Bearings in Centrifugal Pumps - Part I

Application Handbook

Summary

This SKF handbook provides a detailed and in-depth look at bearings as used in centrifugal pumps. **Part I** includes: general centrifugal pump operation principles, pump bearing details including bearing life, bearing lubrication, etc. **Part II** includes: application of ball and roller bearings in detail, comparative viscosity classifications, unit conversions, and references. The handbook is a reprint of the SKF handbook of Bearings in Centrifugal Pumps, publication number 100-955, 1998, 42p.

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Table of Contents

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 fig. 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 fig. 1.2. Radial and mixed flow pumps are either single or double suction designs.

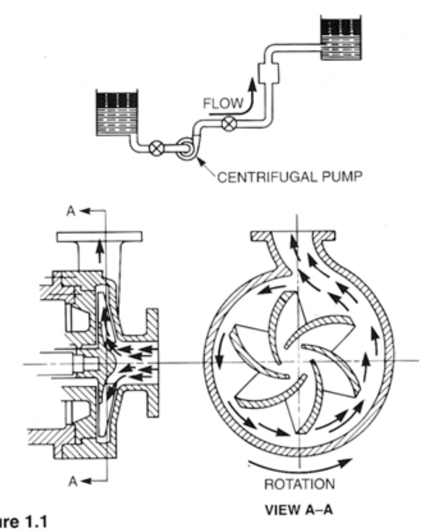

Figure 1.1

Classes of centrifugal pumps

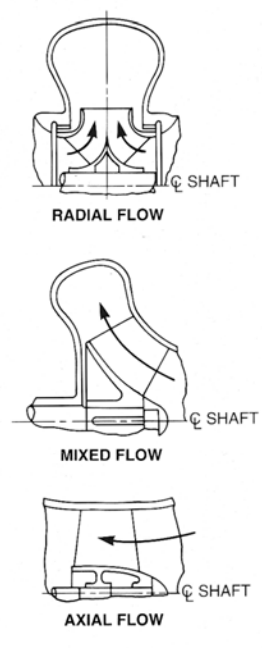

Figure 1.2

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General

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

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 fig. 1.4. The specific speed number of a pump is calculated by the following equation:

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

where

n_s = specific speed
n = pump rotational speed, r/min.
Q = pump flow rate, m^3/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 follows.

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 fig. 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 petro-chemical applications, the pump industry has developed standards for the manufacture and supply of centrifugal pumps. Two important standards are the ASME/ANSI B73.1 [1] for chemical process pumps and API 610 [2] 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 petro-chemical 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.

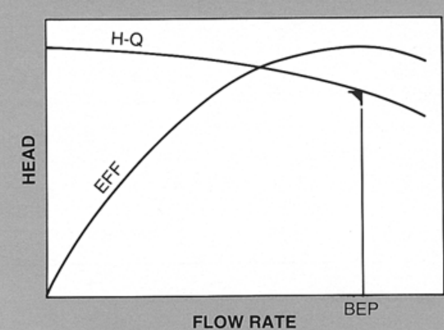

Figure 1.3

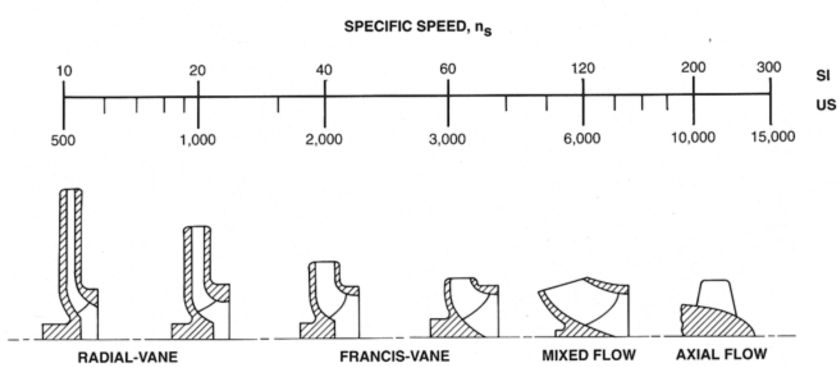

Figure 1.4

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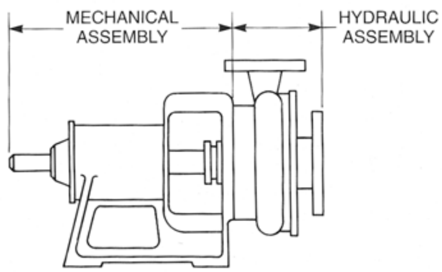

Hydraulic Assembly

Impeller/propeller
Suction inlet
Volute
Seal rings

Mechanical Assembly

Shaft seal
- Labyrinth
- Mechanical
- Packing

Shaft
Bearings
Housing/frame
Drive coupling/sheave

Figure 1.5

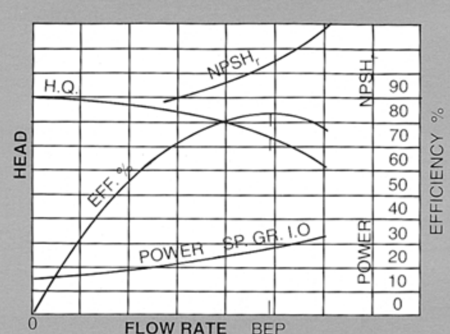

Figure 1.6

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 fig. 1.6. The curve shows the head, efficiency, power requirements, and net Positive Suction Head—required (NPSHr) of the pump versus the flow.

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 NPSHr 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 (NPSHa), is less than the pump's NPSHr, 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 NPSHa, 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 re-enter 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 1500 and 3000 r/min at 50 Hz frequency and 1800 and 3600 r/min at 60 Hz frequency. Other rotational speeds are possible with belt and gear driven pumps etc.

