

Bearings in centrifugal pumps



Application handbook

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Preface

This application handbook is one of a series of application handbooks designed to provide specific application recommendations for SKF® customers when used with the SKF General Catalog.

It is not possible, in the limited space of this handbook, to present all the information necessary to cover every application in detail. SKF application engineers should be contacted for specific bearing recommendations. The higher the technical demands of an application and the more limited the available experience, the more advisable it is to make use of SKF's engineering service.

We hope you find this handbook interesting and useful.



General

Principles of centrifugal pumps

A pump is a device for lifting, transferring, or moving fluids by suction or pressure from one position to another.

The centrifugal pump is a type of pump that uses the kinetic energy of a rotating impeller to impart motion to the fluid, see **figure 1.1**. The rotating impeller accelerates the fluid through its vanes and into the pump casing where the kinetic energy

of the moving fluid is converted to potential energy at higher pressure. As the fluid leaves the impeller through the pump discharge, more fluid is drawn into the pump inlet where the pressure is lowest. This fluid passes through the impeller as still more fluid enters the impeller.

There are three classifications of centrifugal pumps: radial flow, mixed flow and axial

flow based on the direction the fluid enters the inlet (eye) of the impeller, see **figure 1.2**. Radial and mixed flow pumps are either single or double suction designs.

A centrifugal pump produces head, H as a function of the rate of fluid flow, Q through the impeller, see **figure 1.3**. Head is the energy content in the pumped fluid, expressed in meters, m (ft).

Figure 1.1

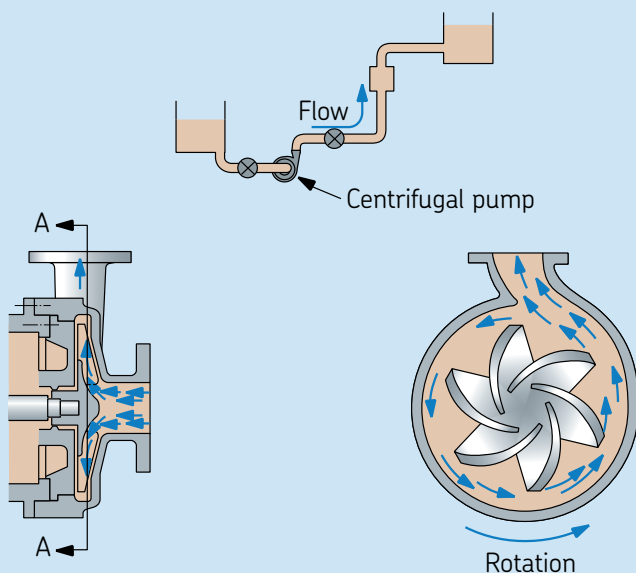


Figure 1.2

Classes of centrifugal pumps

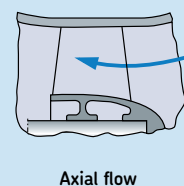
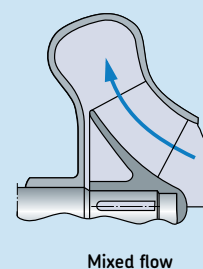
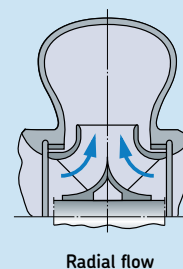
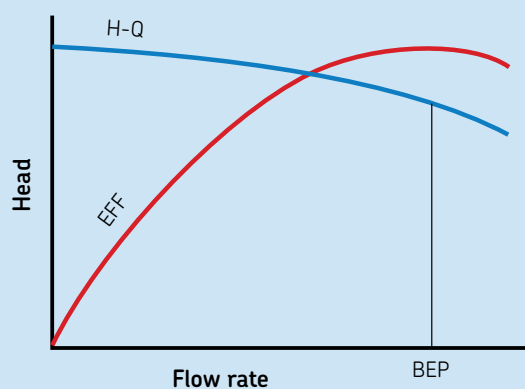


Figure 1.3



The hydraulic performance of a centrifugal pump is characterized by the mechanical shape and size of the impeller, using an index number called specific speed, n_s , see **figure 1.4**.

The specific speed number of a pump is calculated by the following equation:

$$n_s = \frac{n Q^{1/2}}{H^{3/4}}$$

where

n_s = specific speed

n = pump rotational speed, r/min

Q = pump flow rate, m³/s (US gallons/min)
at best efficiency point, BEP

H = pump total head, m(ft) at the BEP

*US units are in parenthesis

The characteristics of a pump based on specific speed are approximately as according to the table.

Pump characteristics		
Specific speed	n_s	Characteristic
low	10-35 (500-1750)	low flow high head
medium	35-85 (1750-4250)	medium flow medium head
high	85-160 (4250-8000)	high flow low head
highest	160-300 (8000-15000)	maximum flow minimum head

A centrifugal pump consists of a hydraulic assembly and a mechanical assembly, see **figure 1.5**. The components of the hydraulic assembly are the impeller, casing (volute), inlet and discharge piping, and shaft seal. The components of the mechanical assembly are the shaft and bearings, pump frame and housing seals, baseplate, and drive coupling or belt sheaves.

For petrochemical applications, the pump industry has developed standards for the manufacture and supply of centrifugal pumps. Two important standards are the ASME/ANSI B73.1 for chemical process pumps and API 610 for general refinery service pumps. These standards define the minimum technical requirements for the mechanical design of the pumps and bearings etc. Because of the strong American influence on petrochemical plant engineering, these standards have worldwide implications.

The centrifugal pump impeller is most typically supported on its own shaft and bearings and driven by an electric motor, and less often by an engine or a turbine. The pump shaft is connected to the driver either directly through a flexible coupling or indirectly by a belt drive. The impeller can also be rigidly connected to the motor shaft.

Pump Operation

A pump is selected for an application to produce a desired flow and head. The performance curve of a typical radial centrifugal pump is illustrated in the **figure 1.6**. The curve shows the head, efficiency, power requirements, and net Positive Suction Head required (NPSH_r) of the pump versus the flow.

Figure 1.4

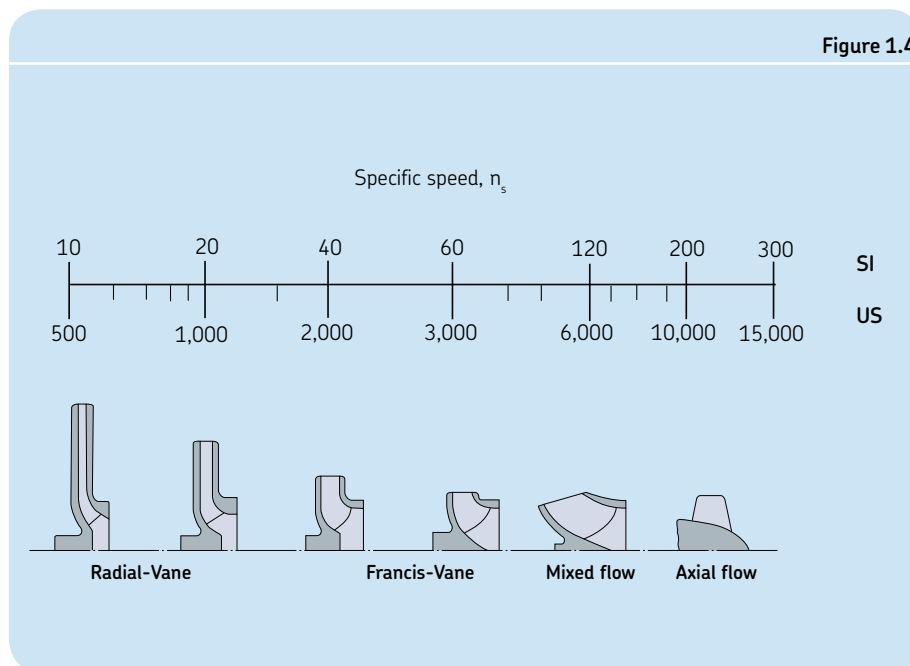
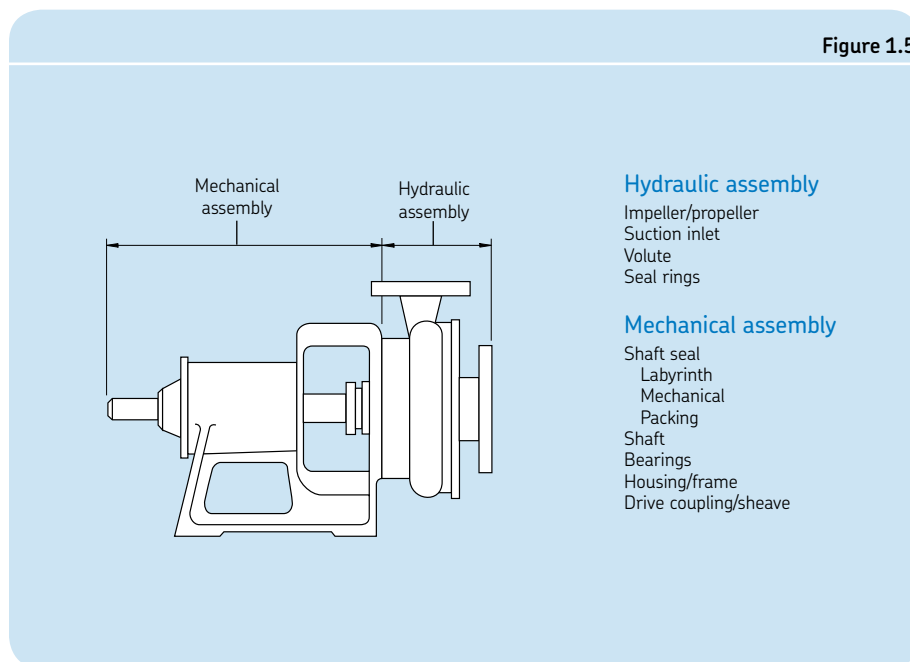


Figure 1.5



The point of highest pump efficiency is called the “Best Efficiency Point” or BEP. This is the pump design point and the operating point where the flow has the least friction and disturbance as it passes through the pump. For lowest power consumption, the pump is operated between 80 and 100% of BEP. Because of practical considerations, it is common for a pump to operate in the range of 50 to 120% of BEP. Pump operation at a flow rate below the BEP causes poor hydraulic performance and increased hydraulic impeller loads. Pump operation above the BEP can result in cavitation and increased vibration.

The $NPSH_r$ is the head at the pump inlet (suction) needed for the pump to satisfactorily draw the fluid into the impeller. If the available head at the pump inlet, called net Positive Suction Head available ($NPSH_a$), is less than the pump’s $NPSH_r$, cavitation will occur and performance will be reduced.

Cavitation is the phenomenon that occurs when the local pressure of the fluid is less than its vapor pressure and local vapor is formed from the fluid. A pump operating with insufficient $NPSH_a$, experiencing cavitation, develops small vapor bubbles near its inlet that grow in size as they move further into low pressure areas of the impeller. This causes unbalanced flow and pressure on the impeller. As the vapor bubbles reenter high pressure areas of the impeller, they collapse, exerting forces on the impeller that cause impeller damage, shaft deflection and increased bearing loading.

The common nominal pump rotational speeds for small and medium size pumps are 1,500 and 3,000 r/min at 50 Hz frequency and 1,800 and 3,600 r/min at 60 Hz frequency. Other rotational speeds are possible with belt and gear driven pumps, etc.

Pump Bearing Loads

The pump bearings support the hydraulic loads imposed on the impeller, the mass of impeller and shaft, and the loads due to the shaft coupling or belt drive. Pump bearings keep the shaft axial end movement and lateral deflection within acceptable limits for the impeller and shaft seal. The lateral deflection is most influenced by the shaft stiffness and bearing clearance.

The hydraulic loads comprise of hydrostatic and momentum forces from the fluid. The forces on the impeller are simplified into two components: axial load and radial load.

Axial Load

The axial hydraulic pressures acting on a single stage centrifugal pump are illustrated in the **figure 1.7**. The axial load is equal to the sum of the forces:

1. the hydrostatic force acting on the impeller’s front shroud and hub (back) shroud due to the hydraulic pressures acting on the surface areas of the shrouds
2. the momentum force due to the change in direction of the fluid flow through the impeller, and
3. the hydrostatic force due to the hydraulic pressure acting on the impeller (suction) opening. The hydrostatic forces dominate the impeller loading.

The magnitude and direction of the axial force may change during the pump start process owing to varying flow conditions in the side spaces between the impeller shrouds and casing walls. The changes in flow conditions and the consequential changes in pressure distributions on the impeller shrouds result in changes to the axial load.

In single stage end suction pumps, the magnitude and direction of the net axial load is most influenced by the design of the impeller. Four typical impeller designs are illustrated in the **figure 1.8**. The semi-open impeller with pump-out vanes and the closed impeller with two wear rings and balance holes are most common in petrochemical and paper mill process applications.

Figure 1.7

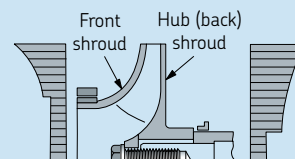
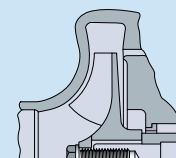
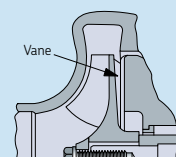


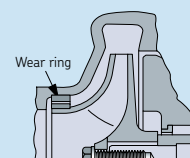
Figure 1.8



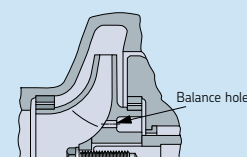
Semi-open impeller



Semi-open impeller with vanes

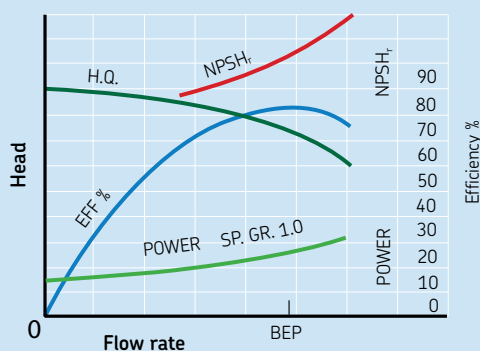


Closed impeller with one wear ring



Closed impeller with two wear rings and balance holes

Figure 1.6



In pumps with open and semi-open impellers, the axial load is normally directed towards the suction side owing to the pressure on the large area of the hub shroud. Closed pump impellers with wear rings can have near balanced (zero) axial load or more usually low axial load directed towards the suction. With increased suction pressures, the axial load can be directed opposite to the suction.

Impeller pump-out vanes and balance holes are employed to balance the axial load.

Pump-out vanes (also called back vanes) are small radial vanes on the hub shroud used to increase the velocity of the fluid between the hub shroud and the casing wall. This reduces the pressure of the fluid and results in reduced axial load on the impeller. The ability of pump-out vanes to reduce axial load is dependent on their clearance with the back casing surface. Balance holes are holes in the hub shroud used to equalize (balance) the pressure behind the impeller with that of the pump suction. Balance holes help to balance the two hydrostatic forces acting in opposite directions on the impeller shroud surfaces. SKF performed tests in which results illustrate the influence of the balance holes on pump axial load, see **figure 1.9**. The impeller without balance holes has greater axial

load than the impeller with balance holes.

The magnitude and direction of the axial load can change from its design value if pump-out vane clearance changes due to wear or is not set within tolerance and if balance holes become plugged with debris. Pump-out vanes and balance holes reduce pump efficiency by several percentage points.

The axial load in double suction impeller pumps is balanced except for possible imbalance in fluid flow through the two impeller halves.

In multistage pumps, impellers are arranged in tandem and back-to-back to balance the axial load.

