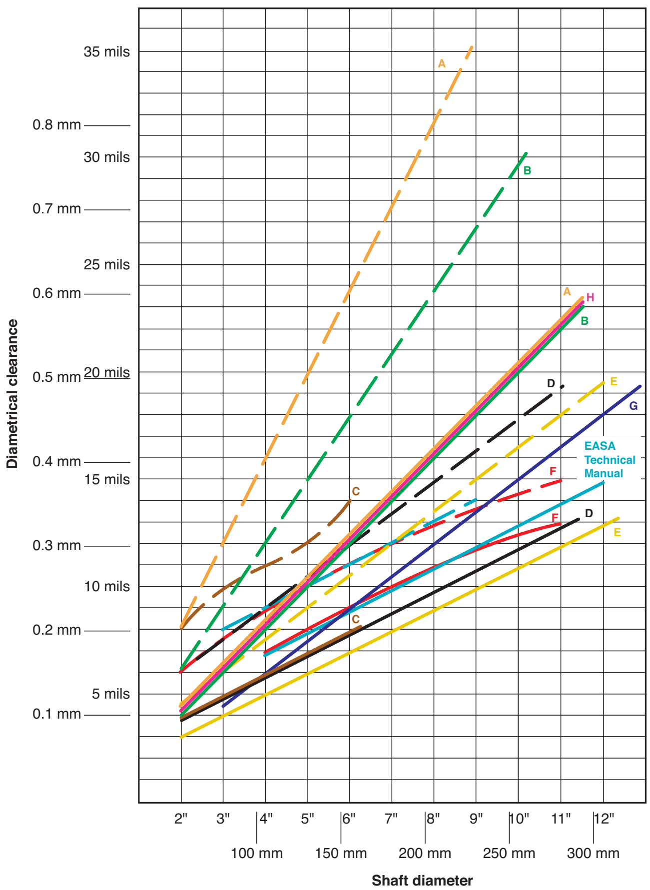
Some of these rules of thumb may look familiar:

- “One thousandth, plus 1 per inch of diameter”
- “Two thousandths, plus 1 per inch of diameter”
- “0.0015 per inch of diameter”
- “0.002” per inch of diameter”

They can’t all be right, yet many of us may have used one of these rules with great success. Which one, if any, is correct? The answer depends on the application.

If your customer base includes slow-speed synchronous motors, you have probably seen motors operating satisfactorily with more than twice the recommended clearance. The person who works primarily on 2-pole petrochemical motors knows that they can vibrate when

FIGURE 4: SLEEVE BEARING CLEARANCES



the bearing clearance is even slightly excessive.

To understand why each of those rules-of-thumb exists, let's sum up the factors a designer of sleeve bearing motors must take into account. Remember the relationship between power, torque and rpm?

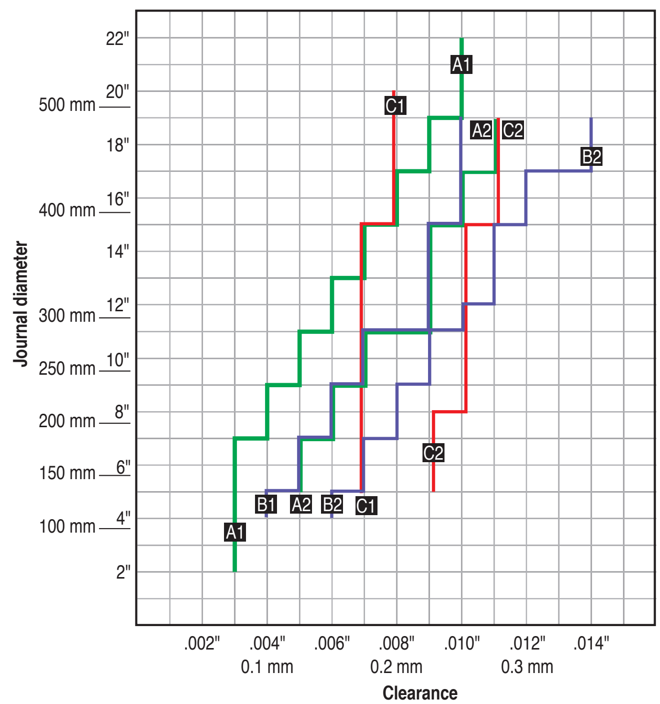
Torque (lb-ft) = hp x 5252 / rpm
or
Torque (N·m) = kW x 9550 / rpm

The higher the torque (lower speed and/or higher hp/kW rating), the larger the required shaft diameter. The heavier the rotor, the larger the bearing must be. The faster the speed, the smaller the allowable journal diameter. The longer the bearing, the more clearance is required to get the oil out.

VERTICAL MACHINES: WHY ARE THEY DIFFERENT?

To begin with, the same spindly shaft and heavy rotor does not result in the shaft deflection we experience in horizontal machines. While the shaft rests on the bottom of a horizontal bearing, it hangs more-or-less centered in the vertical thrust bearing. Radial sag, therefore, is not a concern. As long as there is sufficient radial clearance for the oil film, the vertical guide bearing needs no additional clearance. Guide bearings for vertical sleeve bearing machines are not the cylindrical overshot design, and should have much less clearance than horizontal machines with similar journal diameters. Figure 5 documents the recommended vertical bearing-shaft clearance for two major manufacturers.

FIGURE 5: VERTICAL BEARING/SHAFT CLEARANCE



A1: Length to bore ratio less than or equal to 1:1 (minimum)
A2: Length to bore ratio less than or equal to 1:1 (maximum)
B1: Length to bore ratio greater than 1:1 (minimum)
B2: Length to bore ratio greater than 1:1 (maximum)
C1: Manufacturer (minimum)
C2: Manufacturer (maximum)

LABYRINTH SEALS

Labyrinth seals, for sleeve bearing machines, have some special considerations. There are several styles of labyrinth seals, illustrated below. Some seal better than others, which is important with high speed machines. The simplest arrangement, with a straight clearance fit over a straight shaft (Figure 6), offers the least effective seal. The toothed labyrinth, interrupted by a vacuum break (Figure 7), is the most effective.

The end seal of the sleeve bearing serves to retain 90% of the oil vapor / splashing oil in the chamber. The remaining 5-10% is retained by the labyrinth seals. Clearance between the shaft and labyrinth is critical. The closer the clearance, the better the seal. Because the shaft also moves radially (vibration, bearing clearance, shaft runout), clearance is required as a safety margin. Running contact between the shaft and laby-

FIGURE 6: STRAIGHT PATH SEAL

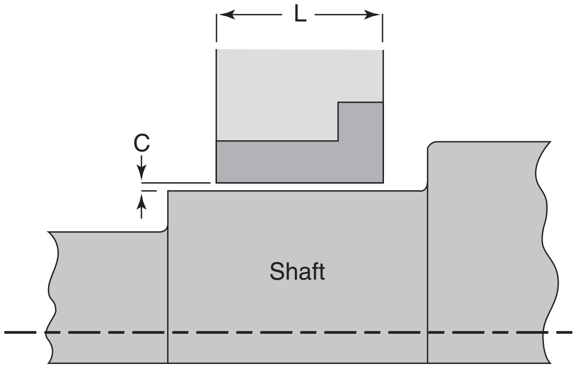
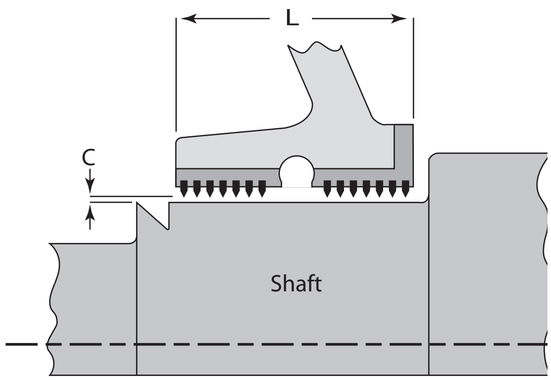


FIGURE 7: LABYRINTH SEAL



rinth seal can damage the shaft and the labyrinth, as well as create high vibration levels.

With a toothed labyrinth seal, each seal is usually divided into at least two sections, with the space between functioning as a vacuum- or pressure-break. The inner section, closest to the bearing, retains the oil. The outer section, closest to the rotor, forms a vacuum break to prevent suction from drawing air through the bearing chamber and labyrinth seal. In most designs, the relief between those two sections is also vented to atmosphere. This is to bypass the bearing chamber, in case a vacuum is present at the labyrinth section closest to the rotor. In 2-pole machines this is especially important, because of the significantly higher airflow.

The best rule of thumb for labyrinth seal clearance is that there should be a little more clearance than the bearing clearance. The closer the labyrinth seal is to the shaft, the better it will seal. If it touches the shaft both may be damaged, and you can expect rapid increases in temperature and vibration levels (especially axial on the end that is rubbing.) Rather than determining labyrinth seal clearance from the shaft diameter, it is better to work from the sleeve bearing clearance. A good guideline, used by several manufacturers, is that the labyrinth seal should have 0.004” – 0.008” (0.10 – 0.20 mm) more diametrical clearance than the bearing.

It should be obvious that the labyrinth seal clearance for a vertical machine can be set closer than for a comparable horizontal machine.

Inspection of labyrinth seals

The area of contact with the labyrinth seal may give further clues about the failure. Contact anywhere other than at the bottom may indicate misalignment. Dirty, oil-soaked windings are a good indication of an ongoing oil leak, caused by excessive clearance of the labyrinth seal or by a pressure-differential between the oil chamber and atmosphere. The longer the leak has been present, or the dirtier the environment, the more dirt will be found mixed into the oil (i.e., the oil will become like mud). (This mud also restricts air flow through the windings, and the oil can damage insulation.)

Perhaps a vent has been inadvertently blocked, or “mud-daubing” insects have nested in the vent opening (not uncommon in cast-in vents, especially in warmer climates). It is important to inspect these areas before any parts are cleaned. Evidence lost may prevent correct interpretation of the failure.

Excess clearance of labyrinth seals can result from a bearing failure that permits the shaft to contact the seal. Typical diametrical clearance for the labyrinth seal of a sleeve bearing machine is 0.007” to 0.020”(.18 to .51 mm), depending on speed and shaft diameter.

Removable labyrinth seals should be sealed during assembly using non-hardening products to facilitate future disassembly.

Lubrication

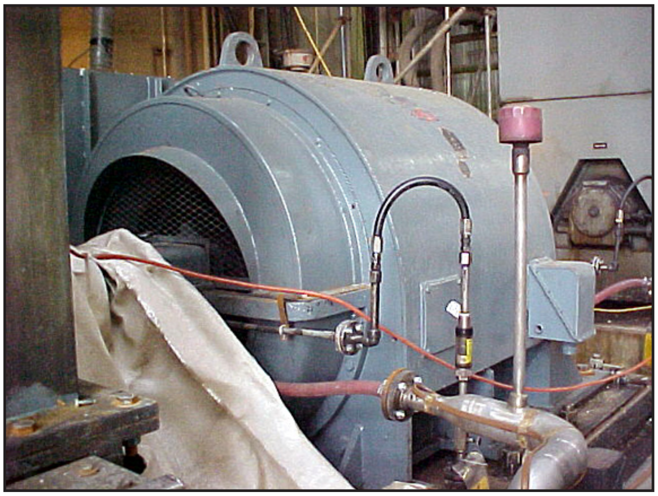
The key to sleeve bearing life is adequate lubrication to maintain minimum friction. A continuous flow of oil is provided by one or more oil rings or a forced-oil system (Figure 8). In each case, oil is moved to the top of the shaft where it fills the oil distribution groove (Figure 9). As the shaft rotates, the oil rings (resting on the shaft) also turn, lifting oil from the sump and transferring it to the bearing and shaft. The oil exits the drain groove at either end, and is cooled by mixing with oil in the reservoir/sump.

