SLEEVE VS. ANTI-FRICTION BEARINGS: SELECTION OF THE OPTIMAL BEARING FOR INDUCTION MOTORS

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Abstract **-** The decision between which bearing type to utilize is not always easy or even obvious. There is no choice in bearing selection on small motors (less than 200 HP), where only anti-friction bearings are readily available. Likewise, a choice does not always exist on larger motors in excess of 2000 HP, where various design requirements leave only the sleeve bearing (or tilting pad bearing) as a viable option. On intermediate size motors, a choice will have to be made. In general anti-friction bearings (AFB) are less expensive, and result in a more compact motor. However, there are potential disadvantages associated with anti-friction bearings. The 'best' bearing decision depends on the details of the particular application. This paper explores the advantages/disadvantages of both types of bearings. Additionally, it provides guidelines to help select the best bearing for the application. Judicious selection of bearings will result in motor purchasing savings without sacrificing process reliability.

I. INTRODUCTION

Induction motors can be purchased with two basic types of bearings. These types are commonly referred to as anti-friction or sleeve bearings. In each type there are various bearing designs. Each design is intended for different applications or has different performance characteristics, which may be necessary in different applications. There are also maintenance and reliability reasons why the user may be interested in one type of design over another. Many of the most common applications can satisfactorily use either type of bearings. The initial cost of the sleeve bearing is much greater than the anti-friction bearing design, but this may not be as significant when you take into account the total life cycle cost.

II. ANTI-FRICTION BEARING MOTOR DESIGN & CONSTRUCTION

Selecting the proper anti-friction bearings for a motor application is a design balance. Factors such as bearing speed limit, bearing life (L10) requirements, lubrication practices, load carrying capability, and so on, will have to be considered.

A. Anti-Friction Bearing Types

Anti-friction bearings used in horizontal motors can be designed for direct connect applications, applications that impose heavy radial loads on the motor shaft extension (such as in belted applications), or in applications that impose a large axial thrust load (such as vertical motors). Obviously, different designs would be required for each type of application.

In direct connect applications the bearings are designed to support the motor rotor weight only. Deep groove ball bearings are typically used. In these motors, either bearing can be fixed (axially constrained), but one of the bearings must be. However, both bearings cannot be fixed or the bearings will load up against each other due to thermal shaft growth. It should be noted that both the rotor and stator frame would grow, so even if you lock the drive end of the motor, the motor frame can grow towards the driven equipment binding the drive end bearing. Coupling must be designed to accept this axial movement or at least be soft applying minimal force to restrained bearing.

Motors in applications that involve heavy radial loads use a cylindrical roller bearing on the drive end a deep groove ball bearing on the non-drive end. By design, a cylindrical roller bearing allows for float of the inner race relative to the outer race. Therefore, the opposite drive end bearing must be fixed in these motors. It should be noted that although not very popular, spherical roller bearings have been used for heavy radial load applications instead of cylindrical roller bearings. These bearings have higher load carrying and misalignment capability. However, their re-lubrication requirements are much mort stringent. These bearings typically require relubrication three to five times more often than similarly sized cylindrical roller bearings. In the past, these more stringent lubrication requirements often were not adhered to, and as a result, lubrication failures with these types of bearings were common. Consequently, these bearings are rarely found in motors today.

Motors designed to take sustained external axial thrust (such as many vertical motors), will have thrust bearings on the opposite drive end, and a deep groove ball bearings on the drive end guide bearing. The type and size of the thrust bearing will depend on the motor speed and magnitude of thrust loads. Spherical bearings will take the highest loads, angular contact bearing(s) will take intermediate thrusts, and deep groove ball bearings will take little to no thrust.

Fig. 1 illustrates the various anti-friction bearing motor constructions. Fig. 1A illustrates vertical motor bearing constructions. The motor to the left side of the Fig. 1A is a typical low/zero thrust bearing configuration, as the upper thrust bearing is a grease lubricated deep groove ball bearing. The motor on the right side of Fig. 1A is a 2-pole high thrust motor. The upper thrust bearing is an oil lubricated back-to-back angular contact bearing. Fig. 2 illustrates a high thrust slower speed (i.e. 4 pole and slower) vertical motor thrust bearing. In this illustration, a spherical roller bearing is shown, although heavier section/larger angular contact bearings may be used as well. Most vertical motors built today employ a deep groove ball bearing for the guide bearing, and this construction is depicted in Fig. 1A as well. Fig. 1B illustrates horizontal motor bearing constructions. These bearings are most often grease or oil mist lubricated. The motor on the top of Fig. 1B is a 2-pole motor that would be used in direct connect applications. Both the inboard and outboard bearings are deep grove ball bearings. The motor on the lower part of Fig. 1B illustrates construction that would be employed when heavy side loads are encountered (such as in belting applications). The outboard bearing is a deep groove ball bearing, and the inboard, a cylindrical roller bearing.

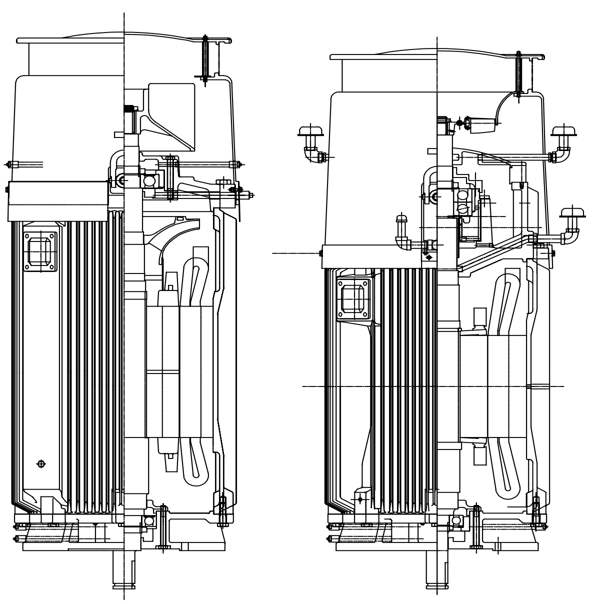


Fig. 1A – Vertical Motor Bearing Constructions.

B. *Bearing Life (L_{10}) Requirement*

Design criteria used for bearing life are based on fatigue of the bearing metal and are usually in terms of L10 life. Life requirements range from an L10 life of one-year to one exceeding 100,000 hours, depending upon the application. Table I can be used as a guide for typical L10 requirements:

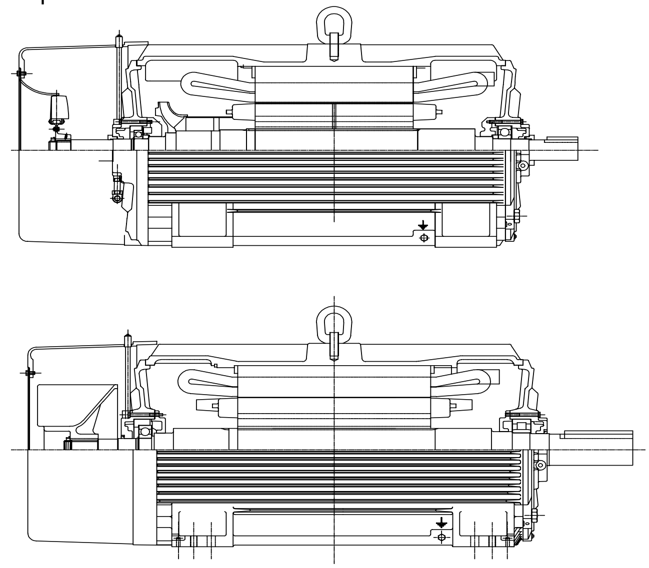


Fig. 1B – Horizontal Motor Bearing Constructions.