Radial Load

The hydraulic radial load is due to the unequal velocity of the fluid flowing through the casing. The unequal fluid velocity results in a non-uniform distribution of pressure acting on the circumference of the impeller. The radial load is most influenced by the design of the pump casing.

The casing is designed to direct the fluid flow from the impeller into the discharge piping. In a theoretical situation at BEP, the volute casing has a uniform distribution of velocity and pressure around the impeller periphery, see **figure 1.10**.

In a real volute at the BEP, the flow is most like that in the theoretical volute except at the cutwater (or tongue) which is needed for the volute construction, see **figure 1.11**.

The disturbance of flow at the cutwater causes a non-uniform pressure distribution on the circumference of the impeller resulting in a net radial load on the impeller. The radial load is minimum when the pump is operating at the BEP and is directed towards the cutwater. The radial load increases in magnitude and changes direction at flows greater than and less than the BEP, see **figure 1.12**.

Figure 1.10

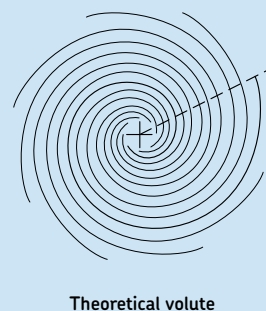


Figure 1.11

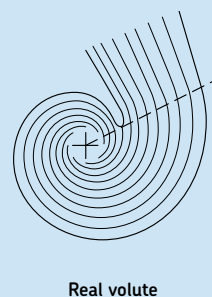


Figure 1.12

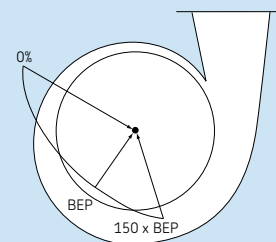
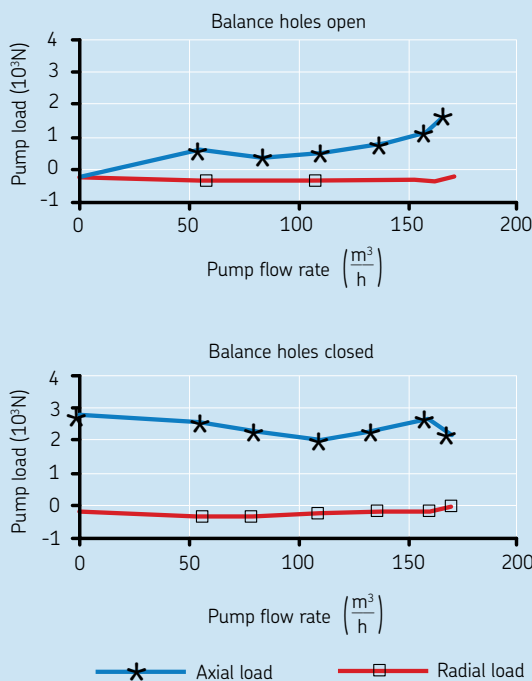


Figure 1.9

Hydraulic loads



Four typical casings are illustrated in the **figure 1.13**. The single volute casing is commonly used in small process pumps. The diffuser and circular volutes are also commonly used and, owing to their diffuser vanes or more open design, have more uniform velocity distribution around the impeller and therefore have lower radial impeller loads. The radial load in a circular volute is minimum at pump shut-off (zero flow) and is maximum near the BEP.

Double volute casings are commonly used in larger pumps when this construction is possible. A double volute casing has two cutwaters which radially balance the two resulting and opposing hydraulic forces. This significantly reduces the hydraulic radial load on the impeller.

Fluctuating and unbalanced radial loads superimpose on the steady radial load. The fluctuating load is sometimes due to the interaction of the impeller vanes passing the casing cutwater. The frequency of the fluctuating force is equal to the number of impeller vanes times the rotational speed. Unbalanced forces can be due to unevenness in the flow through the passages of the impeller or to mechanical imbalance.

The hydraulic loads are dependent on the type and size of impeller and casing, the pump operating conditions such as fluid suction pressure, and the point of pump operation. The magnitude and direction of the hydraulic loads can change greatly with changes in these factors. In most instances, the lowest hydraulic loads exist only at pump operation at the BEP. Pump cavitation influences operation and consequently the pump hydraulic loads.

Bearing loads should be evaluated at the BEP condition and at the maximum and minimum pump rated conditions.

Belt drives and flexible couplings also exert force on the pump shaft. The force from a belt drive is greater than that from a flexible coupling. The forces resulting from flexible couplings are minimized with improved pump shaft and drive motor shaft alignment.

The magnitude and direction of hydraulic loads and the load from the belt or coupling drive are best obtained from the pump manufacturer or user. **Figure 1.14** illustrates a typical bearing arrangement for an end suction centrifugal pump. From the figure, using the equations of engineering mechanics or SKF computer programs, the bearing reactions can be calculated.

Figure 1.13

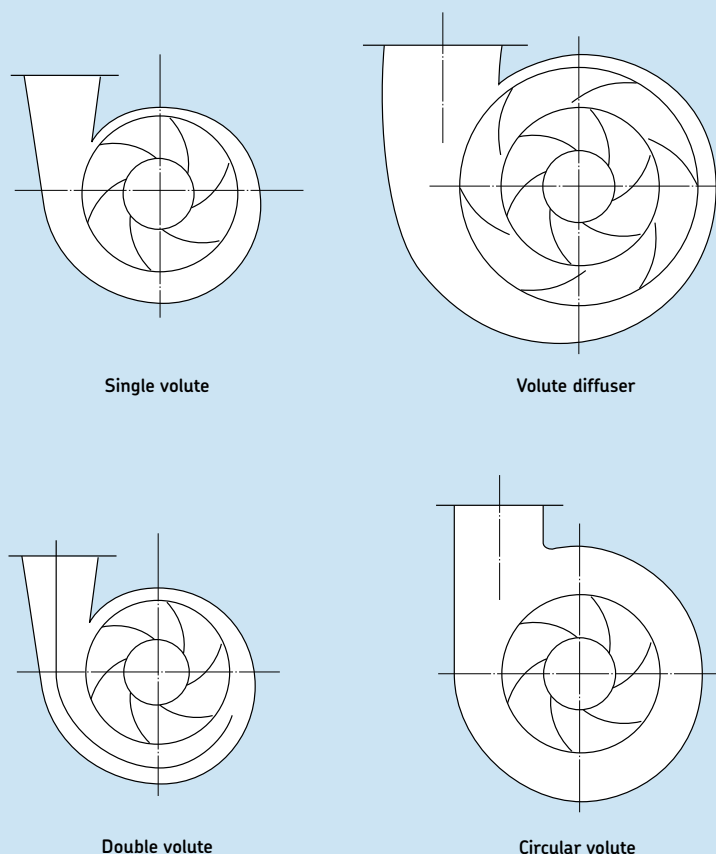
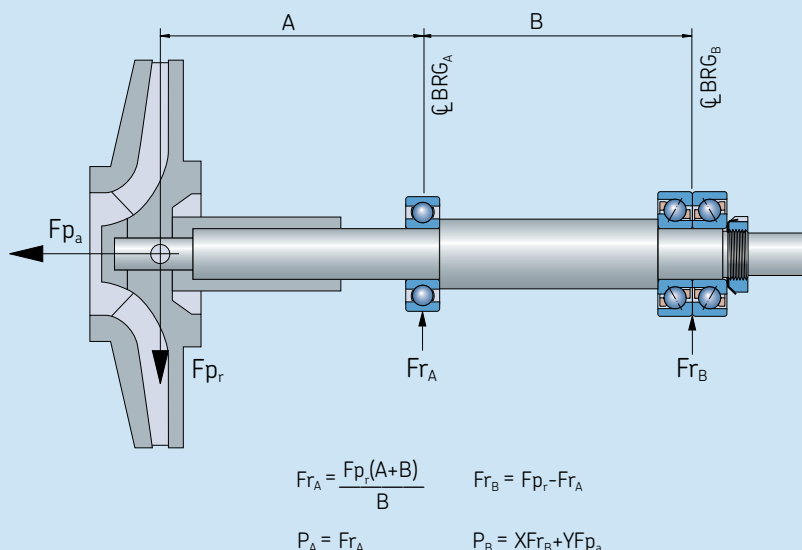


Figure 1.14



Pump bearings

Bearing types used in centrifugal pumps

Figure 2.1 illustrates the rolling bearings common to centrifugal pumps. The three most used ball bearing types are the single row deep groove ball bearing, double row angular contact ball bearing and universally matchable single row angular contact ball bearing.

Ball bearings are most commonly used in small and medium sized pumps because of their high speed capability and low friction.

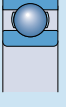
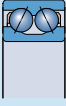
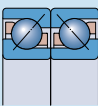
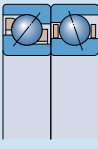
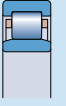
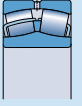
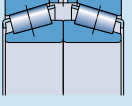
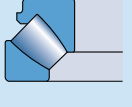
The SKF single row deep groove ball bearings and double row angular contact ball bearings are produced in Conrad (i.e. without filling slots) and filling slot type designs. For pump applications, Conrad bearings are preferred over the filling slot type bearing given that Conrad ball bearings operate at lower temperatures than filling slot bearings in similar pump conditions. If double row angular contact ball bearings with filling slots are used, they must be specially oriented so that the axial load does not pass through the filling slot side of the bearing. The API 610 Standard does not allow filling slot bearings of any type.

SKF single row angular contact ball bearings of the BE design (40° contact angle) are used where high axial load capabilities are needed for greater pump operational reliability. Universally matchable single row angular contact ball bearings can be arranged as pairs to support loading in either axial direction. SKF's MRC brand combines a 40° contact angle ball bearing with a 15° contact angle ball bearing to produce a bearing set called PumPac®. The PumPac bearing set can be used when the pump axial load acts predominately in only one axial direction.

SKF spherical, cylindrical, matched taper roller bearings and spherical roller thrust bearings are used in larger, slower speed pumps where the greatest bearing load carrying capacity is needed.

Figure 2.1

Approximate relative load, speed and misalignment capabilities

	Radial load	Axial load	Speed	Misalign-ment
 Single row deep groove ball bearing	X	X	XXXX	XX
 Double row angular contact ball bearing	XX	XX	XXX	X
 Single row angular contact ball bearing pair	XX	XXXX	XXX	X
 PumPac® bearing set	XX	XXXX (one direction)	XXX	X
 Cylindrical roller bearing	XXX	—	XXX	X
 Spherical roller bearing	XXXX	XX	XX	XXXX
 Taper roller bearing set	XXXX	XXXX	XX	X
 Spherical roller thrust bearing	—	XXXX	XX	XXXX

— No capacity
 X Low
 XX Moderate
 XXX High
 XXXX Very high

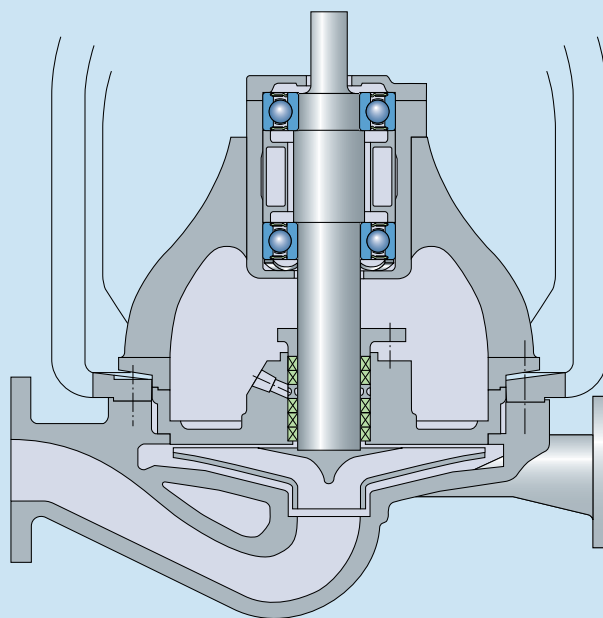
Pump bearing arrangements

The most common pump and pump bearing arrangements are shown in the following figures:

The vertical inline pump, **figure 2.2a**, and the horizontal process pump, **figure 2.2b**, are used in light duty chemical and paper mill process applications. The pump impellers are typically open or semi-open designs. The bearings of the vertical inline pump shown are grease lubricated and “sealed for life.” The bearings are spring preloaded to control the endplay of the shaft.

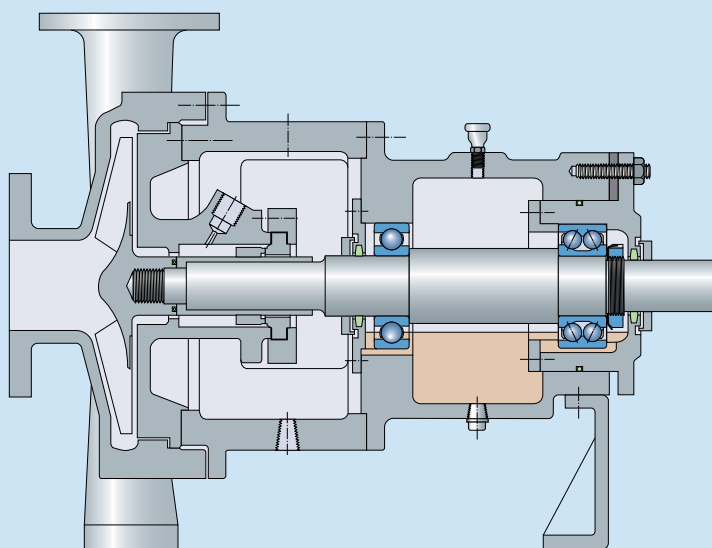
The bearings of the horizontal process pump are most frequently oil bath lubricated. In some cases (as shown in **figure 2.2b**) the bearings supporting the axial load are mounted in a bearing housing, separate from the pump frame, to allow adjustment of the impeller in the casing. In these cases, the adjustable housing is shimmed with the frame to ensure good bearing alignment.

Figure 2.2a



Vertical inline pump

Figure 2.2b

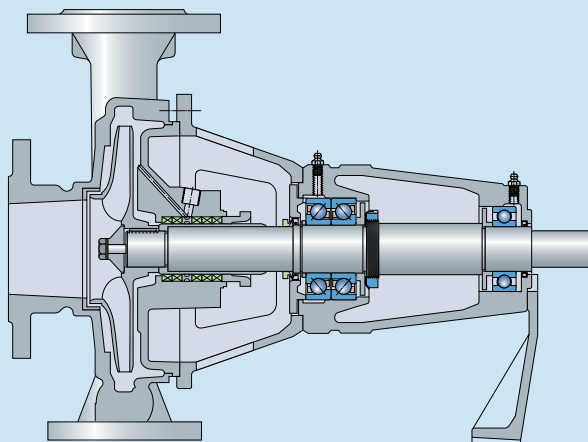


Process pump

The medium duty, **figure 2.2c**, and heavy duty, **figure 2.2d**, process pumps are used in refinery services where the highest reliability is required. The impellers are typically closed designs with one or more wear rings. The axial load is supported by universally matchable single row angular contact ball bearings. The bearings are most frequently oil bath or oil ring lubricated.

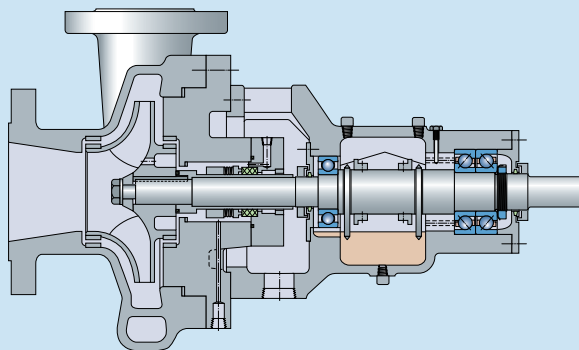
Two heavy duty slurry pump arrangements are shown in **figure 2.2e** and **figure 2.2f**. Roller bearings are used to support the heavier loading common in these applications. Matched taper roller bearings with steep contact angles, arranged face-to-face or back-to-back are well suited to support the combined axial and radial loads in these applications.

