ROLLING BEARINGS IN CENTRIFUGAL PUMPS

by

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ABSTRACT

The primary considerations for the selection and application of rolling bearings in centrifugal pumps are discussed. These considerations apply to both the pump manufacturer and the pump user.

The various types of bearings used in different pump arrangements are illustrated. Bearing life and how it is influenced by lubrication, cleanliness, and bearing operating temperature, as well as the different methods of pump bearing lubrication, are discussed. Guidelines for the use of 7300 series angular contact ball bearings and the 8300 series bearing set are provided. Further, specific recommendations are provided for shaft and housing mounting practices, and guidelines are provided for selecting bearing internal clearance and cage design.

INTRODUCTION

Many different factors affect the successful long term operation of centrifugal pumps; in particular, the selection and use of rolling bearings. This presentation reviews the following considerations involved in properly selecting and using rolling bearings in centrifugal pumps.

- Bearing load and speed capability.
- Pump bearing arrangements.
- L-10 fatigue life.
- Lubricant selection.
- Lubrication methods.
- Bearing temperature.
- Pump bearing system life.
- 7300 series angular contact ball bearings.
- 8300 series bearing set.
- Triplex bearing arrangements.
- Shaft and housing fits.
- Bearing internal clearance/end play.
- Bearing cage type.

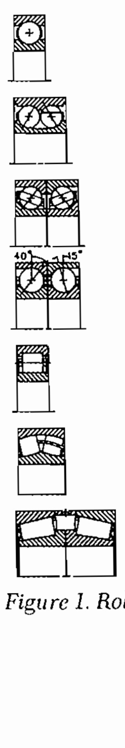
BEARING LOAD AND SPEED CAPABILITY

The most common types of ball and roller bearings used in centrifugal pumps, along with their approximate relative load and speed characteristics are illustrated in Figure 1. Single row deep groove ball bearings (SRDGBB) are used for their good overall load and high speed capabilities. Double row angular contact ball bearings (DRACBB) are used for heavier radial and thrust load applications, while 7300 series angular contact ball bearings and 8300 series bearing sets are used for very heavy thrust load applications. Cylindrical (CRB), spherical (SRB), and tapered roller bearings (TRB) are used in heavy radial load applications. Besides being chosen on the basis of load and speed capabilities, bearings may also be chosen for their capability to reduce shaft deflection and to control shaft stability. Roller bearings inherently have greater stiffness than ball bearings and are sometimes chosen for this reason.

PUMP BEARING ARRANGEMENTS

There are many different pump bearing arrangements, which vary according to the type of pump service. Descriptions of

X - LOW
XX - MODERATE
XXX - HIGH
XXXX - VERY HIGH



Approx. Relative Load and Speed Capability

	Radial Load	Axial Load (both directions)	Speed
SRDGBB	X	X	XXXX plus
DRACBB	XX	XXX	XXX
two 7300 ACBB	XX	XXXX	XXX
8300 Series Set	XX	XXXX one direction XX other direction	XXX
CRB	XXX	X	XXXX
SRB	XXX	XX	XX
paired TRB	XXX	XXX	XX

Figure 1. Rolling Bearings Typically Used in Centrifugal Pumps.

pump bearings in the most common arrangements, and methods for bearing lubrication are as follows.

Vertical In-line Pumps

A typical vertical in-line pump, illustrated in Figure 2, uses grease lubricated single row deep groove ball bearings. The bearings are spring loaded to reduce possible pump shaft end play, and to reduce noise and vibration. The recommended minimum spring load is defined in Equation (1) for reducing bearing noise and vibration.

$$f = 5 \times d \tag{1}$$

where f = spring force, Newtons (N)
d = bearing bore, mm

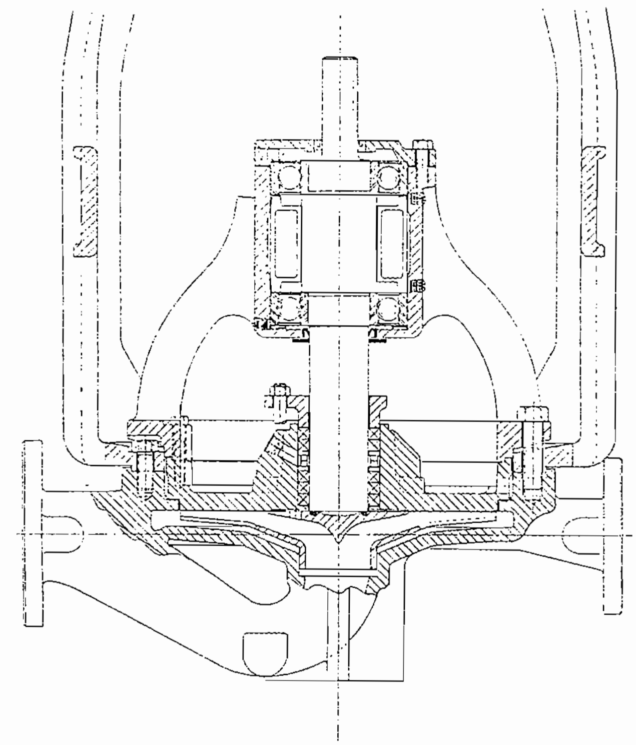


Figure 2. Vertical In-line Pump (Courtesy Duriron Co.).

American National Standard Institute (ANSI) Chemical Process Pump

A typical ANSI process pump is illustrated in Figure 3. The thrust bearing is a double row angular contact ball bearing and the radial bearing is a single row deep groove ball bearing. Bearings used in this application are either grease or oil lubricated, and usually have C3 (loose) internal clearance because of their usual high speed and high temperatures.

American Petroleum Institute (API) General Refinery Service Pump

A typical API process pump is illustrated in Figure 4. The thrust bearing typically consists of two 7300 series 40 degree angular contact ball bearings, mounted back-to-back. An alternative to 7300 series bearings is an 8300 series bearing set. The radial bearing is typically a single row deep groove ball bearing, usually provided with C3 (loose) internal clearance. Another choice for the radial bearing is a cylindrical roller bearing, which may be used when greater load capacity and radial stiffness are required. Bearings in an API process pump are oil-ring lubricated or alternatively, oil-mist lubricated.

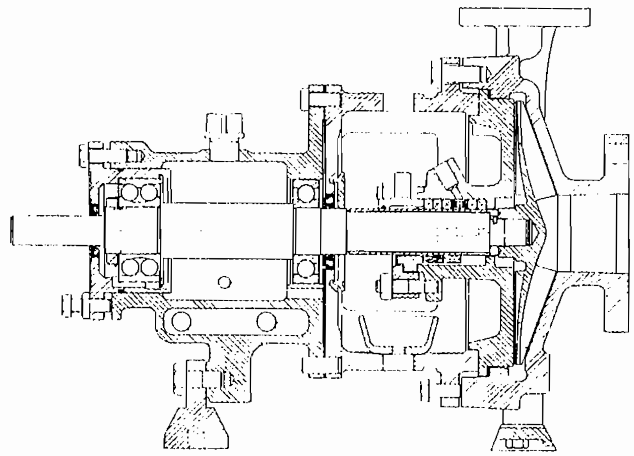


Figure 3. ANSI Pump (Courtesy Goulds Pumps, Inc.).

