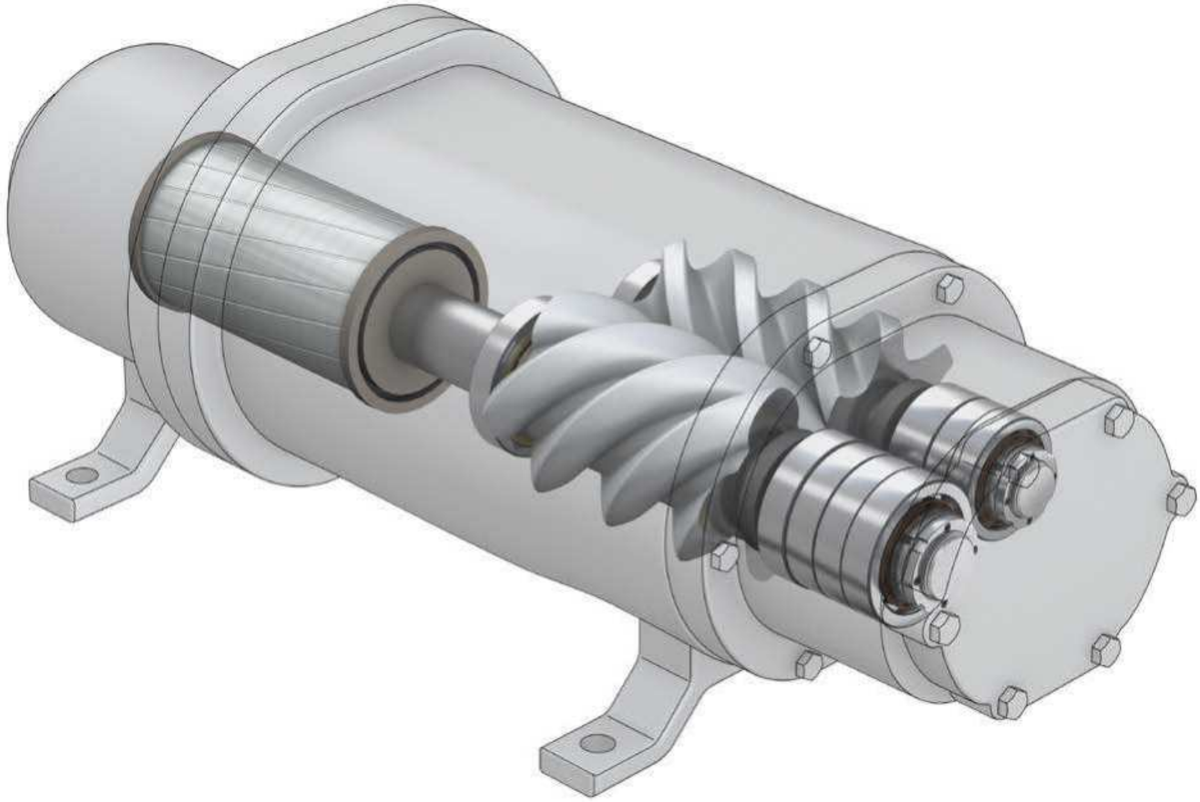
Bearings in twin screw compressors

Application handbook



Bearings in twin screw compressors

Application handbook

Contents

Preface

This application handbook is designed to provide specific application recommendations for SKF customers. It should be used in conjunction with the SKF catalogue *Rolling bearings*. It is not possible, in the limited space of this handbook, to present all the information necessary to cover every bearing application of twin screw compressors in detail. SKF application engineering service 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 application engineering service. The handbook, however, will give you a starting point and guide you as to what questions to ask. We hope you find this handbook informative and helpful.

Developments and advantages of twin screw compressors

The first operating twin screw compressor was developed by Svenska Rotor Maskiner (SRM) in Stockholm, Sweden in the 1930s. SRM acquired several key patents on the new compressor. The first application was a supercharger for airplane engines. After further developments, an industrial air compressor was introduced in the mid 1940s. At that time, SRM also started to sell technology licences.

When first launched, the twin screw compressor competed with the reciprocating piston compressor. The twin screw compressor design offered potential advantages in terms of smaller physical size, lower vibration and noise level, and improved reliability, efficiency and cost. Reciprocating piston compressors are still used today in many applications, when the volume flow is low and the pressure level is high.

Twin screw compressor function

In a twin screw compressor, two meshing rotors are turning in opposite directions inside the compressor housing. The rotor profile can be either symmetric or asymmetric. Asymmetric rotors allows for higher performance. Increased number of lobes increases the rotor stiffness and allows for higher pressures which is common in refrigerant compressors (**fig. 1**).

On the suction side of the compressor, gas is drawn into the suction opening in the housing and into the cavity produced between the housing wall and the two rotors. As the rotors turn in opposite directions, the cavity increases in size and moves forward, drawing in more gas until the cavity has passed the suction opening. At this point the cavity begins to decrease in size as it continues moving forward. As the cavity reaches the discharge side, the compressed gas is pushed through the discharge opening in the housing (**fig. 2a** to **d**, **page 12**).

Fig. 1

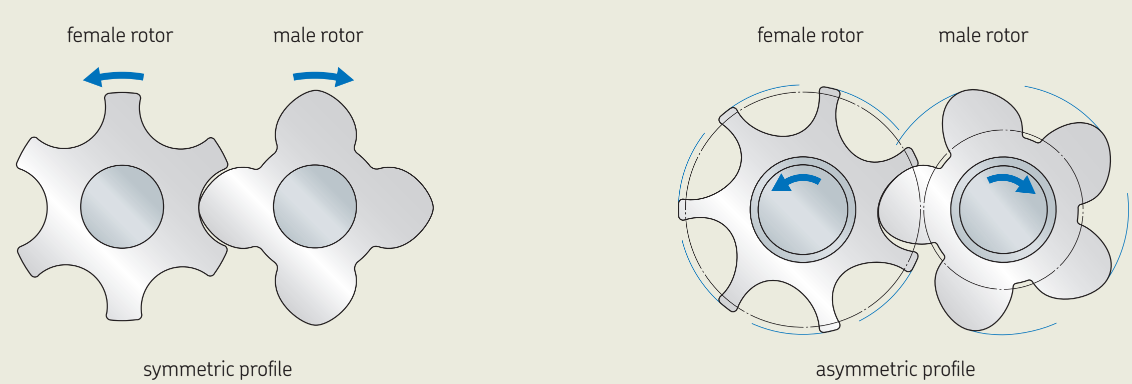
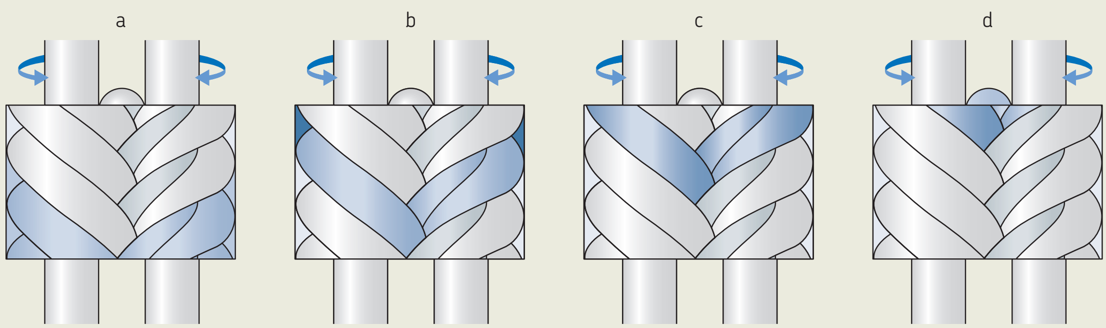


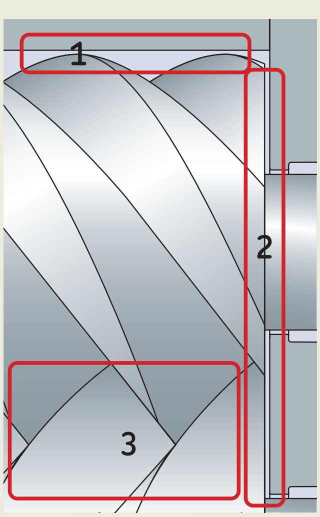
Fig. 2



A slide valve can vary the position of the suction and discharge openings. This makes it possible to control both the volume and the pressure ratio. The pressure increase depends on the volume ratio, but for a given volume ratio, the pressure ratio depends on the thermodynamic properties of the gas. An alternative way to control volume flow only, is to use variable speed control of the drive motor.

Several cavities in various stages of compression are being compressed simultaneously. The most common drive arrangement is that the male rotor drives the female. In such case, the number of cavities equals the number of lobes of the male rotor. Since each cavity has a different pressure, a small amount of gas will leak from a cavity with high pressure to one with a lower pressure. The leakage results in loss of efficiency. The leakage will be minimized by reducing the clearance between the rotors and the housing. Three different clearances (**fig. 3**) must be considered; the clearance between the tips of the rotors and the cylindrical surface in the housing (**1, fig. 3**), the clearance between the end faces of the rotors and the housing ends (**2, fig. 3**), and the clearance between the rotors (**3, fig. 3**). The rotor end clearance is adjusted by adjusting the axial positioning of the thrust bearings during compressor assembly. The clearance between rotor and housing and between the rotors is influenced by the clearance in the radial bearings.