Some sleeve bearing designs incorporate guides or wipers that improve the transfer of oil from each ring to the shaft and bearing. Guides also keep the rings tracking straight, which is especially important in high-speed machines. A ring that tracks erratically turns slower and moves less oil, thereby increasing bearing temperature. Oil rings (fig 8, labeled pic) must be round within 0.010” to .015” (or 0.25 - .38 mm) in order to rotate at a consistent speed, and flat. The erratic movement of bent oil rings may aerate the oil and cause foam.

Oil viscosity

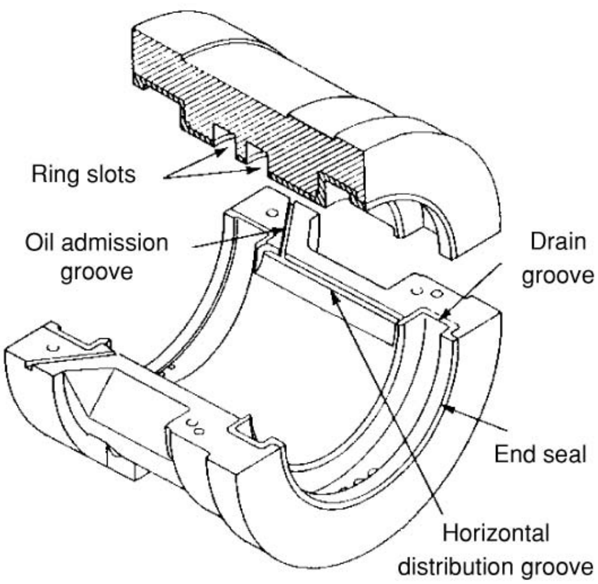
The oil viscosity selected can affect the stiffness of

FIGURE 8: FORCED OIL SYSTEM



The drain line of the forced-lubrication system is larger than the pressure line.

FIGURE 9: SLEEVE BEARING CUTAWAY

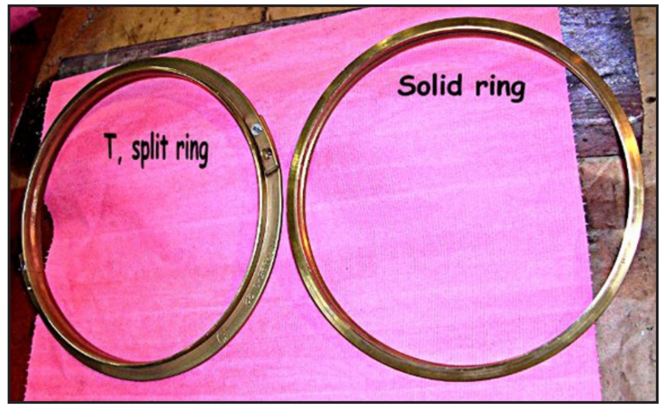


Oil from the rings (or pump) maintains a constant supply of oil in the distribution groove.

the shaft/bearing assembly. Some designs are prone to vibration problems if the wrong oil weight is used. Consult the OEM manual for specific information about recommended oils, especially if a vibration problem seems to defy logic. Sometimes an application limits the choice of lubricants. The food industry, for example, requires the use of vegetable oil for reasons of food safety.

The oil distribution groove, also called a fly-cut or side-pocket, holds in reserve a continuous supply of oil. This reserve ensures a constant oil film between the

FIGURE 10: OIL RINGS



Oil rings may be T-section (left) or trapezoidal (right)

bearing and shaft. The end seal improves recovery of oil as it exits the bearing. When a bearing lacks this drain groove / end seal combination, the labyrinth seal must “work harder” to capture the oil droplets exiting the bearing. Such a bearing design (Figure 11) is prone to oil leaks.

The size of the distribution groove (Figure 11) must be large enough it does not run dry, a problem that can be especially critical in 2-pole machines. If too small, it will not hold enough oil, causing brief interruptions of lubrication. A “patchy” looking babbitt (Figure 12) surface and difficulty in obtaining a good wear pattern suggest overheating due to a distribution groove that is too small. The solution is to enlarge the groove so it holds more oil.

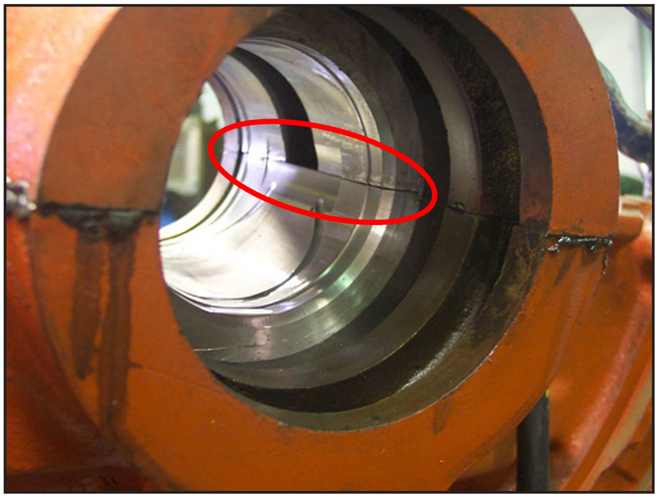
Babbitt grade

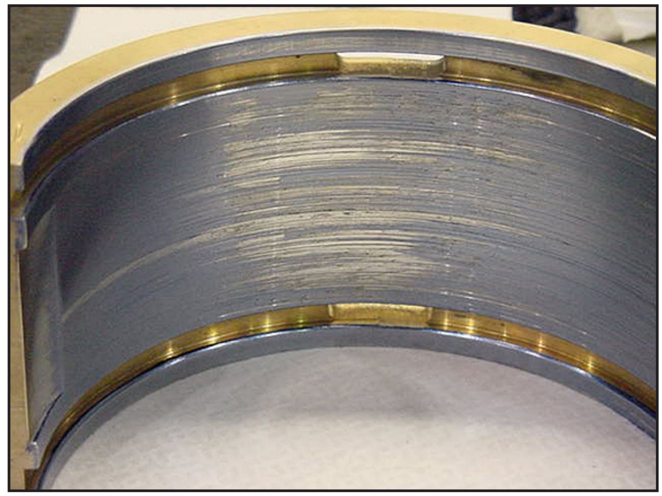
Babbitt grades are selected for specific applications, based on such factors as shaft surface speed, lubrication type and dynamic load. Other considerations include load, temperature and the ease with which dirt can be embedded (e.g., contaminants are much more prevalent in a cement mill than in a food processing plant).

Babbitt grades are classified according to the relative amounts of tin, antimony, lead and copper they contain (Table 1). ASTM alloy grades range from 1 to 19, although babbitt grades 1, 2 and 3 are the most frequently encountered. Tin is the major component of grades 1 through 5, whereas lead is the main ingredient in grades 6 through 19. Lead babbitt has a lower load-carrying capacity than tin babbitt, and is much less resistant to corrosion.

In general, tin-based babbitt bearings for electric motors have load-carrying capacities of 800 to 1500 psi (5760 to 10800 kPA). Lead-based babbitt bearings have capacities of 800 to 1200 psi (5500 to 8270 kPA). The babbitt grade used for a lightweight, high-rpm in-

FIGURE 11: END SEAL, DRAIN GROOVE





Left: This bearing has a oil distribution groove (circled) but has no drain groove or end seal, and has an ineffective labyrinth seal. Right: The drain groove and end seal are critical to oil recovery.

FIGURE 12: INSUFFICIENT SIZE OF DISTRIBUTION GROOVE

The patchy contact pattern is one indication that the bearing is "oil starved." A deeper, wider distribution groove is required.

duction motor will differ from that used in a large, low-speed synchronous ball-mill motor.

Some service centers find it convenient to use the same grade of babbitt for all bearings. Companies specializing in babbitt bearing repair are more likely to have the equipment and the babbitt inventory to duplicate the original babbitt grade. To confirm babbitt grade, have a sample analyzed by a lab or contact the OEM for the original grade. As column 3 in Table 1 indicates, melting temperature is not a reliable indicator of babbitt grade.

Bearing-to-housing clearance

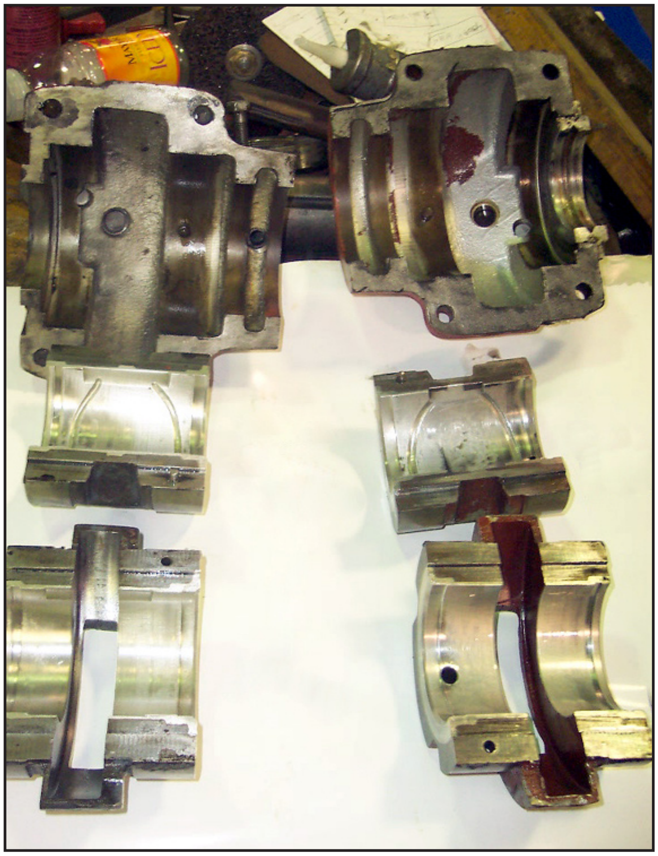
Although bearing-to-shaft clearance receives considerable attention (Figure 13), bearing-to-housing clearance is often overlooked. The different coefficients of expansion for the steel shaft/brass shell/babbitt bearing/cast-iron housing make this clearance necessary. If the bearing-to-housing fit has zero clearance, the bearing shell cannot expand outwards as it heats up. Thermal expansion will therefore cause the bearing to

TABLE 1: BABBITT GRADE

ASTM grade number	Percent tin	Percent antimony	Percent lead	Percent copper	Melting point	Pouring temperature
1	91	4.5	—	4.5	433° F (223° C)	825° F (441° C)
2	89	7.5	—	3.5	466° F (241° C)	795° F (424° C)
3	83	8.3	—	8.3	464° F (240° C)	915° F (491° C)
5	65	15	18	2	358° F (181° C)	690° F (366° C)
7	10	15	75	1.5	464° F (240° C)	640° F (338° C)

While melting temperatures are similar, the correct casting temperatures vary considerably among babbitt grades. Pouring babbitt at tool low a temperature reduces the chance for a good bond between the babbitt and the shell.

FIGURE 13: BEARING AND HOUSING DISMANTLED



The fit between bearing and housing is as important as between bearing and shaft.

grow "in," reducing the bearing-to-shaft clearance. If the bearing-to-shaft clearance becomes too tight, the bearing will fail. Too much clearance between the bearing and housing will cause high vibration.

Most electric motor sleeve bearings perform best with housing clearances of 0.001" to 0.003" (.025 to .040 mm). This clearance may be determined by using a micrometer to measure the bearing OD and housing ID.

One procedure for determining the clearance between the sleeve bearing and the housing is to place a soft wire of lead on top of the bearing. The top half of the bearing housing is then installed, which compresses the malleable lead. After removing the top half of the bearing housing, a micrometer can be used to measure thickness of the compressed lead.

A modern alternative is Plastigage–a plastic extruded wire–available from most auto parts stores. Plastigage is produced in several sizes (0.002" to 0.006"; .004" to 0.009") and has graduated markings on its packaging to simplify measurement.

One advantage of Plastigage rather than micrometer measurement is that it is more likely to reveal elliptical parts. It also gauges the clearance only in the actual fit area. This provides an accurate measurement of clearance for bearings supported by a narrow saddle.