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General

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 fig. 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

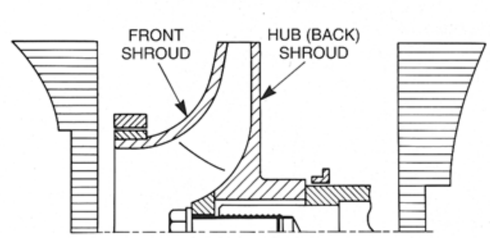

Figure 1.7

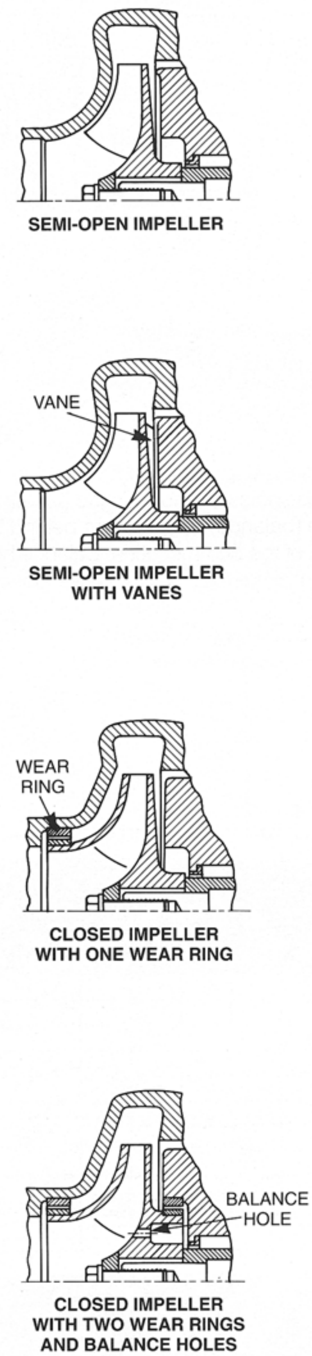

Figure 1.8

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are illustrated in fig. 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.

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 fig. 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.

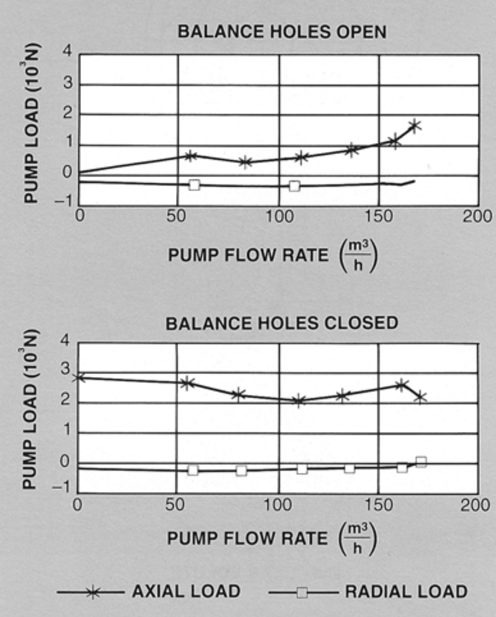

Figure 1.9

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 fig. 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 fig. 1.11.

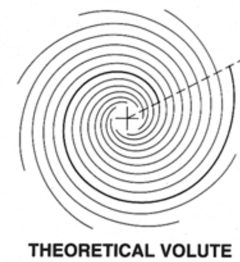

Figure 1.10

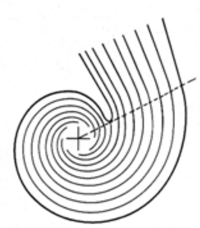

Figure 1.11

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General

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 fig. 1.12.

Four typical casings are illustrated in fig. 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.