Fig. 3



Compressor designs

It is possible to classify screw compressor types in many different ways. The following distinctions between compressor designs are important:

- Fluid injected compressors
- Dry running compressors

In fluid injected compressors, a fluid is injected into the rotor cavities during compression. The purpose of the fluid injection is:

- Seal the leakage gaps between the two rotors by filling the gaps with fluid
- Absorb compression heat from the gas
- Lubricate the contacts between the two rotors
- Prevent corrosion

Oil-injected screw compressors

In oil-injected compressors, oil performs all four functions. The sealing of leakage gaps and lubrication of the rotor contacts is very efficient. The injected oil passes out of the compressor with the discharge gas, which then goes through an oil separator. This separates most of the oil from the gas. The return oil from the separator is delivered to an oil reservoir, to be used again for injection in the compressor and to lubricate the bearings (**fig. 4**).

Oil-injected compressors operate with rotor tip speeds in the range 30 to 60 m/s, which means bearing nd_m[1)] values of 250 000 to 650 000. Typical shaft speeds are in the range of 3 000 to 6 000 r/min. Typical discharge pressures for single stage oil-injected screw compressors are 7 to 13 bar, and for two stage compressors 25 bar.

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

where
n = speed [r/min]
d_m = bearing mean diameter [mm]
d = bore diameter [mm]
D = outside diameter [mm]

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

Water-injected compressors

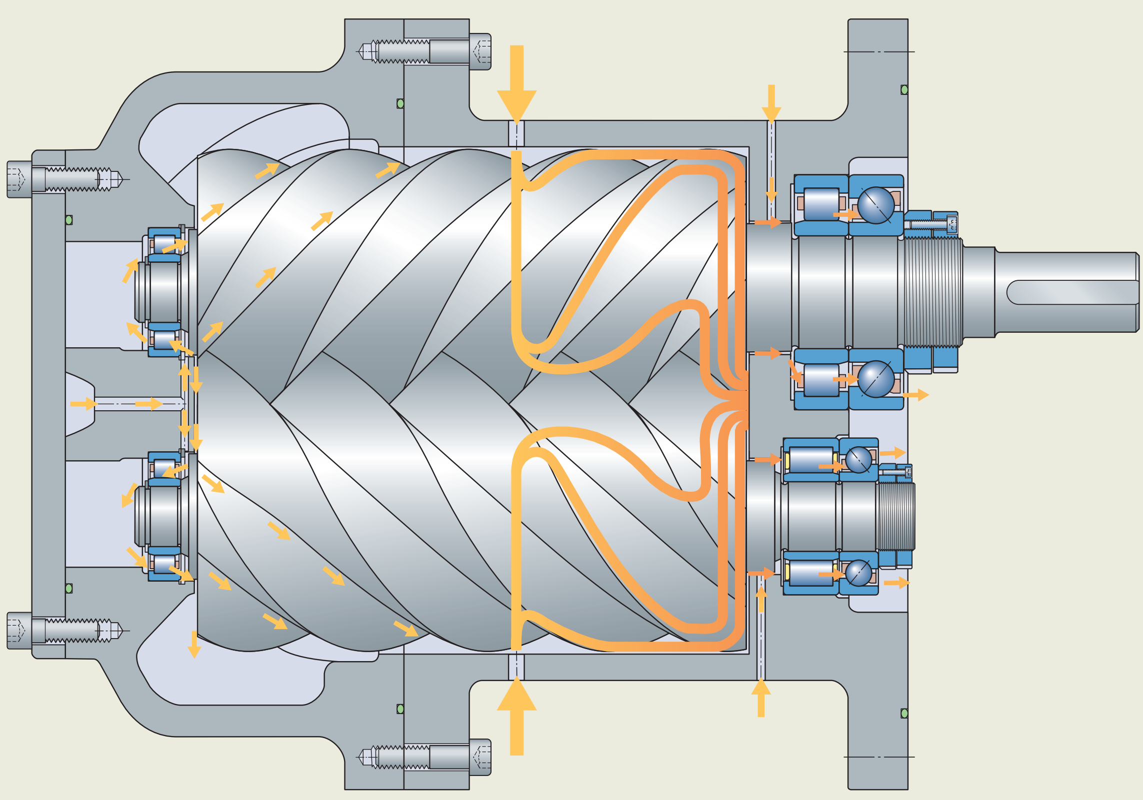
In air compressors, water can also be used for injection into the rotor cavity to absorb compression heat and to seal the leakage gaps. Since water has high specific heat, water injection is more efficient than oil injection at absorbing the heat and thereby reducing the discharge temperature. However, as water is corrosive and not an efficient lubricant, the rotors have to be coated with a polymer or ceramic material. An alternative design uses stainless steel rotors which do not touch, and external timing gears to facilitate appropriate meshing. Water-injected compressors with rolling bearings must have efficient seals between the rotors and the bearings to prevent water from leaking into the bearing lubricating oil.

Liquid refrigerant injected compressors

In refrigerant compressors, it is possible to inject liquid refrigerant instead of oil into the compression cavities. The heat necessary to vaporize the liquid refrigerant is absorbed from the compression process. The compressor design can either be similar to the water-injected compressor, with the bearings separated from the screws, or similar to an oil-injected compressor, with the use of SKF Pure Refrigerant Lubricated bearings.

Fig. 4

Oil flow paths in an oil injected screw compressor
One path is for the injection of oil in the compression cavity. This oil also lubricates the suction side bearings, and some is leaking into the discharge bearing cavity. The other path is for lubrication of the discharge side bearings.



Dry running compressors

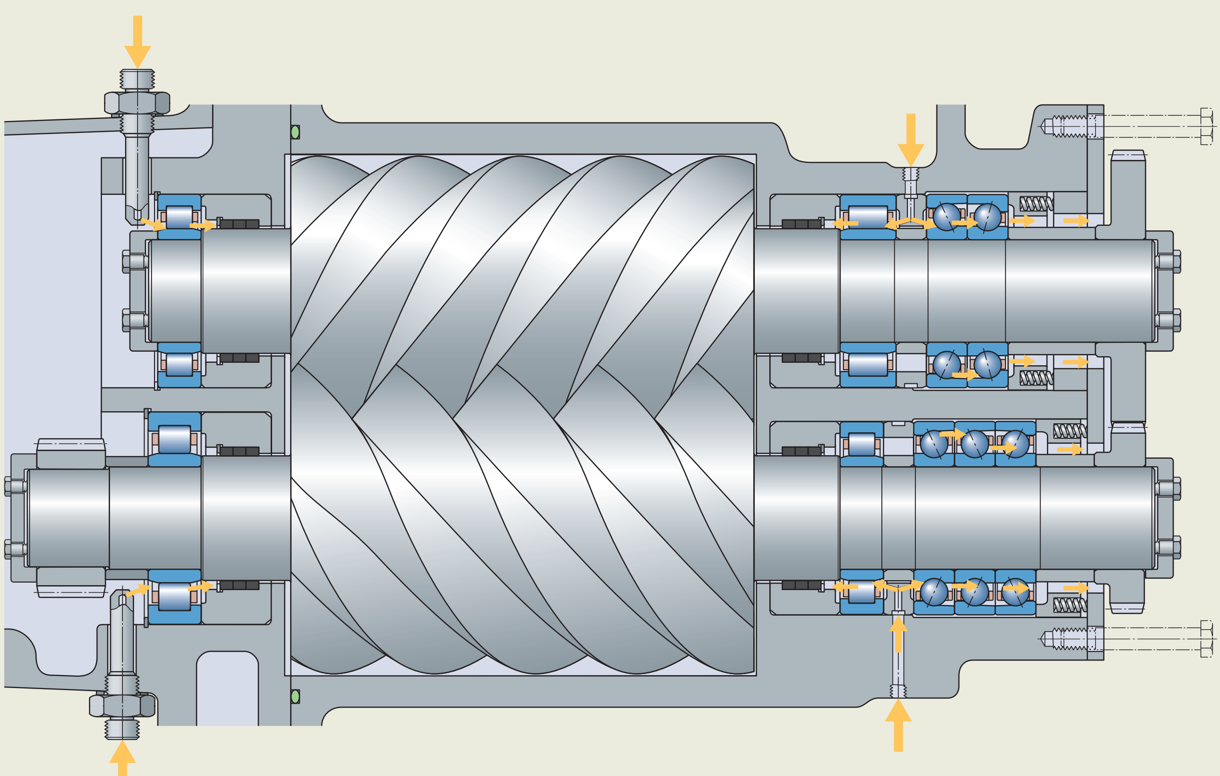
Dry running screw compressors use external timing gears mounted on the extended rotor shafts for accurate meshing of the two rotors. The rotors are designed never to touch. For this reason, the clearances between the two rotors and between each rotor and the housing, have to be larger in dry running compressors. Dry running compressors operate at higher temperatures than injected compressors because no liquid is injected between the rotors for cooling.

The bearing housings and the screw housing are separated by effective seals, assuring that no oil is leaking from the bearing housings to the screw housing. The oil is only used for bearing lubrication (**fig. 5**).

Because of the larger clearances and the lack of fluid for sealing the clearances, the leakage rate is higher for dry running compressors. For this reason, they are designed to run at high speed. By running at high speed, the compression is faster and there is less time for leakage to occur during each revolution. Dry running air compressors operate at rotor tip speeds above 60 m/s and bearing nd_m values in the range of 800 000 to 1 200 000. Typical shaft speeds are in the range of 10 000 to 25 000 r/min. Since there is no contamination of the air by injected oil, dry running air compressors are used in applications such as medical, food and beverage and electronics, as well as other applications where air contamination with oil is sensitive or prohibited. Typically the discharge pressure is 3 to 7 bar. To reach 7 bar, two compression stages are needed, with intermediate cooling.