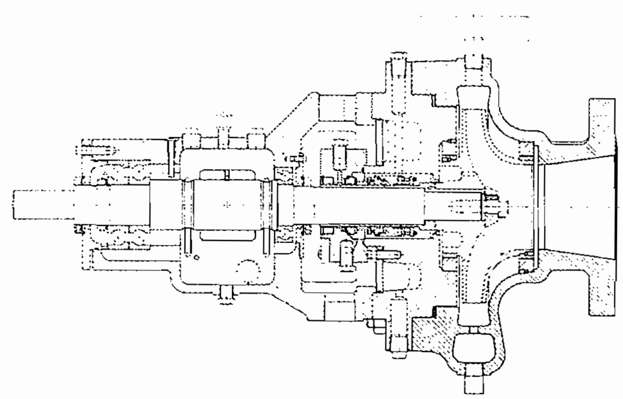


Figure 4. API Pump (Courtesy Goulds Pumps, Inc.).

Heavy Duty Pump

A typical heavy duty centrifugal pump is illustrated in Figure 5. The thrust bearing consists of a matched pair of tapered roller bearings, mounted face-to-face, and the radial bearing is a spherical roller bearing. These bearings have the advantage of having

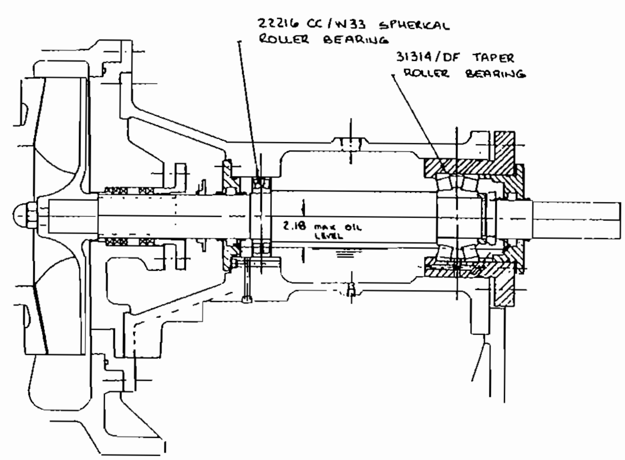


Figure 5. Heavy Duty Pump.

higher load capabilities. The bearings can be either static-oil or grease lubricated.

Vertical Cantilever Pump

A vertical cantilever centrifugal pump is depicted in Figure 6. The thrust bearing is an 8300 series bearing set, and the radial bearing is a single row deep groove ball bearing. The thrust and radial bearings are grease lubricated.

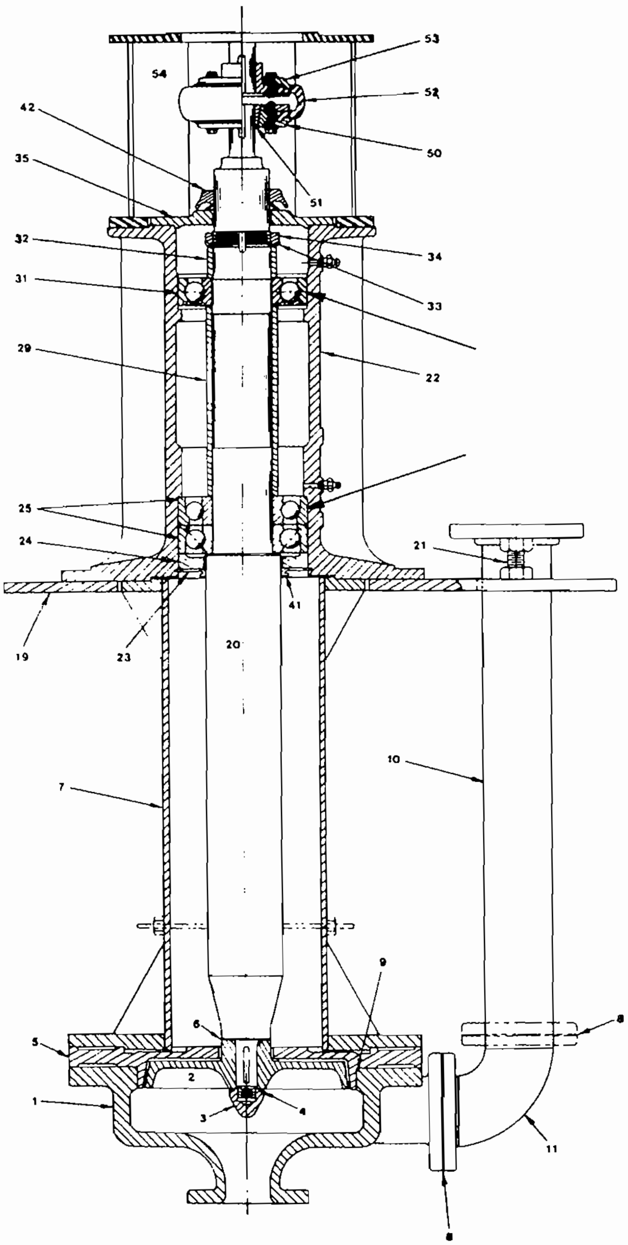


Figure 6. Vertical Cantilever Pump (Courtesy Barrett, Haentjens & Co.).

L-10 FATIGUE LIFE

One criterion for choosing the size of rolling bearings in centrifugal pumps is the bearings' minimum required L-10 fatigue life. Both ANSI and API Standards for centrifugal pumps each specify the minimum required L-10 fatigue lives at defined conditions. The recommended minimum bearing lives in centrifugal pumps of different types of operation are shown in Table 1. A bearing's L-10_a fatigue life can be calculated using Equation (2).

$$\text{L-}10_a = A_{23}\left(\frac{C}{P}\right)^y \times \frac{16667}{n} \qquad (2)$$

where L-10_a = bearing fatigue life, adjusted for material and and lubrication, 90 percent reliability, Hrs
a_{23} = life adjustment factor for lubrication (from Figure 7)
C = bearing basic dynamic load rating, Newtons (N)
P = dynamic equivalent load, Newtons (N)
y = exponent for the life equation
= 3 for ball bearings
= 10/3 for roller bearings
n = bearing rotational speed, rpm

Table 1. Recommended Minimum Bearing Lives in Centrifugal Pumps.

Type of Operation	L_{10} Operating Hours
Intermittent	8,000-12,000
8 Hours per Day	20,000-30,000
24 Hours per Day	40,000-50,000
24 Hours per Day with High Operational Reliability	100,000

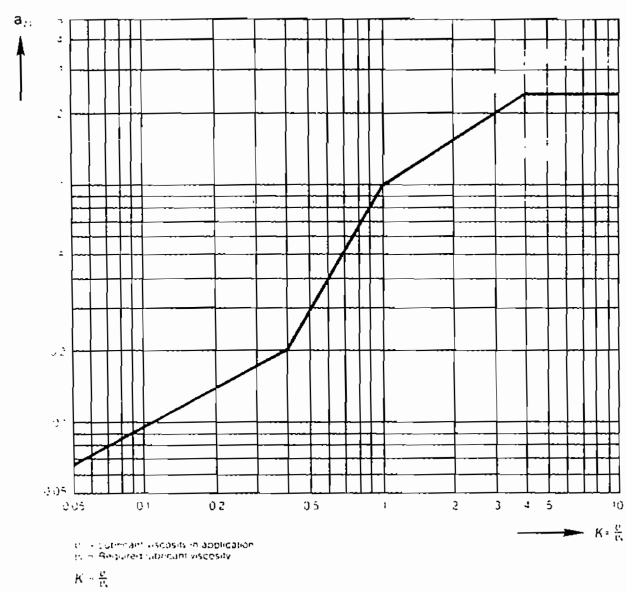


Figure 7. Life Adjustment Factor, a_{23}.