TABLE I
TYPICAL L10 REQUIREMENTS

Application	L10 (hours)	L10 (years)
Vert. Motors	8760	1
Belted Loads	17500	2
Direct Connection	100,000	11.4

Simply stated, the L10 life is of a bearing is the life in hours that 90% of a statistically significant sample of identical bearings would be able to achieve under a given set of operating conditions. The operating conditions include bearing speed and load (i.e. force on the bearing), temperature, and state of the lubricant. The average life (i.e. L50 life) is approximately five times the L10 life. In other words, a motor with bearings designed for a 100,000 hour L10 life would have an average life of 500,000 hours (over 57 years!).

The bearing L10 life is a theoretical life based on bearing operating stresses. Oftentimes this life is not reached. When this occurs many misinterpret the situation as the bearing having a reduced L10. In reality the L10 is unaffected, but the bearings service life did not reach its theoretical L10 life.

C. *Bearing Speed Limits*

The bearing speed limit is the speed at which there is a balance between the heat that can be removed from the shaft/bearing housing and the heat that is generated in the bearing, under a load corresponding to a 150,000 hour L10 life [1]. Generally there are two speed limits: a speed limit for an oil lubricated bearing, and a speed limit for a grease lubricated bearing. The grease-lubricated speed limit is lower than the oil lubricated speed limit. In addition, in oil sump lubricated bearings it is possible to cool the oil such the bearing speed limit can be safely increased.

The highest horsepower rating in which anti-friction bearings can be used is indirectly limited by the bearing speed limit. As larger horsepower ratings are encountered, larger shaft sizes will have to be used to keep torsional shaft stresses within acceptable levels. However, as shaft size (and thus bearing size) goes up, the bearing speed limit goes down. Eventually the required shaft size will result in anti-friction bearings that have a lower bearing speed limit than the motor operational speed. At this point a sleeve bearing would be required.

Table II can be referenced as a guide for typical speed limits for various ratings and applications [1]:

TABLE II
TYPICAL APPLICATION & SPEED LIMITS

Frame	HP	Speed	Application	DE Brg	Grease Spd. Lim.
320	50	3600	DC	6312	5000
580	2000	3600	DC	6315	4300
580	1500	1800	Belted	NU2228	2000
680	4000	1800	DC	6232	1900
800	5000	1200	DC	6334	1700

D. Bearing Housing and Shaft Fits

Bearing manufacturers provide recommendations for bearing housing and shaft fits. In general, the rotating inner ring should have an interference fit on the shaft, and the non-rotating outer ring, a slight clearance in the housing. Although the bearing manufacturers provide recommendations, it is up the motor manufacturers to temper this information and determine what fits work best. Ideally, the fit with a clearance should be as loose as possible without sacrificing vibration performance.

E. Anti-Friction Bearing Temperatures

Bearing temperatures vary a great deal depending on motor design and speed. On open horizontal machines, which are identical on both ends, the bearing temperatures on both ends of the motor will be similar. On totally enclosed fan cooled (TEFC) motors, the non-drive end bearing is much cooler than the drive end bearing due to increased cooling. Most anti-friction bearings them self have temperature limits of 150 Deg. C (300 F) thought they normally will never be allowed to run that warm. If this temperature is exceeded than permanent bearing damage may result. It is possible to get bearings that are heat stabilized to a much higher temperatures. However, bearing lubrication requirements at these higher temperatures will have to be reviewed to make sure that the bearing is properly lubricated at these elevated temperatures. The motor manufacturer can provide maximum safe bearing operating temperature taking all these factors into account.

F. Bearing Temperature Detection

The most common method of bearing temperature detection is thru measurement of the outer race temperature by a Resistance Temperature Device (RTD). This measurement of outer race temperature will provide good indication of overall bearing temperature. Anti-friction bearings can fail quickly, without much prior warning. The true benefit of temperature detection is when used in a trending fashion. Any unexplainable change in bearing temperature should be thoroughly investigated.

G. Bearing Vibration Detection

The most common method of bearing vibration detection is thru measurement of bearing housing vibration. A seismic vibration transducer (i.e. accelerometer) is most commonly used, and the most common units for this type of measurement is in velocity in inches per second. It is important to understand the difference between RMS and peak. Normally in the United States the velocity measurement is taken and expressed in inches per second (ips) peak value. European convention is RMS (root mean square value of peak) expressed in mm per second (mm/s). A common error is to/from convert from ips to mm/s neglecting the peak vs. RMS conversion.

H. Anti-Friction Bearing Lubrication

In addition to choosing the type of bearing to use one most also choose the type of lubrication and type of bearing cavity (cleanliness) protection. Anti-friction bearings are typically lubricated by one of three methods: grease lubrication, oil mist lubrication, and oil sump lubrication.

Motors with grease-lubricated bearings have the lowest initial cost (compared to other anti-friction bearing lubrication methods). They however require the most maintenance. Additionally, this method is the most problematic. Over-lubrication, under-lubrication and lack of cleanliness during re-lubrication are common problems that impact not only cost, but motor reliability as well.

Motors with oil mist lubricated bearings have very similar construction to motors with grease lubrication. As long as the oil mist system is correctly set up and working properly, the bearings are always assured a clean supply of lubricating oil. Although initial cost is higher, in many cases the life cycle cost is lower as the maintenance requirements are much less. In addition, the motors see much greater reliability and increased mean time between failures.

Oil sump lubrication is usually employed when the bearings need additional cooling than what can be offered by grease or oil mist lubrication. Typically only vertical motors use this type of lubrication arrangement. A cooling coil is often submersed in the oil for additional cooling. Fig. 2 illustrates such an arrangement.

I. Shaft Seals

Anti-friction bearing motors typically employ one of four shaft sealing mechanisms: close running bearing housing/bearing cap fit, v-ring seal, felt seal (grease seal) or rotating labyrinth seal. The primary function of these seals is to keep contaminants from the lubricant.

A close running bearing cap fit is the most common. The diametric clearance of this type of fit is typically 15-40 mils, and will afford an IP44 level of protection.

The felt seal is a slight variation on the close running bearing seal. There would be a groove cut into the housing in which a felt seal ring can be inserted into the housing and ride in close contact to the shaft.

A v-ring is the next step up in terms of protection. This seal makes contact at rest, but as motor comes up to speed, centrifugal force pulls the lip away from contact.

A rotating labyrinth seal offers the highest degree of protection. These seals offer an IP55 level of protection while the shaft is stationary, or rotating. These seals can be specially designed to help keep oil mist from escaping down the shaft and into the atmosphere.
This type of sealing arrangement will typically offer an IP54 level of protection.