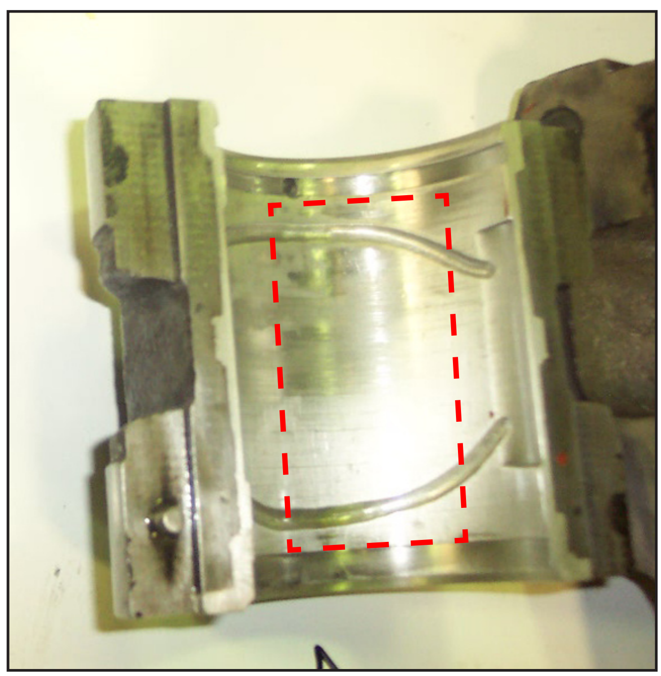
Spherical bearing housings can be especially difficult to measure, making Plastigage more practical. When using Plastigage, it is important to clean any sealant from the flat fit between the bearing bracket and cap, and to fully torque the bolts to insure an accurate reading.

NEW AND REBUILT SLEEVE BEARINGS

Inspection

Inspection of new or rebuilt babbitt bearings should include nondestructive testing (NDT). Ultrasound inspection is the best way to evaluate the bond between bearing shell and babbitt. MIL-spec minimums adopted by some end-users require 80% minimum bond for the load zone (Figure 14), and 40% for the overall bearing. This is somewhat of a judgement call, as the percent bond in the top half of a bearing is not as critical as in the load zone. Likewise, the percent bond for a 2-pole machine is more critical than for a very low-speed application. Common problems affecting the bond are: presence of oil in the bearing shell (or in the material used to seal openings in the shell), failure to tin the shell before rebabbitting, or pouring the babbitt at the wrong temperature (Table 1).

FIGURE 14: LOAD ZONE



Contact in at least 40% of the load zone (delineated by the dashed rectangle) is desirable.

Fitting

Fitting a new sleeve bearing is an important part of the assembly process. Install the bottom half of each bearing and then spin the rotor with the bearing journal dry (or with a small amount of oil wiped onto the journal) to establish a wear pattern quickly. Thrust the shaft axially several times while it rotates. Scraping is generally done using a babbitt knife or bearing scraper (Figure 15), followed by polishing with a Scotch-Brite pad.

FIGURE 15: BEARING SCRAPER

A bearing scraper can be made by annealing and shaping a half-round file. The flat side should be hollow-ground.

The bearing should be thoroughly cleaned after each fitting before being rolled back in for further evaluation. The objective is a minimum of 60% contact centered in the bottom half, with no contact at the corners or top. Why do we want no contact at the top half of the bearing? In operation the shaft rides on a thin oil film, raising the shaft. If the top of the bearing is rubbing the shaft during static fitting, the pressure will increase with the shaft lifting during operation, and smearing of the babbitt is likely.

As a practical matter, most technicians concentrate on the bottom half, then install the bearing top half and securely bolt the top cap in place to retain it for a final spin-and-inspection. This allows the technician to verify that no pinch-points exist. Too tight a bearing-to-housing fit may distort the bearing shell and cause bearing-to-shaft contact that was not evident during the initial fitting process. When the shaft centerline is not perpendicular to the stator-bracket fit, the top bracket half may further alter the bearing-to-shaft alignment.

When replacement sleeve bearings are installed, they must be carefully fitted to ensure adequate contact with the shaft. Because each bearing and bearing housing are slightly different, very few are perfectly parallel with the shaft. Hand-fitting involves repetitive adjustment, often referred to as “scraping in the bearings.” This is because the fitting process involves using a babbitt knife or similar tool to shave/scrape small quantities of babbitt from high regions that would otherwise contact the shaft. Regions of high pressure – even small ones – break down the oil film and elevate the bearing temperature. **Caution:** Never allow a technician to use lapping compound for this. The lapping compound will imbed in the soft babbitt surface and continue to wear the shaft journal.

The usual process is the assemble the motor to the point at which the bearings can be rolled into position. The shaft, bearing and housing must be clean. Raise the shaft just enough to permit the bottom half of each bearing to be inserted. Place a small amount of oil on the bearing journal. A proven method is to dip a finger in oil, smear it across the surface of the bottom half of the bearing, and install the bearing. (See Appendix I.)

Use a strap wrench or spanner wrench to rotate the shaft. As the shaft rotates, thrust it axially to the full limits of its travel. After spinning the shaft for 20-30 seconds, allow it stop with the shaft positioned near the midpoint of its axial limits. Repeat the procedure with the opposite end bearing. Raise the shaft of one end, just enough to roll out the bearing. Visual inspection will reveal the amount of contact between shaft and bearing.

High points of contact are readily identifiable by their shiny appearance. The objective is to obtain a reasonable percent contact area across the bottom half. There are unrealistic “rules” extant, some stating that 80% contact on the bottom half is required. Ignore those; they are neither realistic nor desirable.

In most cases, even 30% contact across the load-bearing zone is adequate, using this fitting method. Use a babbitt knife to scrape the high spots (areas with heavy contact), then polish the entire bearing surface using fine (gray color) Scotch-Brite. Some mechanics use the red (medium) Scotch-Brite, followed by the fine. After polishing, the bearing should be thoroughly cleaned by steam-cleaning. Apply a light coat of oil to the surface, and re-insert the bearing.

Follow the same process for the other bearing. Repeat the entire process as outlined above, until the size of the contact area is acceptable.

For the final fitting, assemble the top half of the bearing and bearing housing, and repeat the spinning / inspection procedure one more time before attempting to run the motor. This step is often overlooked, but it is important. If the bearing bracket is misaligned, or if the bearing-to-housing fit is too tight, this step should reveal contact in the top half of the bearing.

Contact on the top half of the bearing is not desirable. Significant change to the bottom contact regions may indicate that the top half of the bearing housing & bracket are not properly aligned. Possible causes include:

- The split-line of the bracket may not be perpendicular to the face of the stator.
- The stator face might not be perpendicular to the shaft axis of rotation. Observe the top half of each bracket for looseness while tightening/loosening the bolts.
- Dowel pins – or their holes – may be distorted.

General guidelines

Contact from 4:00 to 8:00 is preferable to 100% contact, or to contact on the sides alone. The shaft should ride on a film of oil. In practice, the shaft will contact the bearing just to one side of the center, depending on the direction of rotation. The shaft tends to "climb" the bearing. That is, if the shaft rotates clockwise, the contact is likely to be centered around 5:00, whereas a shaft rotating CCW tends to center around 7:00. The actual position is affected by the speed, rotor weight, shaft diameter and other factors.

Once the bearings are adequately fitted, finish assembling the bearing housings, with oil-rings, etc. Fill to the correct oil level, and prepare the motor for a test run. For 2-pole machines, it is especially critical to monitor the bearing temperature. If a region of contact remains, the resulting friction can raise bearing temperatures so quickly that there is little time to react before bearing damage occurs. Use a vibration analyzer, placing the accelerometer on the bearing housing (axially), to monitor velocity. If the bearing friction increases, the velocity reading will suddenly begin to climb. Consider that as an early warning, before the bearing temperature is observed to increase, that might save a bearing. When possible, use an accelerometer on each end of the motor, one on each bearing housing. This early-warning system is local to the end with the failing bearing.

Labyrinth seal contact will also be revealed as a high axial reading on the end with the additional friction. Axial vibration levels should be nearly identical on both ends of the machine. Higher magnitude of axial velocity on one end of the machine is a strong indication of hard bearing contact or a labyrinth seal rubbing the shaft during operation.

Start-up

Test-running a motor with a newly-fitted bearing requires some caution, because actual contact between the bearing and shaft will result in rapid heating. A high spot on the babbitt bearing surface will interrupt the oil film on which the shaft rides. The resulting friction produces heat that can damage the bearing.

Two-pole machines require special care on start-up, because of the higher surface speeds. When test running a low-speed machine, an increase in bearing temperature can be detected and the machine stopped before damage occurs. With a 2-pole machine, cause and effect are almost simultaneous.

The fitting tip for early detection of a wiping bearing employing a vibration analyzer also applies here. Place the accelerometer probe axially on the bearing housing, with the instrument set to read velocity. The friction caused by a wiping bearing will be indicated as a mechanical rub as the velocity starts to climb, even before the temperature changes. The extra few seconds of warning obtained in this manner have saved many 2-pole sleeve bearings from catastrophic failure.

End float and magnetic center

When a sleeve bearing motor is energized, the rotor should remain centered between the thrust shoulders. During coast-down the rotor may float and contact either thrust shoulder, especially if the motor is not level. To avoid possible damage during extended coast-down, plug-reverse the motor (with reduced voltage) to stop it quickly. When this is not practical, use mechanical means to position the shaft near its mechanical center.

End float and magnetic center are important considerations when rebuilding sleeve-bearing machines. The magnetic center should be clearly marked during the final test run. The recommended procedure is to coat the shaft extension with blue layout spray and then clearly scribe the magnetic center. Never use a file to scribe a deep mark in the shaft – the resulting stress raiser can contribute to shaft breakage in high-torque applications.

TABLE 2: END PLAY AND ROTOR FLOAT FOR COUPLED SLEEVE BEARING HORIZONTAL INDUCTION MOTOTS

Machine hp	Synchronous speed	Min. rotor end float	Max. coupling end float
500 hp and below	1800 rpm and below	0.25" (6.5 mm)	0.09" (2.3 mm)
300 to 500 hp	3600 and 3000 rpm	0.50" (13 mm)	0.19" (4.8 mm)
600 hp and higher	All speeds	0.50" (13 mm)	0.19" (4.8 mm)
NEMA MG 1-2006			

Total end float should be documented. The mechanical center and the magnetic center should closely coincide, although there is no standard for how closely. Table 2 is a compilation of NEMA MG1 limits for minimum end float for sleeve bearing motors by HP and RPM, and for maximum allowable coupling end float. Note that, when a machine is properly aligned when installed, there should not be axial thrust wear.

In some cases the magnetic center may "hunt," especially with 2-pole machines, which have weaker axial centering forces. See Appendix II for the May 2005 EASA *CURRENTS* article, "Axial 'Hunting' of 2-Pole Motors: Causes and Cures."

Bearing temperature

Bearing temperature is influenced by several factors,

including the quality of the fitting process, rotational speed and oil used. Sleeve bearing temperatures above 150° F (65° C) can usually be improved by fitting. While some motor designs are subject to inherently higher temperatures, in rare cases as high as 220° F (105° C), such machines are usually fitted with auxiliary cooling via forced-lubrication.

Babbitt bearings require extra attention during inspection, especially when the purpose is to determine the cause of failure. Treating the symptoms rather than the problem is common because of the difficulty in interpreting evidence. Working with the end-user is essential when trying to determine why a sleeve bearing failed. Machine history and knowledge of the application system—not just the motor—are key to satisfactory service.