Fatigue life can be increased by selecting a lubricant which provides a sufficiently high viscosity. The effect of the lubricant on the L-10 fatigue life is determined by the a_{23} factor, which is a function of Kappa, K. Kappa is the ratio of the lubricant's actual viscosity, ν, to the bearing's minimum required lubricant viscosity, ν_1, (i.e. $K = \nu/\nu_1$). A Kappa value greater than or equal to one indicates that the lubricant provides adequate or more than adequate lubrication to the bearing. A Kappa value less than one indicates that the bearing is not receiving sufficient viscosity to develop a satisfactory oil film, and a reduction in fatigue life will result. Both values of lubricant viscosity, v and v_1, are referenced at the bearing's actual operating temperature.

LUBRICANT SELECTION

Among a lubricant's functions are separation of the rolling elements and raceway contact surfaces, minimization of sliding friction, and corrosion prevention. A bearing's lubricant viscosity requirements depend very little on the load; rather, they depend more on bearing size, dm ((bearing bore + O.D.)/2), and the bearing's rotational speed, n. A bearing's minimum required lubricant viscosity can be determined from Figure 8. In addition, the chosen lubricant should provide a higher viscosity than the minimum required viscosity, v_1, as an added safety margin or precaution and to minimize the sliding friction in the bearing.

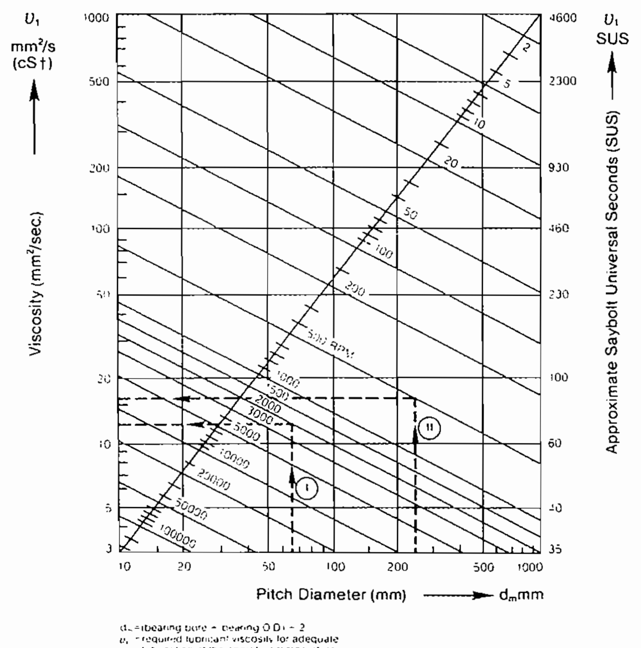


Figure 8. Minimum Required Lubricant Viscosity, v_1.

The actual viscosity of the lubricant, v, at any given operating temperature can be determined from the typical temperature-viscosity chart in Figure 9.

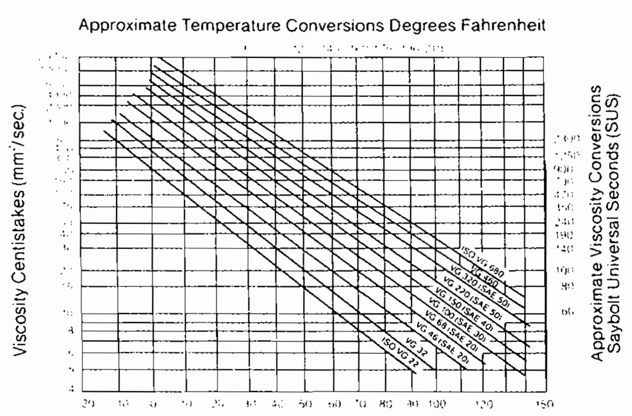


Figure 9. Temperature-Viscosity Chart.

Lubricating oils are identified by ISO Viscosity Grade (VG), which denotes the oil viscosity at 40°C (104°F); e.g., an ISO VG 100 oil has a viscosity of 100 cSt (480 SUS) at 40°C (104°F). Each oil grade can provide satisfactory lubrication up to a limiting temperature, depending on its viscosity. An ISO VG 46 oil, for instance, can provide satisfactory lubrication of a certain bearing to a temperature of 72°C (162°F). An ISO VG 68 oil can provide satisfactory lubrication of the same bearing to a temperature of 82°C (180°F), and an ISO VG 100 oil does so to 90°C (194°F). The ISO VG 68 or the ISO VG 100 oils are generally recommended for most pump applications because each provides a greater safety margin to the bearing lubrication.

How frequently a lubricant must be changed depends on the operating conditions and the quality of the lubricant. The lubricant must be free of solid and liquid contaminants. When the following general guidelines for contaminant levels are exceeded, the lubricant should be changed.

solid contaminants	0.2 percent (max.)
water	0.002 - 0.2 percent
acidity	Max. 1 unit increase

Specific limits of allowable contamination level are functions of various properties of both the contaminant and the lubricant. Details should be discussed with the manufacturers of the bearings and lubricants.

Quality mineral oils with a minimum Viscosity Index of 95 are recommended; however, multigrade, detergent, and lubricants with viscosity improvers are not recommended because of possible poor additive shear stability. Generally, mineral oils oxidize at a continuous operating temperature of 100°C (212°F) and should be changed every three months, while synthetic oils are more resistant to temperature effects and therefore require less frequent change.

LUBRICATION METHODS

The method of bearing lubrication has a tremendous effect on the performance and life of a bearing. The following statements review the four major methods of pump bearing lubrication: static oil, oil-ring, oil-mist, and grease lubrication.

Static Oil Lubrication

In a horizontally oriented pump, the oil level should be set at the center line of the lower-most rolling element (Figure 10). This is the static oil level.

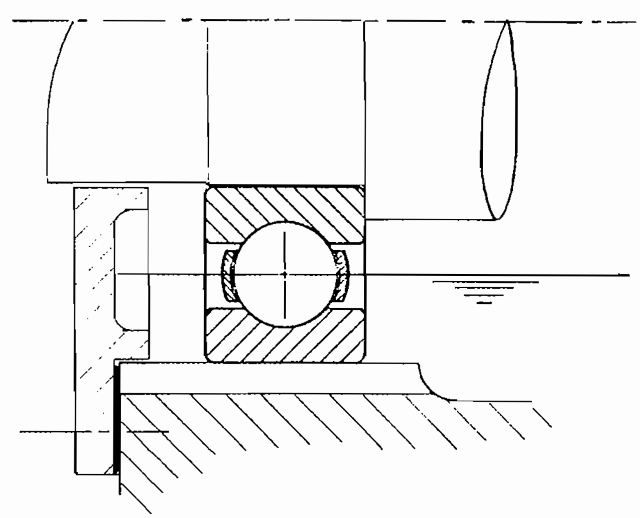


Figure 10. Static-Oil Lubricated Bearings.