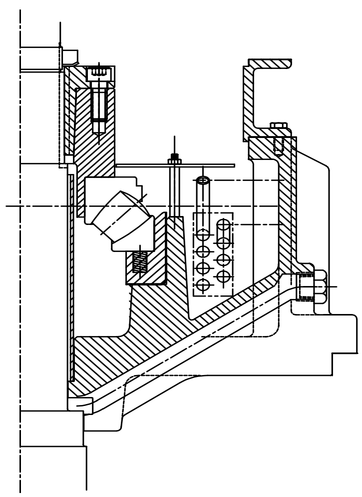


Fig. 2 Oil Sump Lubricated Bearing

J. *Anti-Friction Bearing Motor Construction*

Horizontal motors with anti-friction bearing are less complex with only a few parts required to contain the bearing.

- Bearing Housing
- Internal end cap (optional)
- External End cap (optional)

Oftentimes, the bearing housing will integrally contain either the internal or external end cap. The Fig. below illustrates a bearing housing with an integral external end cap. Refer to Fig. 3 for cutaway diagrams of some of the different horizontal motor constructions.

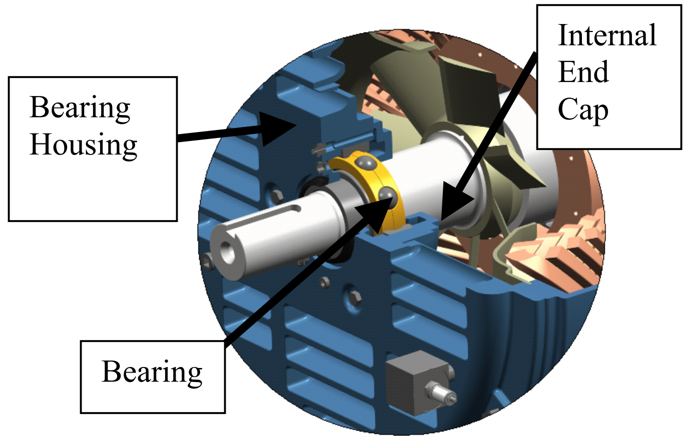


Fig. 3. Anti-friction Bearing Arrangement

Construction of vertical motors with anti friction bearings varies a great deal depending on the particular bearing arrangement used. The construction of motors utilizing grease or oil mist lubricated deep groove ball bearing bearings is similar to horizontal anti-friction bearing motors. Higher thrust motors utilizing angular contact or spherical bearings are usually oil sump cooled. Refer to Fig. 1 for cutaway diagrams of some of the different vertical motor constructions.

III. SLEEVE BEARING MOTOR DESIGN & CONSTRUCTION

The scope of selecting sleeve bearings is greatly reduced as sleeve bearings are used only in direct connect applications, and that theoretically, sleeve bearings have infinite life. However, there are other factors that need to be considered. For example, bearing and oil selection may have significant impact on vibration performance. If the motor is required to operate at extremely slow speed, hydrostatic jacking may be required, etc.

A. Basic Operating Principal & Design Considerations

Sleeve bearings operate under the principal of hydrodynamic lubrication. As the shaft rotates, it builds up a wedge of oil between the shaft and bearing. Notice that the shaft is necessarily moved into an eccentric position. Also, a clearance is required between the shaft and bearing, otherwise it would not be possible for the load-carrying wedge to be established. Fig. 4 illustrates the pressure distribution of a sleeve bearing that has fully established the oil wedge.

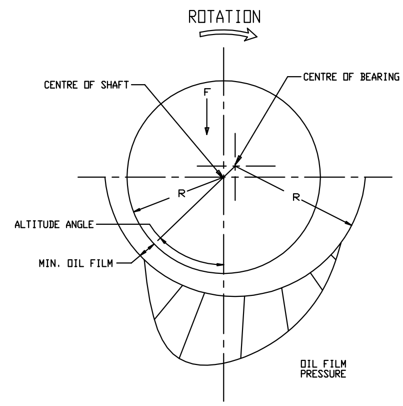


Fig. 4 Pressure Distribution for Sleeve Bearing

B. Sleeve Bearing Types

There are five basic types of sleeve bearings: plain cylindrical bore, two-lobe bore (lemon shape), four-lobe bore, four pad tilting pad, and five pad tilting pad bearing. These bearing types are illustrated in Fig. 5.

The standard cylindrical sleeve bearing is the simplest and most universally used. In actuality, the cylindrical bearing only has a partial arc to support the bearing with two additional partial arcs on the upper portion of the bearing to limit upper shaft movement. This type of bearing is

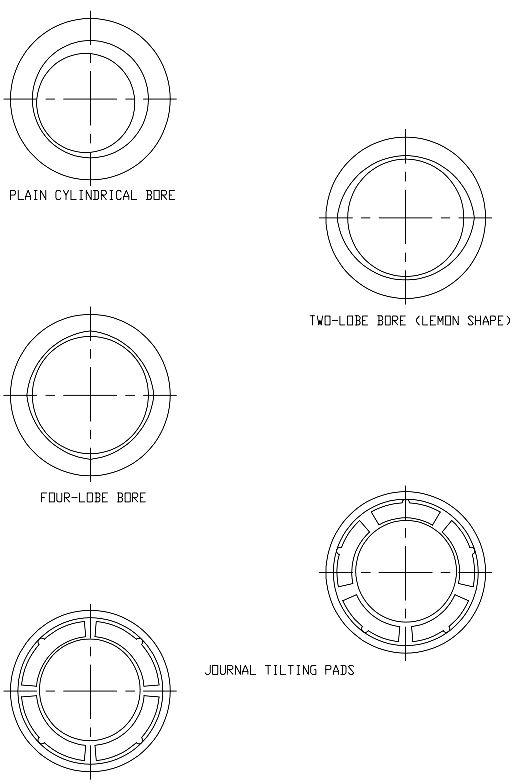


Fig. 5 Different Types of Sleeve Bearings

technically referred to as a partial arc cylindrical sleeve bearing, and is illustrated in Fig. 6

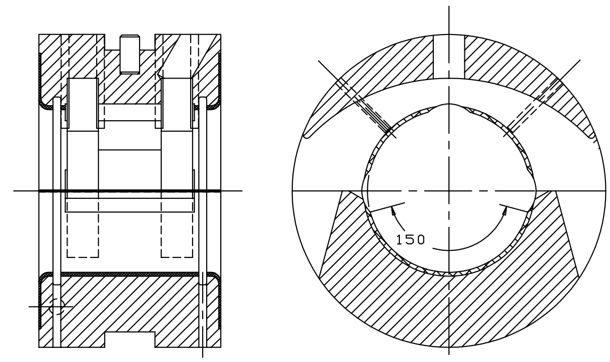


Fig. 6 Partial Arc Cylindrical Sleeve Bearing Section

The two horizontal reliefs are present to assure oil flow into the loaded portion of the bearing. The upper portion of the bearing is broken up into two portions for symmetry. As this is not the loaded portion of the bearing, the area can be reduced to minimize heating.