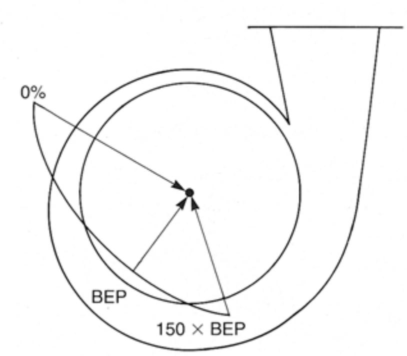

Figure 1.12

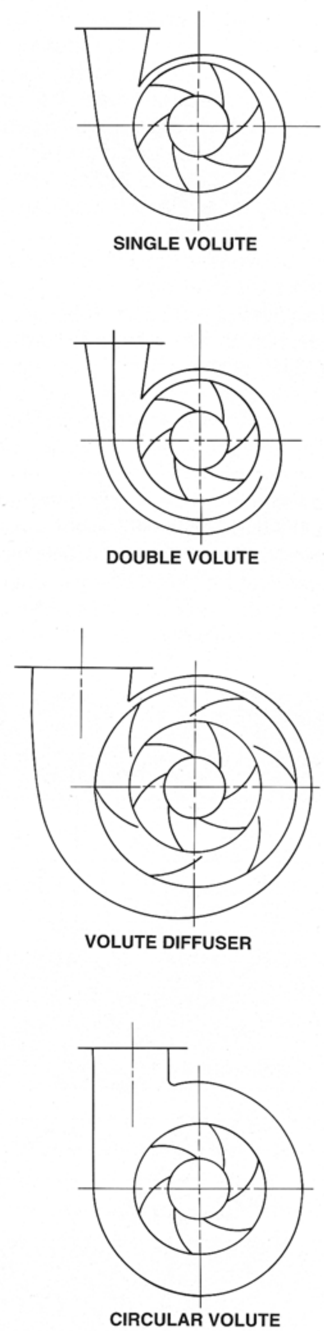

Figure 1.13

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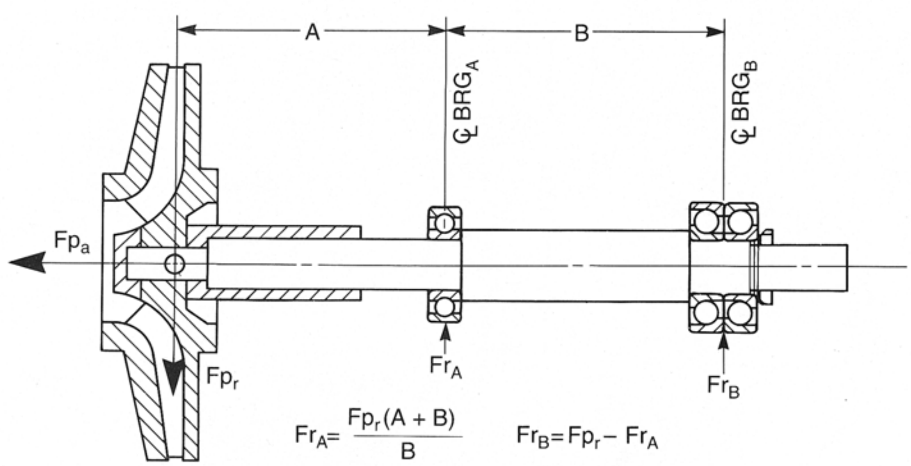
The hydraulic loads are dependent on the type and size of impeller and casing, the pump operating conditions such as fluid suction pressure, and the point of pump operation. The magnitude and direction of the hydraulic loads can change greatly with changes in these factors. In most instances, the lowest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



$$Fr_A = \frac{Fp_r(A+B)}{B} \qquad Fr_B = Fp_r - Fr_A$$

$$P_A = Fr_A \qquad P_B = X\,Fr_B + Y\,Fp_a$$

Figure 1.14

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Pump bearings

Bearing types used in centrifugal pumps

Fig. 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.

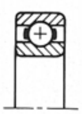
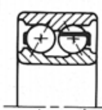
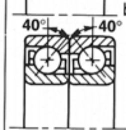
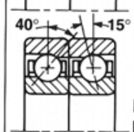
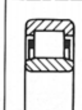
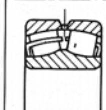
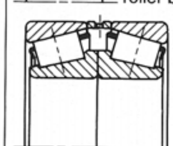
The SKF single row deep groove ball bearings and double row angular contact ball bearings are produced in Conrad (i.e. without filling slots) and filling slot type designs. For pump applications, Conrad bearings are preferred over the filling slot type bearing. Conrad double row angular contact ball bearings operate at lower temperatures than filling slot double row bearings in similar pump conditions and are not influenced by the filling slot. Double row angular contact ball bearings with filling slots should 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. MRC Bearings* has combined a 40° contact angle ball bearing with a 15° contact angle ball bearing to introduce a bearing set call 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.

*MRC and PumPac are registered trademarks of SKF USA Inc.

Approximate relative load, speed and misalignment capabilities

	Radial load	Axial load	Speed	Mis-alignment
 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	—	XXXX	X
 Spherical roller bearing	XXXX	XX	XX	XXXX
 Taper roller bearing set	XXXX	XXXX	XX	X

—: No Capacity
X: Low
XX: Moderate
XXX: High
XXXX: Very High

Figure 2.1

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Pump bearing arrangements

The most common pump and pump bearing arrangements are shown in figs. 2.2a to 2.2h.

The vertical inline pump, fig. 2.2a, and the horizontal process pump, fig. 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 fig. 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.

The medium duty, fig. 2.2c, and heavy duty, fig. 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 fig. 2.2e and 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.

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 preloaded to ensure that sufficient load is applied to the bearing during conditions when the axial load reverses at pump start-up or stoppage etc. This arrangement is most commonly oil-bath lubricated.

For vertical deep well pumps, the spherical roller thrust bearing is a good choice, fig. 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, fig. 2.2h. The impeller and its shaft are supported in 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.