Fig. 5

Oil flow paths in a dry air screw compressor
The oil injected by jets and is only for bearing lubrication. To prevent leakage of oil into the air stream, the compression cavity and the screws are sealed from the bearing cavities.



Future trends

Energy efficiency, reliability, noise level and cost continue to be the main drivers in screw compressor development. For bearings, this means requirements for improved rotor positioning accuracy, low friction, varying speeds and low cost.

Smaller bearing operating clearances, narrower tolerances, energy-efficient compressor designs and higher load capacities enable bearings to meet those requirements. Optimized bearing arrangements will enable compressor designs to take advantage of high performance bearings.

Variable speed drive (VSD) technology has a strong influence on screw compressor design and performance. This technology is not in itself new, but cost reductions have made it economically feasible to use the technology in screw compressors. VSDs make it possible to operate at peak efficiency as operating conditions vary. The speed of direct drive compressor designs is no longer limited to 3 000 or 3 600 r/min, so compressors are able to run at both higher and lower speeds. Gear drive designs can be replaced by direct drive designs. This requires bearings that can also run at varying speeds, both high and low.

Another future trend is the introduction of new refrigerants, with lower global warming potential (GWP). The new refrigerants, such as R1234ze, have different pressures and cooling capacities, that will affect bearing loads, speeds and lubrication

Bearing applications in twin screw compressors

Compared to bearing applications in other types of machinery, applications in twin screw compressors are more challenging for several reasons. One is that bearing size is limited because of the fixed centre distance between the two rotors. Another factor is that the rotor positioning accuracy must be high and so bearing clearance must be small and bearing runout tolerances small. Bearing mounting and adjustment of rotor end clearance are other factors to consider in the selection of the bearing and bearing arrangement. This chapter discusses the factors affecting the bearing application and offers recommendations on how to select a good bearing arrangement and suitable bearing type.

Bearing function and selection criteria

Bearings in twin screw compressors provide accurate radial and axial positioning of the rotors and support rotor loads. These functions must be performed reliably, with low friction and low noise generation. With accurate rotor positioning, it is possible to design compressors with small clearances for high efficiency. A high radial positioning accuracy is achieved by using bearings with small operating clearances and small runout tolerances. Axial positioning accuracy is accomplished by small axial bearing clearance or preload.

Axial positioning is also affected by the fit between the thrust bearing inner ring and the rotor shaft and bearing deflection and displacement due to centrifugal forces. Interference fits will change the relative axial position of the bearing rings after mounting. Axial positioning of the rotor is further affected by the accuracy of the end clearance adjustment during assembly. Thrust bearings mounted with interference fits make the end clearance adjustment more difficult.

The rotor centre distance can be a limitation in the bearing selection. Since the rotor design affects the centre distance, the rotor design and bearing selection process is sometimes iterative.The centre distance limits the outer diameters of the bearings. Therefore, if one rotor carries a higher load, it is possible to select a larger bearing for this rotor and a smaller bearing for the other rotor. This design option, however, conflicts with the desire to minimize the number of different bearings in the compressor. It also requires additional tooling for the production of the compressor housing.

The rotors can be supported on rolling bearings or on a combination of hydrodynamic and rolling bearings. Small operating clearances are the main advantage of rolling bearings. Rolling bearings also have lower friction than hydrodynamic bearings, require less oil for lubrication and cooling, and are less sensitive to momentary loss of lubricant and flooding of refrigerant than hydrodynamic bearings.

Twin screw compressor bearing loads

Bearing loads in twin screw compressors are produced by:

- Gas pressure on the rotors
- Gear forces from input and timing gears
- Rotor forces from transmission of torque from one rotor to the other
- Induced loads from rotor inertia at startup
- In semi-hermetic compressors, electric motor rotor weight and magnetic force
- Loads from incompressible fluids being trapped between the rotors
- Induced loads from centrifugal forces in thrust bearings
- Spring preload or balance piston forces
- Loads from belts drive

The gas pressure is low at the suction side and high at the discharge side (**fig. 1**). The gas pressure along the length of the rotor produces radial forces on the rotors. These forces are higher at the discharge side. The gas also produces axial forces from the pressure acting on the projected areas at both the suction and discharge end of the rotors. The difference between these two forces is the net axial gas force on the rotor. The axial gas force is always directed towards the suction side and is larger on the male rotor.

Balance pistons

Stationary or rotating balance pistons can be used to reduce the net axial force acting on the thrust bearings. A rotating balance piston is a disc mounted at the suction end of the rotor. Gas at discharge pressure from the compressor is allowed to act on the end face of the disc, producing an axial force directed towards the discharge side. This force balance axial gas force on the rotor (**fig. 2**).

A stationary balance piston uses a bearing for transmission of the balancing force to the rotor (**fig. 3**).

Fig. 1

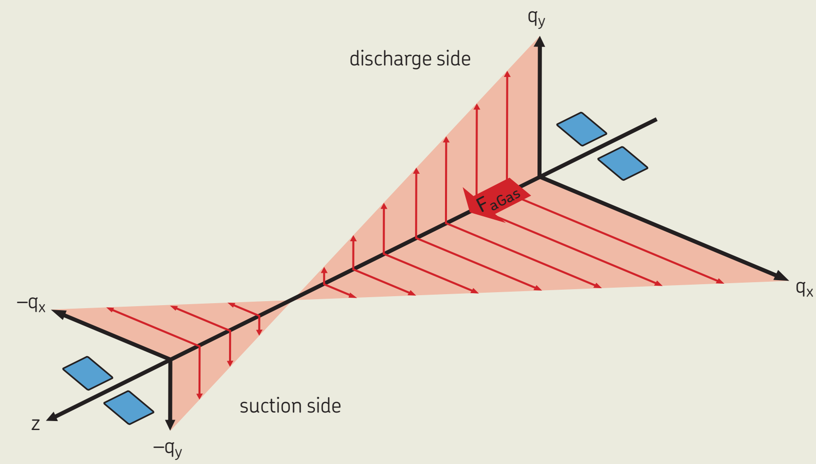


Fig. 2

In order to reduce the axial force from the rotor, stationary or rotating balance pistons are sometimes used.
A rotating balance piston is a disc mounted at the discharge end of the rotor.

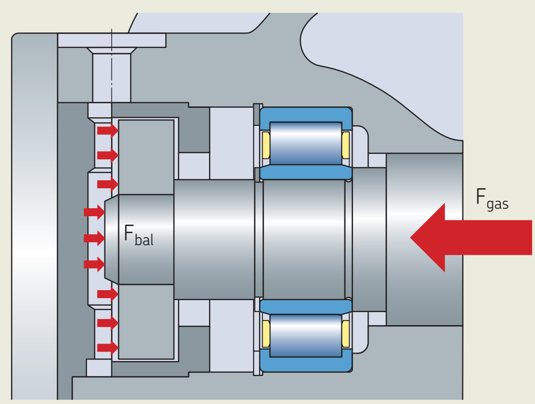
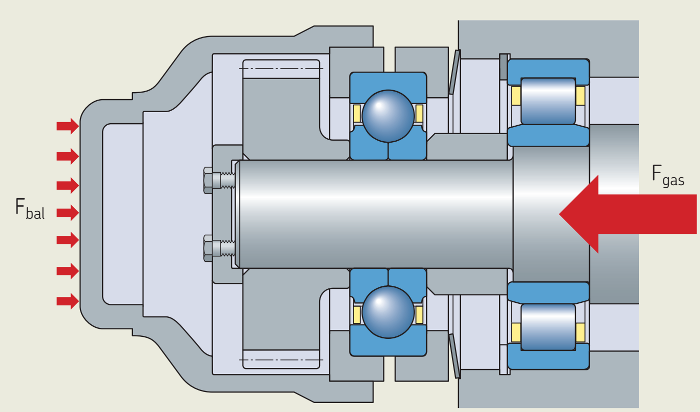


Fig. 3

A stationary balance piston uses a bearing for transmission of the balancing force to the rotor



Gear forces

In a gear-driven compressor, (**fig. 6**, **page 20**) the forces from the gears are also supported by the rotor bearings. By varying the gear helix angle, it is possible to control both the magnitude and direction of the gear axial forces. This can cause the net axial force on the rotors to reverse and can cause rubbing between the ends of the rotors and the housing on the discharge side. It can also cause instability. A reverse thrust bearing can be used to avoid rotor rub and instability. Reverse thrust bearings can either be spring loaded or arranged such that the axial bearing clearance is limited or eliminated completely. If there is an axial bearing clearance, then the rotor end clearance must be larger than the axial bearing clearance.

Too low an axial force on the bearings can be detrimental if the loads become less than the minimum required load for satisfactory operation. The timing gear forces and inertial forces from transmission of torque between the rotors are usually small, except at compressor startup. Analysis of bearing loads in screw compressors is complicated and should be performed through detailed analysis of compressor design parameters and operating conditions.

Belt drive forces

An alternative way to change the speed between the drive motor and the compressor input shaft is to use a belt drive. This is only possible in an open drive arrangement and typically used in oil injected air compressors.

The addition of the belt load must be accounted for in the selection of compressor input shaft bearings.