As the arc of contact of the cylindrical sleeve bearing increases, so does its horizontal stiffness, but so does the heating. An additional factor that impacts the stiffness and heating is the bearing clearance. All things being equal, the smaller the bearing clearance, the greater the stiffness and heating. This needs to be balanced in the bearing design.

From a hydrodynamic lubrication perspective, the minimum oil film thickness at the loaded portion of the bearing must be three to five times the shaft journal roughness (peak to peak). With a 32 microinch finish shaft, this would correspond to a minimum film thickness of 0.3 mils (0.0003"). Oil films this thin do not take into account any potential contamination. The finishes are smooth enough that a thinner oil film could be allowed than what is typically used, but with a thinner oil film, the margin of error in application would be reduced, and the bearing can be more easily damaged. For reference, typical sleeve bearings have an oil film thickness ranging from .8 mils to 4 mils.

Tilting Pad Bearings come in either 4 or 5 pad bearings. This bearing has the greatest stiffness since the clearance is less and the shaft journal rests between the two or three lower pads. Oil is delivered to the bearings between the pads through a force feed lubrication system. In both cases of tilting pad bearings the horizontal and vertical bearing stiffness is nearly the same and any significant difference seen in horizontal and vertical system stiffness would be a result of the bearing bracket or main frame or housing (yoke). This is very important when it comes to system response and rotor dynamics, which will be discussed in greater detail later.

Most tilting pad bearings are built without oil rings. Special tilting pad bearing can be built without the upper pads, or modified upper pads, allowing the addition of oil rings. These oil rings can be used to provide oil to the bearings for a short period of time such as during a coast down due to a loss of power.

C. Bearing Pressures

Bearing pressure is defined as the load on each bearing divided by the area supporting the shaft journal. The area supporting the journal is usually taken as the projected area of the journal over the effective area of the bearing. Or, it is the product of the journal diameter times the effective length. The effective length is less than the geometric length because of oil grooves and edge effect. Many specifications limit motor bearing pressures to 200 PSI. In actuality, this pressure is rarely exceeded because of other constraints. For example, journal diameter will be defined by how much torque the motor shaft has to transmit. Most bearings will safely tolerate pressures well beyond 300 PSI without any deleterious effects to the bearings [3].

It is important to point out that the motor sleeve bearings are designed as if the only load they will see comes from the rotating system weight (i.e. rotor and coupling weight). This is important not only from the magnitude of weight supported, but direction as well. Sleeve bearings have different characteristics in different directions. Arbitrary horizontal loads cannot be placed on them!

D. Lubrication

Sleeve bearings are designed with a specific oil viscosity in mind. The viscosity is selected based on assumptions about the bearing operating temperature ranges. Things such as oil film thickness, heat generation, and bearing property have to be considered.

E. Oil Ring Lubrication

Sleeve bearings can be oil ring or flood lubricated. Oil ring lubrication is used on cylindrical, two lobe, and four lobe sleeve bearings. It is generally not done on tilting pad bearings.

The oil ring is rotated by contact on the shaft. The lower portion of the oil ring dips into the oil, and as this portion rotates to the top, it runs down the surface of the shaft. This is illustrated in Fig. 7.

While Fig. 7 depicts a bearing with two oil rings, many bearings have only one oil ring. The depth of the oil ring in the oil is critical. If the oil level is too low, it won't pick up the oil. If the oil ring is submersed too deep, proper rotation of the oil ring will be inhibited, and it won't rotate. In either case, the outcome will be the same: inadequate oil to the bearing. Normal rotational speed for oil rings is from one-tenth of the shaft rotational speed at low speed, to 1/20-1/30th of shaft speed for 3600-rpm motors.

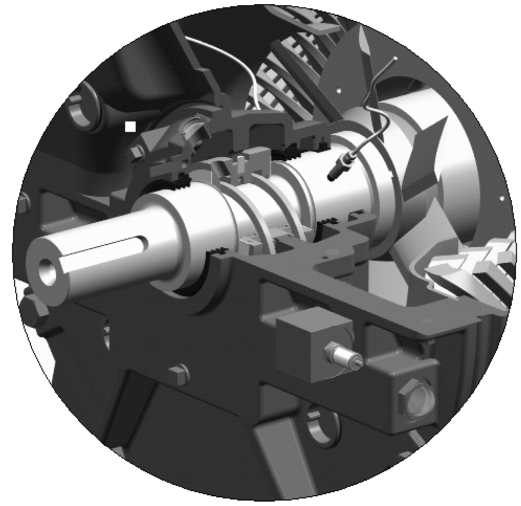


Fig. 7 Sleeve Bearing and Oil Ring Operation

F. Flood (or Force Fed) Lubrication

Flood lubrication is used where it is not possible to use an oil ring(s) (such as in tilt pad bearings), or in cases where additional bearing cooling is required. Although other means of bearing cooling exist, as indicated in the Sleeve Bearing Cooling subsection, flood lubrication is the most popular.

The flood lubrication system must be able to deliver a certain volume of oil at a certain pressure. An orifice within the sleeve bearing housing will throttle the oil to deliver required volumetric flow rate. The oil exits at atmospheric pressure back to the flood lubrication system supply reservoir through the outlet pipe.

Even when flood lube systems are used, oftentimes the oil ring is maintained (if the bearing design allows it). This redundant feature will enable the motor to safely coast down in the event that the flood lubrication system fails. It would be recommended on tilting pad bearings (or on bearings which don't have oil rings) to provide a control circuit to shut the motor down if there is a loss of oil pressure in the flood lube system, as the bearings would be severely damaged before the motor could safely coast down.

The oil sump can be either a wet or dry sump. In a dry sump all the oil is immediately returned to the force feed lubrication reservoir. These bearings are not intended to run when the force feed lubrication system is not on. Some times the bearings can be designed to maintain enough lubrication to handle a coast down if power is lost. A dry sump would be recommended in applications such as ship board where the motor/ship will rock and roll and possibly splash out oil in a full oils sump.

Oil is supplied to the bearing surface by oil rings, which deliver oil from the bearing oil sump, and on larger machines or in high ambient temperature areas by auxiliary oil supplied to the bearing by an external source, and drained from the oil sump back to the external source. The external source may be a motor-pump-cooler unit, or often is pressurized supply, and drain lines, from the driven equipment.

G. *Bearing Temperatures*

The total bearing temperature will be a function of the heat generated in the bearing, the heat absorbed by the bearing (caused by external and internal heat sources in proximity of the bearing), and the bearings ability to shed the heat. Refer to the Sleeve Bearing Cooling section for details on how heat from the bearing is removed.

Sleeve bearings by their very nature are heat-producing devices. The rotating shaft journal is separated from the stationary bearing bushing only by an oil film. Considering that the oil adjacent to the shaft journal is moving at shaft journal speed, while the oil adjacent to the stationary bearing bushing is stationary, it is apparent that the oil between the two surfaces is being sheared at a very high rate. Shearing of oil requires force, which in a time period equates to power, which being entirely absorbed in the bearing, equates to heat. A three-inch bearing journal, in a 3600-rpm motor, has a surface speed of 2826 ft/mm. The minimum oil film thickness between the rotating journal and stationary bearing will vary from approximately 0.0025 in a cool bearing to 0.0008 in a warm bearing, which is operating near 85^{o}C. The shearing action of the oil thus takes considerable energy.