In a vertically oriented pump, the oil level is generally set at the center of the upper-most bearing, except for spherical roller bearings. In the latter case, the bearing should be completely submerged in oil, which may cause higher viscous friction and higher bearing temperatures. In this case, it may be necessary to provide external cooling to the oil. A vertical thrust bearing arrangement is illustrated in Figure 11.

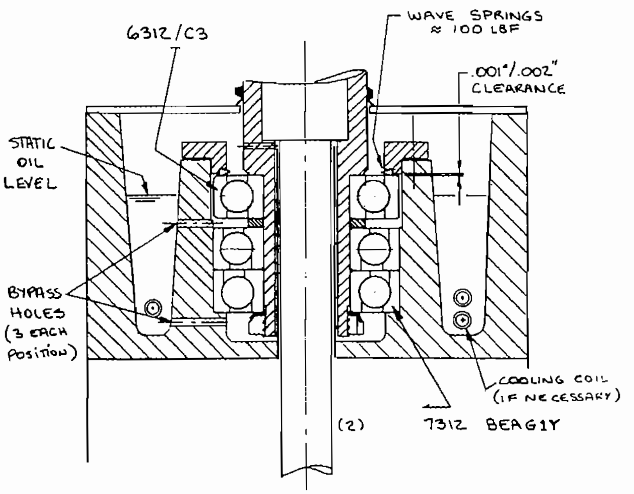


Figure 11. Vertical Pump Thrust Bearing Arrangement.

The oil level, established in the sight glass with the pump idle, may vary slightly with shaft rotation, depending on how the sight glass is positioned on the housing and in which direction the shaft is rotating.

A constant level oiler is a device commonly used in conjunction with static oil lubrication. Provided the housing seals are effective, the constant level oiler may not be necessary. This device adds oil to the bearing housing after initial startup, which causes slightly more oil churning and friction. A constant level oiler is ineffective as a sight glass; thus, the sight glass should still be used with a constant level oiler.

A bearing housing should be designed to allow oil to enter the bearing freely from both sides. This is particularly important with pairs of angular contact ball bearings and double row bearings, which at high speeds can pump the lubricant from themselves, possibly causing lubricant starvation.

The bearing housing should provide sufficient volume of oil to the bearings. The recommended minimum oil volume, V, for each bearing in the housing is defined by Equation (3).

$$V = k \times D \times B \quad (3)$$

where V = Minimum oil volume per bearing, cc
D = Bearing outer diameter, mm
B = Bearing width, mm
k = Range from 0.02 to 1.0

Static oil lubrication represents the baseline of moderate load-independent (viscous) bearing friction, M_0, from which the other lubrication methods are measured.

Oil-Ring Lubrication

A typical oil-ring lubrication arrangement is illustrated in Figure 12. In this arrangement, a ring, usually made of brass, is suspended from the shaft and into the oil sump below. The ring bore diameter is 1.6 to 2.0 times the diameter of the shaft. The ring rotates to a great extent with the shaft, pulls oil from the sump, and throws the oil into the housing and bearings. Oil is directed to the bearings by the shaft and housing construction.

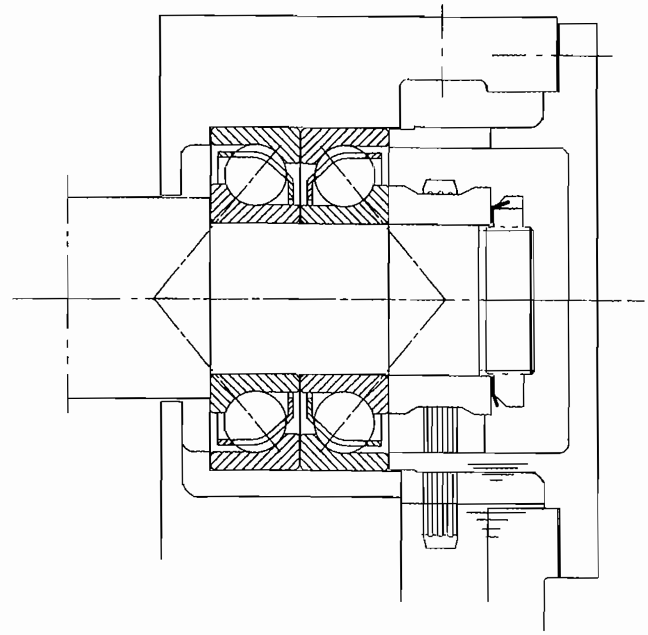


Figure 12. Oil-Ring Lubricated Bearings.

The sump's oil level should be below the bearing to an approximate minimum ring submergence of 10 to 20 percent of the ring's diameter. Increased bearing friction results if the oil level is within the bearing.

An advantage of oil-ring lubrication is that it reduces the load-independent bearing friction, allowing higher speed operation. A disadvantage, however, of oil-ring lubrication is the wear of the ring components.

Oil-Mist Lubrication

In the oil-mist lubrication system, a mist of atomized oil droplets is conveyed to the bearing by compressed air and is reclassified (i.e., precipitated) into larger droplets by the bearing itself or by a reclassifier fitting. The oil mist is produced by a mist generator and is supplied to the bearings at a pressure slightly above standard atmosphere.

The bearings can be either "purge" oil-misted or "pure" oil-misted. Purge oil-mist includes a static oil bath for the bearings, which results in higher bearing friction. Pure oil-mist, on the other hand, directly minimizes the load-independent bearing friction. Both methods help exclude contaminants from the bearing housing.

The mist can be applied to the housing or directed to the bearings themselves. Directed oil-mist is recommended at bearing n × dm values (speed, rpm, × mean diameter, mm) greater than approximately 300,000 and if the bearings are heavily thrust-loaded. A set of angular contact ball bearings lubricated by directed oil-mist is depicted in Figure 13. Manufacturers of oil-mist equipment should be consulted for specific recommendations.

The oil-mist must be vented to the ambient atmosphere on the bearing side opposite the oil-mist application to allow an effective oil-mist flow across the bearing. A labyrinth seal is an effective vent for this purpose.

Special dewaxed oils are recommended for use in oil-mist systems to prevent clogging of the mist fittings and other related problems.

Grease Lubrication

Lubricating greases are semi-fluid to solid dispersions of a metallic soap thickening agent in a mineral or synthetic oil.