Separation of radial and axial forces

In cylindrical roller and angular contact ball, or four point contact ball bearing arrangements, the loads are separated such that the radial loads are taken by the cylindrical roller bearing (**fig. 4**), and the axial loads by the angular contact ball or four point contact ball bearing (**fig. 5**). The load separation is accomplished by a radial gap between the outer ring of the angular contact ball bearing or the four point contact ball bearing and the housing (**fig. 4** and **5**). With this design it is not possible for the angular contact ball bearing or four-point contact ball bearing to take radial load, or for the cylindrical roller bearing to take axial load, since it is axially compliant.

The advantages with this arrangement are:

- The angular contact ball or four point contact ball bearing bearing operates with axial load only; all balls have the same contact loads and contact angles. In this way, cage forces are minimized and load capacity maximized.
- The load capacity of the arrangement is optimized with the separation of axial and radial loads into two bearings.
- It is easy to set the rotor end clearance when the angular contact ball or four-point contact ball bearing is mounted with a light shaft fit. There are many ways to do set the rotor end clearance.

Reverse thrust and backup bearings

The compressor generates pressure as soon as the rotors start rotating. That ensures that there is always an axial gas force on the rotors acting towards suction.

There are situations, however, when the net axial force can reverse. One such situation is when the compressor starts up. There are then inertial moments acting on the rotors. The inertial moments generate axial forces. On the female rotor the direction of the inertial axial force counteracts the direction of net gas force and the direction of the net axial force can be towards discharge.

There is also a speed dependent reverse thrust force generated by the thrust bearings. This force is produced by centrifugal forces acting on the rolling elements. Because of the contact angle with the outer ring, when the rolling elements are forced against the outer ring, an axial component of the centrifugal force is produced. This force tends to separate the bearing rings. If there is not enough axial gas force present, rotor rub can occur. This is seldom a problem, but can happen at nd_m speeds greater than 450 000. It can be avoided if a backup bearing is used. **Diagram 1**, **page 49** shows the ring separation (and rotor displacement) in a single angular contact ball bearing at various speeds. In **diagram 2**, **page 50** a backup bearing has been added to the single bearing.

In tapered roller bearings used to take combined loads, there is also an induced reversed axial force from the radial force (**fig.6**, **page 66**).

Fig. 4

Radial bearing loads

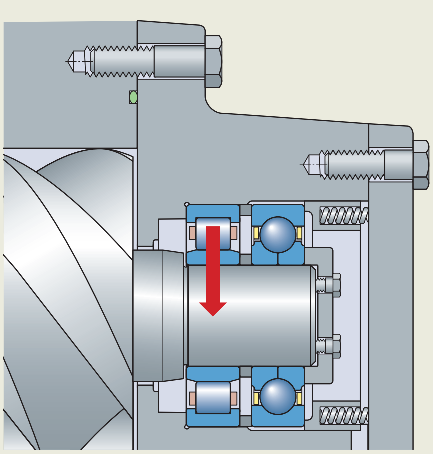


Fig. 5

Axial bearing loads

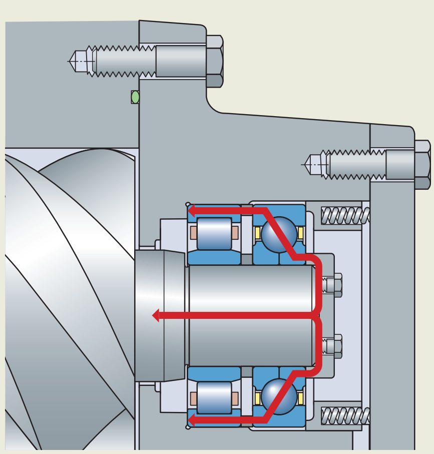
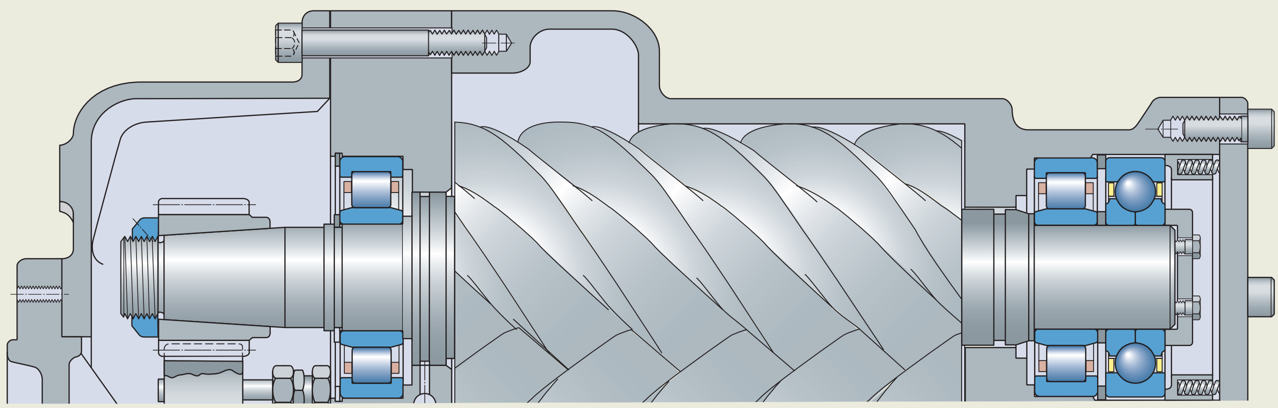


Fig. 6



Setting of rotor end clearance

In principle, there are two ways to set the rotor end clearance. One is to use shims or spacer rings with precise width. The spacer width or shim thickness should be determined by measuring bearing widths and compressor components. The spacers or shims can be positioned between the radial and axial bearing (**fig 7**) or between the radial bearing and shaft shoulder. (Alternatively, bearing sets with controlled widths and stand-out are available SKF.) The bearings should be clamped axially on the rotor (**fig. 7**). The other way is to leave a small (1 mm) gap between the shaft abutment and the thrust bearing, then drive up the thrust bearing to the desired position on the shaft, to produce the desired rotor end clearance. A locking device is then placed behind the angular contact ball bearing inner ring (**fig. 8**).

Fig. 7

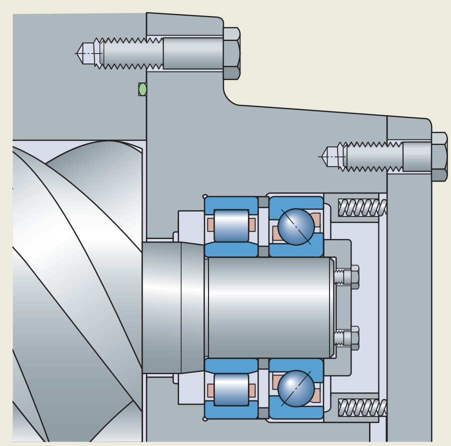
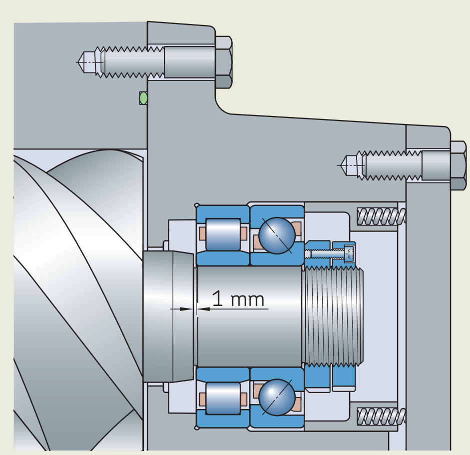


Fig. 8



2

Bearing arrangement examples

A pair of two angular contact ball bearings at discharge-side

In **fig. 9**, two universally matchable angular contact ball bearings mounted face to face are used. The inboard bearing carries the axial and most of the radial load. The outboard bearing is a back up bearing. The rotor end clearance is set by a spacer between the abutment on the rotor and the inboard bearing. The load capacity is limited since only a single angular contact ball bearing carries most of the load. The arrangement has reverse load capability. Angular contact ball bearings with different contact angles can be considered, but two 40° bearings, BE design in combination with CB clearance, is the simplest arrangement. If the axial load is light and the radial high, and the load ratio $F_a/F_r < 1,0$ a bearing with 25° contact angle, AC design, would be the optimum for both bearings.

Cylindrical roller and angular contact ball bearing discharge-side arrangements

The simplest cylindrical roller bearing/angular contact ball bearing arrangement is one single cylindrical roller bearing and one angular contact ball bearing (**fig. 10**). The axial load capacity is limited since there is only one thrust bearing. The rotor end clearance is set by matched spacers or shims and the bearings are clamped axially on the shaft. There is no reverse thrust bearing and no reverse thrust capability.

Fig. 11 is similar to **fig. 10** except that the inner rings are not clamped on the shaft. Instead there is a small gap (1 mm) between the cylindrical roller bearing and the shaft abutment as described in the section *Setting of rotor end clearance*, **page 20**. The gap allows setting of the rotor end clearance by pushing the cylindrical roller bearing and angular contact ball bearing inner rings into the correct position. A lock nut or other means can be used to push the bearings on

NOTE: when angular contact ball bearings are used in double or triple arrangements, they should be universally matchable, with GA or CB clearance.