Figure 2.2c



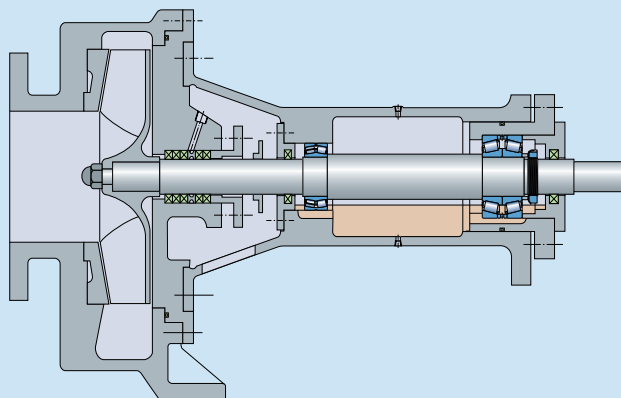
Medium duty process pump

Figure 2.2d



Heavy duty process pump

Figure 2.2e



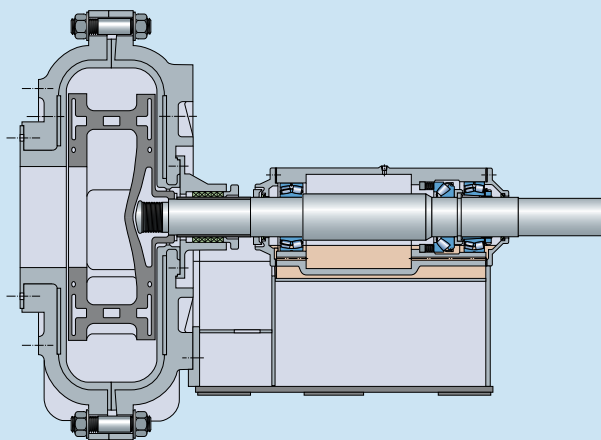
Slurry pump

Spherical roller bearings are used in slurry pumps having very heavy loads. The radial loads are supported by spherical roller bearings. A spherical roller thrust bearing supports the axial load. It is spring pre-loaded to ensure that sufficient load is applied to the bearing during conditions when the axial load reverses at pump start-up or stoppage for example. This arrangement is most commonly oil-bath lubricated.

For vertical deep well pumps, the spherical roller thrust bearing is a good choice, **figure 2.2g**. It easily accommodates the misalignment usual in these applications having long slender shafting.

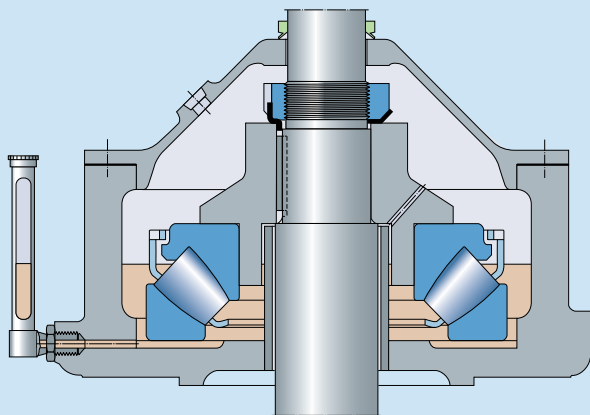
Poor reliability of the shaft sealing has increased the application of magnetic drive pumps, **figure 2.2h**. The impeller and its shaft are supported by plain bearings lubricated by the pumped fluid. Rolling bearings are used to support the drive shaft. Deep groove ball bearings are most commonly used in these types of pumps. The bearings can be spring preloaded to limit shaft end movement and maintain adequate load on the bearings. The spring preload prevents outer ring rotation in the often lightly loaded bearings.

Figure 2.2f



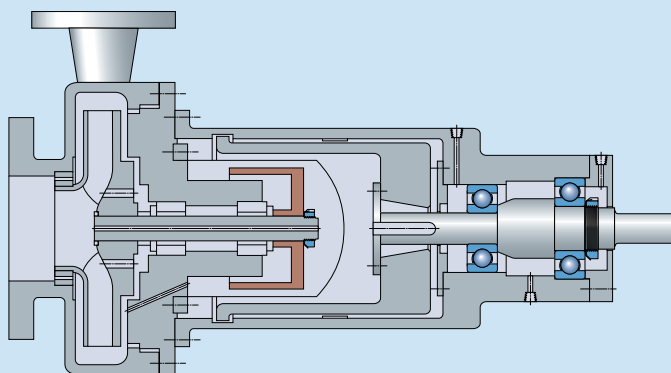
Slurry pump

Figure 2.2g



Vertical deep well pump

Figure 2.2h



Magnetic drive pump

Bearing life

Basic rating life

According to ISO 281:1990 the basic rating life of a rolling bearing is defined as the number of revolutions which the bearing is capable of enduring before the first sign of metal fatigue (flaking, spalling) occurs on one of its rings or rolling elements. This is expressed as

$$L_{10} = \left(\frac{C}{P} \right)^p$$

If the speed is constant, it is often preferable to calculate the life expressed in operating hours, using the equation

$$L_{10h} = \frac{10^6}{60n} L_{10}$$

The ASME/ANSI B73.1 Standard for process pumps specifies that rolling bearings shall have rating lives L_{10h} greater than 17,500 h at maximum load conditions and rated speed. The API Standard 610 for refinery service pumps specifies that rolling bearings shall have rating lives greater than 25,000 h at rated pump conditions and not less than 16,000 h at maximum load conditions at rated speed.

SKF rating life

Practical experience shows that field failures are not generally caused by fatigue, but are more often caused by contamination, wear, misalignment, corrosion, or as a result of lubrication or seal failure.

Therefore, ISO 281:1990/Amd 2:2000 contains a modified life equation to supplement the basic rating life to account for the lubrication and contamination condition of the bearing and the fatigue limit of the material.

The equation for SKF rating life is in accordance with ISO 281:1990/Amd 2:2000

$$L_{10m} = a_{SKF} L_{10} = a_{SKF} \left(\frac{C}{P} \right)^p$$

If the speed is constant, the life can be expressed in operating hours, using the equation

$$L_{10mh} = a_{SKF} \frac{10^6}{60n} \left(\frac{C}{P} \right)^p$$

where

- L_{10} = basic rating life (at 90 % reliability), millions of revolutions
- L_{10h} = basic rating life (at 90 % reliability), operating hours
- L_{10m} = SKF rating life, millions of revolutions
- L_{10mh} = SKF rating life, operating hours
- a_{SKF} = SKF life modification factor
- C = basic dynamic load rating, kN*
- P = equivalent dynamic bearing load, kN
- n = rotational speed, r/min
- p = exponent of the life equation
 - = 3 for ball bearings
 - = 10/3 for roller bearings

SKF life modification factor a_{SKF}

This factor represents the relationship between the fatigue load limit ratio (P_u/P), the lubrication condition (viscosity ratio κ) and the contamination level in the bearing (η_c).

Kappa is the ratio of the lubricant viscosity (ν) at the operating conditions to the minimum required lubricant viscosity (ν_1) at the operating conditions. The Kappa value should ideally be greater than 1.5.

Figure 2.3

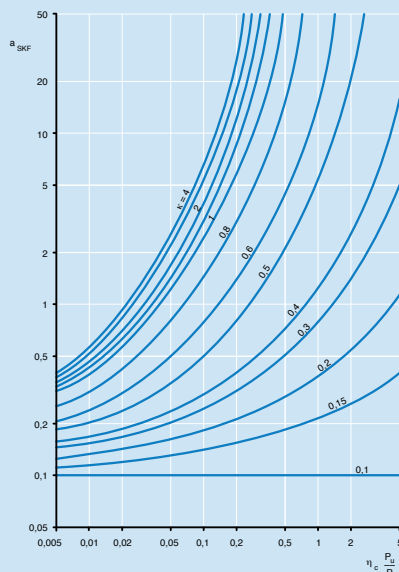
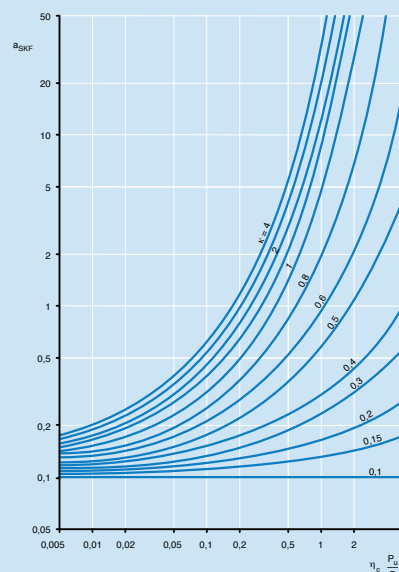


Figure 2.4



* See product tables in SKF General Catalog 6000EN.

The contamination factor (η_c) ranges from 0 for very severe contamination to 1.0 for extreme cleanliness. A typical η_c value for oil lubricated pump bearings falls between 0.3 to 0.6. The contamination factor (η_c) generally considers only solid particle contamination of the lubricant. Contamination of the lubricant by water and other fluids can also reduce the life of the bearings. Preferably, the water content should be below 200 ppm. There is risk of reduced bearing life if the water content is in excess of this value.

Values for the factor a_{SKF} can be obtained from the "SKF Interactive Engineering Catalogue" or "SKF Bearing Select" available online at www.skf.com or from **figures 2.3** and **2.4**, depending on bearing type, as a function of η_c (P_u/P) for SKF standard and SKF Explorer bearings and different values of the viscosity ratio κ : When calculating the rating life of SKF bearings, it is recommended that the basic rating life L_{10h} and the SKF rating life L_{10mh} each be evaluated, provided sufficient information is available to satisfactorily evaluate the SKF rating life. The added complexity of the SKF rating life reduces the uncertainty and risk by better predicting the bearing service life.

Bearing lubrication

The lubricant separates the rolling and sliding contact surfaces within the bearing. The lubricant also provides corrosion protection and cooling to the bearings. The principal parameter for the selection of a lubricant is viscosity, v .

In general, the allowable operating temperature of a bearing is limited by the ability of the selected lubricant to satisfy the bearing's viscosity requirements (i.e. kappa). Rolling bearings can achieve their rated life, even at high temperatures, provided the lubrication is satisfactory, and other precautions such as the correct selection of internal clearance, cage material, etc. are taken.

The effectiveness of the lubricant is primarily determined by the degree of surface separation between the rolling contact surfaces. The condition of the lubricant is described by the viscosity ratio kappa, κ , as the ratio of the actual viscosity v to the required viscosity v_1 for adequate lubrication. Both values are considered when the lubricant is at operating temperature.

$$\kappa = v/v_1$$

where

κ = viscosity ratio

v = actual operating viscosity of the lubricant, mm^2/s

v_1 = rated viscosity depending on the bearing mean diameter and rotational speed, mm^2/s (see **figure 2.5**)

In order to form an adequate lubricant film between the rolling contact surfaces, the lubricant must retain a certain minimum viscosity when the lubricant is at operating temperature. The rated viscosity v_1 , required for adequate lubrication, can be determined from **figure 2.5** using the bearing mean diameter $d_m = 0.5 (d + D)$, mm, and the rotational speed of the bearing n , r/min. This diagram has been revised taking the latest findings of tribology in rolling bearings into account. The actual lubricant selected for an application should provide greater viscosity v than the minimum required viscosity v_1 (i.e. $\kappa > 1.0$).

When the operating temperature is known from experience or can otherwise be determined, the ISO Viscosity Grade (VG) Number of the appropriate lubricant can be determined using **figure 2.6**. The VG Number is the viscosity of the oil in centistokes (mm^2/s) at 40°C (104°F). This figure is compiled for a viscosity index of 95.

The most common methods of pump bearing lubrication are: oil bath, oil ring, oil mist and grease. Circulating oil lubrication is also optionally used.

Figure 2.5

Estimation of the required viscosity v_1 at operating temperature

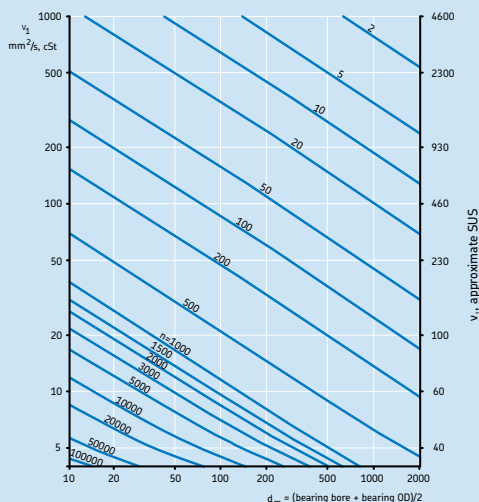
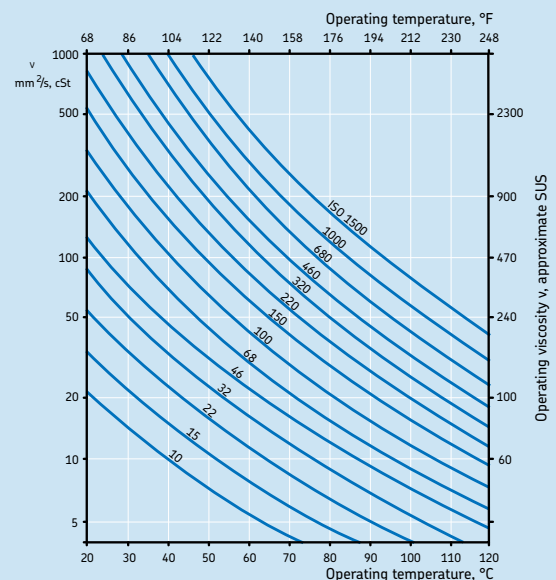


Figure 2.6



Oil-bath lubrication

Horizontal oil-bath lubrication represents the baseline of moderate bearing friction. The friction with other lubrication methods can be compared with that of oil-bath lubrication. Vertical oil bath lubrication produces high friction if one or more bearings are fully submerged, possibly limiting the operating speed.

In horizontally oriented applications, the oil bath level is set at the center of the bearing's lowest rolling element when the pump is idle, see **figure 2.7**. A sight glass or window is needed to visually set the oil level in the bearing. The oil level observed in the sight glass will vary slightly when the shaft is rotating due to the splashing of the oil in the housing.

The housing should allow the oil to freely flow into each side of the bearing. The housing should have a bypass opening beneath the bearings to allow the oil to flow freely. The cross-section area of the opening can be estimated according to the following equation:

$$A = 0.2 \text{ to } 1.0 (nd_m)^{1/2}$$

where

A = bypass opening cross-section area, mm²

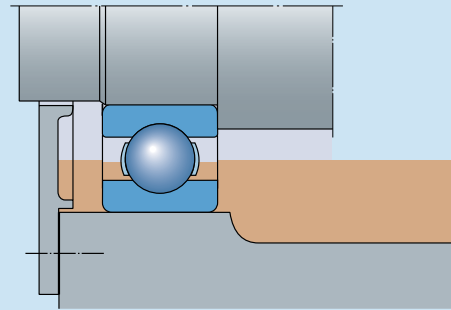
n = rotational speed, r/min

d_m = mean diameter of bearing, 0.5 (d + D), mm

The small value from the above equation applies to ball bearings and the large value to spherical roller thrust bearings. Intermediate values can be used for other bearing types. If the bypass opening is not provided, the oil may not pass through bearings having steep contact angles (angular contact ball, taper roller and spherical roller thrust bearings) operating at high speeds, in which case a pumping action caused by the bearing internal design may cause starvation of the bearing or flooding of the shaft sealing.