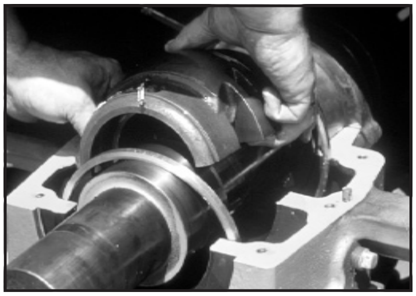
CONCLUSION

There are a lot of sleeve bearing clearance tables circulating around our industry, but some of them are suited to a specific type of motor – like low-speed synchronous motors – and should never be applied universally. In broad terms:

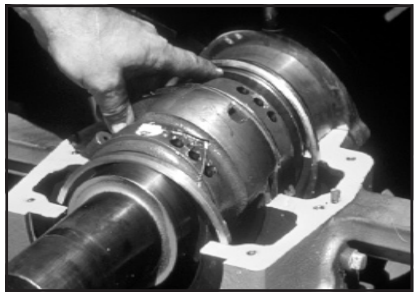
- Low speed motor bearings can operate with more clearance.
- Longer bearings require more clearance.
- Vertical bearings require less clearance.
- Hermetics require tighter clearance.
- Labyrinth seals should be as close as possible, without contacting the shaft.

About those rules of thumb: If you have had good success with one for your routine jobs, watch out for sleeve bearings that differ from your usual work. Examples would include 2-poles versus low-speed machines, a significantly different length/diameter ratio, or vertical machines. In those cases, contact the manufacturer or EASA Technical Support for help.

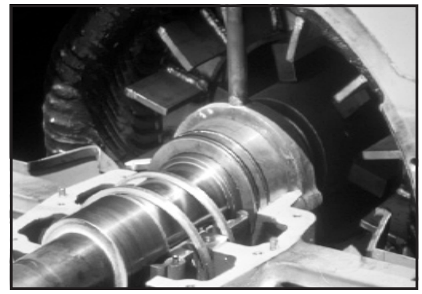
APPENDIX I: INSTALLATION OF A SLEEVE BEARING



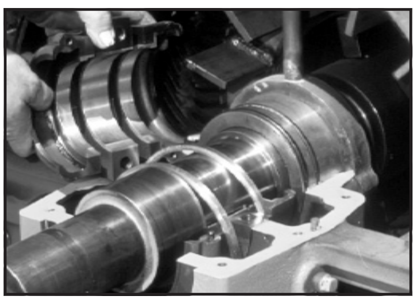
1 Check the bearing markings. Be sure any holes are properly oriented. Verify that the insulated bearing, if any, goes on the opposite drive end. With the bracket mounted, place the bearing lower half on the shaft. Smear a small amount of oil on the bearing surface.



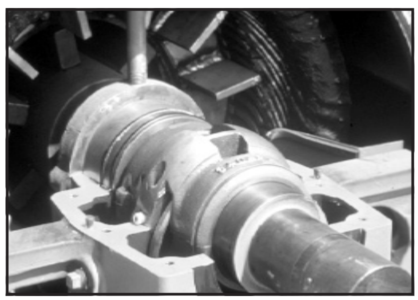
2 Use a crane jack to lift the shaft slightly. Keep control of the bearing and roll it into position. Level both sides of the bearing flush with the bracket split line.



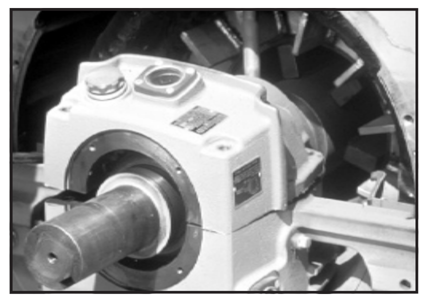
3 Align the oil rings with the relief grooves of the upper bearing half.



4 Install upper bearing half. Carefully line up the dowel pins with the corresponding holes.



5 Bolt bearing halves together. Verify that the oil rings are free to turn.



6 Install top bearing cap and cross tighten the bolts. When assembled for the final time, a non-hardening sealant should be applied to the bracket split line.

7 Install the labyrinth seal. The mating fit to the bracket should be sealed using a non-hardening sealant. Use shimstock wrapped around the shaft to center the labyrinth seal.

8 With the top bearing cap and labyrinth seals installed, double-check the clearance between the labyrinth seal and shaft. Be sure to use pipe sealant on the pipe pugs and sight glasses.

Courtesy of U.S. Electrical Motors

APPENDIX II: AXIAL "HUNTING" OF 2-POLE MOTORS: CAUSES AND CURES

By Chuck Yung
EASA Technical Support Specialist

This article was original published in EASA *CURRENTS*, May 2005.

A common observation about 2-pole machines fitted with sleeve bearings is the inherent weak magnetic centering force. The classic symptom is chronic axial movement: a 2-pole rotor drifting "to and fro" from the established magnetic center position. This article addresses the numerous causes of this phenomenon, colloquially referred to as "hunting." Although the focus is on 2-pole motors, much of this information applies to sleeve bearing motors of any speed rating. Identifying the cause of a problem is good, but solutions are a lot more useful, so I've included those as well.

We can use magnets to describe how a motor works. Opposite poles attract; like poles repel. The magnetic field rotating within the stator turns the rotor, and magnetic force affects the axial position of the rotor relative to the stator core.

The basics required to explain axial "hunting" are:

- Opposites attract; like poles repel
- Inverse relationship between distance and magnetic force
- Gravity
- Rotor bar skew
- Aerodynamic forces
- Various combinations of the above forces

ATTRACTION/REPULSION

Consider an electric motor as a collection of magnets: a north and south pole constitute each pole-pair. Opposite poles attract. The more magnets (of equal strength), the stronger the attracting force. Axial centering forces are proportional to the number of poles. Two-pole machines—with only 1-pole-pair—have weak axial centering forces. The more pole-pairs, the stronger the forces acting to hold the rotor in its axial position.

The components that influence magnetic centering force are:

- Position of stator core ends relative to the rotor core ends
- Airgap between rotor and stator
- Level of the shaft
- Skew, if any, of rotor bars
- Vent ducts, if present
- Line voltage
- No-load current
- Endring extension of the rotor, beyond the rotor core

The axial centering force can be determined by:

$$F = (K \times E \times I \times [E_f + D_f]) / L$$

Where: K = a constant, 0.02
E = line-to-line voltage of stator
I = No-load line current
L = Stack length (inches)
E_f = Sum of core end forces
D_f = Sum of individual stator-rotor vent force factors

If the shaft is not level, gravity acting on the rotor produces an axial force towards the low end. If the machine has no vent ducts, then D_f would be zero, while for a machine with a large number of vent ducts D_f can be relatively large.

NEMA MG1 establishes minimum end float limits for sleeve bearing machines (See Table 2, Page 9).

Some applications hamper our ability to use limited endfloat couplings, as stipulated in the third column of Table 1, Page 9, (that's the part of MG1 many motor users miss.) Axial movement during operation can be far more than a nuisance. Expensive bearings and possibly couplings may be damaged, production interrupted, and vibration-monitoring equipment might give nuisance alarms.

The relative axial position of the rotor and stator is often mistakenly attributed to physical centering of the rotor within the stator core. In fact, the magnetic centering effect is on the center of mass of the laminated core. In most cases, the two—center of core length and center of mass—coincide. But the exceptions can be challenging.

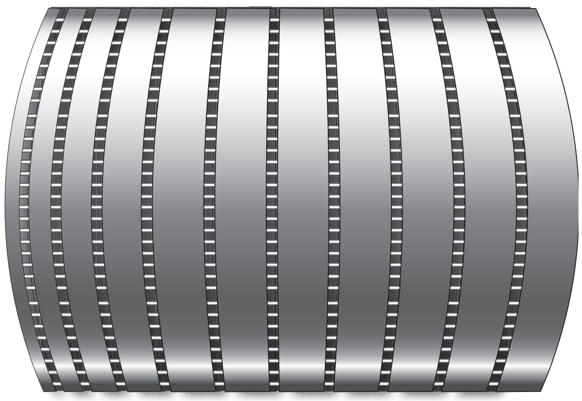
VENT DUCTS

Vent ducts in the stator and rotor must be symmetrical with respect to the midpoint of the stacked core. When practical, manufacturers align the stator and rotor vent ducts to maximize the effectiveness of airflow required to cool the machine. For machines with higher peripheral speeds, a siren effect sometimes results from the close proximity passing of vent duct supports in the stator and rotor. Objectionable noise forces the designer to offset the vent ducts of the stator and rotor.

The rotor will seek the center of mass of the laminated stator core. If a laminated stator core has vent ducts and those ducts are not symmetrically spaced, the rotor will shift towards the end with the most iron.

There are two obvious ways this might occur. There are designs where the center of the iron is not the same as the midpoint of the stator core length. Irregular vent duct spacing, perhaps due to unique ventilation requirements, may be the reason (Figure 1). There are cases where an OEM designs a non-symmetrical motor, and someone later reverses the stator to change the mount-

FIGURE 1: IRREGULAR VENT DUCT SPACING



ing (e.g., F1 to F2). Such a dissymmetry might also be caused by an error of stacking during manufacture, or an improperly done restack of the stator (or rotor) core. In either case, the magnetic center position when loaded is liable to move from the unloaded position.

In the second example, it might be necessary to restack the stator core. An alternative is to machine false vent ducts in the rotor to correspond to the stator vent ducts, as described in *EASA Tech Note 15.*

LEVEL

For 2-pole machines, with their weak magnetic centering forces, accurate leveling of the shaft is important. With most equipment installed decades ago, perhaps by contractors unfamiliar with the need for extremely accurate leveling, and foundations settling over time, it is not uncommon for a motor shaft to be off level. A 2-pole motor coupled to a pump can, upon shutdown, coast for upwards of an hour. If the motor is not level, significant sleeve bearing damage is possible during coast-down.

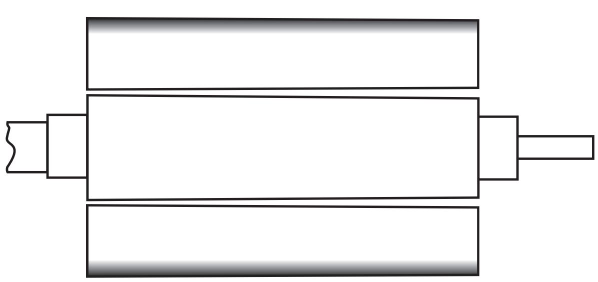
TAPERED AIRGAP

The airgap is the physical distance between the stator and rotor. If the airgap differs from one end to the other (tapered rotor or tapered stator bore), the rotor will be pulled towards the end with the smaller airgap (Figure 2). The explanation is simple: There is a square relationship between magnetic force (F) and the inverse of the distance (d) between the parts.

$F = (1/d)^2$

If the airgap is 20% closer at one end of the machine, the magnetic attracting force is 144% greater (1.2 x 1.2 = 1.44) at the end with the closer airgap. The rotor will be pulled axially towards the closer airgap, moving the shaft off the expected magnetic center. There are crane motor designs, with tapered stator bore and rotor, which take advantage of this fact to release the brake.

FIGURE 2: TAPERED AIR GAP



The result of a tapered air gap is the same, whether it is the stator bore or the rotor that is tapered.

A tapered rotor can result from machine tool wear during the manufacturing process or an offset tailstock on a lathe. For the service center, correction may be as simple as a very light skim cut of the rotor surface. Consultation with the OEM is strongly recommended to determine the maximum allowable airgap before machining the rotor OD. Tool speed must be controlled to avoid surface shorting of the rotor laminations. Two recommendations from manufacturers are:

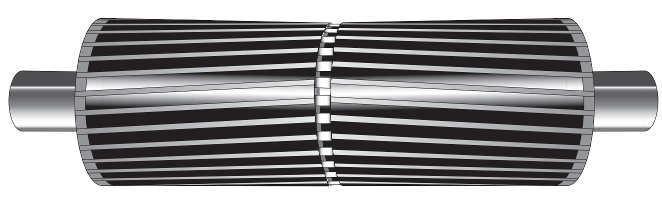
- Sharp tool at 400 – 450 feet per minute
- #4 radius tool at 850 feet per minute

A tapered stator bore is more difficult to correct. Machining the stator bore is fraught with the risk of a grounded winding. To rewind and restack a stator core is costly and time-consuming. Mounting a rotating grinder on a tool post minimizes lamination smear that could increase core losses and reduces the chance of a lamination being pushed into a coil.