Fig. 9

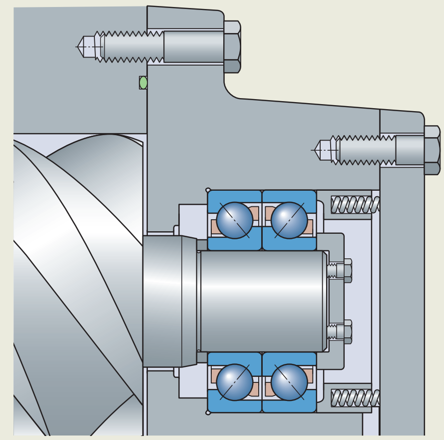


Fig. 10

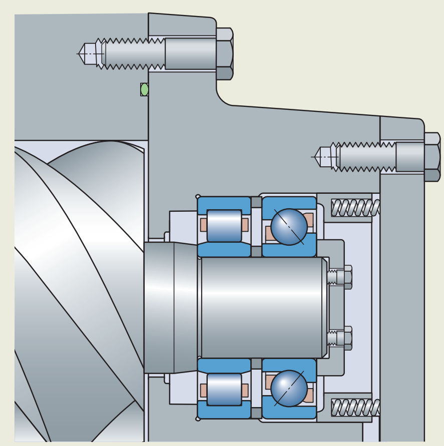
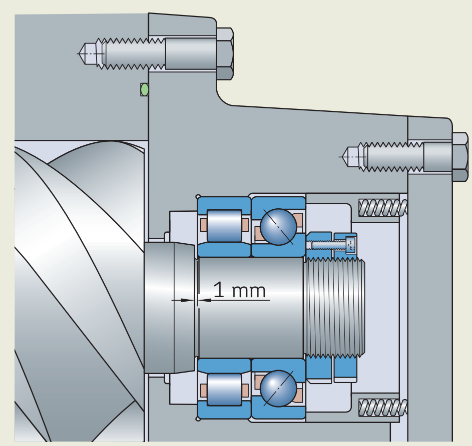


Fig. 11



the shaft into the right position. The locknut shown in **fig. 11** is the SKF KMD nut, which can be locked at an exact axial position by means of locking screws.

Fig. 12 shows one cylindrical roller bearing and two angular contact ball bearings in tandem. This is similar to the arrangement in **fig. 10**, except that there are two angular contact ball bearings in tandem to take the axial load and thus provide higher axial load capacity. The rotor end clearance is set by matched spacers or shims and the bearings are clamped axially on the shaft. There is no reverse thrust bearing and no reverse thrust capability.

Fig. 13 shows one cylindrical roller bearing, two angular contact ball bearings in tandem and one deep groove ball bearing as reverse thrust bearing. The deep groove ball bearing is spring loaded on the outer ring. There is a spacer between the deep groove ball bearing and the angular contact ball bearing to ensure that the outer ring of the deep groove ball bearing will not touch the outer ring of the angular contact ball bearing. The reverse thrust load capacity is equal to the spring force. The load of the deep groove ball bearing is always equal to the spring force and the spring force will add load to the angular contact ball bearings (**diagram 6**, **page 53**). For determination of spring load, see chapter *Bearing preload*, **page 51**.

Fig. 14 is similar to **fig. 11** except that there are two angular contact ball bearings in tandem and a deep groove ball bearing as reverse thrust bearing. The reverse thrust deep groove ball bearing must be located behind the lock nut. The bearing should have a smaller inner ring diameter and being abutted by a shoulder on the shaft.

Both **fig. 14** and **16** show a PTFE seal between the screw and the bearing housing. For information about PTFE seals see **page 69**.

NOTE: when angular contact ball bearings are used in double or triple arrangements, they should be universally matchable, with GA or CB clearance.

Fig. 12

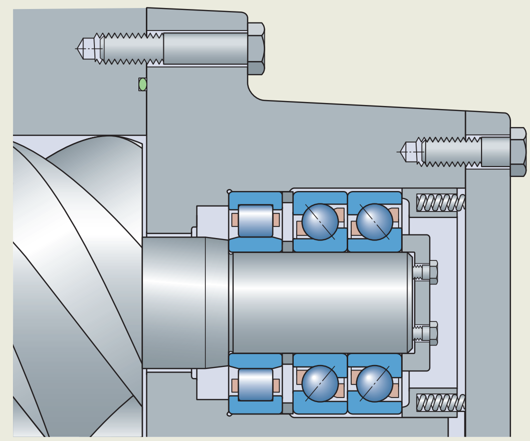


Fig. 13

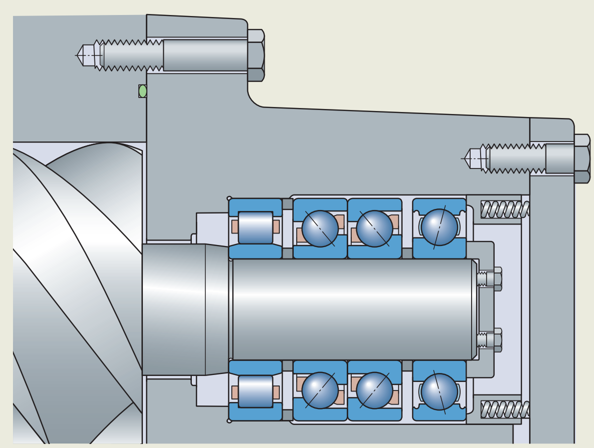


Fig. 14

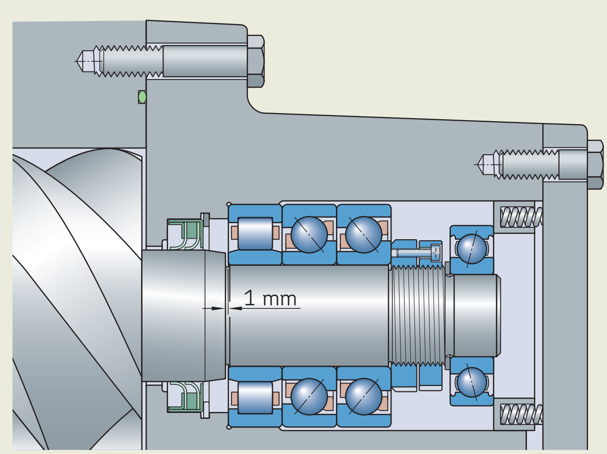


Fig. 15 shows one cylindrical roller bearing and two angular contact ball bearings arranged face-to-face. In this arrangement the additional angular contact ball bearing is the reverse thrust bearing. The outer rings are clamped with a spring. The spring load will not add load to the thrust bearing. The load of the reverse thrust angular contact ball bearing is not easily determinable, and depends on rotor axial load, the clearance between the angular contact ball bearings and the speed (centrifugal forces on the balls) (**diagram 5**, **page 53**). The general clearance recommendation for the angular contact ball bearing set is GA. For determination of spring load, see *Bearing preload*, **page 51**.

Fig. 16 is similar to **fig. 15**, except that there are two angular contact ball bearings in tandem to take the axial load, providing higher axial load capacity. The third bearing in face-to-face arrangement is the reverse thrust bearing.

Fig 17 shows one cylindrical roller bearing and one four-point contact ball bearing. This is similar to **fig. 7**, **page 20**, except that the thrust bearing is a four-point contact roller bearing instead of an angular contact ball bearing. Unlike the angular contact ball bearing, the four-point contact ball bearing has a built-in axial clearance and equal axial load capacity in both directions. In this way, it functions as both main and reverse thrust bearing. The axial clearance should be small to avoid excessive axial displacement in case of reverse thrust. For twin screw compressors, the clearance is typically C2L. If the speed is very high, a larger bearing clearance should be selected (**table 2**, **page 83**) to avoid excessive three point ball contact caused by centrifugal forces on the balls. The inner rings must be clamped on the shaft to ensure correct clearance of the four-point contact roller bearing.

Fig. 15

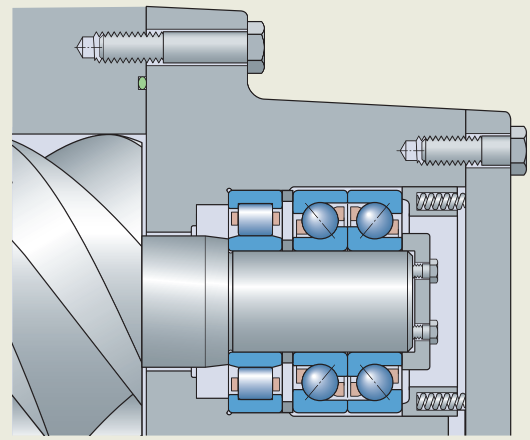


Fig. 16

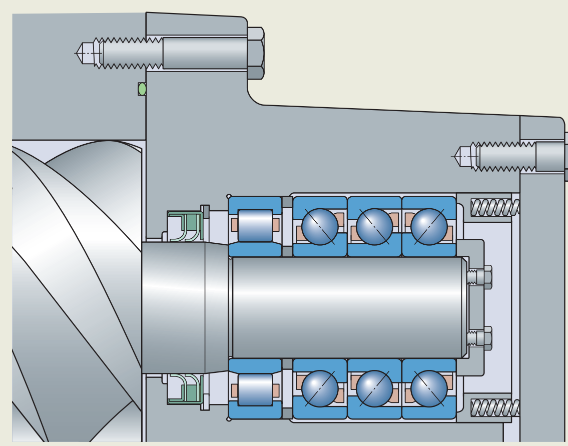
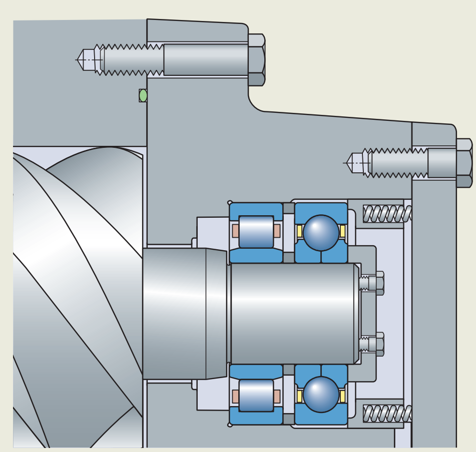


Fig. 17



2

Tapered roller bearing discharge-side arrangements

Tapered roller bearings should be arranged face-to-face so that the inboard tapered roller bearing takes both the radial and the axial load. The outboard tapered roller bearing controls the axial clearance between the bearings and serves as a reverse thrust bearing.