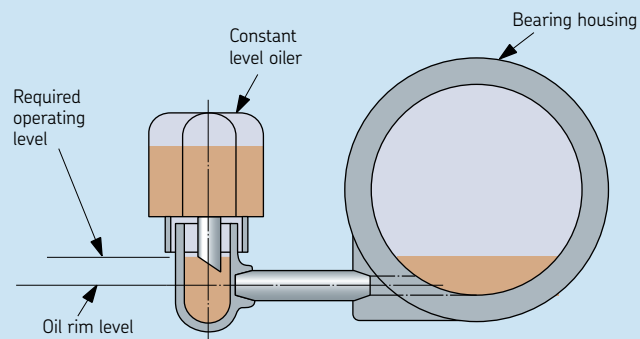
A "constant level oiler," such as an SKF LAHD 500, is an oil reservoir mounted to the bearing housing to maintain the correct oil level and replenish the oil lost from the bearing housing, see **figure 2.8**. A sight glass is recommended along with these devices to enable the correct setting and examination of the lubricant level in the bearing housing.

Figure 2.7



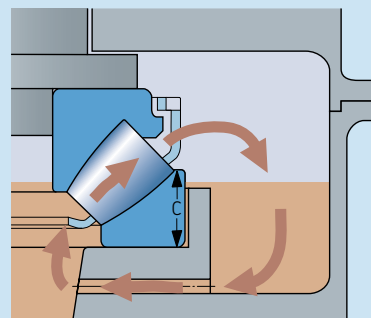
Oil-bath lubrication

Figure 2.8



Oil-bath lubrication

Figure 2.9



Oil-bath lubrication

The recommended minimum oil volume V for each bearing in the housing is estimated from:

$$V = 0.02 \text{ to } 0.1 D B$$

where

V = oil volume per bearing, ml

D = bearing outside diameter, mm

B = bearing width, mm

For applications with vertical shaft orientation, the oil level is set at or slightly above the vertical centerline of the bearing or bearing set. Spherical roller bearings operating in a vertical oil bath should be completely submerged. For spherical roller thrust bearings, the oil level is set at 0.6 to 0.8 times the bearing housing washer height, C see **figure 2.9**. Shaft sealing in these applications is best provided by a thin cylindrical sleeve inside the bearing inner ring support, see **figure 2.2g**.

The frequency of oil changes depends on the operating conditions and the quality of the lubricant. Quality mineral oils with a minimum Viscosity Index (VI) of 95 are recommended. Multigrade oils, and lubricants with detergents and viscosity improvers are not recommended. Mineral oils oxidize and should be replaced at three month intervals if operated continuously at 100° C (212° F). Longer intervals between replacements are possible at lower operating temperatures and when contamination is held at acceptable levels. Synthetic oils are more resistant to deterioration from exposure to high temperature and may allow less frequent replacement. Oil analysis at regular intervals is recommended to define the optimum replacement frequency.

Oil-ring lubrication

An oil ring is suspended from the horizontal shaft into an oil bath positioned below the bearings, see **figure 2.10** (see also **figure 2.2d**). The rotation of the shaft and ring flings oil from the bath into the bearings and housing. The housing channels the oil to the bearings.

Oil ring lubrication reduces the oil volume to the bearing and therefore reduces the viscous friction in the bearing system. The large size of the bearing housing needed for the oil ring improves the heat transfer from the bearings and oil. Higher shaft speeds and lower viscosity lubricants are possible with oil ring lubrication because of the lower friction and better cooling.

The performance of an oil ring is dependent upon the viscosity of oil that the ring is splashing. Therefore caution should be used when changing to a higher viscosity grade of oil.

The oil ring is made of brass or steel and sits on the shaft. The inner diameter of the oil ring is generally 1.6 to 2.0 times the diameter of the shaft and can be grooved for best oiling efficiency.

Some sliding may occur between the oil ring and the shaft causing wear. The shaft surface requires a fine finish to minimize this wear.

Oil-mist lubrication

A mist of atomized oil droplets is conveyed by compressed air to the housing where it is reclassified (precipitated) into larger oil droplets by a condensing fitting and the bearing, see **figure 2.11**. The mist is produced by a mist generator and is pressurized slightly above ambient pressure.

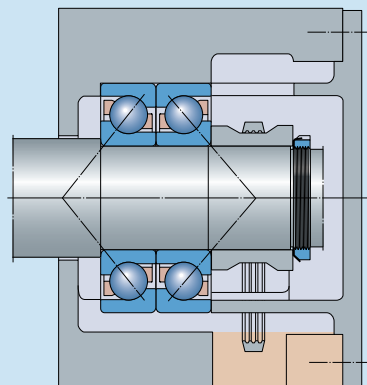
Oil mist minimizes the viscous friction in a bearing by providing fine droplets of clean, fresh, and cool lubricant. Contaminants are excluded from the bearings by the oil mist pressure inside the bearing housing. The mist can also be supplied to the bearings when the pump is idle for maximum bearing protection from contamination and condensation.

The oil mist may be introduced into the bearing housing (indirect mist) or directed at the bearing by a reclassifier fitting. In both cases, the housing must be provided with a small vent, 3 mm (0.125 in.) diameter, opposite the point where the mist enters the housing or bearing to allow free oil-mist flow. Directed oil mist is recommended if the bearing nd_m value is greater than 300,000 and if the bearing supports high axial load.

Synthetic or special de-waxed oils are often used for oil-mist lubrication. Paraffins in standard oils may clog the small oil mist fittings.

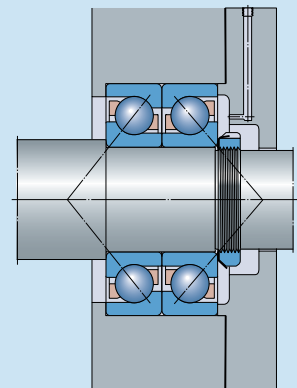
Bearings can be "purge" oil-mist lubricated or "pure" oil mist lubricated. Purge oil mist combines oil mist lubrication for bearings already lubricated with an oil bath. The "purge" oil mist purges contamination from the bearings and safeguards against the possible loss of oil bath lubrication.

Figure 2.10



Oil-ring lubrication

Figure 2.11



Oil-mist lubrication

“Pure” oil mist lubrication is without an oil bath, i.e. a dry sump. The bearings are lubricated only by the clean mist lubricant and less likely exposed to contamination. Pure oil-mist lubrication has been shown to significantly improve bearing life. The generation of oil-mist must be adequately safeguarded with alarms etc. to avoid bearing failure in the event of mist failure. It is recommended to prelubricate the bearings with similar oil or connect the bearings to the mist for a period of time before pump start-up to ensure satisfactory initial lubrication.

Environmental concerns may limit the use of oil mist lubrication. The bearing housings can be fitted with special labyrinth seals or magnetic face seals and oil mist collectors to limit the emissions to the environment.

Air-oil

Air-oil lubrication involves the delivery of a small amount of metered oil directly to each individual bearing using compressed air to carry the oil. The small amount of oil enables the bearings to run at lower operating temperatures and higher speeds than conventional static oil and grease lubrication. Unlike the oil mist system, the oil is not atomized, but actually coats the inside of the tubing lines and streaks along them (see **figure 2.12**). The oil is projected to the bearing, via a nozzle or direct passage. The compressed air helps cool the bearing slightly and also produces positive pressure within the housing to help prevent contaminants from entering. The transporting air stream is practically oil-free upon leaving the bearing before exiting through the seals or drain.

Grease lubrication

Lubricating greases are semi-liquid to solid dispersions of a soap thickening agent in a mineral or synthetic oil. The thickening agent is a “sponge” from which small amounts of the oil separate to lubricate the bearing.

Greases are selected for their consistency, mechanical stability, water resistance, base oil viscosity and temperature capability. Lithium soap thickened greases are good in all these respects and are recommended for general pump applications.

Grease consistency is graded by the National Lubricating Grease Institute (NLGI). The selection of the consistency grade is based on the shaft orientation, vibration, speed, size and type of bearing. The most common greases for roller bearings and medium to large ball bearings have a consistency grade of 2. NLGI 1 consistency greases are recommended for bearings operating at low temperatures or for large bearings operating at low speeds. NLGI 3 consistency greases are recommended for small-to-medium ball bearings, pumps operating with vertical shaft orientation and pumps having considerable vibration.

The grease base oil viscosity is selected in a similar manner to that of lubricating oils. The viscosity of the base oil at the bearing operating temperature should be greater than the minimum required lubricant viscosity v_1 .

Greases of different thickener types and consistencies should not be mixed. Some thickeners are incompatible with other type thickeners. Mixing different greases can result in a grease with unacceptable consistency. Polyurea thickened greases have historically been incompatible with

other metallic thickened greases, mineral oils, and preservatives. However recent formulations of polyurea greases have been updated to reduce incompatibility problems with other greases. But it is always recommended to check with the lubricant manufacturer for compatibility information.

The bearing and the adjacent housing cavity are generally filled 30 to 50% with grease at assembly. Excess grease is purged from the bearing into the housing cavity. The period that the grease can provide satisfactory lubrication (i.e. grease life) is dependent on the quality of the grease, operating conditions, and the effectiveness of the sealing to exclude contamination.

The SKF General Catalog 6000EN provides guidelines for the regreasing interval and the quantity of the grease to be added at regreasing. The relubrication interval (t_r) is based on the use of a lithium grease with mineral base oil at 70° C (158° F). The regreasing interval can be increased if the operating temperature is lower or if a premium quality grease is used. In contrast the regreasing interval is reduced when the bearing temperature is higher to account for the accelerated ageing of the grease.

The regreasing interval is reduced by half if the bearing orientation is vertical. It is best to provide a shelf beneath the bearing to help retain the grease. The shelf should have clearance with the shaft to allow excess grease to purge, see **figure 2.13**.

Excessive bearing temperatures may result if the bearing and the space around the bearing are completely packed with grease. The bearing housing should be designed to purge excess grease from the bearing at start-up and at regreasing.

Figure 2.12

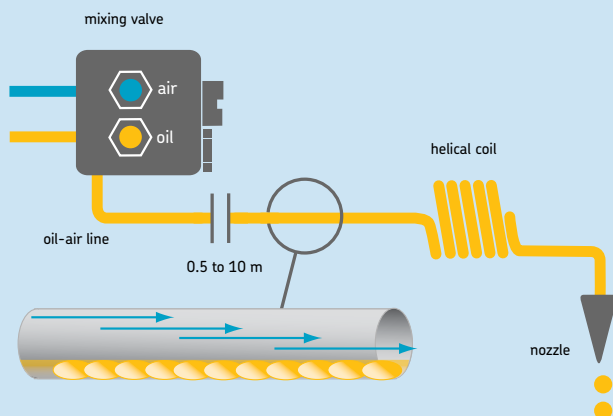
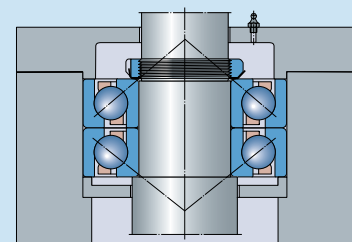


Figure 2.13



Vertical grease lubrication

SKF centralized lubrication systems for pumps

SKF offers a complete line of centralized total-loss and circulating oil lubrication systems:

- Grease lubrication units
- Air+oil system
- Circulating oil

Bearing temperature

In some instances, the operating temperature is the limiting factor determining the suitability of a bearing for an application.

Bearing operating temperature is dependent on the bearing type, size, operating conditions, lubrication, and rate of heat transfer from the shaft, bearing housing, and foundation. Operating temperature is increased when heat is transferred to the bearings from external sources such as high temperature pump fluids and rubbing contact housing seals.

Bearing internal heat generation is the product of the rotational speed times the sum of the load dependent and load-independent friction moments.

The General Catalog provides guidelines for calculating the bearing frictional moment which is comprised primarily of load-dependent and load-independent friction moments. The load-dependent frictional moment is the result of elastic deformations and partial sliding in the contacts. The load independent frictional moment is influenced by the hydrodynamic losses in the lubricant and depends on the viscosity and quantity of the lubricant.

For determining the frictional moment or power loss, SKF has developed a "Power loss" calculator available online at www.skf.com, using the latest SKF models.

The tolerances above are recommended for bearings mounted on solid steel shafts. Heavier fits than normal, resulting in greater interference, may be necessary if the bearing is mounted on a hollow shaft or sleeve.

Lighter fits using ISO j5 or h5 (k5 for large size bearings) tolerances may be necessary for bearings mounted on shafts made of stainless steel and having a large temperature difference between the bearing inner and outer ring. Stainless steels have lower conductivity than carbon steels and some stainless steels (AISI 316) have high coefficients of thermal expansion. High temperature in a bearing mounted with too heavy interference on a stainless steel shaft may cause too great stress in the bearing inner ring and excessively reduce the internal clearance. ISO j5 and h5 may also be used for bearings supporting pure axial loads.

An ISO j6 shaft tolerance can be used for all types of bearings supporting only axial load. An ISO k5 tolerance is commonly used with paired universally matchable single row angular contact ball bearings supporting only axial load to control bearing internal clearance or preload.

Housing fits

The standard recommended housing tolerance for all bearing types is ISO H6. This tolerance results in a slight clearance between the bearing outer ring and housing. This allows for easy assembly and radial clearance for bearing expansion with increases in temperature. The risk of ring rotation is minimal with this tolerance. The ISO H7 tolerance is recommended for larger bearings.

A looser ISO G6 housing tolerance is recommended for larger bearings ($d > 250\text{mm}$ [10 in.]) if a temperature difference greater than 10°C (18°F) exists between the bearing outer ring and the housing.

If the bearing is lightly loaded, it is recommended to spring preload the bearing outer ring. For radial bearings, the recommended spring preload is estimated from the following equation

$$F = k d$$

where

F = spring preload force, N

k = factor ranging from 5 to 10

d = bearing bore, mm

The housing material is recommended to have a hardness in the range of 140 - 230 HB, minimum. Too low material hardness can result in wear of the housing where the bearing seats.

The inner rings of double row ball bearings and paired universally matchable single row angular contact ball bearings arranged back-to-back should be clamped on the shaft with a locknut and washer. The outer rings of these bearings can be loosely clamped or preferably provided with slight axial clearance, 0.0 to 0.05 mm (0.002 in.), in the housing.

For all bearing types, the axial clamp force on the bearing rings should not exceed one quarter of the basic static load rating of the individual bearing ($C_0/4$). In the case of the double row angular contact ball bearings, the clamp force should not exceed one eighth of the static load rating ($C_0/8$). The clamp force must uniformly clamp the bearing rings without distortion. The above recommendations for shaft and housing fits are in accordance with the ANSI/AFBMA Standard 7, a requirement of the API Standard 610 pumps.

Bearing mounting and radial clearance

Shaft fits

The standard recommended shaft tolerances for ball and roller bearings in centrifugal pump applications supporting radial load or combined axial and radial loads are shown in **figure 2.14**.

These tolerances result in an interference between the bearing inner ring and shaft. This is needed if the bearing supports radial load.