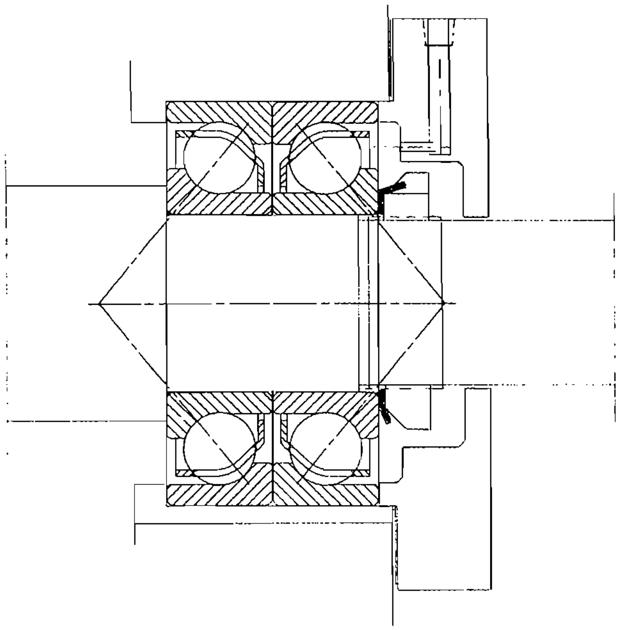


Figure 13. Oil-Mist Lubricated Bearings.

Lithium soap thickening greases are recommended for general rolling bearing applications. A grease should be selected so that its base oil can satisfy the bearing's lubrication requirements.

At normal temperatures, NLGI (National Lubricating Grease Institute) number 3 consistency greases are recommended for small- to medium-size ball bearings. An NLGI number 3 consistency grease should also be used in vertically oriented pump applications and in pumps having considerable vibration. NLGI number 2 consistency greases are recommended for roller bearings and medium- to large-size ball bearings. NLGI number 1 consistency greases are recommended for large bearings operated at low speeds.

Greases of different thickener types and consistencies should not be mixed together, because some thickener types are incompatible with one another. Mixing, thus, can result in a grease with an unacceptably low consistency. Specifically, polyurea base greases are generally incompatible with other metal soap base greases, mineral oils, and preservatives.

In general, the bearing and housing cavities should be filled 30-50 percent with grease at assembly. The interval between bearing regreasings can be calculated using Equation (4).

$$T = K\,[(14.0 \times 10^6/n\sqrt{d}) - 4\,d] \tag{4}$$

where T = Grease service life and relubrication interval, hr
K = Factor depending on bearing type
= 10 for ball bearings
= 5 for cylindrical roller bearings
= 1 for spherical roller bearings
n = Bearing rotational speed
d = Bearing bore, mm

The relubrication interval calculated by using Equation (4) is based on a bearing operating temperature of 70°C (158°F) and the use of average quality greases. The interval between regreasings can be increased for bearings operating with lower temperatures and if higher quality and/or temperature greases are used. On the other hand, the relubrication interval should be reduced for bearings operating at higher temperatures, if the possibility of contamination is present, and if the bearing axis is oriented vertically.

At each regreasing a quantity of grease, G should be added to the bearing. The quantity can be calculated using Equation (5).

$$G = 0.005 \times D \times B \tag{5}$$

where G = Quantity of grease to be added, gm
D = Bearing outer diameter, mm
B = Bearing width, mm

Bearings may be supplied with single or double shields and seals. These shield and seal arrangements for single row deep groove ball bearings are shown in Figure 14. These same arrangements are available for double row angular contact ball bearings.

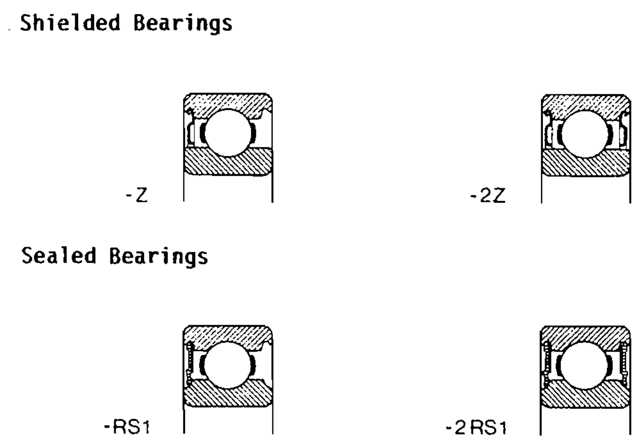


Figure 14. Sealed and Shielded Ball Bearings.

Bearings with two seals are considered "sealed for life," that is, they are not regreasable. Manufacturers typically supply sealed bearings with grease filling 25 to 35 percent of the bearing cavity. The life of the double sealed bearing depends on the L-10_a fatigue life of the bearing and the grease's life.

A single shielded bearing can be installed in a sealed pump to control the amount of grease entering the bearing. The bearing should be oriented with the shield toward the grease supply (Figure 15). Sufficient grease will pass through the shield clearance to lubricate the bearing and excess grease will pass out of the bearing. The user should regrease the bearing when the pump is idle.

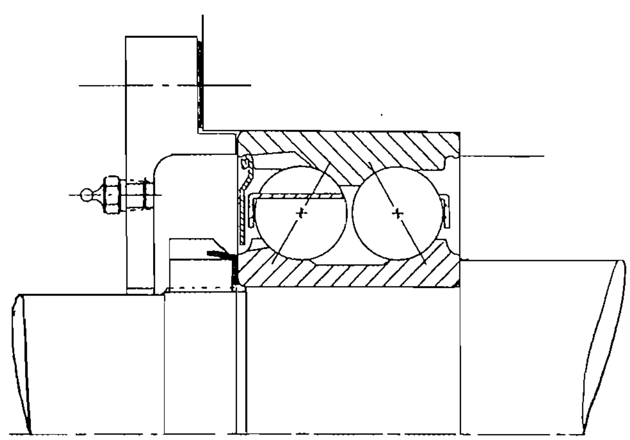


Figure 15. Grease Lubricated Double Row Bearing with Single Shield.

BEARING TEMPERATURE

Single row deep groove ball bearings are generally heat-stabilized to withstand temperatures to 120°C (250°F) without metallurgical or resulting dimensional change. Roller bearings are heat-stabilized to withstand higher temperatures: to 150°C (300°F) or higher. Individual bearing manufacturers should be consulted for details about their products.

Bearing operating temperature is usually limited by the lubricant's ability to provide sufficient viscosity. Ideally, operating temperature should not exceed 93°C (200°F). Special lubrication systems or features, such as circulating oil lubrication or oil cooling, may be necessary above this temperature.

Bearings with outer diameters greater than 100 mm (4 in) should be heated for mounting on the shaft, but the inner race temperature should not exceed 120°C (250°F). Bearings that are greased and sealed should not be heated for mounting because this may damage the grease and/or seals.

Bearing operating temperature can be estimated using empirical formulas for bearing internal friction and heat transfer. One such formula is found in Equation (6), which is the heat balance equation for heat generated by the bearing and heat loss to the ambient atmosphere.

$$\frac{n(M_0 + M_1)}{10{,}000} = K \left(\frac{dm}{1000}\right)^{1.65} (T_b - T_a) \qquad (6)$$

where M_0 = Load-independent (viscous) friction moment, N-mm
M_1 = Load-dependent friction moment, N-mm
n = Bearing rotational speed, rpm
K = Housing cooling factor, (typically 180-220)
dm = Bearing mean diameter, mm
T_b = Established bearing temperature, °C
T_a = Ambient temperature, °C

The bearing friction moments, M_0 and M_1, are functions of bearing type, loading, and speed; lubricant viscosity; and lubrication method. Values for M_0 and M_1 can be obtained from the bearing manufacturer.