Although sleeve bearings produce a significant amount of heat the biggest variable on bearing temperature is ambient temperature and motor load. Assuming a 40^{o}C ambient temperature, the only remaining heat generation variable is simply the motor load. Correlation between motor load and bearing temperature will vary from one motor design to another. Fig. 8 shows the dependence on bearing temperature for a particular motor design.

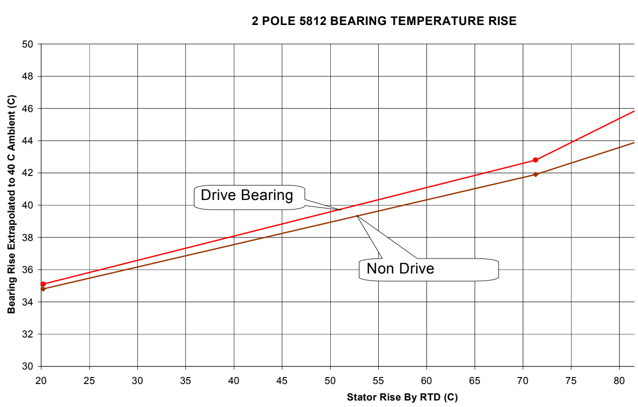


Fig. 8 bearing temperature as Function of Stator Temperature for one particular Motor

There are two concerns in exceeding design temperatures. First of all, the oil viscosity is strongly a function of temperature. If the heat becomes excessive than minimum oil film thickness will not be maintained. This could accelerate wear. Additionally, as the oil film thickness changes significantly, the rotor-dynamic performance may be affected. Secondly, the oil itself will break down faster at elevated temperatures, requiring more frequent oil change intervals. In extreme cases, the oil itself can start to coke. The bearing babbit material also has temperature limits. Absolute bearing temperature limits vary from one motor manufacturer to another. This temperature limit will vary based on how the temperature is being measured. This will be discussed in greater detail in the following subsection, Bearing Temperature Detection. API 541 limits bearing temperature to 93C absolute temperature.

H. *Bearing Temperature Detection*

Bearing temperatures may be detected by monitoring the sump temperature, or by monitoring the bearing metal temperature. The bearing metal temperature may be detected by means of embedded detector or by a bayonet type of detector. The embedded detector can usually be placed closer to the bearing surface than the bayonet detector. Fig. 9 illustrates both types.

When detecting bearing metal temperature, the loaded portion of the bearing must be monitored. Both types of bearing detectors in Fig. 9 are shown in this orientation.

Bearing detectors can be either thermocouples or resistance temperature devices (RTD's).

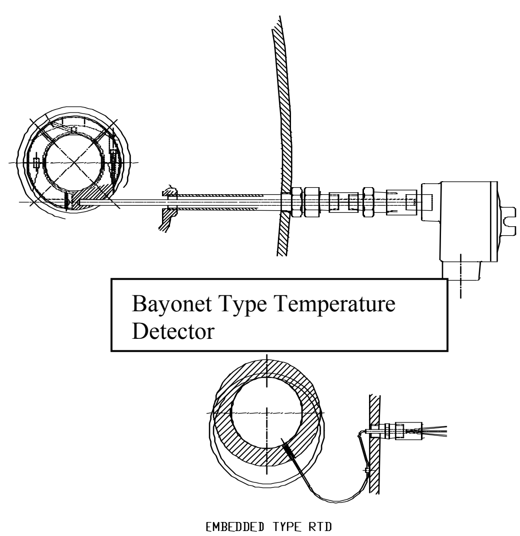


Fig. 9 Bearing Temperature Detection Devices

I. *Vibration Monitoring*

Ideally, in sleeve bearing motors shaft vibration and housing vibration would be obtained. In anti-friction bearing motors, housing vibration data is sufficient because the bearing is sufficiently stiff and has low enough damping properties that any movement in the shaft will be transmitted to the housing. In sleeve bearing machines this is not the case. Because of the larger bearing clearance, lower (and asymmetric) stiffness characteristic, and a high degree of damping, it is possible to have significant movement in the shaft that would no be detected at the bearing housing.

J. Speed Limit

Although sleeve bearings do have speed limits, they are very high. Depending on the sleeve bearing type, peripheral speeds up to 492 ft/sec are obtainable [2]. As a reference, a 3600-RPM motor with a 4.5" journal diameter has a peripheral speed of 71 ft/sec!

K. Sleeve Bearing Life

Sleeve bearings are theoretically considered to have an infinite life, and, when properly maintained the life may be extremely long. The only time that a properly maintained sleeve bearing should get any significant wear is at prolonged low speed operation. During low speed operation the oil film will not be at its desired thickness and subsequently, increased bearing wear results. At standstill, the oil film thickness is zero. Such conditions may be encountered during extended startup or coast down periods. If the frequency of such events is often enough, hydrostatic jacking is recommended to minimize bearing wear. Hydrostatic jacking is where the shaft journal is lifted off the bearing by oil pressure from the lower part of the bearing before the shaft begins to turn, thus ensuring adequate film thickness at extremely low speeds (down to zero speed). Hydrostatic jacking requires special sleeve bearings and an oil pressurization system and is normally only available on very large machines.

L. Load Capability

Motor sleeve bearings are not intended for use in applications where there is either side loading or axial loading. In regards to side loading, cylindrical sleeve bearings are only intended to take the weight of the rotor in the nearly vertical direction. In side loading (i.e. horizontal loading) there would be no oil film wedge developed to maintain the proper separation between the journal and the bearing. Tilting pad bearings are more tolerant of side loading.

As far as axial loading is concerned, there is normally an axial babbitted surface. In most sleeve bearings designed for horizontal application, the axial surface is designed to tolerate only momentary axial loading as could be seen on start up. A sufficient oil film is never developed; therefore, sustained axial loading is not permissible. Some sleeve bearings are designed to take continuous axial load but because of the greatly reduced surface area, the axial load capability is very slight. Sleeve bearings are so sensitive in this regard (i.e. axial thrusting), that maintaining level motors is extremely important. A motor that is not level can axially load the bearing if the coupling is incapable of holding the rotor in place.

M. Sleeve Bearing Cooling

The heat generated at the bearing surface is removed through oil being supplied to and flowing through the loaded bearing surface. This oil then conducts away the heat. Three methods are in common usage: self cooled bearings, force feed lubrication, and water cooling.

In self-cooled bearings the heat in the oil sump is transmitted to the bearing housing, which itself is cooled by external airflow around the externally finned surfaces. This is always the first choice of cooling methods, but oftentimes more heat is generated/absorbed by the bearing than what a self-cooled bearing can get rid of. If that is the case, one of the other two mechanisms must be employed.

In force feed lubrication systems the oil is taken out of the bearing housing. There is a bearing sump, but the key here is that the oil itself is cooled before it returns to the bearing. See the Flood (or Force Fed) Lubrication subsection for additional details.

Water-cooled bearings simply imply that a cooling coil is inserted in the oil sump. Cool water is run through the cooler thus removing heat from the oil.