When the airgap is uniform on one end but offset on the other, the effect is similar to a tapered airgap. Although the axial displacement force is not as strong as with the tapered airgap, it can still be problematic.

It has long been known that the rotor should be concentric to the stator within 10% of the average airgap. For 2-pole machines, other factors—including reduced shaft stiffness—a tolerance of 5% can reduce problems with noise and vibration.

FIGURE 3: EXAMPLE OF HERRINGBONE SKEW



ROTOR CAGE SKEW

Rotor cage skew is frequently used to smooth out the torque curve and/or reduce electrical noise (Figure 3). Torque developed in the rotor is perpendicular to the rotor bars. When the rotor bars are skewed, the result is an axial component to the torque. The greater the skew, the greater the axial portion of torque results from the

skew and the more likely that the rotor will be displaced axially when the machine is loaded. The OEM skew should never be changed without discussing the situation with the manufacturer or a qualified engineer. Elimination of rotor bar skew can cause significant changes in the torque curve, as well as electrical noise.

AERODYNAMIC FORCE

Aerodynamic force is especially strong in 2-pole machines, which often have directional fans located inboard of both bearing housings. Steep fan blade pitch not only increases the airflow but also the axial aerodynamic force.

When the aerodynamic forces resulting from opposing fans are not equal, the stronger fan may pull the rotor off magnetic center. The relative position of fan and baffle are factors, as is the distance from fan to bracket, and the duct friction in each end. WP-I and WP-II enclosures are sometimes designed with nonsymmetrical ventilation passages. Blocked openings can upset the balance of airflow.

As the rotor is displaced in the axial direction, the force required to restore the magnetic center increases. The rate at which the required force increases decreases with the distance from magnetic center. The increase in axial restoring force can be approximated by raising the increase in axial offset distance to the 0.75 power. For example, if the rotor is offset 0.04” vs. 0.08” the force increases by about the $2^{0.75}$ power, or 1.7 times. If the offset were a factor of 4 instead, the increased force would be roughly 2.8 times as great ($4^{0.75} = 2.8$).

THRUST-LIMITING AUXILIARY BEARING

When the only negatives resulting from axial float are cyclical vibration and the actual movement, there is a mechanical solution. The solution is to install—on the opposite drive end—a ball bearing in an oversized housing, to prevent axial movement of the rotor.

The ODE bracket should be machined with a flat mounting surface perpendicular to the axis of rotation and a locating fit machined before the machine is assembled. During the test run, a magnetic center is marked, usually by scribing the drive end shaft where it exits the labyrinth seal. After running the machine, the shaft is positioned with the rotor on magnetic center and dimensions taken so that the ball bearing housing and stub shaft can be constructed and installed. The housing should be designed with enough depth to permit shimming of the captive bearing to hold the shaft on magnetic center.

TABLE 2: CAUSES AND SOLUTIONS FOR AXIAL HUNTING OF 2-POLE ROTOR

Cause	Solution
Tapered rotor	Machine rotor OD at appropriate speed
Tapered stator bore	Restack stator core
Shaft not level	Level the machine and realign equipment
Nonsymmetrical stator stack	Restack and rewind OR false vent ducts
Excess rotor bar skew	Add ball bearing to limit axial movement
Aerodynamic forces (windage)	Adjust baffles, false vent ducts
Unbalanced voltage	Balance the voltage to within 1%

UNBALANCED VOLTAGE

Line voltage and current affect axial centering force. It may come as a surprise to know that unbalanced voltage affects the rotating magnetic stator field. Uneven magnetic force, combined with any of the above-listed factors, can cause axial oscillation of the rotor.

Causes and solutions for axial hunting of 2-pole rotors are summed up in Table 2. When all else fails, the endplay-limiting ball bearing can be relied on to stop axial movement, as well as dual magnetic centers.

There are two additional issues with magnetic centering of a rotor in a sleeve bearing motor. First, the magnetizing current (no-load amps); and second is the load current. If a magnetic centering challenge seems to be load related, we can focus on the load-related factors:

- Rotor skew; since torque is developed perpendicular to the rotor bars, and is proportional to rotor current, rotor skew effect is noticed with increasing load.
- Endring component; like skew, the current carried by the endrings varies with load.

Both of these contribute to the brief axial thrusting that may occur during starting, when the rotor current is several times greater than full-load current.

DUAL MAGNETIC CENTERS

Tapered airgap, a tapered rotor, non-symmetrical stacking of the stator or rotor; these can result in dual magnetic centers. If the magnetic center changes from the no-load test in the service center to the customer’s loaded condition, those are prime candidates.

Unbalanced voltage might explain why the customer’s no-load shaft position differs from the results you had on the test panel. It might just be a case of a shaft that is severely out of level.

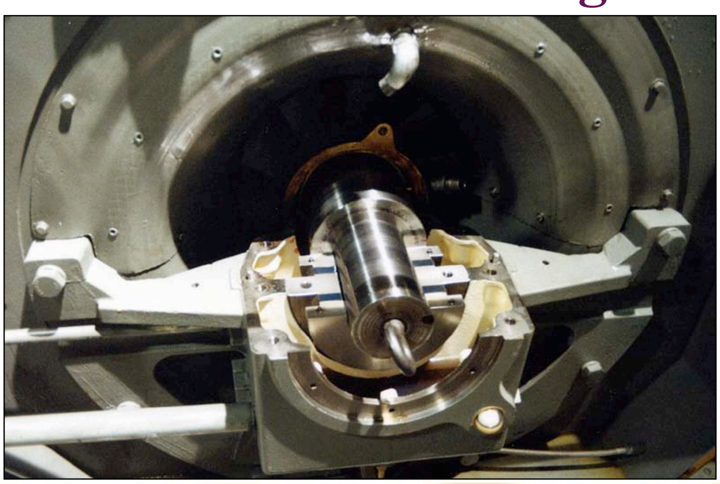
Sleeve Bearing Repair

2007 EASA Convention Minneapolis, MN

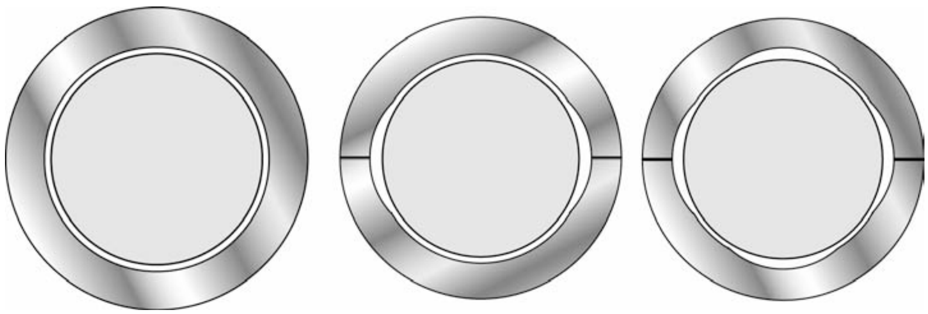
Chuck Yung
EASA
St Louis, MO

Leadership • Vision • Action

Sleeve / Plain metal / White metal Babbitt bearing

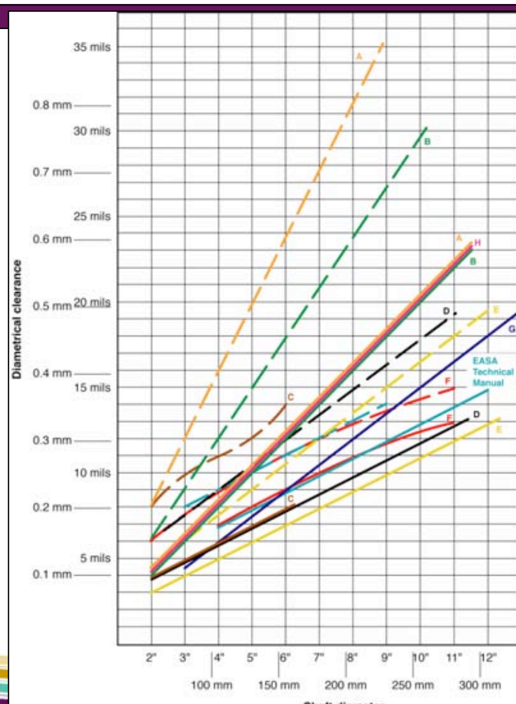


Cylindrical overshot bearings



Cylindrical overshot 2-lobe 4-lobe

- Bearing clearance depends on shaft diameter, bearing length, rpm, shaft weight and lubricant.



Heavy rotors sag, require extra bearing clearance

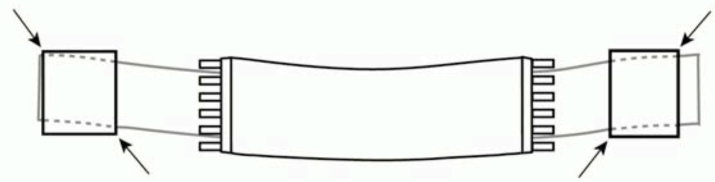


Figure 3. A heavy rotor increases shaft sag. The journal may rub the bearings at the places indicated by arrows. To prevent that, the designer increases the clearance.

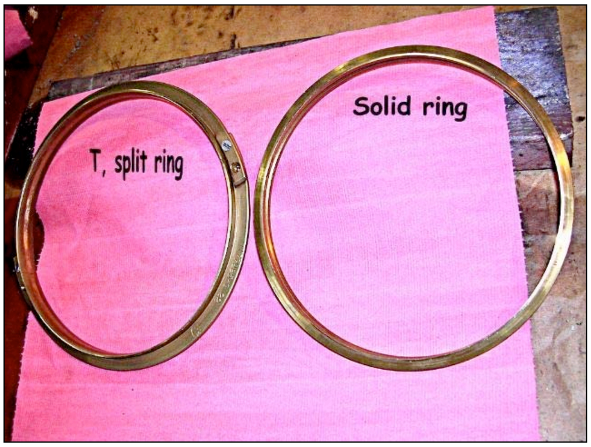
Babbitt grade, pouring temperature

TABLE 1: BABBITT GRADE

ASTM grade number	Percent tin	Percent antimony	Percent lead	Percent copper	Melting point	Pouring temperature
1	91	4.5	—	4.5	433° F (223° C)	825° F (441° C)
2	89	7.5	—	3.5	466° F (241° C)	795° F (424° C)
3	83	8.3	—	8.3	464° F (240° C)	915° F (491° C)
5	65	15	18	2	358° F (181° C)	690° F (366° C)
7	10	15	75	1.5	464° F (240° C)	640° F (338° C)

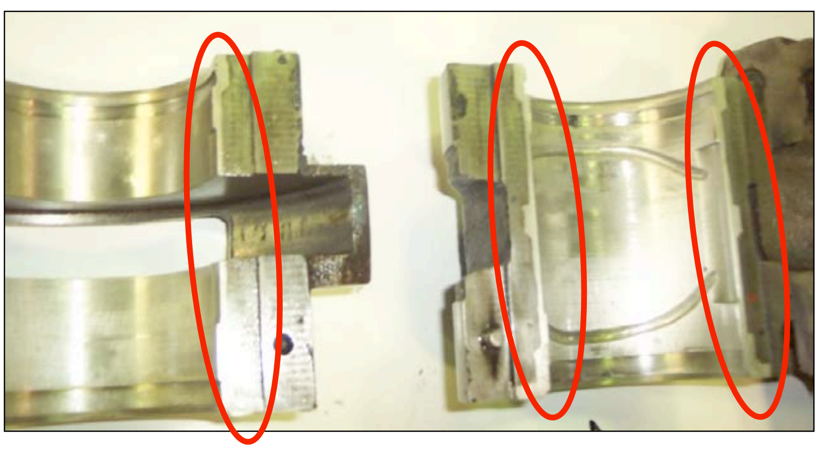
While melting temperatures are similar, the correct casting temperatures vary considerably among babbitt grades. Pouring babbitt at tool low a temperature reduces the chance for a good bond between the babbitt and the shell.