Figure 2.14

Shaft fits				Tolerance
Shaft diameter (mm)				
Ball bearings	Cylindrical roller bearings	Taper roller bearings	Spherical roller bearings	
≤ 17	–	–	–	j5
18 to 100	–	–	–	k5
–	≤ 30	≤ 40	–	k6
101 to 140	31 to 50	–	≤ 40	m5
141to 200	–	41 to 65	–	m6
–	51 to 65	–	41 to 60	n5
–	66 to 100	66 to 200	61 to 100	n6
–	101-200	–	101-200	p6

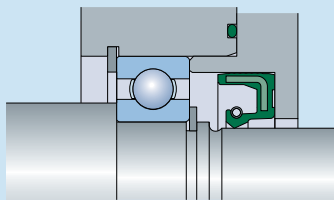
Bearing housing sealing

Sealing of the shaft at the housing is important to exclude solid and liquid contaminants and retain the lubricant. Common seals used on bearing housings are radial lip seals, **figure 2.15** and labyrinth seals, **figure 2.16**.

Radial lip (garter) seals have a synthetic rubber lip spring loaded to contact the shaft surface. The sealing depends on a lubricant supply to the seal and a good surface finish of the shaft. Excessive seal friction can cause high temperatures and seal and shaft wear. The life of lip seals is usually short (2,000 to 4,000 h).

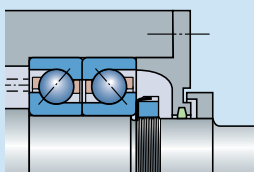
Labyrinth seals are effective in excluding contaminants and in retaining the lubricant. They cause little or no friction and have long life. Depending on their design, labyrinth seals can provide natural venting for oil mist.

Figure 2.15



Garter seal

Figure 2.16



Labyrinth seal

SKF Engineering Consultancy Services

Applications exist where it is desirable to predict the expected bearing life as accurately as possible because machine reliability is extremely important. SKF Engineering Consultancy Services provides calculations and simulations utilizing high-tech computer programs, in combination with over one hundred year global experience in the field of rotating machine components. They can provide support to:

- analyze technical problems
- suggest the appropriate system solution
- select the appropriate lubrication and an optimized maintenance practice.

Service benefits to OEM and end users include:

- faster development processes and reduced time to market
- reduced implementation costs by virtual testing before production start
- improved bearing arrangement by lowering noise and vibration levels
- higher power density by upgrading
- longer service life by improving lubrication or sealing

Advanced computer programs

Within the SKF Engineering Consultancy Services there are highly advanced computer programs which can be used for:

- analytical modeling of complete bearing arrangements, consisting of shaft, housing, gears, couplings, etc.
- static analysis, i.e. determination of elastic deformations and stresses in components of mechanical systems
- dynamic analysis, i.e. determination of the vibration behavior of systems under working conditions
- visual and animated presentation of structural and component deflection
- optimizing system costs, service life, vibration and noise levels.

The standard high-tech computer programs used within the SKF Engineering Consultancy Services for calculation and simulations are briefly described in the following sections.

SKF Interactive Engineering Catalog

The SKF Interactive Engineering Catalog (IEC) is an easy-to-use tool for bearing selection and calculation. Bearing searches

are available based on designation or dimensions, and simple bearing arrangements can be evaluated as well. It enables the generation of CAD bearing drawings and contains the complete range of rolling bearings, bearing units, bearing housings, plain bearings and seals. The SKF Interactive Engineering Catalog is published on the Internet at www.skf.com.

SKF bearing beacon

SKF bearing beacon is the new mainstream bearing application program used by SKF engineers to find the best solution for customers' bearing arrangements. The program combines the ability to model generic mechanical systems (using also shafts, gears, housings, etc.) with a precise bearing model for an in-depth analysis of the system behavior in a virtual environment. It also performs bearing rolling fatigue evaluation using the SKF rating life in particular. SKF bearing beacon is the result of several years of specific research and development within SKF.

Orpheus

Orpheus enables studying and optimizing the dynamic behavior of noise and vibration in critical bearing applications (e.g. electric motors, gearboxes). It can be used to solve the complete non-linear equations of motion of a set of bearings and their surrounding components, including gears, shafts and housings. It can also provide profound understanding of and advice on the dynamic behavior of an application, including the bearings, accounting for form deviations (waviness) and mounting errors (misalignment). This enables SKF engineers to determine the most suitable bearing type and size as well as the corresponding mounting and preload conditions for a given application.

Beast

Beast is a simulation program that enables SKF engineers to simulate the detailed dynamics inside a bearing. It can be seen as a virtual test rig performing detailed studies of forces, moments, etc. inside a bearing under virtually any load condition. This enables the "testing" of new concepts and designs in a shorter time and with more information gained compared with traditional physical testing.

Ball bearings in centrifugal pumps



Deep groove ball bearings

Deep groove ball bearings

Deep groove ball bearings are commonly used in small and medium duty centrifugal pumps, either as the radial bearing, or in some cases as the thrust bearing.

SKF Explorer deep groove ball bearings – the high performance class

SKF Explorer single row deep groove ball bearings are produced to higher precision than the ISO Normal tolerances. The dimensional accuracy corresponds to P6 tolerances, except the width tolerance. The running accuracy depends on the bearing size and corresponds to:

- P5 tolerances for $OD \leq 52$ mm
- P6 tolerances for $52 \text{ mm} \leq OD \leq 110$ mm
- Normal tolerances for larger bearings.

In addition to improved precision level other Explorer feature include:

- Optimized internal geometry and rolling contact surface
- Upgraded ball quality
- Higher cleanliness steel

Such features provide superior performance, quiet running, increased speed capability, and longer service life.

The API Standard 610 specifies that radial bearings shall be of Conrad design (no filling slots) and have greater than Normal (suffix C3) radial internal clearance. In general, greater than Normal radial internal clearance is recommended when:

- bearings are mounted with heavier than normal interference
- high operating temperatures are expected from heat conducted to the bearing from an external source
- operating at greater than 70% of the speed ratings listed in the product tables.

Angular contact ball bearings in centrifugal pumps

Single row angular contact ball bearings

Single row 40° angular contact ball bearings are widely used in medium and heavy duty centrifugal pumps, either as pure thrust bearings or for combined radial and



Angular contact ball bearing with brass cage

axial loads. The most important features are high radial and axial load carrying capacity combined with a high speed rating. Single row angular contact ball bearings operate with a small clearance or a light preload, providing good positioning accuracy of the shaft.

SKF Explorer single row angular contact ball bearings – the high performance class

SKF Explorer single row angular contact ball bearings are manufactured as bearings for universal matching with P6 dimensional accuracy and P5 running accuracy. In addition to improved precision level Explorer feature include:

- optimized internal geometry and rolling contact surface
- upgraded ball quality
- improved materials
- new heat treatment
- improved cages

These features provide superior performance, quiet running, increased speed capability, and longer service life.

Bearing minimum axial load

For satisfactory operation, angular contact ball bearings, like all ball and roller bearings, must always be subjected to a given minimum load, particularly if they are to operate at high speeds or are subjected to high accelerations or rapid changes in the direction of load.

Under these conditions centrifugal forces on the balls can cause a change in the contact angle between the inner and outer raceways. This contact angle difference can cause damaging sliding movements (skidding) to occur between the balls and raceways, leading to increased friction and cage stresses.

The requisite minimum axial load to be applied to the primary thrust bearing (or tandem arrangement) can be estimated using;

$$F_{am} = k_a \frac{C_0}{1,000} \left(\frac{nd_m}{100,000} \right)^2$$

In addition for bearing pairs arranged back-to-back or face-to-face, the requisite minimum radial load can be estimated using;

$$F_{rm} = k_r \left(\frac{vn}{1,000} \right)^{2/3} \left(\frac{d_m}{100} \right)^2$$

Figure 3

Minimum load factors		
Bearing series	Minimum load factors	
	K_a	K_r
72 BE	1.4	0.095
72 B	1.2	0.08
73 BE	1.6	0.1
73 B	1.4	0.09

where

F_{am} = minimum axial load, kN

F_{rm} = minimum radial load, kN

C_0 = basic static load rating of single bearing, or bearing pair, kN*

k_a = minimum axial load factor according to **figure 3**

k_r = minimum radial load factor according to **figure 3**

v = oil viscosity at operating temperature, mm²/s

n = rotational speed, r/min

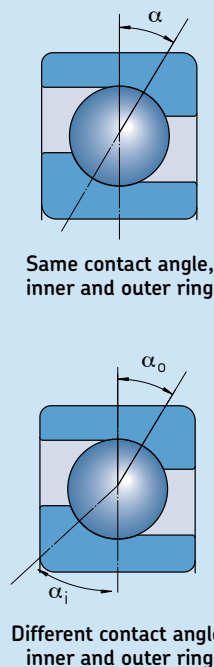
d_m = bearing mean diameter = 0.5 (d + D), mm

For determining the requisite minimum loads, SKF has developed a "Minimum load" calculator available online at www.skf.com.

During operation, the minimum required axial load in a bearing pair can be internally maintained by limiting the internal axial clearance. With small axial clearance, the balls are loaded by centrifugal force against the raceways with nearly equal inner and outer ring contact angles. As the axial clearance increases, so does the difference in the inner and outer ring contact angles. This allows increased internal sliding. This is illustrated in **figure 3.1**.

For pumps having a horizontal shaft, the weight of the components supported by the bearings, together with external forces, generally exceeds the requisite minimum radial load. If this is not the case, the angular contact ball bearing must be subjected to an additional load. In the case of pumps having vertical shaft orientation, the minimum required axial load may be satisfied by the weight of the shaft and pump impeller. When starting up at low temperatures or when the lubricant is highly viscous, even greater minimum loads may be required.

Figure 3.1



Bearing preload

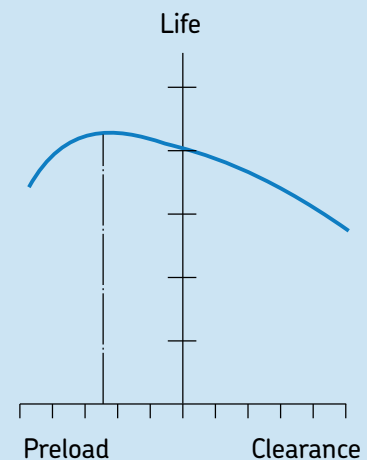
In centrifugal pump applications, bearing preload can:

- minimize the potential of ball skidding
- control contact angle variation
- improve internal load distribution
- increase bearing stiffness
- improve shaft positioning accuracy e.g. for mechanical seals

Bearing preload can increase the fatigue life of a bearing by improving internal distribution of the applied external loads. However, too great a preload will:

- reduce the bearing fatigue life as illustrated in **figure 3.2**.
- lead to loss of heat balance, ie. heat is generated faster than it can be dissipated
- increase rotational torque and power consumption
- increase sensitivity to misalignment and incorrect mounting (than bearings with clearance)

Figure 3.2



* See product tables in SKF General Catalog 6000EN.

Figure 3.3 shows the static load-deflection diagram for two preloaded angular contact ball bearings. This diagram is typical of 40° bearings arranged either back-to-back or face-to-face. Preload P^i in this example is achieved by the elastic deflection of the bearings against one another. The initial deflection of the bearings due to the preloading is δ_0 .

When an axial load is applied to the shaft, only one bearing supports this load. This bearing is denoted the “active” bearing. The deflection, δ , of the active bearing, as a result of the applied axial load, reduces the preload in the adjacent “inactive” bearing.

The load-deflection diagram for a pair of preloaded bearings rotating at 3,600 r/min is shown in **figure 3.4**. Under rotation, the preload force is increased, and the force in the inactive bearing does not fully reduce to zero due to centrifugal forces.

At increased speeds (nd_m values 250,000 and greater), gyroscopic spinning of the balls will occur if the residual preload in the inactive bearing is less than the minimum required axial load, F_{am} .

Bearing preload can also be applied by compression springs. The load from axial springs is constant and not affected by differences in the bearing mounting or temperature. The figures illustrate two examples of bearings preloaded by springs, face-to-face and back-to-back.

Figure 3.3

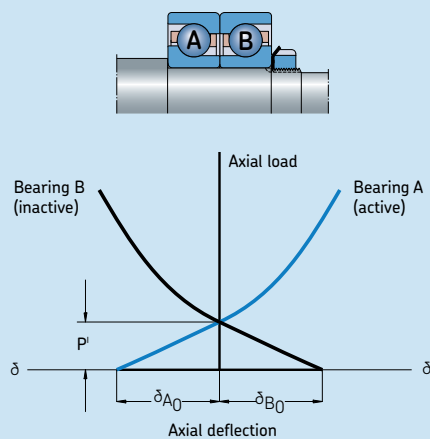
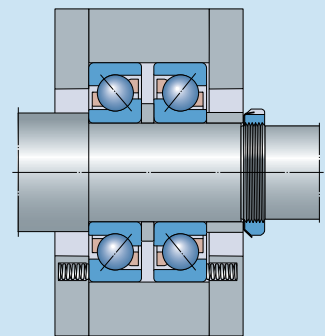


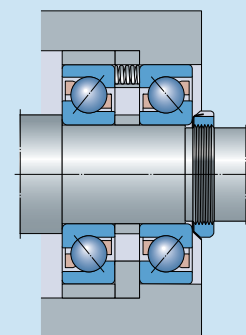
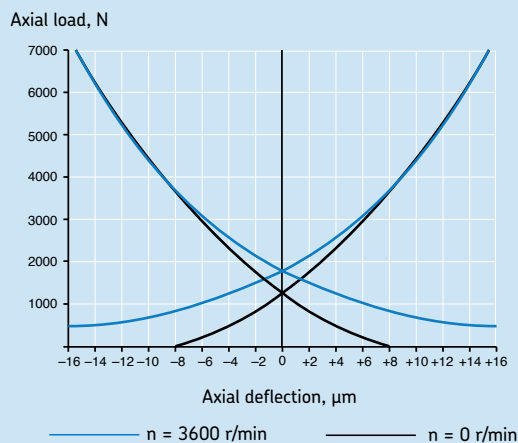
Figure 3.5



**Spring loaded
face-to-face arrangement**

Figure 3.4

Axial load versus axial deflection - (2) SKF 7310 BEGB nominal bearing condition / mean k5 shaft fit



**Spring loaded
back-to-back arrangement**

Universally matchable single row 40° angular contact ball bearings

Since a single angular contact ball bearing can only accommodate axial load in one direction, see **figure 3.6**, angular contact ball bearings are typically mounted as pairs either in a back-to-back or face-to-face arrangement.

Bearing pairs can support combined axial and radial loads and will assure accurate positioning of the pump shaft. A bearing pair will support an axial load equally in either axial direction. If the axial load of the pump is very heavy in one direction, the bearing pair can be arranged with a third bearing mounted in tandem.

To be mounted in paired arrangements, the bearings must be manufactured for universal mounting. The standard SKF bearings available for universal matching have the CB or GA suffix, e.g. 7310 BECBM or 7310 BEGAM. The CB suffix denotes that the bearing is universally matchable, and that a pair of these bearings will have a certain axial clearance when mounted in any of the three arrangements, shown in **figure 3.7**. The GA suffix also denotes that the bearing is universally matchable, but a pair of these bearings will have a light preload when mounted in any of the three arrangements shown in **figure 3.7**.

SKF universally matchable bearings are also available with smaller or greater clearances (suffixes CA and CC, respectively) and with moderate and heavy preloads (suffixes GB and GC, respectively).

Greater axial clearance (CC suffix) bearings may be necessary for operation at high temperatures, with heavy interference shaft and/or housing fits, and significant temperature differences between shaft and housing. Shaft material with a significantly greater coefficient of thermal expansion than carbon steel may also require CC clearance.

Preload (GA or GB suffix) may be necessary in bearings supporting predominantly axial load operating with light shaft and housing fits and at increased speeds (nd_m values approximately 250,000 and greater). Care must be taken with preloaded bearings to ensure correct shaft and housing fits and alignment, as preloaded bearings are more sensitive to fitting errors than clearance bearings.