The contact angle of the inboard bearing should be selected such that the bearing is seated toward the suction end of the compressor. This means that the ratio of axial to radial load should preferably be:

$$\frac{F_a}{F_r} > \frac{0,75}{Y}$$

but at least

$$\frac{F_a}{F_r} > \frac{0,5}{Y}$$

where
F_a = total axial load on bearing [N]
F_r = applied radial load [N]
Y = bearing axial load factor, according to the SKF catalogue *Rolling bearings*

If not the rotor will move within the bearing axial clearance towards the discharge end of the compressor.

If the outboard bearing is spring loaded, **fig. 19**, the axial load on the inboard bearing can be increased by increasing the spring load to satisfy the formulae above.

This means that the male rotor bearing should usually have a large contact angle (**fig. 18**) and the contact angle of the female rotor bearing should be smaller (**fig. 19**). The reverse thrust tapered roller bearing is lightly loaded and can be smaller than the load-carrying bearing (**fig. 18**).

The reverse thrust bearing can be either spring loaded or set with shims to produce a small clearance between the two bearings. (**fig. 20**). Because of difficulties in controlling heat generation and maintaining a controllable preload, SKF does not recommend negative clearance (preload). With too little clearance, or preload, there is a risk of early bearing failure due to excessive heat generation and runaway preload.

Fig. 18

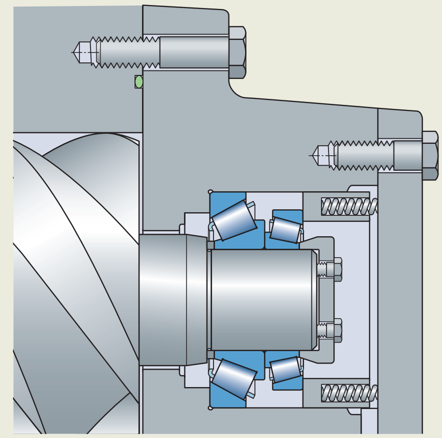


Fig. 19

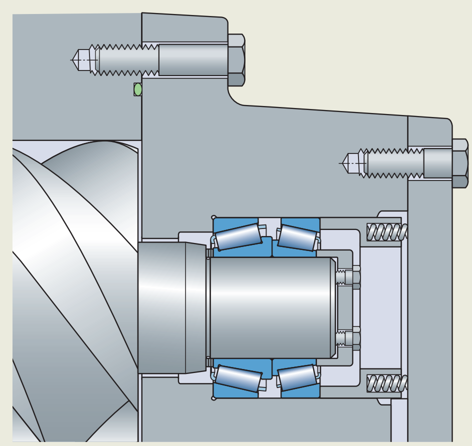
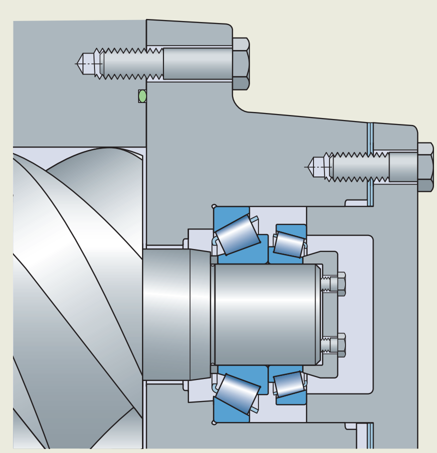


Fig. 20



The axial bearing clearance should be smaller than the rotor end clearance, to allow reverse thrust loading without causing rotor rub. A deep groove ball bearing can also be used as a reverse thrust bearing, but should be spring preloaded since it has low axial stiffness (**fig. 21**).

Traditionally tapered roller bearings of the same size and designation have been used for both the load carrying bearing and for the backup bearing. With this arrangement the load capacity of the backup bearing is not fully utilized (**fig. 22**).

If there is no reverse load, and depending on the ratio of the axial to radial load and the bearing contact angle, it may be possible to use only one tapered roller bearing (**fig. 23**). Because of the contact angle, the radial load will induce an axial load in the reverse direction to the axial gas load. If the gas load is greater than the reverse induced load, one single bearing can be used. In order to minimize the reverse induced load, a tapered roller bearing with a small contact angle, meaning large Y factor, should be used. The reverse induced load can be calculated with the formula:

$$F_a = \frac{0,5\, F_r}{Y}$$

where
F_a = total axial load on bearing [N]
F_r = applied radial load [N]
Y = bearing axial load factor, according to the SKF catalogue *Rolling bearings*

If the reverse load calculated with this formula is equal to the axial gas load, the bearing loaded zone will be 180 degrees. If the ratio of axial to radial load changes, the loaded zone will change and the rotor axial and radial position will change. It is recommended that the axial gas load should be at least double the induced reverse axial load.

Since this bearing is carrying radial load, it should be mounted with an interference shaft fit. This means that the standoff will change after mounting. This shift can be accounted for in the selection of spacer width (**page 67**).

Fig. 21

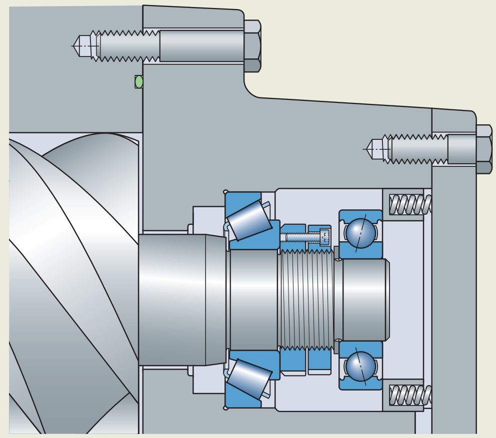


Fig. 22

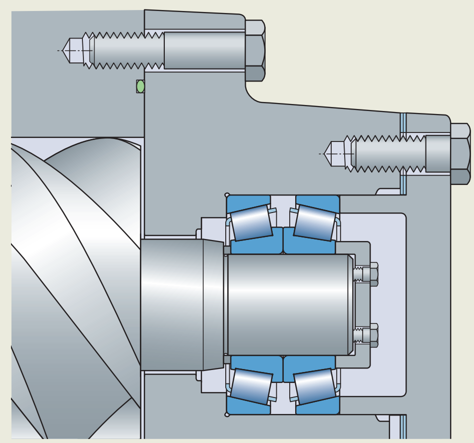
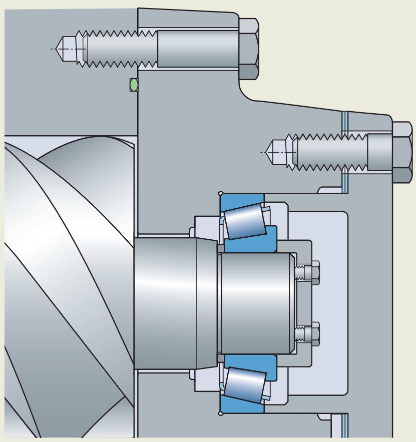


Fig. 23



Suction-side bearing arrangements

The most common suction side bearing arrangement is a single cylindrical roller bearing or needle roller bearing. A snap ring on the shaft can be used to prevent the inner ring from moving axially (**fig. 24**). Alternatively an NJ bearing can be used with the flange of the inner ring is positioned on the inboard side of the rotor, preventing the ring from moving axially (**fig. 25**). If the load is light it is also possible to use a deep groove ball bearing as a suction-side bearing (**fig. 26**) or an angular contact ball bearing with 25 degree contact angle (AC suffix).

The angular contact ball bearing should be spring loaded (**fig. 27**). An angular contact ball bearing will also function as a reverse thrust load bearing. The spring load, (F_s) should be:

$$F_s > 0,6\ F_r$$

where
F_s = Spring load [N]
F_r = Radial load [N]

Gear shaft bearing arrangements

Compressors with speed increasing gears have a gear shaft coupled to the motor shaft, and a gear that drives a pinion which is fitted at the end of one of the rotors, usually the male. The purpose of the gears is to increase the compressor speed to get the specified compressor capacity.

A common bearing arrangement for the gear shaft is tapered roller bearings arranged in cross location face-to-face, with the gear in between. This arrangement is simple and easy to assemble. The axial clearance should be set with shims (**fig. 28**). An alternative arrangement for higher speeds and loads is to use a four-point contact ball bearing as the fixed bearing and two cylindrical roller bearings as radial bearings, with the gears between the cylindrical roller bearings (**fig. 29**).