Oil rings must be round, flat

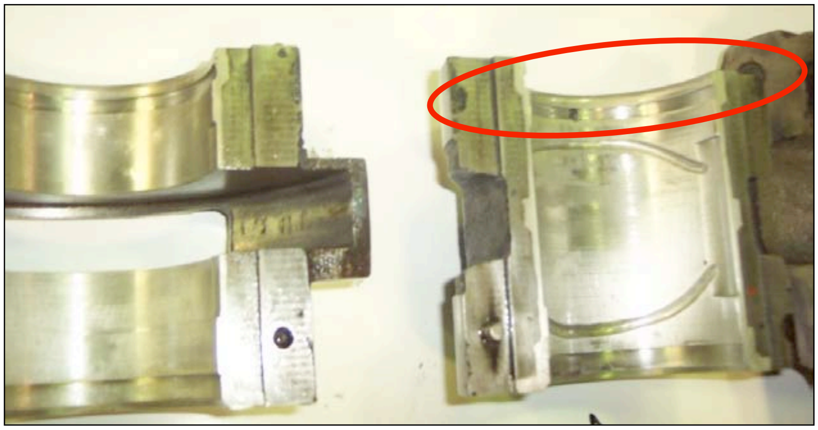


- Round with 0.010” – 0.015” (.25 - .4mm)
- Flatness check on flat surface.

Pinch split line to check bond



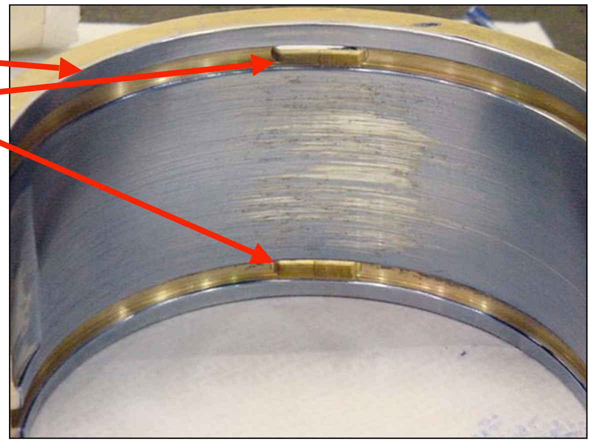
Pinch split line to check bond



Sleeve bearing end bracket

End seal, drain groove

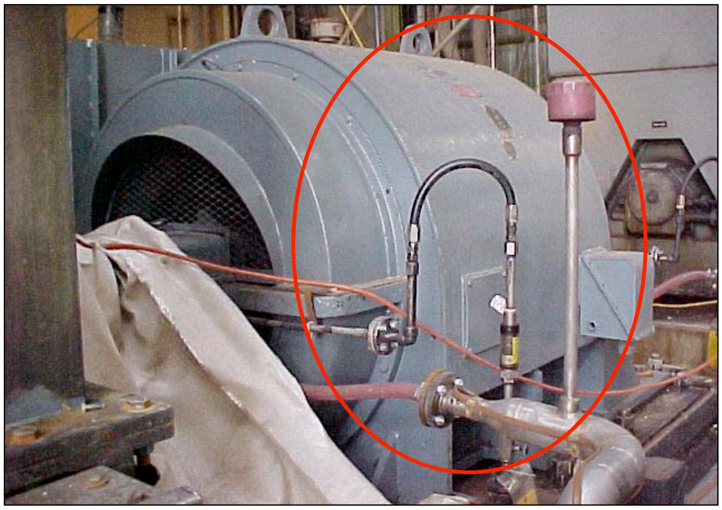
- End seal and drain groove assist the labyrinth seal



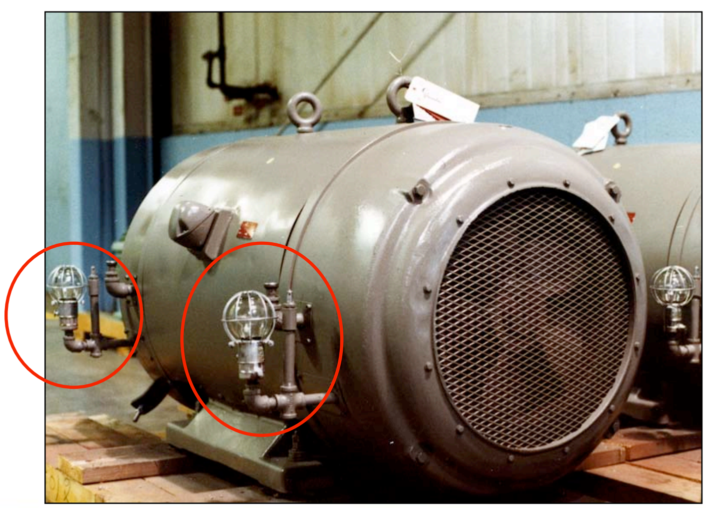
Forced-oil lubrication

- Beneficial for:
  - High ambient temperature
  - Low peripheral speed (Adjustable Speed Drives)
  - Frequent starting
  - Reduce operating temperature of bearings

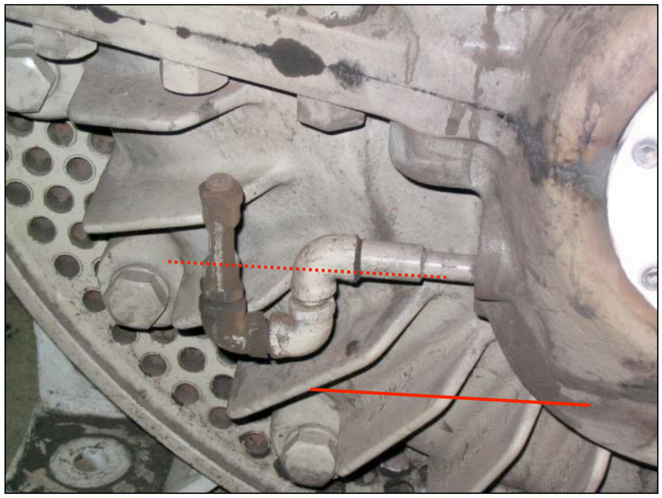
Force-oil lubrication



Automatic oilers



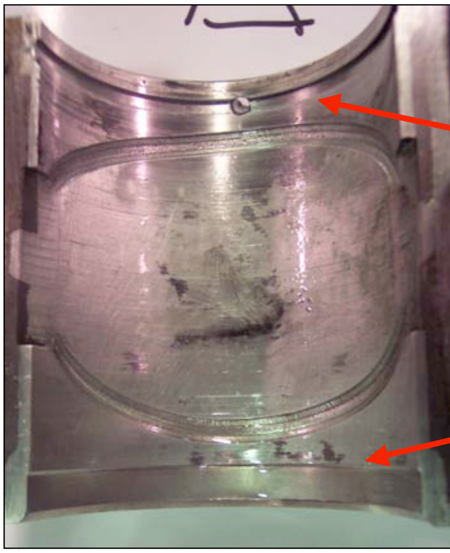
"P" trap defeats the sight glass



Oil retention and labyrinth seals

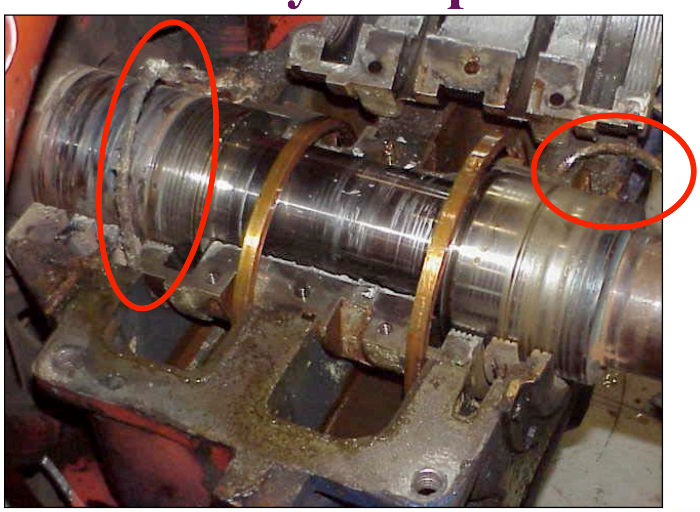
- Drain groove and end seal capture 95% of oil exiting the bearing
- Labyrinth seal captures the rest
- Absence of drain groove/end seal requires a better labyrinth seal

Oil retention



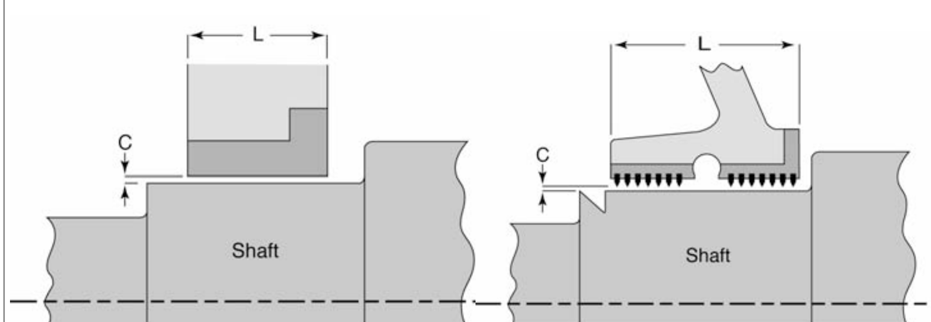
- Drain groove captures 90% + of oil exiting the bearing
- Lack of drain groove requires more intricate labyrinth seal.

Oil leak history and poor solutions



Least effective labyrinth seal

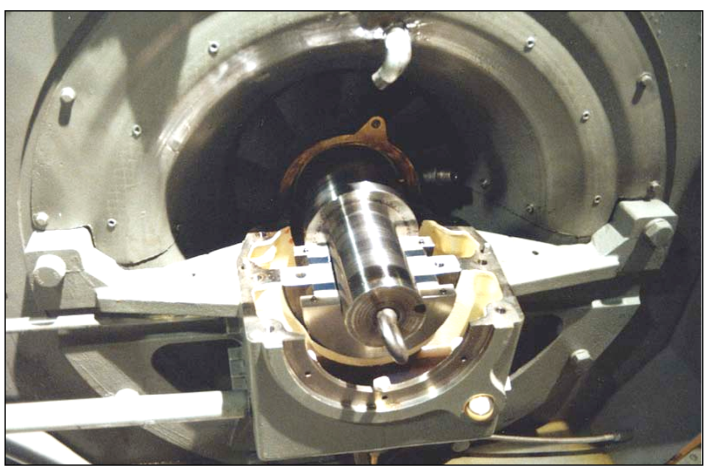
Labyrinth seal effectiveness factors



The toothed labyrinth seal (right) is twice as effective as the straight seal (left) of the same length.