N. Sleeve Bearing Construction

Sleeve bearing motors are inherently more complex than anti-friction bearing motors, requiring many more components. See Fig. 10.

- Bearing
- Oil Rings
- Capsule
- Inner and outer Oil Seals
- Inner air seal
- Atmospheric Vent (Built Into Oil guard &/or Capsule)
- Internal Pressure Lines and Chamber (Depending on machine size and speed) Not Shown

The bracket holds the capsule to the motor frame (yoke). The capsule may contain the oil seals, or the oil seals may be a separately bolted on part. Then inside the capsule you have the bearing itself. The bearing normally comes with 1 or 2 oil rings, which pick up the oil from the reservoir and dispense it on the shaft as the oil ring rides up against it. This is shown in Fig. 7.

The sleeve bearing itself must be constructed in such a fashion that the bearing bushing is self-aligning during the assembly process. If proper alignment is not set up, then it will be impossible to achieve a correct wear pattern. The wear pattern is the wear marks that a rotating shaft leaves on the bearing before an oil film is established. An even, centered pattern with at least 85% contact is desirable.

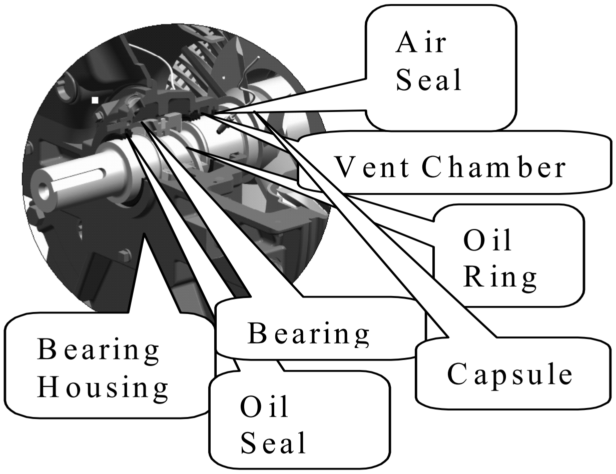


Fig. 10 Sleeve Bearing Arrangement

O. Shaft (Oil) Seals

In sleeve bearings, shaft seals are normally referred to as oil seals. This is because their primary function is to keep the oil from escaping. Recall that the primary purpose of anti-friction bearing shaft seals is to keep contamination out off the bearings. The primary purpose of sleeve bearing shaft seals (i.e. oil seals) is to keep the oil in. Oil seals are required on both sides of the bearing (internal and external to the motor). On some motors (such as open machines) there may be suction internal to the machine, trying to suck the oil into the motor's rotor. Many times the vacuum/pressure is great enough that the labyrinth seal alone is not enough to prevent oil leaks. In these cases there is an external air intake present between the labyrinth seals. In larger/faster motors even this may not be enough. In some larger high-speed machines another chamber will be added which the internal fans pressurize.

The oil seals are critical to the design. There are several types of oils seals in common usage: rigid labyrinth, floating labyrinth, and baffle seal. In addition, some of the seals as described in the anti-friction bearing section may be employed for additional protection from contaminants. Oftentimes several oil seals are used in conjunction with one another to form a sealing system.

On rigid and floating labyrinths, it is important to maintain a tight fit to the shaft but so tight that they will rub. Oil seals can be made out of aluminum, bronze or non-metallic materials such as but not limited to Torlon, Teflon or Peak.

IV. MAINTAINABILITY & RELIABILITY

In this section both sleeve bearings and anti-friction bearings will be addressed. Obviously, some issues are only of concern to one of the bearing constructions,

A. Anti-Friction Bearing: Re-Lubrication & Lack of Lubrication

Proper lubrication is critical with all bearings but it is much more important with anti-friction bearings. The primary reason is that anti-friction bearings give little prior notice before catastrophic failure.

Oil sump and oil mist lubrication is very reliable. Bearings thus lubricated generally achieve their calculated life. However, it is generally much more difficult to properly maintain grease lubricated bearings, which are by far the most common. There are many reasons why grease lubrication may not be as reliable. There are concerns with over-lubrication, under-lubrication, re-greasing with incompatible grease, and lack of cleanliness during re-lubrication, just to name a few.

Contamination will shorten the bearing life significantly. Particular if the contaminant is harder than the bearing metals.

Under-lubrication will result in increased metal-to-metal contact and increased bearing contact stresses. This will result in severely shortened bearing life.

Regreasing the bearings with incompatible grease typically causes the resultant grease mixture to be much softer. Soft enough in many cases to let the lubricants totally run out of the bearing. Once this happens then the bearing fails from under-lubrication. It is always safest to re-grease bearings with the same grease that's already in them. If this is not possible, the degree of compatibility/incompatibility needs to be determined. This can only be conducted by the grease manufacturers. If the greases are at all incompatible, then they cannot be intermixed.

Over lubrication will cause the bearing to run hotter, especially if the bearing cavity has no empty space in the housing in which to dispose of the excess grease. A 20-30 degree C increase in bearing temperatures can cause the bearing to over heat and go into a runaway condition. Over-lubrication can be a serious problem particularly in high-speed motors. High speed is relative to bearing size and type. A good measure of 'high speed' is how close the bearing speed is to its grease-lubricated speed limit. For example, a 6218 deep groove ball bearing has a grease lubricated maximum speed of 3800 RPM. A 6232 deep groove ball bearing has a grease lubricated maximum speed of 1900 RPM. The 6218 bearing at 3600 RPM is at the same risk of failure due to over-lubricated as a 6232 bearing at 1800 RPM. An over-lubricated bearing takes many hours for the excess grease to purge itself out. The temperatures may exceed the bearing heat stabilization temperature, thus ruining the bearing and causing it to fail prematurely

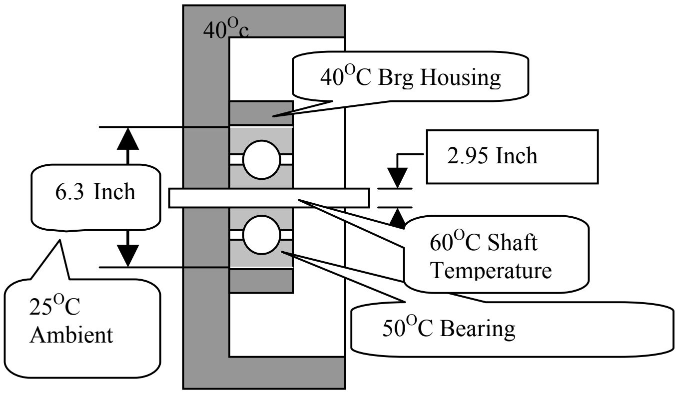


Fig. 11 – Thermal network A/F Brg

B. Sleeve Bearing: Re-Lubrication & Lack of Lubrication

Proper lubrication is important for any bearing type. However, sleeve bearings are more tolerant of lubricant abuse than anti-friction bearings. If the lubrication is non-optimal, than the sleeve bearing will give better warning than anti-friction bearings would, especially if the bearings are equipped with bearing temperature detectors.

Sleeve bearings are lubricated with high-grade turbine oil. It is important to use the proper viscosity oil. If a different viscosity is used than what the bearings were designed for, than several things may occur: inferior rotor-dynamic performance, change in bearing oil film thickness, or increase in bearing heating.