Vertical inline pump

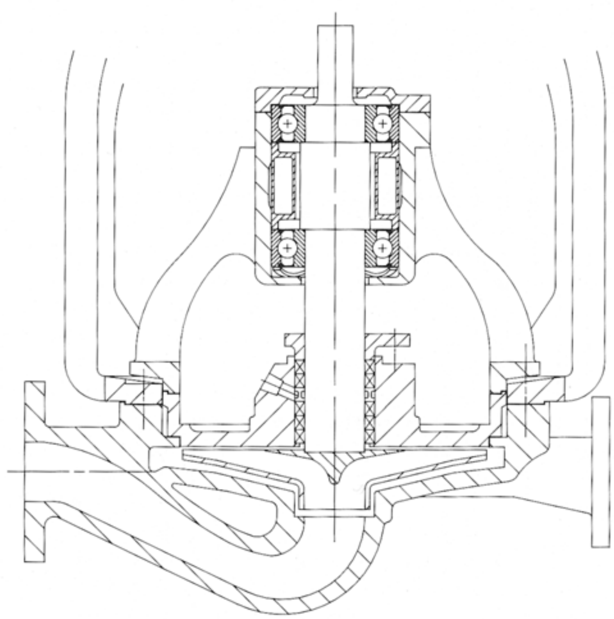

Figure 2.2a

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Process pump

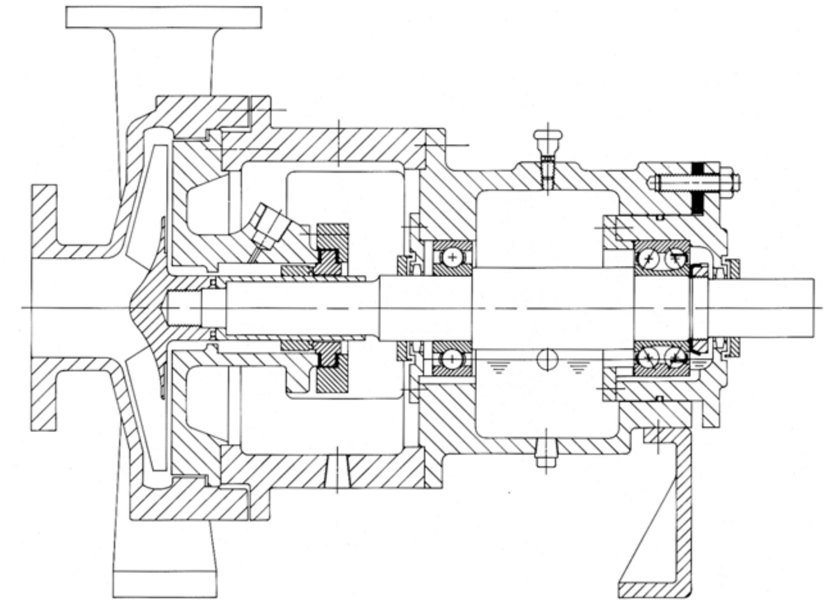

Figure 2.2b

Medium duty process pump

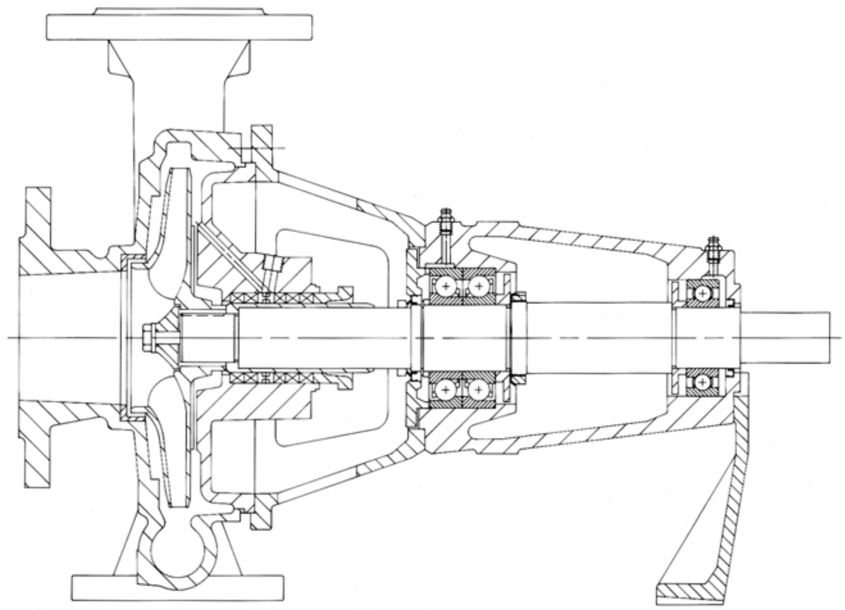

Figure 2.2c

Pump bearings

Heavy duty process pump

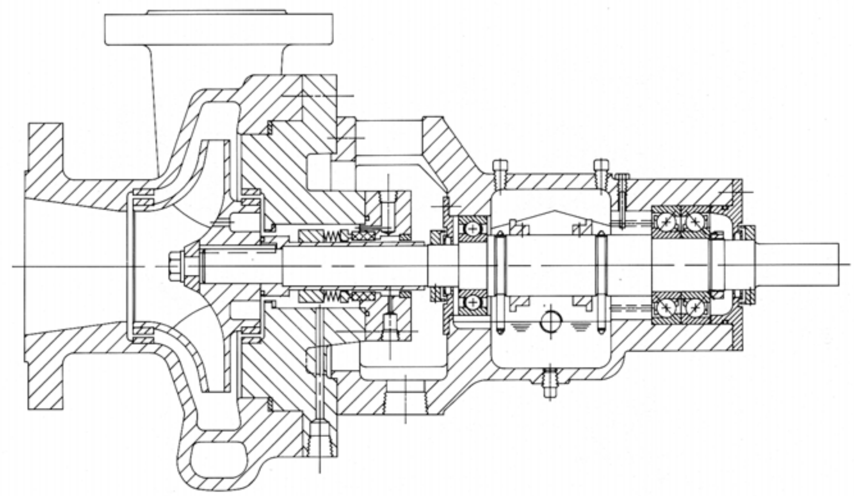

Figure 2.2d

Slurry pump

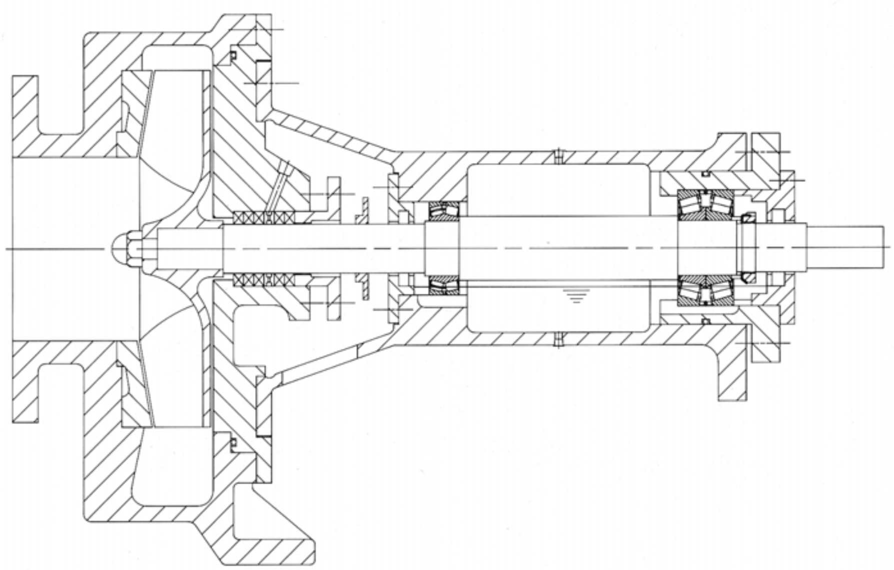

Figure 2.2e

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Slurry pump

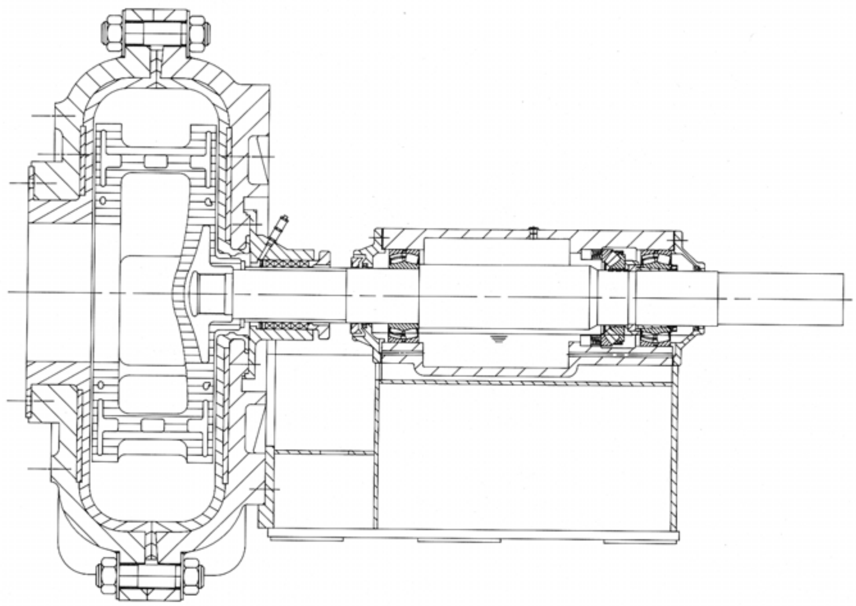

Figure 2.2f

Vertical deep well pump

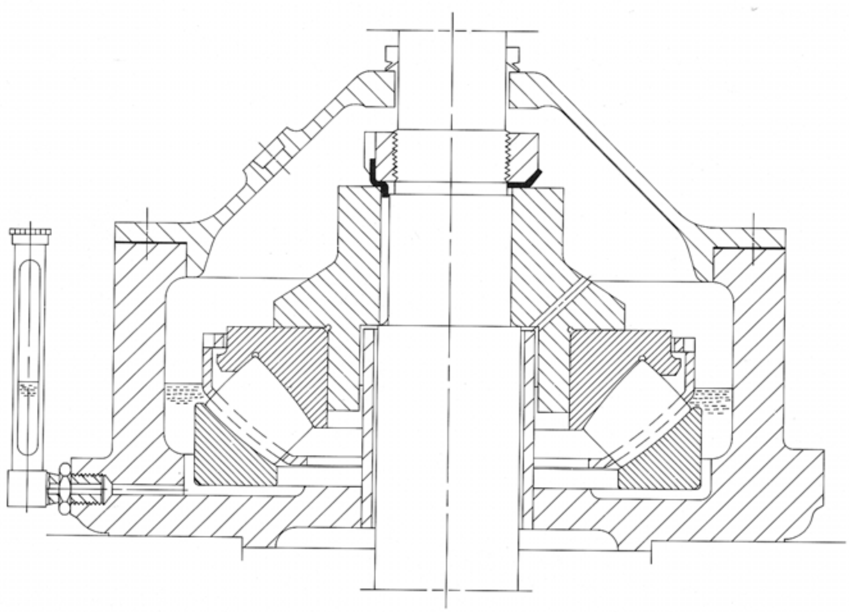

Figure 2.2g

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Pump bearings

Magnetic drive pump

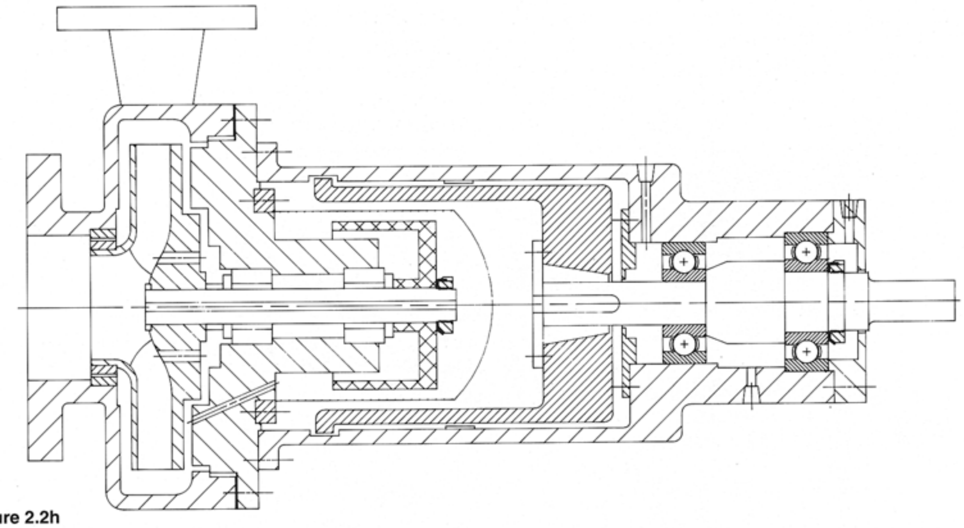


Figure 2.2h

Bearing life

When calculating the rating life of SKF bearings, it is recommended that the basic rating life L_{10h}, the adjusted rating life L_{10ah} and the New Life Theory rating life L_{10aah} each be evaluated, provided sufficient information is available to satisfactorily evaluate the adjusted rating life and the New Life Theory.

The equations to use when calculating the bearing rating life are as follows:

$$L_{10h} = \left(\frac{C}{P}\right)^p \frac{1000000}{60\,n} \qquad \text{EQ. 2}$$

$$L_{10ah} = a_{23}\,L_{10h} \qquad \text{EQ. 3}$$

$$L_{10aah} = a_{SKF}\,L_{10h} \qquad \text{EQ. 4}$$

where

L_{10h} = basic rating life in operating hours
n = rotational speed, r/min
C = basic dynamic load rating, N
P = equivalent dynamic bearing load, N
p = exponent for the life equation
  $p = 3$ for ball bearings
  $p = 10/3$ for roller bearings
h = hours
L_{10ah} = adjusted rating life in operating hours
a_{23} = combined factor for material and lubrication, see diagram in General Catalog
L_{10aah} = adjusted rating life according to New Life Theory in operating hours
a_{SKF} = life adjustment factor based on New Life Theory

The ASME/ANSI B73.1 Standard for process pumps specifies that rolling bearings shall have rating lives greater than 17 500 h at maximum load conditions and rated speed. The API 610 Standard 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. Both Standards allow a basic rating life L_{10h} or adjusted rating life L_{10ah} calculation.