Two vacuum/pressure breaks

Inner labyrinth vented to baffle is characteristic of 2-pole machines

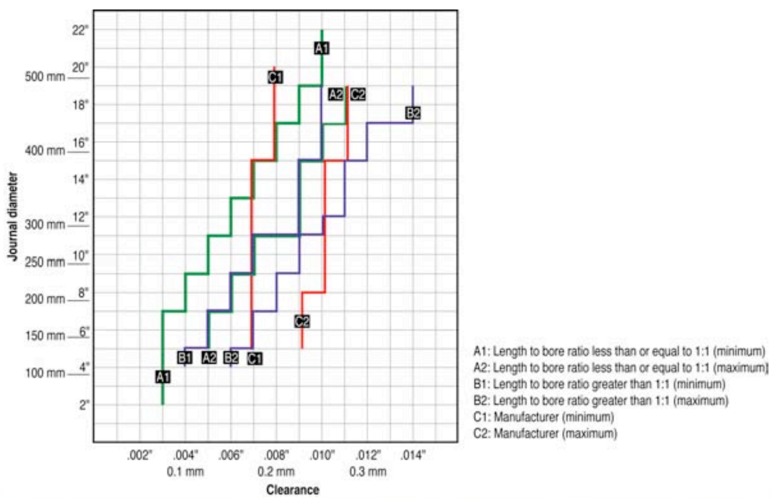


Vertical sleeve bearings

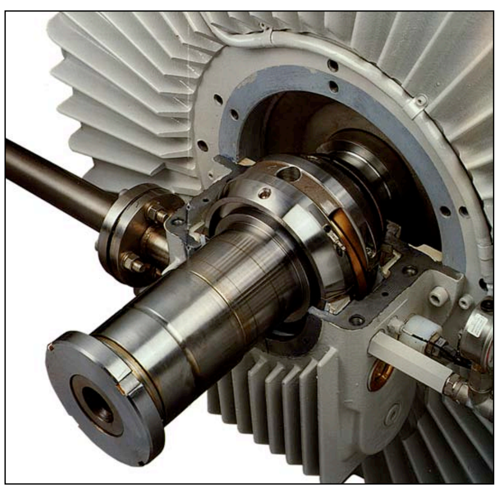
- Vertical shaft = no angular deflection from "sag"
- Less shaft-to-bearing clearance required
- Labyrinth seal clearance also tighter

Vertical sleeve bearing

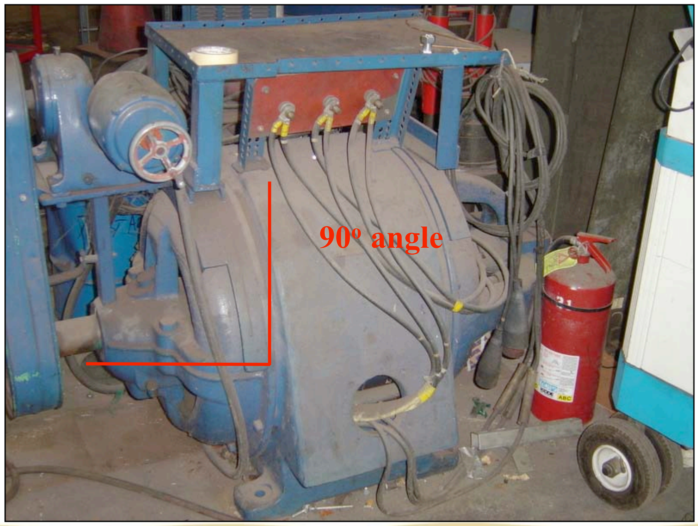
Vertical sleeve bearing clearance



Cylindrical overshot bearing



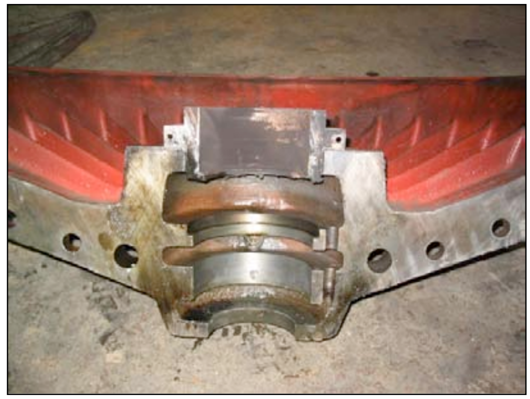
End bracket square?



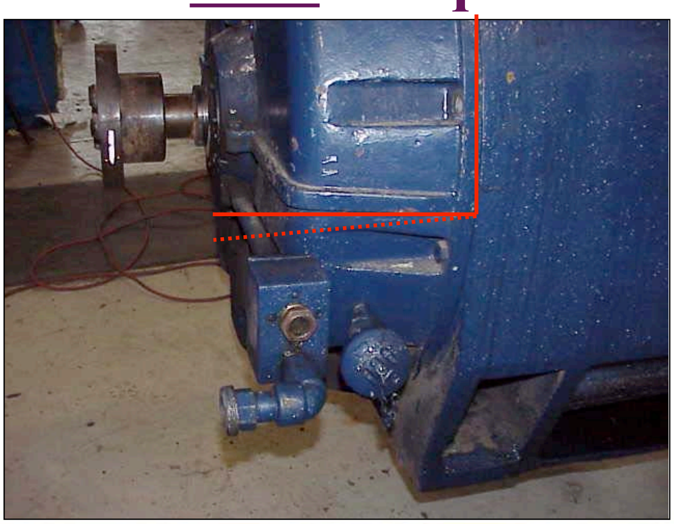
Bracket or stator geometry

- Frame or bracket not square?
- Bracket split-line irregular?
- Emergency situation?
- Machine the frame or bracket to square
- Restore flatness
- Shim gaps near bolt-holes

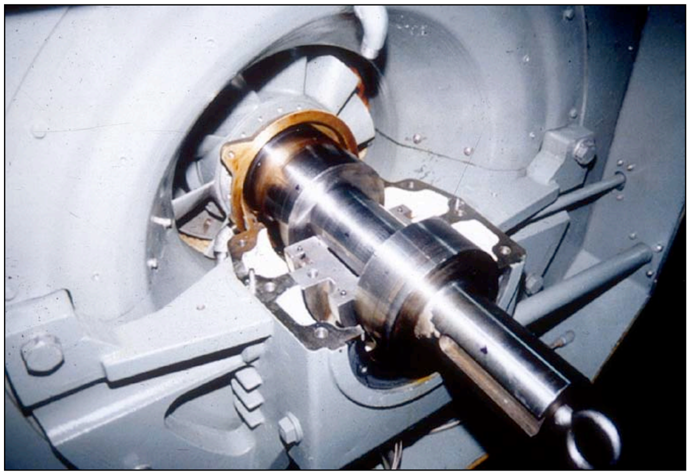
Split brackets = additional opportunities



Brackets must be square to stator



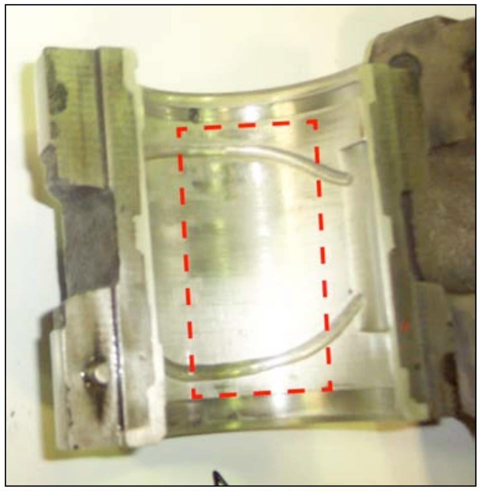
Half-bracket reduces distortion



Fitting of sleeve bearing

- Contact desirable in bottom only
- 40% or more of load zone
- No pinching of sides / corners
- No contact on top half during operating
- Geometry of bearing changes if bracket is distorted

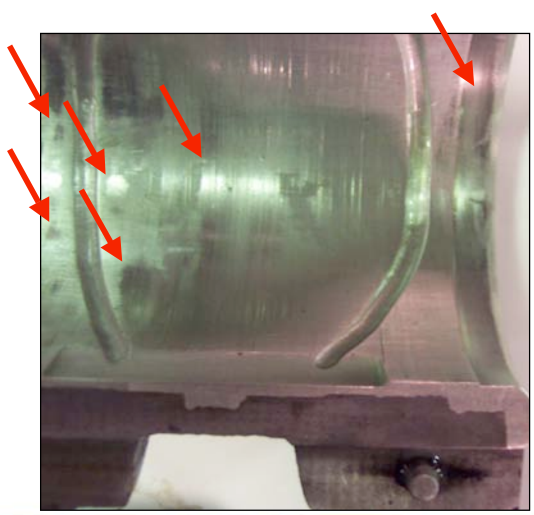
Load zone and contact area



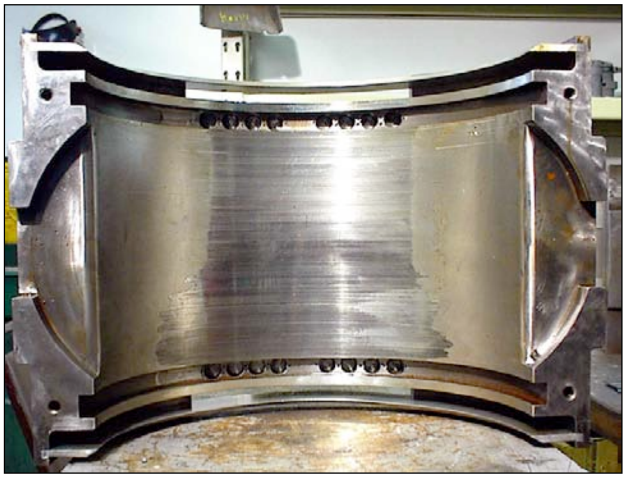
Wiping, too broad contact zone

Bearing scraper / knife

Contact areas

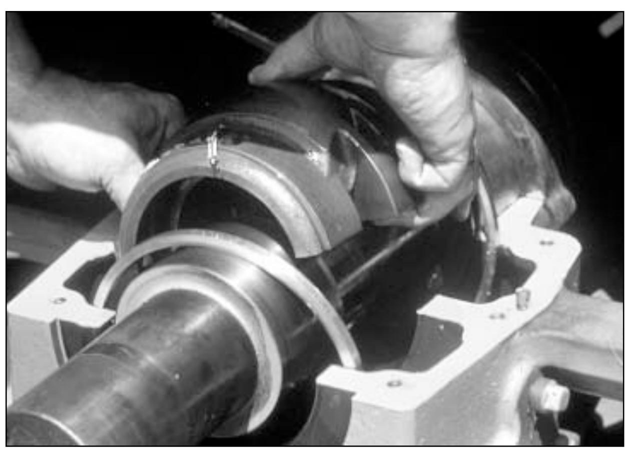


Heavy wear, low peripheral speed



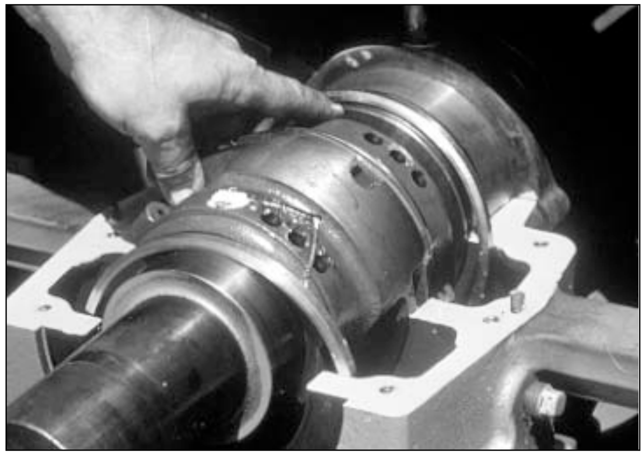
Installation of a sleeve bearing

Check the bearing markings. Be sure any holes are properly oriented. Verify that the insulated bearing, if any, goes on the opposite drive end. With the bracket mounted, place the bearing lower half on the shaft. Smear a small amount of oil on the bearing surface.



Installation of a sleeve bearing

Use a crane jack to lift the shaft slightly. Keep control of the bearing and roll it into position. Level both sides of the bearing flush with the bracket split line.

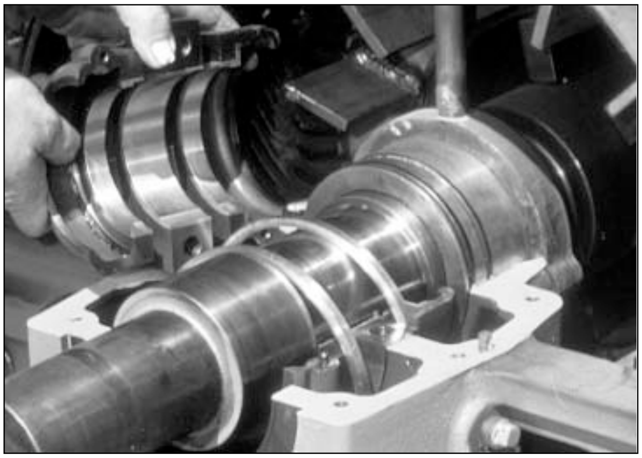


Installation of a sleeve bearing

Align the oil rings with the relief grooves of the upper bearing half.