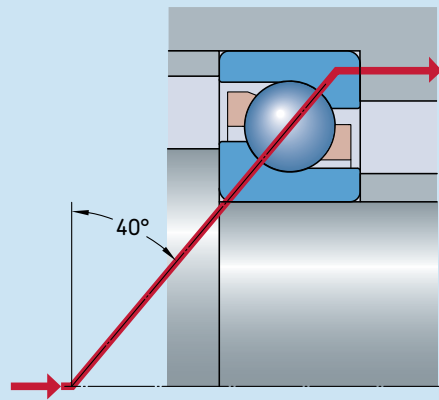


Figure 3.6

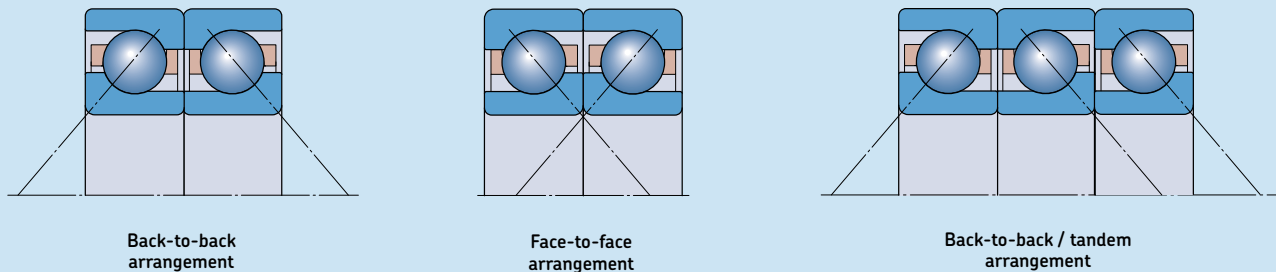


Figure 3.7

Figures 3.8 and 3.9 for the values of unmounted axial clearance and preload for pairs of bearings. The initial bearing clearance or preload is assured when the bearing rings are axially clamped together. The initial clearance in a bearing pair is reduced or initial preload is increased by interference fits and if the shaft and inner ring operate with a higher temperature than the outer ring and housing.

Caution: Single bearings are not to be used where only radial loads are present. For two directional thrust loads, use paired bearings.

The back-to-back arrangement is recommended for most pump applications given its ability to support tilting moments and since the clearance of the pair is controlled by clamping the inner rings, and no clamping of the outer rings is necessary.

The API Standard 610 specifies that the pump thrust loads shall be supported by two 40°, single row angular contact ball bearings, arranged back-to-back. The need for bearing axial clearance or preload is to be based on the requirement of the application.

For proper function, the outer rings of bearings arranged face-to-face must be securely clamped in the housing. The axial clamp force for bearings arranged face-to-face must be greater than the axial load supported by the bearings but less than the limiting clamp load, $C_0/4$. The axial clamp force can be applied to the outer rings either by the clamping of the

housing cover or by springs. Spring loaded arrangements (**figure 3.5**) better control the ring clamp load, and are less sensitive to mounting tolerances, misalignment, and changes in the bearing and pump temperatures.

The permissible misalignment of the shaft relative to the housing is generally limited to only a few minutes of angular misalignment before excessive stresses develop within the bearing. The misalignment limit depends on the:

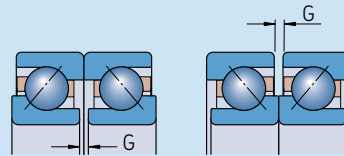
- operating internal clearance of the bearing
- bearing size
- internal design
- forces and moments acting on the bearing.

For bearings mounted in sets, particularly those with small axial internal clearance mounted in a back-to-back arrangement, misalignment can only be accommodated by increased ball loads, which will create cage stresses and reduce bearing service life. Any misalignment of the bearing rings will also lead to increased running noise.

The face-to-face arrangement is used when misalignment is unavoidable, such as in double suction pumps with slender shafts and housings bolted on the pump frame. The main advantage with the face-to-face arrangement is less sensitivity to misalignment, see **figure 3.10**.

Figure 3.9

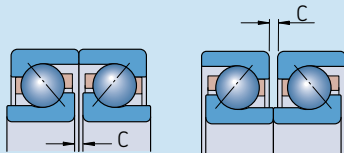
Preload of universally matchable single row angular contact ball bearings arranged back-to-back or face-to-face



Bore diameter d over	incl.	Preload Class GA		max	GB		min	max	GC		min	max
		min	max		min	max			min	max		
mm		μm		N	μm		N		μm		N	
10	18	+4	-4	80	-2	-10	30	330	-8	-16	230	660
18	30	+4	-4	120	-2	-10	40	480	-8	-16	340	970
30	50	+4	-4	160	-2	-10	60	630	-8	-16	450	1 280
50	80	+6	-6	380	-3	-15	140	1 500	-12	-24	1 080	3 050
80	120	+6	-6	410	-3	-15	150	1 600	-12	-24	1 150	3 250
120	180	+6	-6	540	-3	-15	200	2 150	-12	-24	1 500	4 300
180	250	+8	-8	940	-4	-20	330	3 700	-16	-32	2 650	7 500

Figure 3.8

Axial internal clearance of universally matchable single row angular contact ball bearings arranged back-to-back or face-to-face

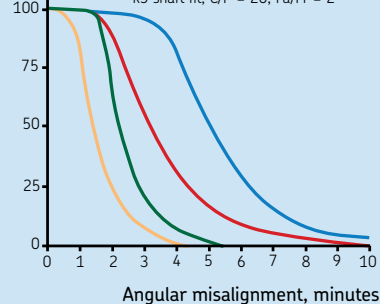


Bore diameter d over	incl.	Axial internal clearance Class CA		CB	max	CC	max
		min	max			min	
mm		μm		min	max	min	max
10	18	5	13	15	23	24	32
18	30	7	15	18	26	32	40
30	50	9	17	22	30	40	48
50	80	11	23	26	38	48	60
80	120	14	26	32	44	55	67
120	180	17	29	35	47	62	74
180	250	21	37	45	61	74	90

Figure 3.10

Relative life, %

k5 shaft fit; C/P = 20; Fa/Fr = 2



- 7310 BECBM, back-to-back arrangement
- 7310 BECBM, face-to-face arrangement
- 7310 BEGAM, back-to-back arrangement
- 7310 BEGAM, face-to-face arrangement

Cages

SKF single row angular contact ball bearings are produced with one of the following cages (**figure 3.11**):

- a machined brass window-type cage, ball centered, suffix M (**a**)
- a pressed brass window-type cage, ball centered, suffix Y (**b**)
- an injection moulded window-type cage of glass fibre reinforced polyamide 6,6, ball centered, suffix P (**c**)
- an injection moulded window-type cage of polyetheretherketone (PEEK), ball centred, designation suffix PH (**c**)

The selection of the proper cage for an application is generally dependent upon speed, temperature, and lubrication. **Figure 3.12** provides speed guidelines for selecting the appropriate cage.

For bearings with the polyamide cage (P suffix), the outer ring temperature should not exceed 100° C (212° F), otherwise premature aging will occur and lead to reduced bearing service life.

The machined brass cage (M suffix) provides the most reliable performance in harsh operating conditions and is therefore recommended for centrifugal pump applications operating in heavy duty service conditions such as the API 610 refinery service pump. The complete suffix designation for the standard universally matchable bearings with the machined brass cage is for example, BECBM or BEGAM, e.g. 7310 BECBM, 7310 BEGAM.

Consult with SKF Application Engineering for details of the recommendations or if the operating conditions are different from those listed in the table.

Figure 3.11

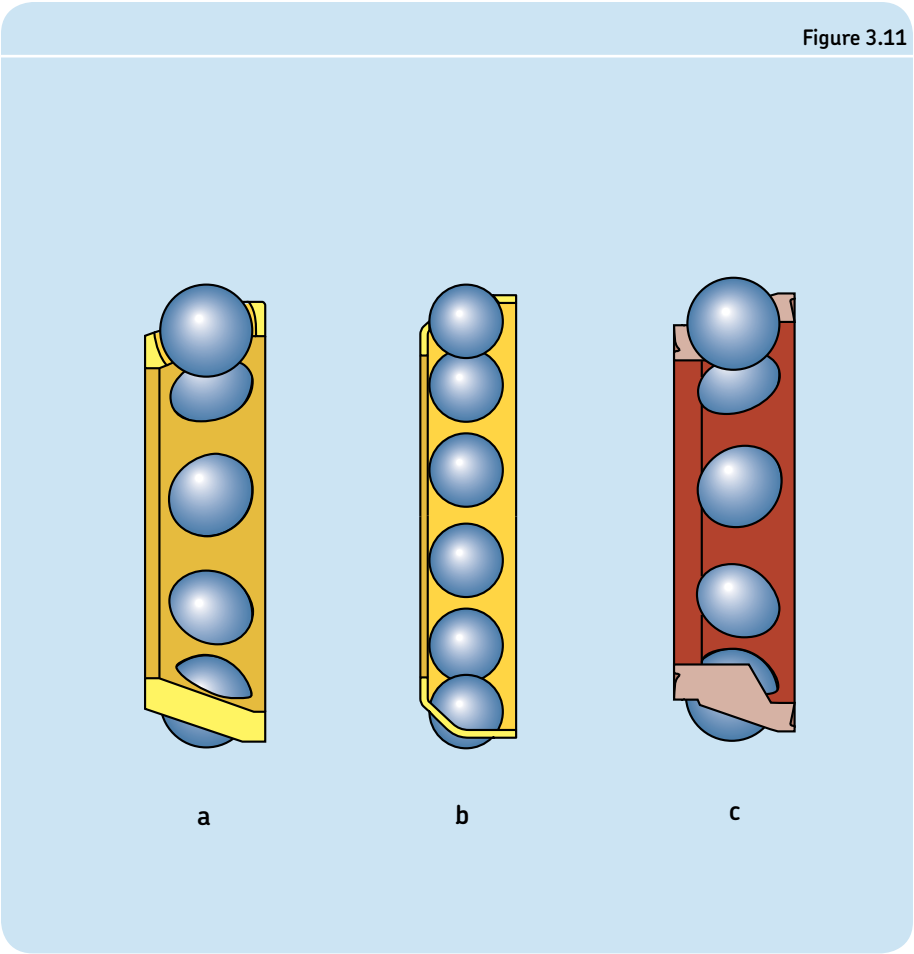


Figure 3.12

Recommended cage designs	
	Cage
ndm < 450000	M, P, Y
450000 < ndm < 650000	M
ndm > 650000	Contact SKF application engineering

MRC PumPac® bearings

MRC PumPac® bearings are a matched set of two angular contact ball bearings, one having a 40° contact angle and the other a 15° contact angle, **figure 3.13**. The bearings are produced as standard with machined brass inner ring land riding cages, and ISO class 6 (ANSI/ABMA Class ABEC 3) tolerances.

Figure 3.13

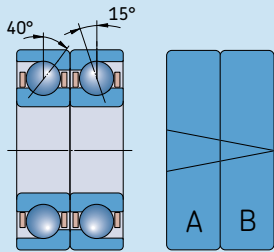


Figure 3.14

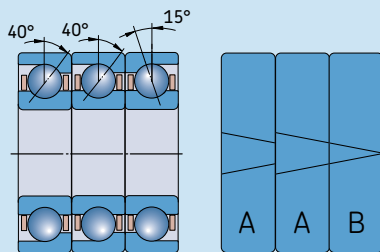
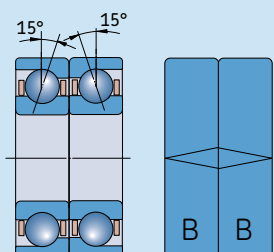


Figure 3.15



The PumPac bearing set is to be mounted in the pump with the 40° bearing supporting the applied axial load as the active bearing. The use of the 15° bearing provides several advantages:

- low sensitivity to gyratory motion and low requirement for axial load
- lower sensitivity to mounting conditions resulting in lower mounted preload
- greater initial preload deflection resulting in greater residual preload with applied axial loads
- higher radial stiffness

The benefit of the greater initial preload deflection is best illustrated in a load-deflection diagram as shown in **figure 3.16**. Greater axial load can be applied to a PumPac bearing set compared with two 40° bearings before the residual load on the inactive 15° bearing is reduced to zero. The PumPac bearing set can support greater axial load than two 40° bearings before unloading the 15° bearing. The result of these features is reduced bearing operating temperature, documented in some cases to be as great as 10° C (18° F).

The PumPac bearing set must be mounted in the pumps so that the 40° bearing supports the applied axial load as the active bearing. The outer rings of the two bearings are scribed together with a V arrow (see **figure 3.14**). This V arrow is to be oriented in the direction of the applied axial load.

Caution: The PumPac bearing set should not be used in pumps where the direction of axial load is unknown. Operation

with axial load in the direction of the 15° bearing can result in bearing failure. It is recommended that the PumPac bearing set be used in applications where the axial load is high in one direction and does not change direction during operation. The PumPac bearing set can accept momentary reversals in axial load, such as those that occur during pump start-up and stoppage etc.

When axial loads become excessive in the primary direction an additional 40° (A) bearing is added in tandem for increased axial capacity. The resulting arrangement is referred to as the PumPac triplex set designated MRC 8000 AAB (**figure 3.14**).

PumPac Diamond (<>)

The PumPac Diamond series consists of two 15° “B” bearings placed back-to-back, so that the etching on the bearing outside diameters forms a diamond (**figure 3.15**).

The PumPac Diamond is used in centrifugal pumps where the thrust loads are light and radial loads predominate such as double suction impeller pumps, multi-stage “between bearings” designs, and pumps with closed impellers. This bearing arrangement, as compared to a conventional 40° bearing arrangement, results in cooler running and substantially reduced vibration.

The inner ring land riding cage of the PumPac bearings requires special attention when grease lubricated. An initial charge of grease must be specifically injected between the cage and inner ring at assembly to ensure satisfactory lubrication.

Figure 3.16

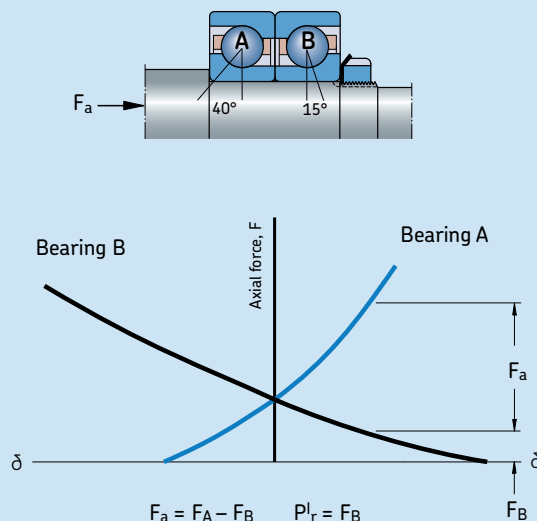
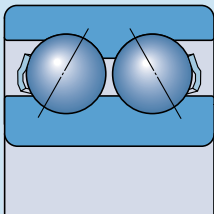


Figure 3.17

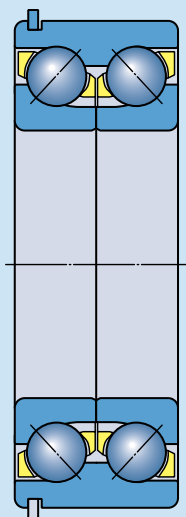


Double row angular contact ball bearings

SKF double row angular contact ball bearings correspond in design to two single row angular contact ball bearings but take up less axial space (**figure 3.17**). They can accommodate radial loads as well as axial loads acting in both directions. They provide stiff bearing arrangements and are able to accommodate tilting moments.