The adjustment factor a_{23} and the adjusted rating life L_{10ah} are dependent on the viscosity of the lubricant at the bearing operating conditions. Fig. 2.3 plots the a_{23} factor versus Kappa κ. Kappa is the ratio of the lubri-

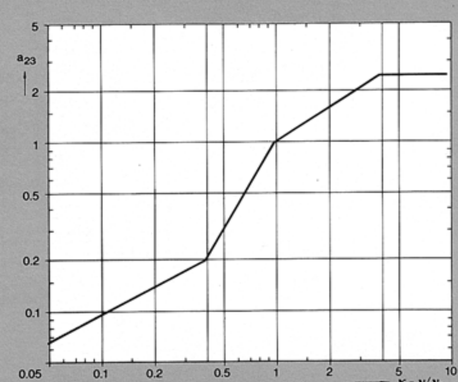


Figure 2.3

SKF

cant 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.

The New Life Theory indicates that a rolling bearing can have infinite fatigue life provided the applied loads are below a fatigue limit, the bearing operates in a sufficiently clean environment, and was manufactured to accurate tolerances. This is a consideration in some pump applications where hydrodynamic bearings are selected for infinite life.

The adjustment factor a_{SKF} for application of the New Life Theory is dependent on the viscosity of the lubricant at the operating conditions (ν), the fatigue load limit of the bearing (P_u), and the contamination level in the application. To enable a systematic and consistent evaluation of the contamination level, SKF has developed a computer program for this purpose. Continued research on the quantification of contamination for use with the New Life Theory will lead to further refinement of this program. For evaluation of a bearing arrangement with the New Life Theory, contact an SKF application engineer.

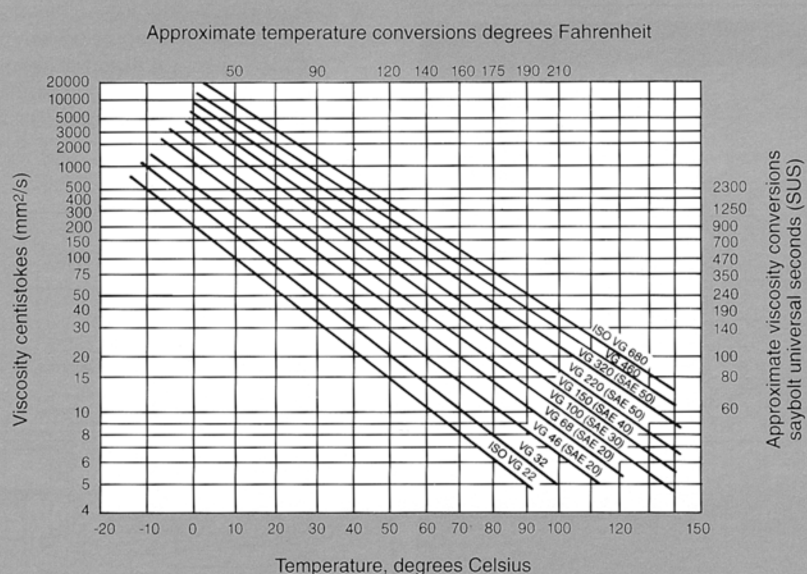
In some cases, the life of pump bearings can be extended by the use of shielded or sealed grease lubricated bearings installed within the sealed pump bearing housing. In the case with sealed bearings, it may be necessary to limit the speed of the pumps because of the additional seal friction.

The New Life Theory 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. The allowable free water content in mineral oil type lubricating oils generally ranges from 200 to 500 ppm by volume depending on the additives supplied in the lubricant. Preferably, the water content should be below 200 ppm. Some synthetic hydrocarbon (polyalfaolefins) oil lubricants have the same limits as mineral oils. There is risk of reduced bearing life if the water content is in excess of these values. Lubricant contamination by water is one of the most common reasons for bearing failures in pumps.

Bearing lubrication

The lubricant separates the rolling elements and raceway contact surfaces and lubricates the sliding 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, ν.

Lubricating oils are identified by an ISO Viscosity Grade (VG) Number. The VG Number is the viscosity of the oil at 40°C (104°F). The common oil grades are shown in fig. 2.4. From this chart, the viscosity of an ISO Grade oil can be determined at the bearing operating temperature.



NOTE Viscosity classification numbers are according to International Standard ISO 3448-1975 for oils having a viscosity index of 95.

Figure 2.4

SKF

Pump bearings

Rolling bearing lubricant requirements depend on bearing size d_m* and operating speed n but little on bearing load. The minimum required lubricant viscosity ν_1 needed at the bearing operating temperature is obtained from fig. 2.5. The actual lubricant selected for an application should provide greater viscosity ν than the minimum required viscosity ν_1 (i.e. Kappa, $\kappa > 1.0$). The table below provides general lubricant recommendations for bearings used in centrifugal pumps. These recommendations are valid for operating speeds between 50% and 100% of the bearing Catalog speed rating. At lower speeds, higher Viscosity Grades should be considered and at higher speeds, lower Viscosity Grades should be considered. The viscosity ratio, Kappa should be the guideline for evaluation of viscosity. Kappa > 1.5 is preferred. The lubricant viscosity should not be too great as this would cause excessive bearing friction and heat.

The frequency of oil change depends on the operating conditions and the quality of the lubricant. Quality mineral oils with a minimum Viscosity Index (VI) of 95 are recommended. Multi-grade 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. Synthetic oils are more resistant to deterioration from exposure to high temperature and may allow less frequent replacement. Lubricants may require more frequent replacement if contamination is present.