Bearings with large numbers of rolling elements, such as extra-capacity filling-slot ball bearings and higher capacity spherical roller bearings, may operate at higher temperatures than similar bearings with fewer rolling elements because the former have a greater load-independent friction moment, M_0. A bearing that has a greater number of balls or rollers is forced to displace greater amounts of lubricant from the raceways, causing greater friction and resulting in higher temperatures. For example, tests conducted on filling-slot and Conrad type double row ball bearings show that filling-slot bearings operate 14-17°C (25-30°F) higher than Conrad bearings under similar conditions. A filling-slot bearing has a slot milled into the face of each of its rings to facilitate assembly of the balls int the bearing. This allows more balls to be assembled in a filling-slot bearing than in a Conrad bearing, which does not have a slot.

BEARING SYSTEM LIFE

The reliability or probability of any one bearing failure in a group of bearings is mostly dependent on the bearing with the shortest fatigue life. A pump with two bearings will have a shorter system life, L-10_s, than the L-10_a life of either individual bearing.

For example, let us discuss a pump that has two bearings. Each bearing has an individual L-10_a fatigue life of 24,000 hrs. This pump would have as a system an overall system life of 12,850 hrs (17.8 months). The influence that one bearing has on the system life of a pump with two bearings is shown in Figure 16. The system life is influenced very little when the L-10 life of bearing B is greater than 100,000 hours.

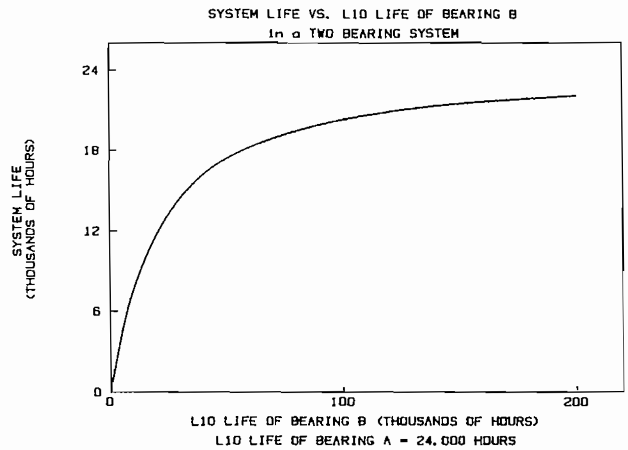


Figure 16. L-10_s System Life as a Function of Bearing Life.

Dosch [1] concludes that increasing the fatigue life of the bearings in an ANSI pump has little effect on the real mean time between failures (MTBF) of that type of pump. Similarly, an evaluation of bearing system life shows that a considerable increase in individual bearing fatigue lives would have to occur to cause a substantial increase in system life. The use of higher capacity bearings does not necessarily result in a more reliable pump, because other factors may contribute to failure. Because of this, total system life should be considered when evaluating pump bearing reliability.

The system life, L-10_s for a two-bearing pump, can be calculated using Equation (7).

$$\text{L-}10_s = \left[\frac{1}{\text{L-}10_A^{\,1.11}} + \frac{1}{\text{L-}10_B^{\,1.11}}\right]^{-.9} \qquad (7)$$

where: L-10_s = System L-10 life, Hrs
L-10_A = L-10 life of bearing "A," hr
L-10_B = L-10 life of bearing "B," hr

7300 SERIES ANGULAR CONTACT BALL BEARINGS

Seventy-three hundred series angular contact ball bearings that are assembled into pairs are supplied from the manufacturer in two configurations: with clearance or preloaded. A clearance and preloaded pair of angular contact ball bearings are illustrated in Figure 17. Bearing clearance/preload is denoted by the manufacturer with a suffix to the basic bearing marking, and bearings of different manufactures and markings should not be paired because of the possibility of mismatches in their internal clearances or preloads. The user should be certain to obtain replacement bearings that fully meet the bearing specifications of the pump manufacturer.

Paired bearings with clearance can be used when the loading is any combination of radial and thrust loads, and the speed is low. Preloaded bearings are recommended when the loading is predominantly thrust and the bearing's rotational speed is high (i.e., usually greater than 1800 rpm). Preloaded bearings are particularly recommended for vertical pump applications, in which the rotor weight usually adds to the applied thrust load.

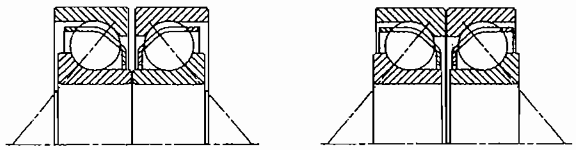


Figure 17. Angular Contact Ball Bearings with Clearance and Preload.

At high speeds, centrifugal forces and gyratory moments acting on the balls can reduce the traction forces that are necessary to maintain the balls rolling on the raceways. If the traction forces become too small due to these additional forces, the balls will have a greater component of sliding and will skid on the raceways. Ball skidding causes high bearing temperatures, metal smearing, and cage wear. Bearing preload maintains load on the bearings to prevent skidding.

Besides the initial clearance or preload manufactured into the bearings, the operating clearance or preload condition of the paired bearings can depend on the following:

- bearing precision (ISO P6 tolerances or normal tolerances).
- shaft and housing fits (ISO k5 and H6).
- locknut and housing cover clamping forces.
- paired arrangement (back-to-back or face-to-face).
- operating temperature.

The mounted preload force, P, is listed in Table 2 for one brand of angular contact bearings commonly manufactured for use in centrifugal pumps. The bearing preload is substantially increased due to its mounting on the shaft with an interference fit.

Table 2. Data on Angular Contact Ball Bearings Commonly Used in Centrifugal Pumps.

Angular Contact Ball Bearings (Preloaded)	Mounted Preload Force (Newton)*	Bearing Factor A	Maximum Clamp Force (Newton)
7306 BEAG05Y	950	8.1	4,750
7307 BEAG05Y	1155	11.1	5,300
7308 BEAG05Y	1280	18.9	7,125
7309 BEAG05Y	1340	29.2	8,875
7310 BEAG1Y	1780	45.2	11,000
7311 BEAG1Y	2110	62.5	13,000
7312 BEAG1Y	2160	84.3	15,000
7313 BEAG1Y	2220	111.4	17,000
7314 BEAG1Y	2260	144.6	19,500
7315 BEAG1Y	2400	184.9	22,000
7316 BEAG1Y	2450	233.8	25,000
7317 BEAG1Y	3130	291.8	28,000

*Based on nominal bearing conditions, mean normal shaft fit (k5 or m5 by size). 1 Newton=0.255 LBF

When thrust load is applied to a pair of preloaded bearings, one bearing supports the load while the residual preload on the adjacent bearing is reduced. The bearing supporting the load is called the "active" bearing and the adjacent bearing is commonly called the "inactive" bearing. The force-deflection diagram for a pair of preloaded bearings is shown in Figure 18.