Double row angular contact ball bearings are used extensively in medium duty centrifugal pumps owing to good load and speed capabilities and ease of mounting.

Figure 3.18



SKF Explorer double row angular contact ball bearings – the high performance class

SKF Explorer double row angular contact ball bearings are manufactured with P6 tolerances. In addition to improved precision level other Explorer features include:

- optimized internal geometry and rolling contact surface
- upgraded ball quality
- improved materials
- new heat treatment
- pressed steel crowned cage

Such features provide benefits of superior performance, quiet running, increased speed capability, and longer service life.

SKF and MRC double row angular contact ball bearings have a 30° contact angle and the raceways are configured in a back-to-back arrangement. Basic design bearings are also available with seals (suffix 2RS1) and shields (2Z). Sealed bearings are lubricated for life and are maintenance free.

Double row angular contact ball bearings in the 32 A and 33 A series as well as the corresponding sealed bearings to (2Z and 2RS1 design) are identical to the MRC 52C and 53C series in the North American market. They have the same performance characteristics and dimensional features, however, MRC uses “C3” as its standard internal clearance without designating it in the suffix.

Cages

The basic design double row angular contact ball bearing is produced with a pressed steel ball centered crown cage as standard. Depending upon the size, an injection molded snap type glass fiber reinforced polyamide 6.6 cage (suffix TN9) is also available.

Bearings with a two-piece inner ring

In addition to the basic design, double row angular contact ball bearings are also available with a two-piece inner ring (**figure 3.18**), series SKF 3300 DNRCBM and MRC 5300 UPG.

These bearings have a 40° contact angle, machined brass cages, and a snap ring on the outer ring. These features provide high axial load carrying capacity and simple space-saving axial location in the housing. They have been designed specifically for operating conditions typical of centrifugal pumps.

Minimum Load

Important! Double row angular contact bearings must always be subjected to a given minimum radial load for satisfactory operation, particularly if they are to operate at high speeds or are subjected to high accelerations or rapid changes in the direction of load. Under such conditions, the inertia forces of the balls and cages, and the friction in the lubricant, can have a detrimental influence on the rolling conditions in the bearing arrangement and may cause damaging sliding movements to occur between the balls and raceways.

Figure 3.19

The minimum required radial load for double row bearings may be estimated from the following equation:

$$F_{rm} = k_r \left(\frac{v n}{1,000} \right)^{2/3} \left(\frac{d_m}{100} \right)^2$$

F_{rm} = minimum radial load, kN

k_r = minimum radial load factor

0.06 for bearings in the 32 A series

0.07 for bearings in the 33 A series

0.095 for bearings in the 33 DNR series

v = oil viscosity at operating temperature, mm²/s

n = rotational speed, r/min

d_m = bearing mean diameter =

0.5 (d + D), mm

Important! Double row bearings should ideally not be subjected to axial load without radial load.

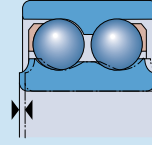
The recommendations of shaft and housing fits for double row angular contact ball bearings are the same as for other ball bearings in pump applications.

Internal clearance

SKF double row angular contact ball bearings in the 32 A and 33 A series are produced as standard with Normal axial internal clearance. They are also available with the greater C3 clearance.

The 33 DNRCBM series is produced exclusively with an axial internal clearance according to the values provided in **figure 3.19**. They are valid for bearings before mounting under zero measuring loads.

Axial internal clearance of double row angular contact ball bearings



Bore diameter d		Axial internal clearance of bearings in series				33 DNRCBM	
		32 A and 33 A Normal		C3			
over	incl.	min	max	min	max	min	max
mm		µm				µm	
–	10	5	21	12	28	–	–
10	18	6	23	13	31	–	–
18	24	7	25	16	34	–	–
24	30	8	27	18	37	–	–
30	40	9	29	21	40	10	30
40	50	11	33	23	44	10	30
50	65	13	36	26	48	18	38
65	80	15	40	30	54	18	38
80	100	18	46	35	63	–	–
100	110	22	53	42	73	–	–

Roller bearings in centrifugal pumps



Cylindrical roller bearing

Cylindrical roller bearings

SKF cylindrical roller bearings are used in centrifugal pumps for their high speed and high radial load capability. They are typically used as the non-locating radial bearing in end suction process pumps. Their separable component design simplifies mounting.

Of the large range of cylindrical roller bearing configurations typically the NU or NUP types are used (see **figure 4.1**). The NU type bearing is preferred because it can easily accommodate axial displacement inside the bearing due to thermal expansion of the shaft. This feature also enables tight shaft and housing fits, even for bearings in the non-locating position in pumps where impeller imbalance is unavoidable, an interference housing fit should be used to avoid bearing outer ring rotation. The NUP type is used where it is desired for the bearing to function as a single unit much like single row ball bearings and spherical roller bearings. When mounting the NUP bearing, the loose inner ring flange is abutted against the shoulder of the shaft.

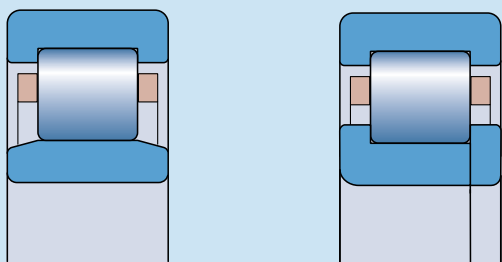
SKF Explorer cylindrical roller bearings – the high performance class

SKF Explorer cylindrical roller bearings are manufactured with P6 running accuracy and increased load carrying capacity. Such features provide benefits of superior performance, quiet running, increased speed capability, and longer service life. They are produced with three optional cages: the glass fiber reinforced polyamide 6,6 cage (suffix P shown in photo), the pressed steel cage (suffix J), and the machined brass cage (suffix M or MA).

The polyamide cage is standard in most sizes. This cage is used very successfully in cylindrical roller bearings operating in centrifugal pumps. In order for bearings with polyamide cages to obtain the longest service life, the outer ring temperature should not exceed 100° C (212° F).

Cylindrical roller bearings are somewhat sensitive to misalignment. The maximum allowable misalignment is three to four minutes, depending on the bearing series. For bearing housings machined in one setup, this is usually not a problem.

Figure 4.1



Minimum load

In order to provide satisfactory operation, single row cylindrical roller bearings, like all ball and roller bearings, must always be subjected to a given minimum load, particularly if they are to operate at high speeds or are subjected to high accelerations or rapid changes in the direction of load. Under such conditions, the inertia forces of the rollers and cage, and the friction in the lubricant, can have a detrimental influence on the rolling conditions in the bearing arrangement and may cause damaging sliding movements to occur between the rollers and raceways. The requisite minimum load to be applied to single row cylindrical roller bearings can be estimated using

$$F_{rm} = k_r \left(6 + \frac{4n}{n_r} \right) \left(\frac{d_m}{100} \right)^2$$

F_{rm} = minimum radial load, kN

k_r = minimum load factor*

n = rotational speed, r/min

n_r = reference speed, r/min*

d_m = bearing mean diameter
= 0.5 (d + D), mm

Taper roller bearings

SKF taper roller bearings are typically used in centrifugal pump applications having high combined radial and axial loads, such as slurry applications. A limiting factor is the speed capability of taper roller bearings. The speed rating is limited by the sliding friction between the rollers and the inner ring flange. Taper roller bearings can be used singly at a bearing position or in matched pairs and are suitable for both oil and grease lubrication.

Matched pairs of single row taper roller bearings with preset axial clearances are used when the load carrying capacity of one bearing is insufficient and accurate axial guidance of the shaft is necessary. SKF taper roller bearing of series 313 are well suited for pump applications having high axial loads because of their steep contact angle. The bearings are often arranged face-to-face (suffix DF) or back-to-back (suffix DB). The outer rings of the two bearings arranged face-to-face must be axially clamped in the housing to ensure correct operation. The axial clamp force must be greater than the applied axial load but less than the limiting clamp load, $C_0/4$, where C_0 is the static load rating of one bearing.

If the clamp force is by necessity high, an outer ring spacer with greater stiffness may be needed for bearings arranged face-to-face to limit the deflection due to this clamping.

Because of the steep contact angle in 313 series bearings, it is important to have free lubricant flow to each face of the bearing. The housing should have a bypass opening beneath the bearing to allow free lubricant flow into each face of the bearing and between the bearings through the holes provided in the outer ring spacer. Higher bearing temperatures will result if this is not possible.

Important! For satisfactory performance, a taper roller bearing should be subjected to a given minimum radial load. The requisite minimum radial load to be applied to SKF standard taper roller bearings can be estimated from

$$F_{rm} = 0.02 C$$

and for SKF Explorer bearings from

$$F_{rm} = 0.017 C$$

where

F_{rm} = minimum radial load, kN

C = basic dynamic load rating, kN*



* See product tables in SKF General Catalog 6000EN.

Radial spherical roller bearings

SKF spherical roller bearings are used in centrifugal pump applications having heavy loads and generally low operating speeds. These bearings are used for their high radial and axial load capabilities and their ability to operate in misaligned conditions. Increased speeds can be accommodated provided the lubrication and cooling are satisfactory. Spherical roller bearings are produced as standard with steel cages.

Because of their special internal design, SKF spherical roller bearings are able to accommodate heavy radial loads and heavy axial loads acting in both directions. The axial load capability is sometimes limited by the axial-to-radial load ratio, bearing friction, and resulting operating temperature.

SKF Explorer spherical roller bearings – the high performance class

SKF Explorer spherical roller bearings are manufactured to P5 running accuracy for bearings up to and including 300mm bore. In addition to improved precision level Explorer features include:

- optimized internal geometry and rolling contact surface
- improved materials
- reduced friction

These features provide superior performance, quiet running, increased speed capability, and longer service life.

SKF spherical roller bearings are available with cylindrical bore or tapered bore for adapter sleeve mounting. The bearings can be oil or grease lubricated. To facilitate efficient bearing lubrication, SKF spherical roller bearings are provided with an annular groove and three lubrication holes in the outer ring.

Important! For satisfactory operation, spherical roller bearings must be subjected to a certain minimum load. The minimum load is necessary to maintain the motion of the rollers and can be estimated from the following equation.

$$P_m = 0.01C_0$$

where

P_m = equivalent minimum load, kN

C_0 = basic static load rating, kN*



Spherical roller bearing

In some applications it is not possible to reach or exceed the requisite minimum load. However, if the bearing is oil lubricated lower minimum loads are permissible. These loads can be calculated when $n/n_r \leq 0.3$ from

$$P_m = 0.003C_0$$

and when $0.3 < n/n_r \leq 2$ from

$$P_m = 0.003 C_0 \left(1 + 2 \sqrt{\frac{n}{n_r} - 0.3} \right)$$

where

P_m = equivalent minimum load, kN

C_0 = basic static load rating, kN*

n = rotational speed, r/min

n_r = reference speed, r/min*

In vertical pump applications, the loading on spherical roller bearings can in some instances be relatively light, and rotating due to imbalances in the machine. This can cause rotation of the outer ring in the housing. Rotation of the outer ring can be prevented or minimized by pinning the outer ring with the housing or by using an O-ring mounted in a groove in the housing bore.

* See product tables in SKF General Catalog 6000EN.



Spherical roller thrust bearing

Spherical roller thrust bearings

SKF spherical roller thrust bearings are used in centrifugal pumps to support very high axial loads. Their internal design allows them to accommodate radial loads in addition to simultaneously acting axial loads.

SKF Explorer spherical roller thrust bearings – the high performance class

SKF Explorer spherical roller thrust bearings are manufactured to closer tolerances than normal. Explorer features include:

- optimized internal geometry and rolling contact surface
- improved materials
- reduced friction
- improved wear resistance
- improved surface texture

These features provide longer service life by optimizing the stress distribution, increasing the bearing toughness and optimizing the oil film between the contact surfaces.

Important! For satisfactory performance, spherical roller thrust bearings must be subjected to a given minimum axial load. The required minimum axial load to be applied can be estimated by the following equation

$$F_{am} = 1.8 F_r + A \left(\frac{n}{1,000} \right)^2$$

where

F_{am} = minimum axial load, kN

F_r = radial component of the load for bearings subjected to combined load, kN

C_0 = basic static load rating, kN*

A = minimum load factor*

n = rotational speed, r/min

If $1.8 F_r < 0.0005 C_0$ then $0.0005 C_0$ should be used in the above equation instead of $1.8 F_r$.

The axial load on the bearing must always be greater than that estimated by the above equation. If necessary, additional axial load must be applied to the bearing to satisfy the requirement for load. Compression springs are often used for this purpose.

* See product tables in SKF General Catalog 6000EN.

Bearing arrangement with a single spherical roller thrust bearing, radially guided on a vertical shaft

When a spherical roller thrust bearing is axially loaded with at least the minimum requisite bearing load it can be used as a single bearing to accommodate both radial and the axial loads (**figure 4.2**). The bearing at the other end of the shaft, however, should be a radial bearing.

This arrangement is suitable when the axial load always acts in one direction. Correctly applied, the bearing will then work smoothly as long as $F_r \leq 0.55 F_a$. If the bearing must accommodate a heavy radial load, $F_r > 0.55 F_a$, the bearing should be combined with another bearing.

Bearing arrangement with a single spherical roller thrust bearing, radially free on a horizontal or vertical shaft

In applications where a spherical roller thrust bearing is mounted radially free and axial loads may not meet requisite minimums, springs must be used to preload the bearing. In this example, the spherical roller thrust bearing is spring preloaded and carries the predominant axial load. The shaft is supported by two radial bearings of which one is locating in the opposite direction (**figure 4.3**). This arrangement is suitable when the axial load in one direction is predominant.

Lubrication

Generally, lubrication with oil or grease containing EP additives is recommended for spherical roller thrust bearings. When lubricating with grease the roller end/flange contacts must be supplied with an adequate amount of lubricant.

Because of their internal design, spherical roller thrust bearings have a pumping action that can be taken advantage of, in order to provide circulation of the lubricating oil. The pumping action must be considered when selecting lubricants and seals. Shown below are examples when:

- the shaft is vertical (**figure 4.4**) or
- the shaft is horizontal (**figure 4.5**).

Figure 4.2

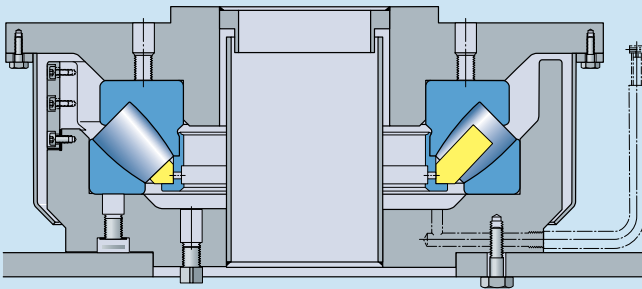


Figure 4.4

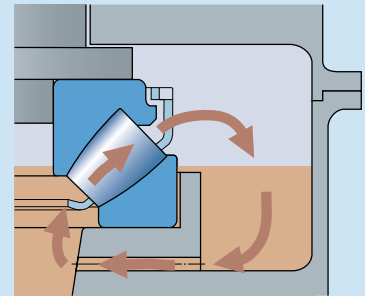


Figure 4.3

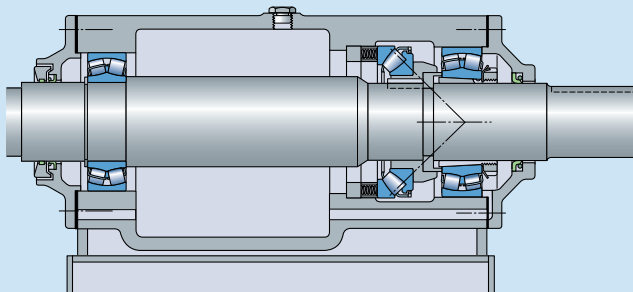
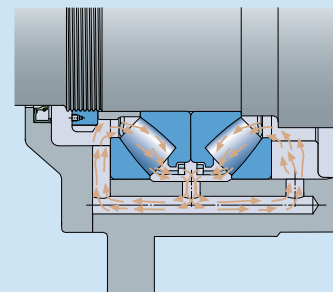


Figure 4.5



Bearing technologies for the next generation pump



Hybrid bearing

Hybrid bearings

Hybrid bearings have rings of bearing steel and rolling elements of bearing grade silicon nitride (Si_3N_4). In addition to being excellent electric insulators, hybrid bearings have a higher speed capability and will provide longer service life than all-steel bearings in centrifugal pump applications.