Fig. 28

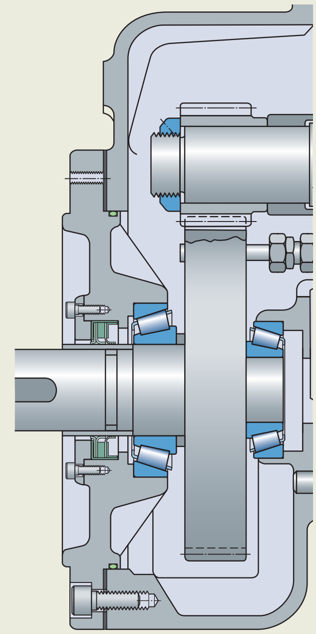


Fig. 29

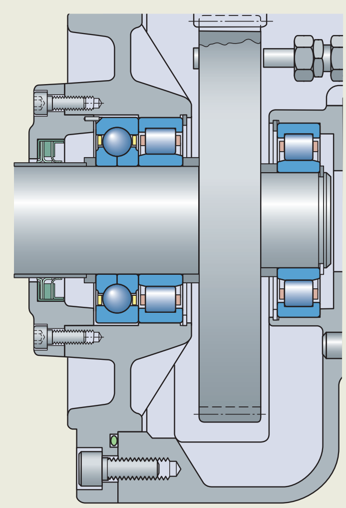


Fig. 27

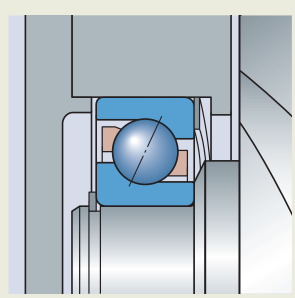


Fig. 24

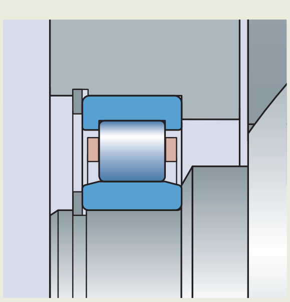


Fig. 25

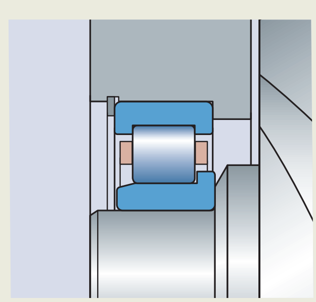
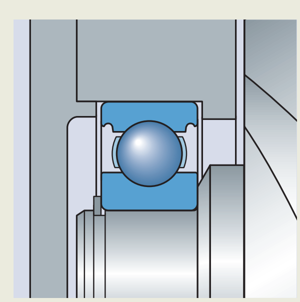


Fig. 26



2

Bearing lubrication

Bearings in twin screw compressors are lubricated by flow of circulating oil. The oil lubricates the rolling contact surfaces and the sliding surfaces within the bearing. The lubricant also provides corrosion protection and cooling to the bearings. In oil-injected compressors, the oil lubricating the bearings is the same oil that is injected into the compressor to lubricate the rotors, remove the heat of compression and seal gaps.

Oil viscosity

Operating viscosity v is the principal parameter for selecting a bearing lubricant. Lubricating oils are identified by an ISO viscosity grade (VG) number. The VG number is the viscosity of the oil in mm^2/s at 40 °C. Standardised viscosity grades are shown in **table 1**. The viscosity index (VI) is indicative of the change in oil viscosity with temperature. From **diagram 1**, the viscosity of an ISO grade oil can be determined at the bearing operating temperature. Synthetic oils are also used in compressors. The main reasons are higher thermal stability, which results in reduced carbon buildup on hot surfaces, and in refrigeration compressors, miscibility characteristics of the oil and the refrigerant.

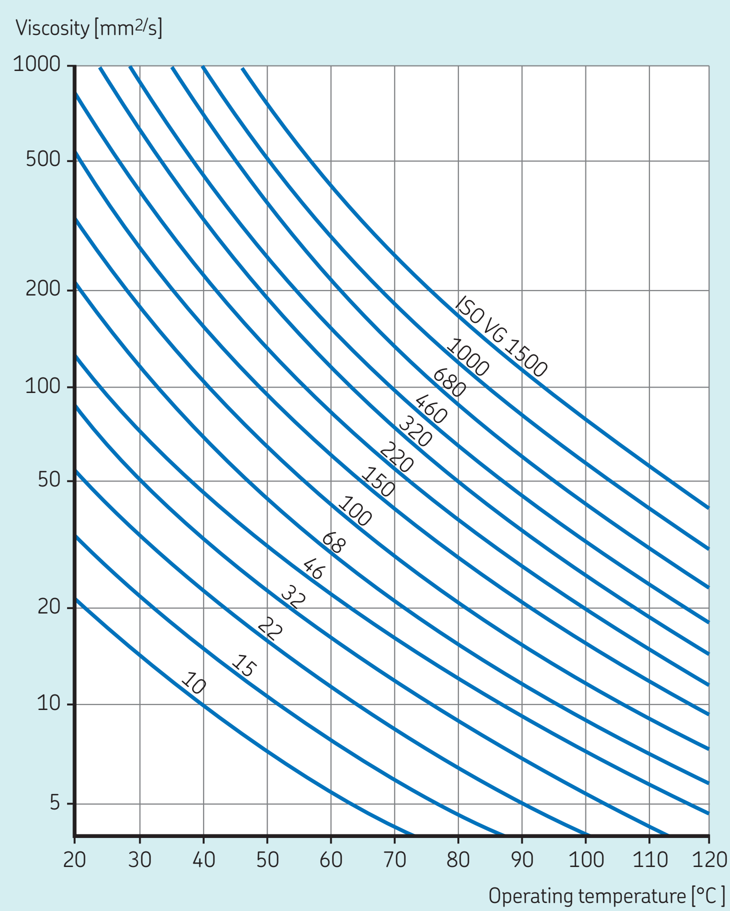
Table 1

Viscosity classification in accordance with ISO 3448

Viscosity grade	**Kinematic viscosity limits at 40 °C** mean	min.	max.
–	mm^2/s		
ISO VG 2	2,2	1,98	2,42
ISO VG 3	3,2	2,88	3,52
ISO VG 5	4,6	4,14	5,06
ISO VG 7	6,8	6,12	7,48
ISO VG 10	10	9,00	11,0
ISO VG 15	15	13,5	16,5
ISO VG 22	22	19,8	24,2
ISO VG 32	32	28,8	35,2
ISO VG 46	46	41,4	50,6
ISO VG 68	68	61,2	74,8
ISO VG 100	100	90,0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1 000	1 000	900	1 100
ISO VG 1 500	1 500	1 350	1 650

Diagram 1

Viscosity-temperature diagram for ISO viscosity grades
(Mineral oils, viscosity index 95)



For example, miscibility with HCFC-134a is the reason why polyolester (POE) oils are used with this refrigerant. Synthetic oils have a higher viscosity index than mineral oils and therefore a higher viscosity at elevated temperatures (**diagram 2**). The viscosity index of synthetic oils can be in the range of 130 to 200. Oils with a high viscosity index have less viscosity decrease with increase in temperature. Common synthetic oil types used in compressors are the polyalphaglycol (PAG), polyalphaolefin (PAO), and polyolesters (POE). Synthetic lubricants have lower effective viscosity in the rolling contacts compared to mineral oils. This is due to a lower viscosity increase under the pressure of the rolling contact. This difference is characterised by the pressure-viscosity coefficient (α) of the lubricant. The net effect of higher viscosity and lower pressure coefficient is that there is no significant influence on oil film thickness. The rated viscosity ν_1 required for adequate lubrication is obtained from **diagram 3**. The actual operating viscosity is obtained from **diagram 1**, **page 27**. The operating viscosity selected for screw compressors should be greater than ν_1, this means κ should be greater than 1,0 and preferably greater than 1,5. The lubricant viscosity should not be too great since this causes excessive bearing friction and heat. The ratio κ of actual viscosity to ν_1 should be used for calculation of the SKF rating bearing life, **page 31**.

$$\kappa = \frac{\nu}{\nu_1}$$

where

κ = viscosity ratio

ν = actual operating viscosity of the lubricant [mm²/s]

ν_1 = rated viscosity of the lubricant depending on the bearing mean diameter and rotational speed [mm²/s]