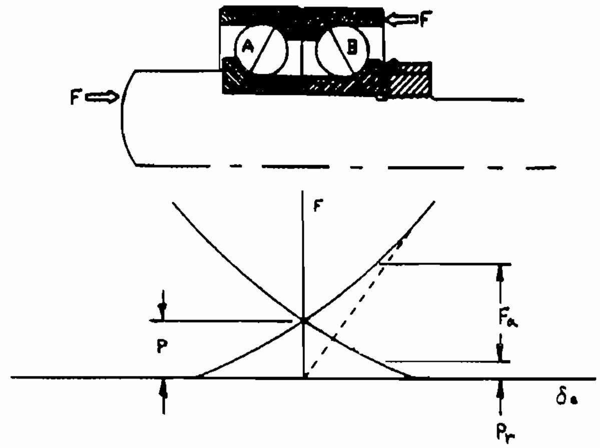


Figure 18. Force-Deflection Curve for Preloaded 7300 Series Angular Contact Ball Bearings.

The inactive 40 degree contact angle bearing will have zero residual preload when a thrust load equal to 2.85 times the preload force is applied to the active 40 degree contact angle bearing. The residual preload force, Pr, on the inactive bearing can be calculated using Equation (8).

$$P_r = P\left[\frac{P - F_a/2.85}{P}\right]^{1.5} \qquad \text{FOR } P > F_a/2.85 \qquad (8)$$

where P_r = Residual preload force on inactive bearing, N
P = Mounted preload force, N (Table 2)
F_a = Applied thrust load, N

Ideally, the residual preload force, Pr, should only be high enough to prevent the unloading of the inactive bearing. The residual preload force adds to the applied thrust load, Fa, to determine the L-10 life of the bearing.

At moderate speeds, provided the axial clearance in the bearing is not excessive, the preload between two angular contact bearings is actually increased during operation. This is due to centrifugal force on the balls, and thus the bearings will operate satisfactorily even though the inactive bearing may have zero residual preload. If the inactive bearing has excessive clearance, the orbital speed of the ball is reduced and the increase in preload is reduced, and thus the balls can skid.

At high speeds, it is necessary to maintain preload on the inactive bearing to prevent skidding. At bearing n × dm values approaching 500,000 the residual preload force, Pr, must be greater than the minimum axial force necessary to prevent skidding. The residual preload should be greater than the Fa(min) at high n × dm values. The minimum force, Fa(min) can be calculated using Equation (9).

$$F_{a(min)} = A\left(\frac{n}{1000}\right)^2 \qquad (9)$$

where $F_{a(min)}$ = Minimum required axial force, N
A = Factor based on bearing geometry (Table 2)
n = Bearing rotational speed, rpm

Back-to-back and face-to-face mounted angular contact ball bearings are illustrated in Figure 19. A back-to-back bearing arrangement is generally recommended for pump applications. Face-to-face mounted bearings can accept angular misalignments better than back-to-back mounted bearings, but in typical pump bearing arrangements angular misalignments are not significant. This is illustrated in Figure 20. Above two minutes of misalignment, the life of back-to-back mounted bearings begin to decrease. Greater than six minutes of misalignment causes face-to-face mounted bearings also to begin to experience a reduction in L-10 life. Two minutes of misalignment is equivalent to a bearing housing having its two bores offset 0.089 mm (0.0035 in) in a 152 mm (6 in) span. Typically, the bores in a housing are aligned within 0.025 mm (0.001 in). At this tolerance, back-to-back and face-to-face mounted bearings perform equally well.

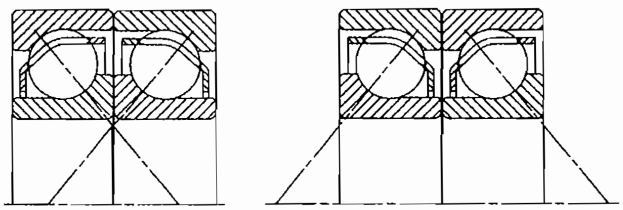


Figure 19. Back-to-Back (Right) and Face-to-Face (Left) Mounted Angular Contact Ball Bearings.

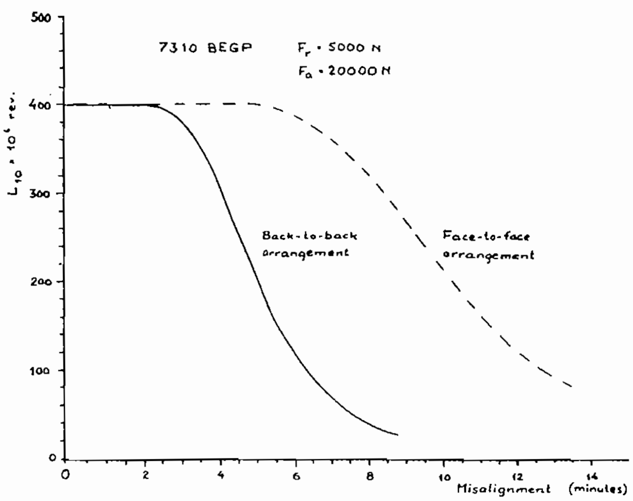


Figure 20. Bearing Life vs Misalignment for Angular Contact Ball Bearings.

In addition, face-to-face mounted bearings generally operate at higher temperatures than back-to-back mounted bearings. The preload in face-to-face mounted bearings can become unstable at high temperatures due to the thermal expansion of the inner rings relative to the outer rings. This is particularly true of grease lubricated face-to-face mounted bearings when grease is pumped to the space between the bearings and cannot escape, causing added friction.

Excessive clamping of the bearings on the shaft or in the housing can deform the bearing rings, adding excessive preload to the bearing. The clamping force should not exceed one quarter of the bearing's static load rating, Co (Table 2 has clamping force information).

8300 SERIES BEARING SET

An alternative to preloaded 7300 series bearings is an 8300 series bearing set (Figure 21). This bearing set can be used when the pump thrust is known to be in one direction. An 8300 series bearing set is a pair of 40 degree and 15 degree angular contact ball bearings. The 40 degree contact angle bearing supports the applied pump thrust load, and the 15 degree contact angle bearing is the inactive bearing. An 8300 series bearing set can accept thrust loads approximately three times heavier than a pair of preloaded 7300 series bearings can without skidding. The 8300 series bearing set should operate cooler because of this reduced skidding. Therefore, an 8300 series bearing set can be used in applications in which a pair of 7300 series bearings might skid.

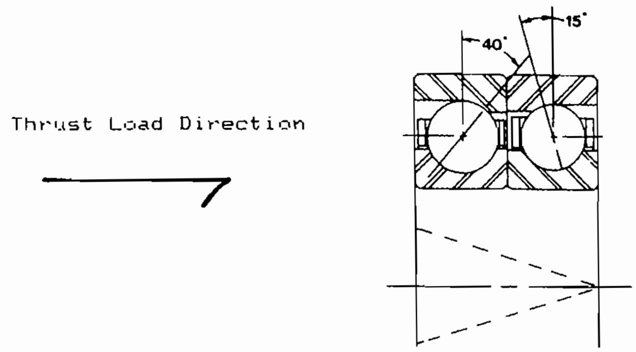


Figure 21. 8300 Series Bearing Set.