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

Recommended ISO Viscosity Grade

Bearing operating temperature °C (°F)	Ball and cylindrical roller bearings	Other roller bearings
70 (158)	VG46	VG68
80 (176)	VG68	VG100
90 (194)	VG100	VG150

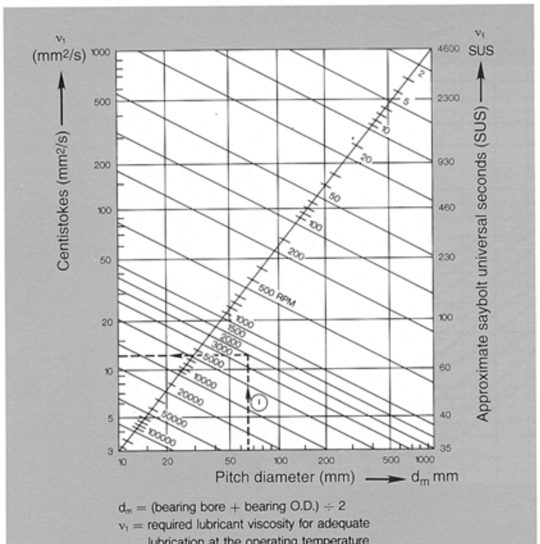


Figure 2.5

* $d_m = \frac{d + D}{2}$, mm

Oil-bath lubrication

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 fig. 2.6. 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 and flinging 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:

$$A = 0.2 \text{ to } 1.0 \sqrt{n\, d_m} \qquad \text{EQ. 5}$$

where

A = bypass opening cross-section area, mm²

n = rotational speed, r/min.

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

Oil-bath lubrication

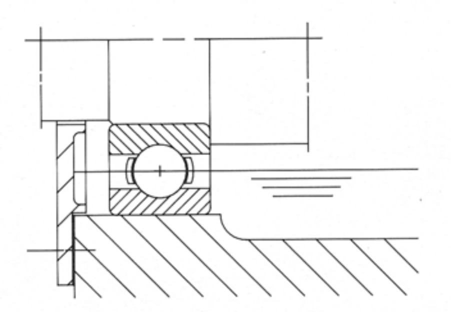


Figure 2.6

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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 or not sufficiently large, the oil may not pass through the bearing. This is particularly true for 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" is an oil reservoir mounted to the bearing housing to replenish oil lost from the bearing housing, see fig. 2.7. A sight glass is recommended along with these devices to enable the correct setting and examination of the lubricant level in the bearing housing.

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

$$V = 0.02 \text{ to } 0.1\, D\, B \qquad \text{EQ. 6}$$

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. 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. Shaft sealing in these applications is best provided by a thin cylindrical sleeve inside the bearing inner ring support, see fig. 2.2g.

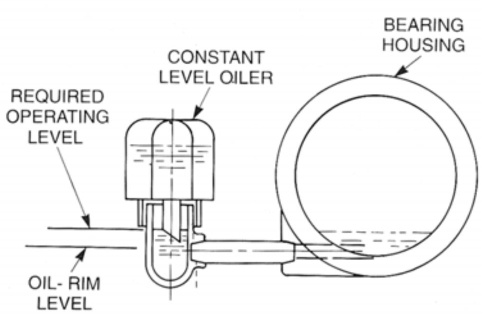

Figure 2.7

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.

Oil-ring lubrication

An oil-ring is suspended from the horizontal shaft into an oil-bath below the bearings, see fig 2.8 (see also fig. 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.

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-ring lubrication reduces the oil volume to the bearing and therefore the bearing friction. 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.

Oil-ring lubrication

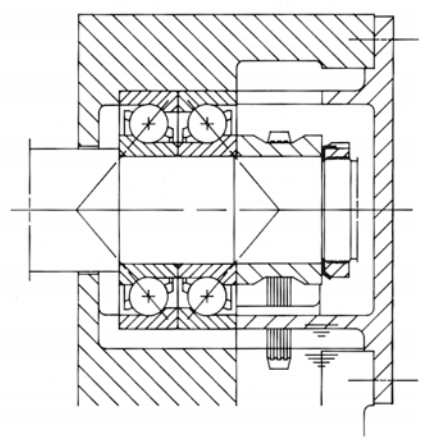

Figure 2.8

SKF

Pump bearings

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 fig 2.9. The mist is produced by a mist generator and is pressurized slightly above ambient pressure.

Oil-mist provides fine droplets of clean, fresh, and cool lubricant to the bearings. 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 dia. (0.125 in.)] opposite the point where the mist enters the housing or bearing. This is 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. A specialist in oil-mist lubrication should be consulted for recommendations.

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.

Oil-mist lubrication

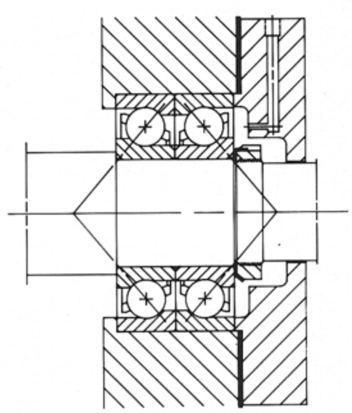

Figure 2.9

"Pure" oil-mist lubrication is without an oil-bath. 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 [3]. 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 long time period 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 magnetic shaft seals and oil-mist collectors to limit the emissions to the environment.

Oil-mist lubrication minimizes the bearing friction.

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). Consistency selection is based on the size and type of bearing used. NLGI 3 consistency greases are recommended for small-to-medium ball bearings, pumps operating with vertical shaft orientation and pumps having considerable vibration.

NLGI 2 consistency greases are recommended for roller bearings and medium to large ball bearings. NLGI 1 consistency greases are recommended for large bearings operating at low speeds.