The density of silicon nitride is only 40% of the density of bearing steel. Thus the rolling elements weigh less and have lower inertia, resulting in less cage stresses during rapid starts and stops and also significantly lower friction at high speeds. Lower friction means cooler running and longer lubricant service life.

Under insufficient lubrication conditions there is no smearing between silicon nitride and steel. This enables hybrid bearings to last much longer in applications operating under severe dynamic conditions or lubrication conditions with low operating viscosity ($\kappa < 1$). For hybrid bearings it is common to apply $\kappa = 1$ for running conditions with $\kappa < 1$ to estimate life under such conditions. Hybrid bearings may perform well



Hybrid bearing

when lubricated with ultra thin film forming media, such as refrigerants, enabling process lubrication or oil-free designs but care needs to be taken in design and material selection. In such cases it is recommended to consult SKF application engineering.

Silicon nitride has a higher hardness and higher modulus of elasticity than steel, resulting in increased bearing stiffness and longer bearing service life in contaminated environments.

Silicon nitride rolling elements have a lower thermal expansion than steel rolling elements. This means more accurate pre-load control and less likelihood of excessive preloading when temperature gradients exist within the bearing.

SKF hybrid bearing rings are made as standard of the same steel as the equivalent all-steel bearing. The standard stabilization temperatures apply.

On request, hybrid bearings can be manufactured with through-hardened rings of stainless bearing steels with good corrosion, wear and high temperature properties. Such bearings can operate at temperatures up to 300° C.

NoWear® bearings

New and advanced applications are placing higher demands on bearings, and increasing the risk of smearing, boundary lubrication, sudden load variations, low loads or high operational temperatures. To withstand these types of severe operating conditions SKF bearings can be furnished with a low friction ceramic coating on the contact surfaces inside the bearing. This coating, trademarked NoWear, was developed by SKF for rolling bearings and is covered by an SKF patent.

The coating is applied using a physical vapor deposition process giving the bearing surfaces the resilience of the underlying material, but with the hardness, low friction coefficient and wear resistance of the NoWear coating.

NoWear bearings can withstand longer periods of insufficient lubrication, sudden variations in load and rapid speed changes, vibrations and oscillations. This creates new possibilities for existing applications operating under severe conditions, without introducing major design changes.



NoWear bearings

The essential properties of the NoWear coating are listed in the table below.

Table 5.1

Properties of NoWear coating	
Properties	NoWear
Hardness	1,200 HV10
Coating thickness - depending on bearing size (µm)	1 ... 3
Coefficient of friction - dry sliding against steel	0.1 ... 0.2
Operating temperature range¹⁾ - NoWear coating	+350° C

¹⁾NoWear coating withstands temperatures up to +350. However, most times the bearing steel is the limiting factor. Please contact SKF application engineering for more information.

Energy efficient bearings

SKF has developed a new family of bearings that reduce energy consumption by at least 30% compared to standard ISO products. This new bearing family includes two bearing types commonly used in centrifugal pumps: deep groove ball bearings and tapered roller bearings. The internal modifications to the bearings substantially reduce their energy consumption while maintaining the service life and load carrying capacity of standard ISO bearings. The technical improvements that deliver the energy savings for the taper roller bearing include modified surface topography, raceway profiles and geometry, together with a special polymer cage and an optimized set of rollers. The optimized roller set means a lighter bearing which itself is more energy efficient because it takes less power to move the rolling elements. It also lowers inertia of the moving parts thus reducing the chance of skidding and smearing, which would otherwise affect performance and service life.

The technical improvements that deliver the energy savings for the deep groove ball bearing include optimized internal geometry, a new polymer cage and lower friction grease for sealed or shielded variants.

Magnetic bearings

In traditional rolling element bearings, the loads being supported by the bearing are transferred through the rolling elements between the inner and outer races as the bearing rotates. Magnetic bearing systems represent a completely different approach to the support of the rotating element of centrifugal pumps.

Magnetic bearings provide attractive electromagnetic suspension by application of electric current to ferromagnetic materials used in both the stationary and rotating parts (the stator and rotor, respectively) of the magnetic bearing. This creates a flux path that includes both parts, and the air gap separating them, through which non-contact operation is made possible. Non-contacting operation results in negligible friction loss and wear, and therefore the elimination of traditional lubrication, increased reliability and the ability to achieve

previously unattainable rotational speeds. Since the natural tendency of the stator is to attract the rotor until it makes contact, a control system is required to modulate the magnetic field and maintain the rotor in the desired position. The most common type of control involves the feedback of shaft position. This information is then used by the control system to modulate the magnetic field through power amplifiers, so that the desired rotor position is maintained even under changing shaft load conditions.

An active magnetic bearing system consists of electromagnet bearing actuators, position sensors, a control system and power amplifiers. The bearing actuators and sensors are located in the machine, while the control system and amplifiers are generally located remotely.

More information on SKF Magnetic Bearings can be found at www.skf.com.



Bearing installation

Bearing mounting

General Information

To provide proper bearing performance and prevent premature failure, skill and cleanliness when mounting ball and roller bearings are necessary. As precision components, rolling bearings should be handled carefully when mounting. It is also important to choose the correct method of mounting and to use the correct tools for the job. A variety of factors such as incorrect mounting techniques or methods, dirty hands or tools, contaminated grease or oil can cause bearing damage. See the SKF Bearing Maintenance Tools Catalog or www.skf.com/mount or www.mapro.skf.com.

When replacing bearings, document the orientation of the old bearings in the pump before they are removed. Compare the part number of the bearings removed from service with what is specified for the application.

Preparations before mounting

A clean working surface, correct mounting methods and appropriate tools are essential elements of a successful bearing installation. The mounting environment needs to be absolutely clean and free from any contaminants or corrosive fluids that might damage the bearing.

Measure the shaft and housing

Prior to an installation, always check to ensure that the shaft and housing seating dimensions and form accuracy correspond to the manufacturer's specifications.

The cylindricity can be checked by measuring the diameter in two cross sections and in four planes by using outside and inside micrometers as shown in **figure 6.1**. The shaft and housing seatings need to be checked for straightness and abutments for perpendicularity. Straight edges and dial gauges can be used for this. Whenever there is reason to suspect that the radial and axial runouts are not appropriate, they should be checked as well. Check the assembly drawings for specifications. Record the measurements for future reference.

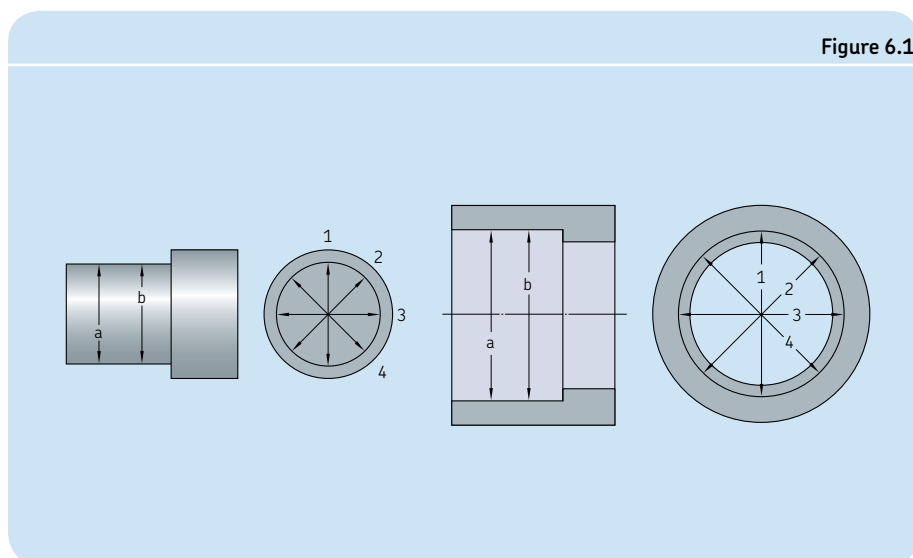


Figure 6.1

Handling bearings

New SKF bearings are well protected in their package. Do not remove them from the package until immediately before mounting. If the mounting process is interrupted, the bearing should be covered with wax paper or foil.

Mounting

Nearly all rolling bearings in pump applications require the use of an interference fit on at least one of the bearing rings, usually the inner. Consequently, all mounting methods are based on obtaining the necessary interference without undue effort and with no risk of damage to the bearing. Depending on the bearing type and size, mechanical or thermal methods are used for mounting. In all cases it is important that the bearing rings, cages and rolling elements or seals never receive direct blows and that the mounting force is never directed through the rolling elements.

Two basic mounting methods are used depending on the number of mountings, bearing type and size, magnitude of the interferences and available tools.

Cold/mechanical mounting

Cold mounting is suitable for cylindrical bore bearings with an outside diameter up to 100 mm. Gently position the bearing so that it lines up with the shaft. Position the mounting tool and apply the mounting force to the bearing ring with an interference fit (see **figure 6.2**). Applying the mounting force to the loosely fitted ring only will transfer the mounting force through the rolling elements and brinell the rolling surfaces. In the case of bearings with an OD > 100 mm, hot mounting is recommended.

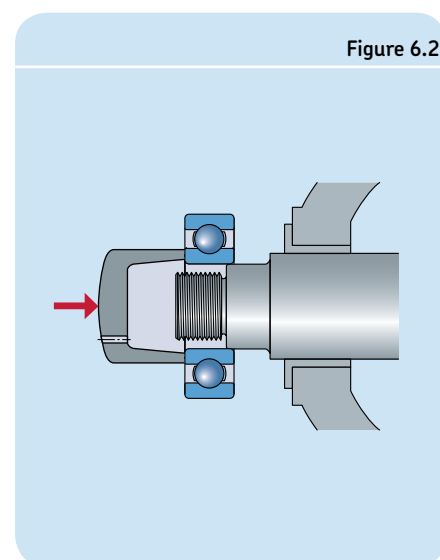


Figure 6.2

Hot mounting

The required force needed to mount a bearing increases rapidly with bearing size. Larger bearings (OD > 100 mm) cannot be pressed easily onto a shaft because of the mounting force required. To overcome the interference fit between the bearing and shaft, the bearing should be heated. Normally a bearing temperature of 65° C (150° F) higher than that of the shaft is sufficient for mounting. Never heat a bearing to a temperature greater than 120° C (250° F). Overheating can destroy a bearing's metallurgical properties, softening the bearing and potentially changing its dimensions permanently. Standard ball bearings fitted with shields or seals should not be heated above 100° C (210° F) because of their grease fill or seal material.

An induction heater equipped with adjustable thermostats and automatic demagnetization is recommended. An open flame is never acceptable for heating a bearing.



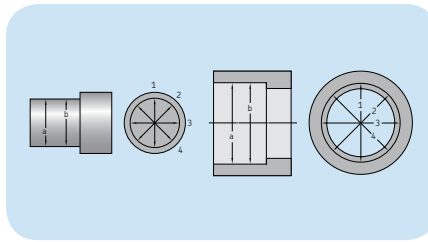
Induction heater

Mounting procedures

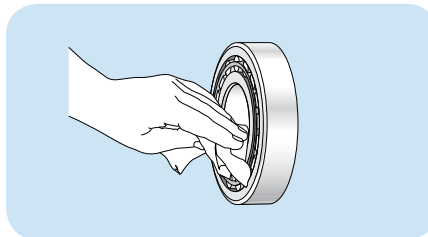
Step by step mounting procedures for ball bearings are provided below. For other bearing types consult www.skf.com/mount.

Step 1: Make sure the mounting environment, shaft, housing, and other components are clean. The bearings should be left in their original packages until immediately before mounting so that they do not become dirty.

Step 2: Measure the shaft and housing for size, roundness, and profile.



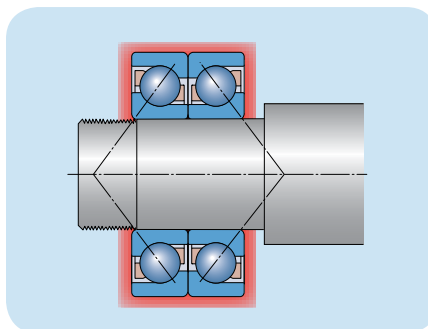
Step 3: Unwrap the bearing and wipe the preservative from the bore and O.D. surfaces. Visually inspect the bearing surfaces.



Step 4: Orient the bearings per the pump manufacturer's specification.

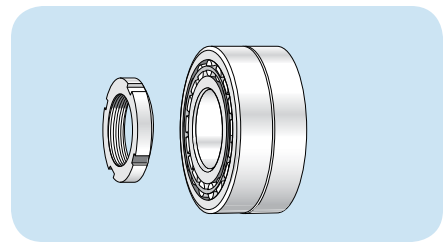
Step 5: Heat the bearings using an induction heater, making sure the temperature probe is on the inner ring. Paired single row angular contact ball bearing should be heated together.

Step 6: Remove the bearings from the induction heater and place onto the shaft. Paired single row angular contact ball bearing should be installed together.



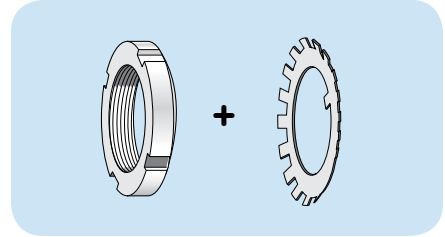
Step 7: Apply the locknut, without lock washer, and hand tighten with a spanner wrench to a snug position making sure that the inner ring is firmly seated against the shaft abutment. For paired single row angular contact ball bearings ensure the inner rings are seated against each other.

Note: The lock washer is not used in this step due to the risk of shearing the locking tab during the tightening process.

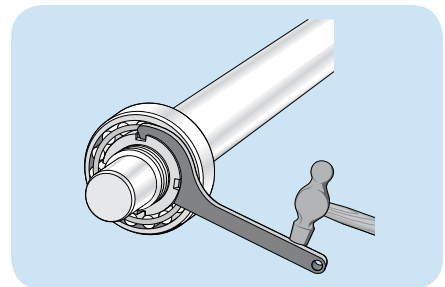


Step 8: Allow the bearings and shaft to cool down to room temperature.

Step 9: Remove the lock nut, apply the lock washer and re-apply the lock nut. Applying a light coating of oil to the chamfered face of the lock nut will make mounting easier.



Step 10: Using a spanner wrench and hammer, tighten the lock nut an additional 1/8 to 1/4 turn beyond the snug position to remove gaps that may have developed from cooling. Failure to remove gaps between paired bearings during installation will add unwanted endplay to the shaft assembly.



Step 11: Find the locking washer tang that is nearest a lock nut slot. Do not loosen the lock nut to align the tang with a slot but instead continue tightening the lock nut to align the next closest tang. Bend the tang into the slot.

Step 12: If the assembly is to remain exposed after the bearings have been mounted, then the bearings should be covered to avoid having debris come in contact with them.



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