A "V" is scribed on the 8300 series bearing set to assure proper mounting orientation; that is, so that the 40 degree bearing will support the thrust load. The bearing set must be mounted so that the "V" points in the direction of the applied pump thrust.

TRIPLEX BEARING ARRANGEMENTS

If the pump thrust is very heavy, a triplex arrangement of angular contact ball bearings may be used. In a triplex mounting arrangement, two bearings are mounted in tandem, and one bearing is opposed in a back-to-back or face-to-face arrangement. The more common triplex back-to-back arrangement is illustrated in Figure 22. Preloaded bearings are required to maintain load on the inactive bearing and to assure good load sharing in the tandem bearings. Mounting a third preloaded bearing with a pair of preloaded bearings increases the preload by 35 percent. The inactive bearing of the triplex bearings will become unloaded when a force equal to approximately 4.2 times the preload is applied to the tandem bearings. The L-10_a life of a triplex bearing arrangement is calculated in a way similar to that calculated for two bearings mounted in tandem when the loading is predominantly thrust.

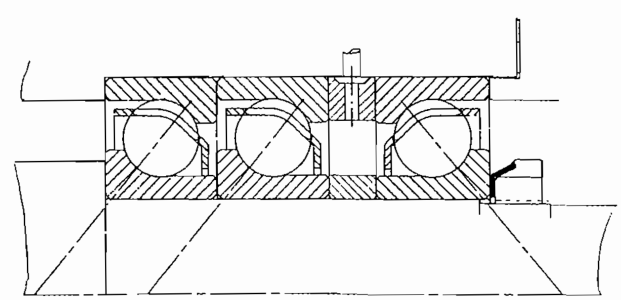


Figure 22. Triplex Mounted Angular Contact Ball Bearings.

Spacers between the active and inactive bearings are suggested to allow oil to freely enter the bearings. The inner and outer spacers should be ground together as a set and should have equal widths and parallel faces within 0.0025 mm (0.0001 in).

SHAFT AND HOUSING FITS

The International Standards Organization (ISO) has established ranges of shaft and housing fits for rolling bearings. Specific values for the fits are available from the ISO and bearing manufacturers. The recommended fits for rolling bearings in centrifugal pumps are as follows.

Shaft Fits

Bearings supporting any combination of radial and thrust load should be mounted on the shaft with an interference fit. The degree of interference fit generally depends on the type of bearing, although lighter or heavier interference fits may be used depending on the loading. Bearings supporting pure thrust load may be mounted with a clearance fit on the shaft, although an interference fit may be desired with angular contact bearings mounted in pairs to maintain a desired range of mounted preload. The recommended ISO shaft fits for the different types of bearings are as follows.

Ball bearings (SRDGBB, DRACBB, ACBB)

- ISO k5 to 100 mm bore
- ISO m5 over 100 mm bore, inclusive

Cylindrical roller bearing ISO m5
Spherical roller bearing ISO m5 to 65 mm bore
ISO m6 over 65 mm, inclusive

The shaft ideally should be machined to the mean of the diameter tolerance and should be round and without taper. Special considerations should be given to a shaft made of stainless steel. Stainless steels generally have a thermal expansion greater than carbon steels, and this can affect bearing internal clearance or preload. If the bearing is mounted on a stainless steel shaft and is near a heat source, a lighter ISO h5 shaft fit and a C3 (loose) internal clearance should be considered.

Housing Fits

Provided the bearing load is relatively constant and sufficiently heavy, bearings can be mounted in the housing with a clearance fit. The recommended ISO housing fits are as follows.

- ISO H6 or H7 (general applications)
- ISO G6 (cool ambient applications)

The housing bore should have a minimum 1,6 - 3,2 micrometers (63 - 125 RMS) surface finish.

BEARING INTERNAL CLEARANCE/END PLAY

Radial bearings and double row angular contact ball bearings should have internal clearances greater than normal (C3 suffix or loose) when the bearing speed is greater than 70 percent of the manufacturer's listed speed limit. This is because the higher operating temperatures that usually exist at these speeds may cause a preloading of the bearing. For the same reason, bearings with greater internal clearance should be used when heat can be transferred to the bearings from an external source.

BEARING CAGE TYPES

The function of the bearing cage is to position the rolling elements equally in the bearing. The major types of cages, ball-riding and outer- and inner-ring riding are illustrated in Figure 23.

Cage material and construction can affect the performance of the bearing. Cages made of polyamide material have performed very satisfactorily in single row deep groove and double row angular contact ball bearings and in cylindrical roller bearings in pumps. Single row angular contact ball bearings with polyamide cages have proved to be more sensitive to proper preloading than those with metallic cages. Spherical roller bearings without guide rings rely on the cage to control roller skew; polyamide cages are less suitable than metallic cages for performing this function. The polyamide material has a temperature limit of 100°C (212°F).

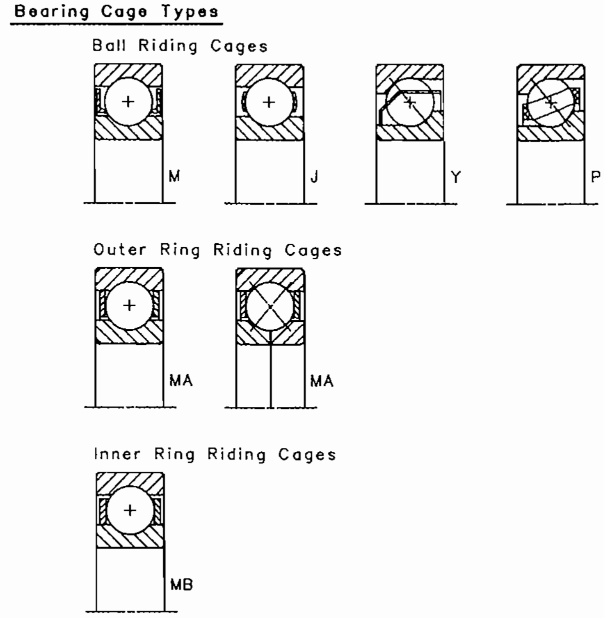


Figure 23. Common Bearing Cage Designs.

Bearings with machined brass cages are recommended when bearing n × dm values are greater than 600,000 for single row deep groove ball bearings and 450,000 for angular contact ball bearings.

Bearings with ball-riding cages can be oil or grease lubricated. Bearings with ring-riding cages preferably should be oil lubricated; however, angular contact ball bearings with counterbored rings can be oil or grease lubricated. A ring-riding cage has relatively small clearance with the guiding ring surface and may not receive satisfactory lubrication.

CONCLUSION

The primary considerations for the selection of rolling bearings for centrifugal pumps have been presented. Other considerations, such as bearing noise, etc. for specific applications may also require review by the pump manufacturer and user. Close cooperation between the bearing manufacturer, the pump manufacturer, and user is necessary for exchanging information and achieving the goal of increased pump and pump bearing reliability.

REFERENCE

1. Dosch, J. B. "Reducing ANSI Pump Maintenance Costs Through Component Upgrades," *Proceedings of the 4th International Pump Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1987).