The grease base oil viscosity is selected in a similar manner to that of lubricating oils. The viscosity ν of the base oil at the bearing operating temperature should be greater than the minimum required lubricant viscosity ν_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 are generally incompatible with other metallic thickened greases, mineral oils, and preservatives.

*nd_m is the bearing speed n in r/min multiplied by the bearing mean diameter d_m in mm.

$$d_m = \frac{d + D}{2}$$

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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 General Catalog provides guidelines for the regreasing interval and the quantity of grease to be added at regreasing. The regreasing interval (t_f) from the General Catalog is based on the use of lithium grease with mineral base oil at 70° C (158° F) temperature. The regreasing interval can be increased if the operating temperature is lower or if a premium quality grease is used. The regreasing interval is reduced if the bearing temperature is higher.

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 fig. 2.10.

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.

Vertical grease lubrication

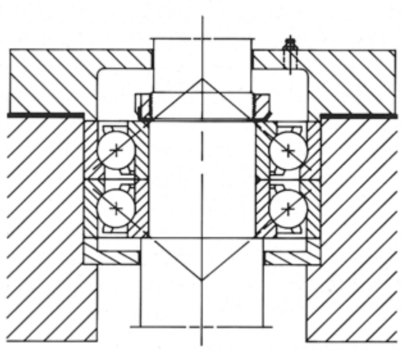

Figure 2.10

Bearing temperature

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 at high temperatures provided the lubrication is satisfactory, and other precautions such as the correct selection of internal clearance etc. are taken.

In some cases, a bearing operated from start-up can not achieve satisfactory low steady state temperature conditions due to incorrect bearing fitting (i.e. high initial internal preloading) or due to excessively fast pump start-up rate. These conditions can cause a thermal imbalance in the bearing, resulting in unintended bearing preload. This latter case is not unusual when the bearing housing is very cold due to low ambient conditions (cold climate or chemical process). The best solution in these instances is to control the bearing fitting (initial bearing clearance and shaft and housing fits) and slow the start-up rate of the machinery to allow the establishment of thermal equilibrium. Machined brass cages (M or MA suffix) may be needed in these applications.

In order for bearings with polyamide cages to obtain longest service life, the outer ring temperature should not exceed 100° C (212° F) in pump applications. Because of this limitation, some pump manufacturers and users do not allow the use of bearings with polyamide cages (TN9 or P suffix).

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, 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.

SKF

Pump bearings

The bearing load-dependent and load-independent friction moments can be calculated in accordance with the General Catalog. Recommended values of the lubrication factor f_0 for the pump bearing lubrication methods are as follows:

Lubrication factor, f_0	oil-mist	oil-ring	oil-bath(1)	grease
Single row deep groove ball bearing	1	1.5	2	0.75-2
Double row angular contact ball bearing				
Conrad	2.3	3.3	4.3	2.7
with filling slot	3.4	5	6.5	4
Single row angular contact ball bearing pair	3.4	5	6.5	4

(1) - double value for vertical shaft orientation

Values of f_0 for other bearing types can be found in the General Catalog. Bearing operating temperature and the viscosity of the lubricant can be estimated using SKF computer programs. (The above values of f_0 are recommended for use with this program.) Higher bearing operating temperatures can be expected when rubbing contact shaft seals are used.

In cases of bearings operating in cold climates or with lubricants having very high viscosity, the radial load may need to be greater than the minimum required radial load estimated by the equations provided in the General Catalog. Bearings having machined brass cages may also be necessary in these applications. In no case should a bearing be operated at temperatures less than the lubricant's pour point temperature.

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 as follows:

Shaft diameter, mm			Tolerance
ball bearings	cylindrical, metric taper roller bearings	spherical roller bearings	
≤ 18	–	–	j5
(18) to 100	≤ 40	≤ 40	k5
(100) to 140	(40) to 100	(40) to 65	m5
(140) to 200	(100) to 140	(65) to 100	m6

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

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.

The minimum required interference between the bearing inner ring and a solid shaft can be estimated by the following equation:

$$i = 0.08 \sqrt{\frac{d\,F}{B}} \qquad \text{EQ. 7}$$

where

i = interference, μm
d = bearing bore, mm
B = bearing width, mm
F = maximum radial load, N

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.

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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 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:

$$F = k\,d \qquad \text{EQ. 8}$$

where

F = spring preload force, N
k = factor ranging from 5 to 10
d = bearing bore, mm

The General Catalog provides guidelines for shaft and housing form tolerance and surface finish.

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 610 Standard pumps.

Internal radial clearance

Bearing internal radial clearance greater than Normal (C3 suffix) is recommended for bearings mounted with heavier than normal interference shaft fits and if high bearing temperatures are expected either from operation at high speed or from heat conducted to the bearing from an external source.

Greater than Normal (C3 suffix) internal radial clearance is recommended for radial bearings operating at greater than 70% of the General Catalog speed rating.

The API 610 Standard specifies that bearings, other than angular contact ball bearings, shall have greater than Normal (C3 suffix) internal radial clearance. Angular contact ball bearings are sensitive to excessive clearance, so special considerations apply to them.

Bearing housing sealing

Sealing of the shaft at the housing is important to exclude solid and liquid contaminants and retain the lubricant. Common seals are radial lip seals and labyrinth seals, see fig. 2.11.

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. Seal friction increases bearing operating temperature. The life of lip seals is usually short (2000 to 4000 h).

Labyrinth seals are effective in excluding contaminants and in retaining the lubricant. They cause little or no friction and have long life. Labyrinth seals provide natural venting for oil-mist lubrication.

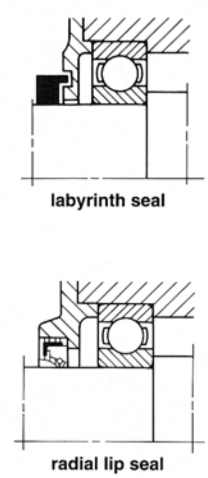

Figure 2.11

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