Unless otherwise stated, it is assumed that this oil is mineral based. Synthetic oils are becoming more common, but their compatibility with paint and sealing systems must be verified. Non-compatible oils will be chemically aggressive to the paint and sealing systems causing leaks and foreign material getting into the bearing. Catastrophic failure may result!

The oil must be maintained at the proper level. Proper level is indicated by having the oil level centered on the bull's eye or to the appropriate mark on a sight glass/gage. Additionally, constant level oilers can be used to assure proper oil level on sleeve bearing motors.

The variation allowance in the oil level should be kept at a minimum. If the level is too high it can leak out, or slow the oil ring down to the point that it is incapable of delivering the required oil to the shaft journal/bearing. If it is more than ¼ in low it can cause failure. Refer to Oil Ring subsection for additional details.

C. Notification of Imminent Failure

Sleeve bearings are more likely to give you a warning imminent failure than anti-friction type bearings. Many times sleeve bearings will begin to wipe under high vibration conditions and while this is happening RTD will show some higher temperature levels and then return to normal, but when you open up the machine you will see heavy wear patterns and bearing wiping. Also when Sleeve bearings begin to fail babbit material will begin to contaminate the oil causing it to turn black. This discoloration can also be seen when shaft currents exist. Anti-friction bearings on the other hand can fail rather quickly with little to no warning. Many cases have been noted where the operator walked past a motor when it was running very well and moments later the bearings failed catastrophically causing the rotor to hit the stator resulting in total destruction of the motor.

One scenario is quite possible. When for any reason the bearing begins to run hotter than the bearing housing the clearance in the bearing (which can be as low as one or two mils on anti-friction bearings) can quickly be used up. . Fig. 11 shows the typical temperatures that can be seen on open motors with ambient air on both sides of the bearing. A 6.3-inch OD bearing that has a 10 degree C rise above the bearing housing temperature will expand 1.12 mils per inch for every 100 deg C. For this 6.3-inch bearing that would be .71 mils for every 10°C. The diametrical clearance between the bearing and the housing ranges from 0-2 mils. The bearing internal clearance ranges from 1-2 mils. This would result in a total average clearance of 2.5 mils. If the differential temperature between the housing and the bearing in this example were 35°C the clearance would be used up. Above that the bearing would be damaged. The bearing would then be subjected to extreme compressive stresses as the bearing starts pushing against itself, the shaft, and the bearing housing. Initially the tighter clearances may actually result with the motor running with less vibration. But once this process begins, a thermal runaway condition will follow. Eventually, the outer race will expand to the point of locking in the bearing housing. The inner race expands and begins to spin on the shaft and the balls get imbedded in the inner and outer race. At this point the bearing has catastrophically failed. If the motor is not shut down very quickly the rotor can quickly hit the stator causing extensive and very expensive damage.

Another condition that may exist is axial loading between bearings. When the rotor assembly and shaft heats up and expands axially, the bearing, which is intended to float in the housing, could possibly, due to excessive heat, expand radially to a point where it could seize in the housing not permitting axial growth. This would result in excessive axial loading of both bearings. This failure mode will look like excessive thrust loading.

D. Bearing Replacement

It is not possible to replace anti-friction bearings without removing the bearing housings and uncoupling the driven equipment. Then a significant amount of access room is still required for the necessary equipment to remove the old bearings. In general the motor must be removed for bearing replacement.

Sleeve bearings, on the other hand, can be replaced with the lower bearing housings intact and driven equipment coupled. This can be a significant feature if rigging requirements make it extremely difficult to remove the motor.

Anti-friction bearings failures tend to be catastrophic in nature, oftentimes providing little prior warning. In some cases the bearings fail to the point of having the rotor drop down into the stator, essentially destroying the entire motor. Even if the entire motor is not damaged, the shaft (and thus rotor assembly) must oftentimes be replaced after a catastrophic bearing failure, as the inner race tends to first spin and later weld itself to the shaft.

When sleeve bearings fail, they tend to be less catastrophic, and generally provide information as to the degrading state of the bearings. When the bearings fail catastrophically, the bearing itself tends to be sacrificial, as the bearing babbit is much softer than the shaft steel. Generally it is possible to clean up the shaft with minor regrinding of the bearing journal surface.

V. PERFORMENCE

A. Efficiency vs. Bearing Type

It is sometime asked whether anti-friction bearings can achieve better efficiency than sleeve bearing machines. In general the answer is that the difference is insignificant.. A/F bearings do normally generate fewer losses but normally the losses are a small percentage of the total losses having little effect on the total efficiency. Regarding this question, the following example for a 500 frame 2 pole 300 HP and 1250 HP motor may help:

HP	Total. Loss W/O Brg. Loss	Brg. Loss in kW A-F / Sleeve.	Efficiency A/F Brg.	Efficiency Sleeve Brg
300	13.5	.52/ .75	94.09	94.00
1250	46.7	.52/ .75	95.17	95.15

As can be seen from the above, the difference is small.

B. Service life vs. Theoretical life

The calculated life expectancy of any bearing is based on four assumptions.

1) Good lubrication in proper quantity will always be available to the bearing.
2) The bearing will be mounted without damage.
3) Dimensions of parts related to the bearing will be correct.
4) There are no defects inherent in the bearing.

In general, sleeve bearings have an extremely long life. Anti-friction bearings that are oil mist or sump lubricated also do and will reach their expected L10 fife. Grease lubricated bearing rarely reaches their expected L10 life.

C. Rotor-Dynamic Performance

Bearing selection can have a significant impact on rotor dynamic performance. Table III shows the results of a case study of a similar motor with different bearings. The rotor under study was for a 3600 RPM 1250 Horsepower open motor. The rotors were similar, with the obvious exception of bearings (and the required subsequent changes in the shaft).
Notice that as the stiffness decreases, so does the resonant frequency. However, highly damped resonances are not considered to be critical speeds. Per API 684, amplification factor is defined as a measure of a rotor-bearing systems vibration sensitivity when operated in the vicinity of one of its lateral critical speed [3]. Furthermore, this same standard defines critical speed as a shaft rotational speed that corresponds to a non-critically damped (amplification factor >2.5) rotor system resonance frequency. This is an important definition as a highly damped resonance should not be misinterpreted as a critical speed. The justification in this lies in the fact that even if a motor were to operate right at this speed, the increase in vibratory response would be so low that no damage to the motor or bearings would occur. This is evident when reviewing the calculated rotor dynamic performance of the 150 Partial arc sleeve bearing to the 6315 anti-friction bearing, in particular, with respect to the rotor response to unbalance as illustrated in Fig. 11:
Fig. 11 compared the predicted increases in vibration due to an unbalance of 4W/N, which is equal to the residual unbalance limit for each end of the rotor in API 541 motors. The peak to the right corresponds to the 6315 AFB bearing, and the peak to the left, the 150 degree partial arc bearing. Note the significantly different scales for the two curves (6315 corresponds to the scale on the right, and the 150 degree partial arc, the scale to the left). Furthermore, note that the vibration of the 6315 rotor would be 34 times higher at its resonant speed compared to operating speed. Clearly, vibration severity of this magnitude cannot be tolerated for any period of time.