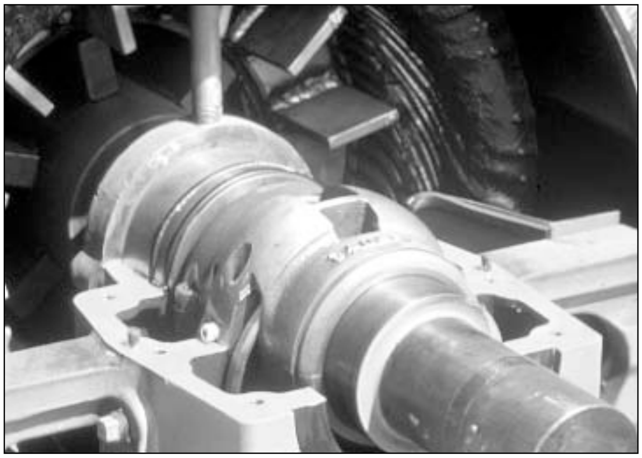
Installation of a sleeve bearing

Install upper bearing half. Carefully line up the dowel pins with the corresponding holes.



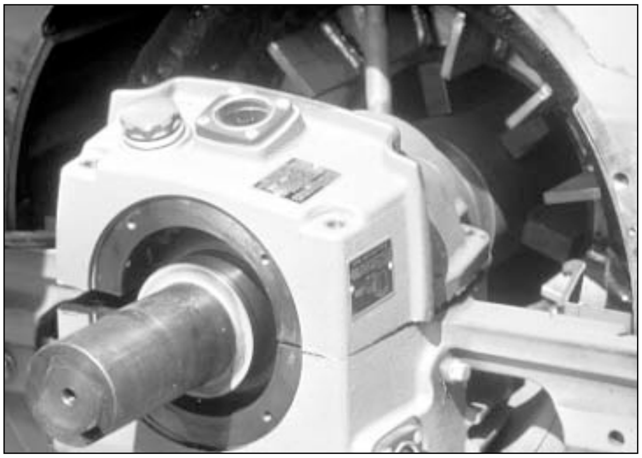
Installation of a sleeve bearing

Bolt bearing halves together. Verify that the oil rings are free to turn.



Installation of a sleeve bearing

Install top bearing cap and cross tighten the bolts. When assembled for the final time, a non-hardening sealant should be applied to the bracket split line.



Installation of a sleeve bearing

Install the labyrinth seal. The mating fit to the bracket should be sealed using a non-hardening sealant. Use shimstock wrapped around the shaft to center the labyrinth seal.

Installation of a sleeve bearing

With the top bearing cap and labyrinth seals installed, double-check the clearance between the labyrinth seal and shaft. Be sure to use pipe sealant on the pipe pugs and sight glasses.

Special considerations

New and rebuilt sleeve bearings

- End float & magnetic center
  - Scribe the magnetic center.
  - Magnetic center must be within the mechanical limits. Near mechanical center is ideal.
- Bearing temperature
  - No-load sleeve bearing temperatures above 150° F (66° C) can usually be improved.

Thrust shoulder wear indicates improper alignment methods, lack of prescribed thrust-limiting device

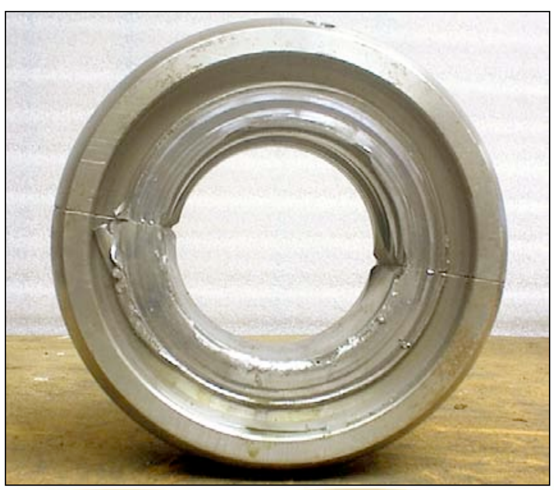


TABLE 2: END PLAY AND ROTOR FLOAT FOR COUPLED SLEEVE BEARING HORIZONTAL INDUCTION MOTOTS

Machine hp	Synchronous speed	Min. rotor end float	Max. coupling end float
500 hp and below	1800 rpm and below	0.25” (6.5 mm)	0.09” (2.3 mm)
300 to 500 hp	3600 and 3000 rpm	0.50” (13 mm)	0.19” (4.8 mm)
600 hp and higher	All speeds	0.50” (13 mm)	0.19” (4.8 mm)
NEMA MG 1-2006			

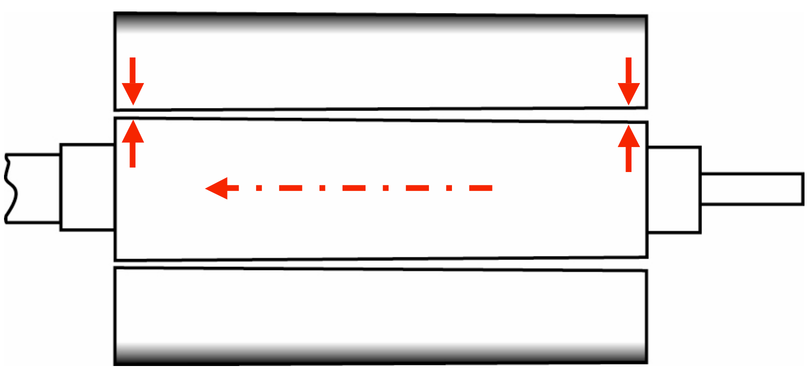
End play and rotor float

Max. coupling end float → 0.19” ←

Rotor end float → 0.50” ←

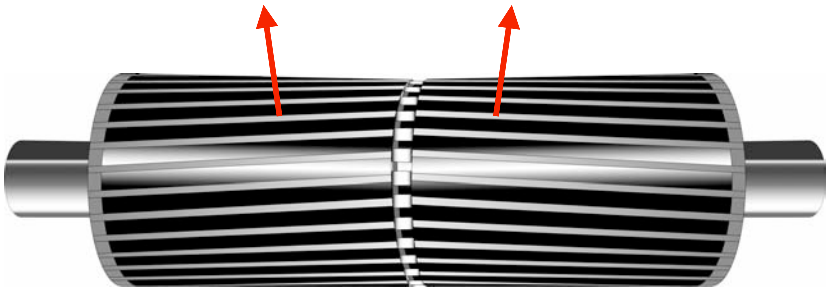
Thrust shoulder wear indicates improper alignment methods, lack of prescribed thrust-limiting device

Tapered airgap = axial force



Magnetic attracting force is inversely proportional to the square of the distance

Herringbone skew = offsetting axial forces



Rotor vent ducts not symmetrical

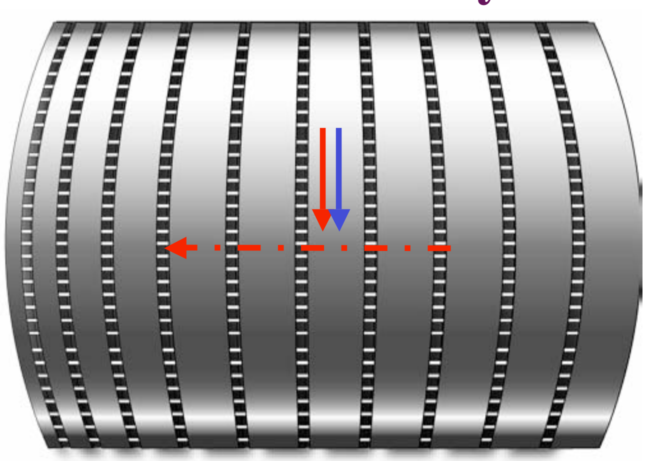


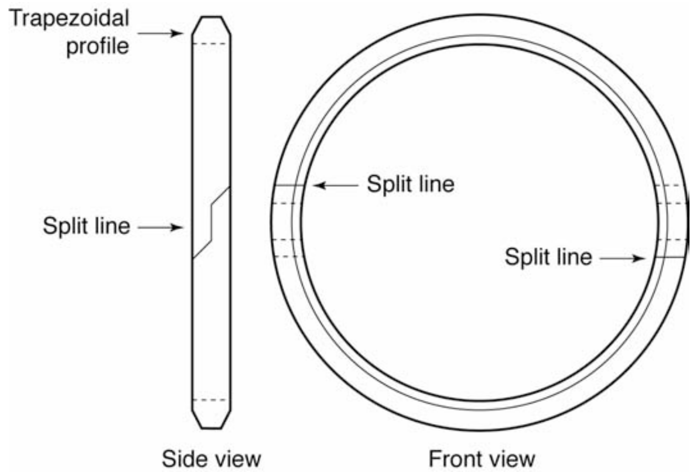
TABLE 2: CAUSES AND SOLUTIONS FOR AXIAL HUNTING OF 2-POLE ROTOR

Cause	Solution
Tapered rotor	Machine rotor OD at appropriate speed
Tapered stator bore	Restack stator core
Shaft not level	Level the machine and realign equipment
Nonsymmetrical stator stack	Restack and rewind OR false vent ducts
Excess rotor bar skew	Add ball bearing to limit axial movement
Aerodynamic forces (windage)	Adjust baffles, false vent ducts
Unbalanced voltage	Balance the voltage to within 1%

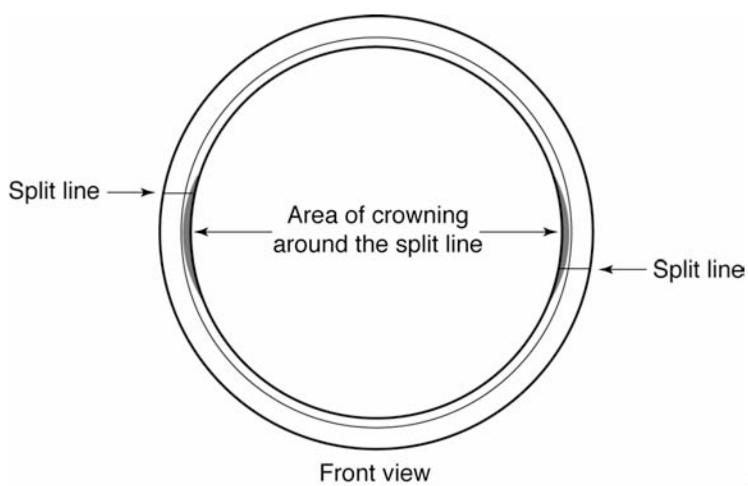
Test run tips

- Monitor vibration (velocity) and temperature
  - Reaction time is proportional to the speed
  - Two poles require extra measures:
    - Monitor axial velocity
    - Velocity increase is first indication of bearing friction = beginning to “wipe” babbitt

Importance of trapezoidal shape

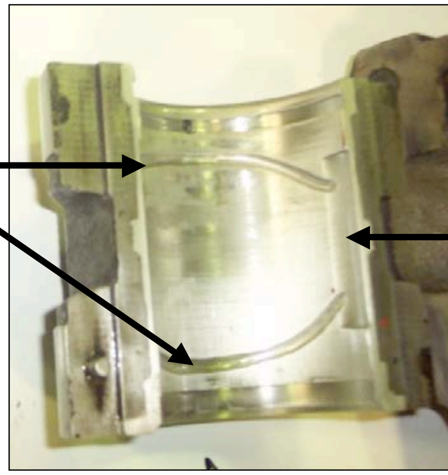


Crowning of oil ring ID



Distribution groove

Extra grooves machined to distribute oil



Horizontal distribution groove

Thank you!