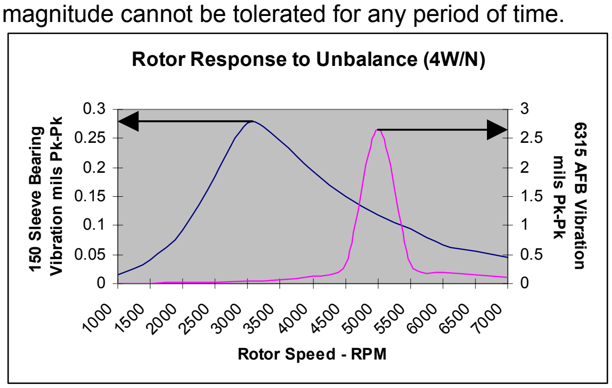


Fig. 11 Rotor response to unbalance

Likewise, the 150-degree partial arc-bearing rotor would be 1.2 times higher at its resonant speed compared to operating speed. The rotor can operate at this speed indefinitely without any harm to the rotor or bearing.

TABLE III
ROTOR RESPONSE OF DIFFERENT BEARNGS

Brg. Type	Oil Viscosity ISO VG	Stiffness (lbs./in. *10^6) Vertical		Horizontal		Resonant Freq. (CPM)		Amp. Factor
		Hsg.	Brg.	Hsg.	Brg.	Vert.	Horiz.	
120 Partial Arc	32	1.0	0.564	0.5	0.215	5000	2800	2.56
150 Partial Arc	32	1.0	0.467	0.5	0.251	5000	3100	2.21
Four Lobe Tilt Pad	32	1.0	0.673	0.5	0.673	4900	4200	5.28
6315 (AFB)	100	1.0	1.000	0.5	1.000	5000	4900	36.9

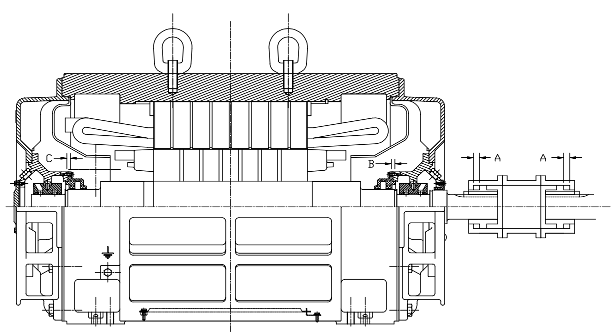
In the past, rotor dynamic performance used to be considered only at the operating speed. The same is true today. However, today the operating speed no longer can be assumed to be a fixed speed. With the advent (and increasing popularity) of adjustable speed drives, operating speed ranges are becoming the norm. All rotors will have a resonance somewhere. From a rotor dynamic perspective, the location of the horizontal and vertical resonance along with their associated amplification factors will have to be carefully evaluated to make sure that the rotor will perform adequately throughout its entire operating range.

D. External Load Carrying Capability

As stated throughout this paper, sleeve bearings are designed to support only the motor rotor weight. Anti-friction bearing motors can be designed to accept significant external load (radial and axial). From that perspective, anti-friction bearings are more versatile.

E. End play

Motors with anti-friction bearings will have self-locating rotors. Since one of the bearings is held in these motors, they are not reliant on the coupling for proper rotor positioning. The same is not true of sleeve bearing motors. Sleeve bearing motors require coupling that either position the rotor, or, limited end float couplings.



Minimum Motor	Limited Coupling	A	B	C
1/2”	3/16”	3/32	5/32	11/32

Fig. 12 Sleeve Bearing Motor with Limited End float Coupling

The limited end float coupling must have an allowable movement not in excess of 3/16”. This will assure that the motor shaft does not move such that the journal shaft contacts the bearing. Design clearances for the drive end and opposite drive end bearing and the coupling is shown in Fig. 12. In applications using sleeve bearings the driven equipment must be capable of withstanding a slight oscillating thrust developed in the induction motor due to slight magnetic variations. The motor itself will not be able to restrain the movement and the movement doesn’t bother the motor.

F. Cost

Although cost is not a traditional performance measure, it is difficult to consider other performance aspects without also considering cost. Sleeve bearing motors are significantly more expensive. Additionally, the vibration monitoring equipment is much more expensive. Full vibration instrumentation for sleeve bearings motors includes proximity probes and accelerometers. Full vibration instrumentation for anti-friction bearing motors needs just the accelerometers. Proximeter probes are much more expensive than accelerometers. The cost comparison in Fig. 13 is shown as a percentage of motor cost.

The top curve represents the increase in motor cost over the base price for addition of Proximeter probes. The next curve represents the increase in cost over base motor cost for sleeve bearings (instead of anti-friction bearing). The final curve represents the increase in motor cost over base price for accelerometers.

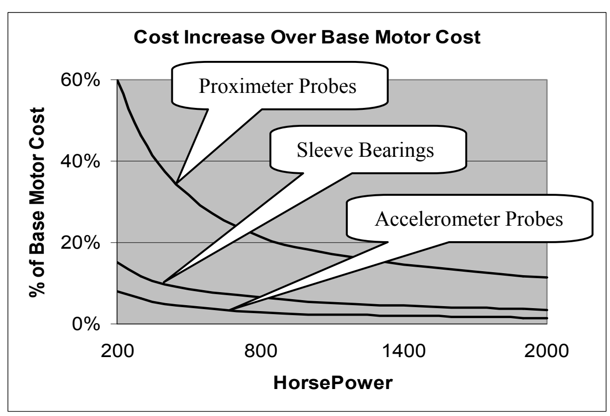


Fig. 13 Cost Vs. HP for Features

VI. COMPARISON

Obviously, both types of bearings have their place. Both bearing constructions have their advantages and disadvantages. Table IV in a very simplistic form lists some of the advantages/disadvantages:

TABLE IV
BEARING SELECTION CHART

1.	Long Life	Sleeve
2.	Availability	A-F
3.	Maintenance	Sleeve
4.	Bearing Replacement	Sleeve
5.	Quietness	Sleeve
6.	Application Flexibility	A-F
7.	Thrust Load	A-F
8.	Belt Drive	A-F
9.	Prior Indication of Failure	Sleeve
10.	Compactness	A-F
11.	Cost	A-F

Unfortunately, the actual selection is somewhat complicated. Also once the type of bearing is chosen the method of lubrication must be established. Within the same application and comparing the same feature or characteristic, arguments can be made for either case. Complicating all of this is the fact that many of the supposed advantages and disadvantages are subjectively based. Some things are just difficult to quantify. Hopefully the reader now has a better understanding of motor bearing design, construction, and maintenance. Weighing all of these factors, a well-informed decision can now be made.

VII. REFERENCES

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[2] "Manual for the Application of RENK Slide Bearings", Renk Aktiengesellschaft

[3] API Publication 684, First Edition, Tutorial on the API Standard Paragraphs Covering Rotor Dynamics and Balancing: An Introduction to Lateral Critical and Train Torsional Analysis and Rotor Balancing, Washington, D.C., 1996

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VIII. VITA

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