Diagram 2

Approximate temperature conversion

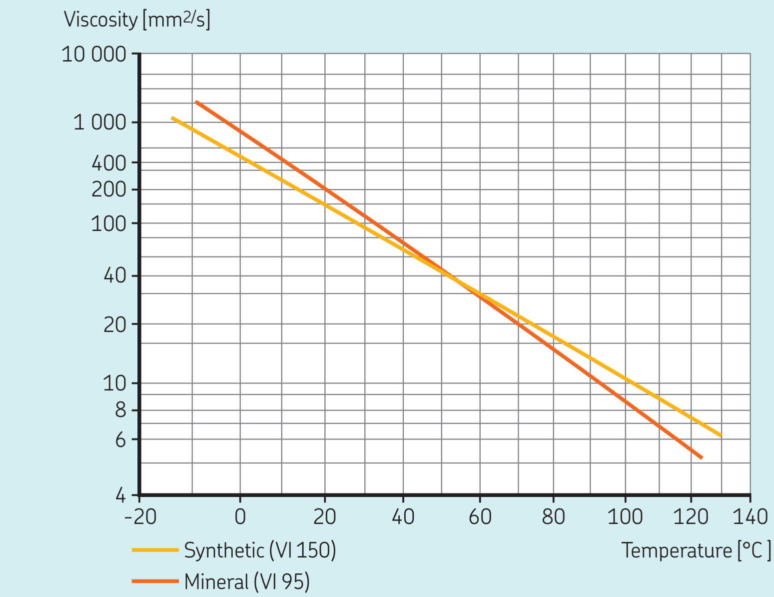
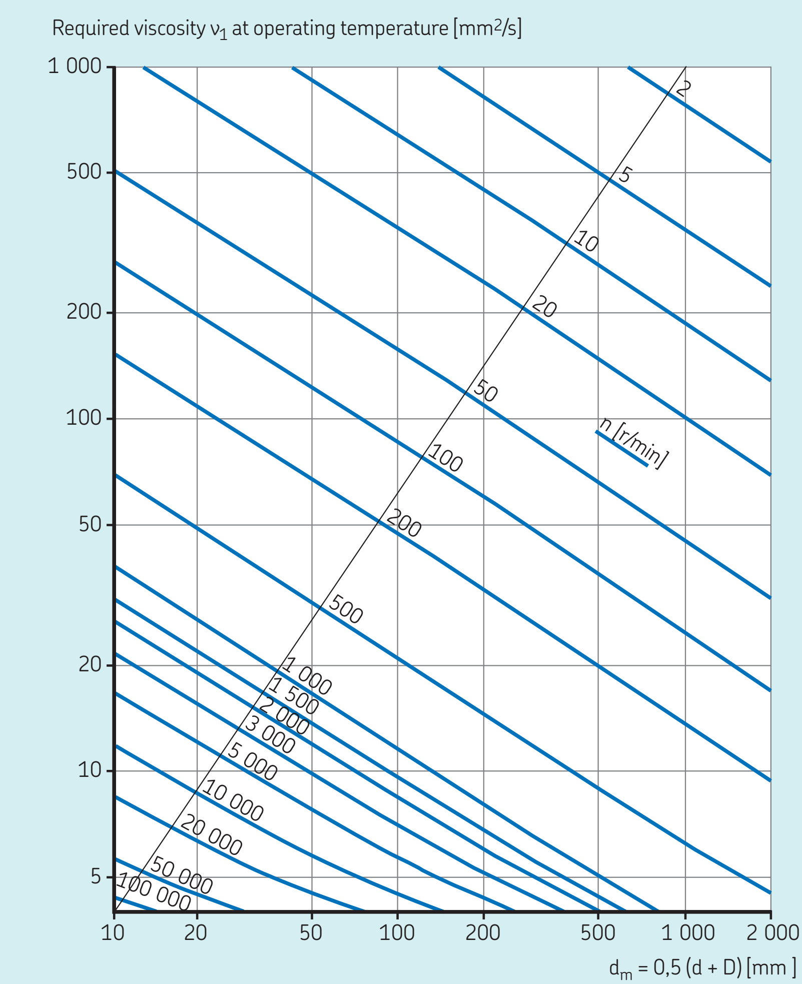


Diagram 3

Estimation of the minimum kinematic viscosity ν_1 at operating temperature



The required oil viscosity is also determined from that needed for injection between the rotors. ISO VG 46 and VG 68 are typical for airscrew compressors. In refrigerant and natural gas compressors, the oil is diluted with dissolved liquefied gases, this reduces the viscosity. For refrigerants, the viscosity of the oil/refrigerant mixture can be determined from two graphs, first for the dilution [%] at equilibrium at a given temperature and pressure, the second (**diagram 4**) for the viscosity of the mixture at the dilution [%] from the first graphs. Such graphs, Daniel plots, are available from oil suppliers.

Oil dilution in natural gas compressors occurs primarily from light hydrocarbon liquids (natural gasoline). The viscosity of the mixture can be calculated by the method described under *Natural and sour gas compressors*, **page 88**.

Because of the lower pressure-viscosity coefficient of dissolved liquefied gases and differences in compressibility and molecular weights, the viscosity of the mixture cannot be used for calculation of κ without an adjustment, **diagrams 5** and **6**.

With the adjustment factor (multiplication factor), these effects can be taking into account.

Diagram 4

Kinematic viscosity of an oil refrigerant mixture as a function of temperature and dilution weight %

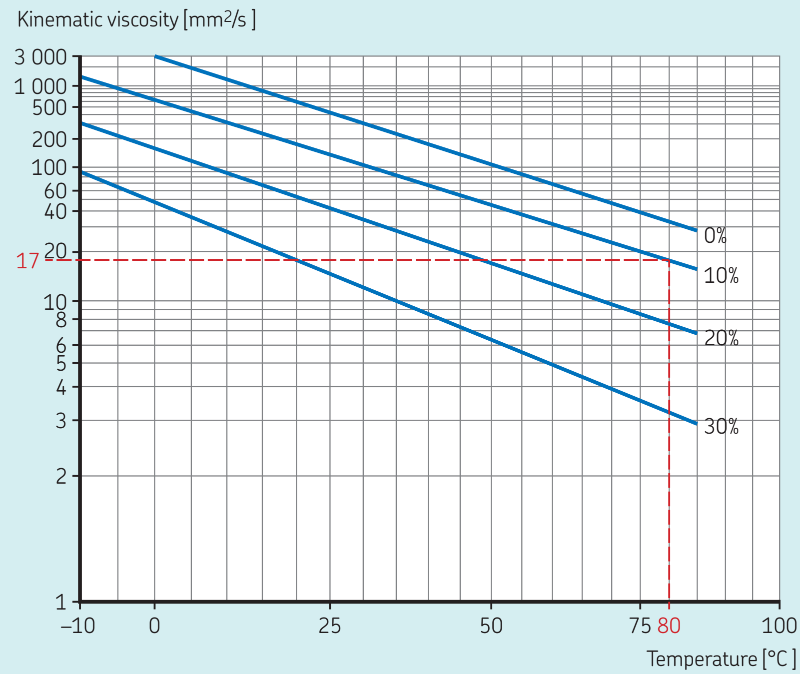


Diagram 5

Correction factor for κ vs dilution weight %, POE oil, ISO VG 68 and R134a

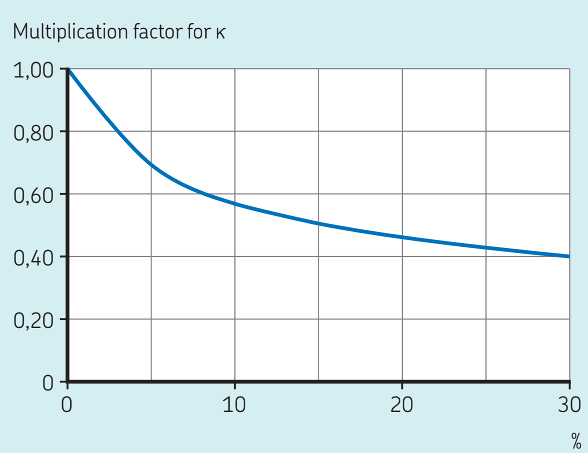
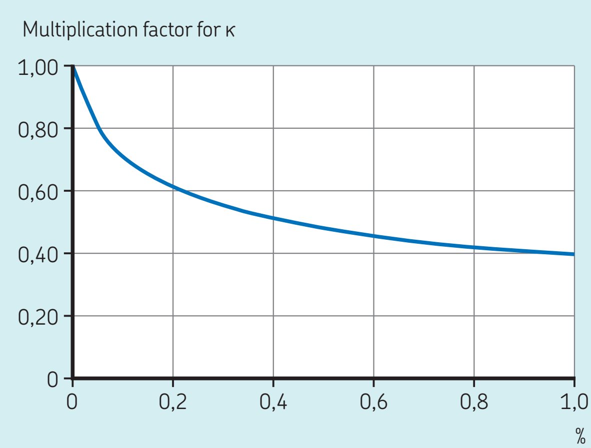


Diagram 6

Correction factor for κ vs dilution volume % of gasoline in mineral oil



Filtration

Definitions

Compressor lubrication systems often include filters to remove solid particle contaminants. Filtration is needed to clean the system of contaminants that can damage the bearings. Bearing life is affected by the cleanliness of the compressor lubricant. Filters are rated according to size and a β ratio. The β ratio defines the efficiency of the filter to remove particles of a given size.

The $\beta_{x(c)}$ ratio is defined as follows:

$$\beta_{x(c)} = \frac{\text{No. of particles entering filter}}{\text{No. of particles leaving filter}}$$

where

x = particle size (c), microns [µm] based on the automatic particle counting method, calibrated in accordance with ISO 11171

For example, a rating of $\beta_{3(c)} = 200$ means that for every 200 particles that enter the filter, only 1 particle greater than 3 microns leaves the filter. The finer the filter, the more quickly the system is cleaned of contamination. When a fine filter is used it may be necessary to have coarser prefilter, to avoid frequent clogging of the fine filter.

Filter selection

The filter specification in compressors is typically in the range of $\beta_{3(c)} = 200$ to $\beta_{40(c)} = 75$. Finer filters increase bearing life, but the degree of increase depends also on the viscosity ratio κ and the bearing load intensity P_u/P. If the κ value is high, a change to a finer filter can give significant improvement in bearing life. If κ is low, a finer filter cannot compensate for the poor lubrication condition and the benefit may be questionable. In such a case it may be more effective to increase the bearing size. For evaluation of filter specifications, SKF computer programs are available which take into account all of the above mentioned factors.

Oil changes

Oil change frequency depends on operating conditions, temperature, lubricant quality, bearing cleanliness, and the cleanliness of the lubrication system. Mineral oils oxidize and require shorter replacement intervals compared to synthetic oils. The actual replacement interval is specified by the compressor manufacturer or determined by oil sample analyses. Longer intervals between replacements are possible at lower operating temperatures. Compressors are sometimes equipped with oil coolers to remove heat. Synthetic oils are more resistant to deterioration from exposure to high temperatures and have long service life.

Lubricants may require more frequent replacement if contamination is present. The gas in the compressor (refrigerant, air, natural gas, etc.) can dilute the lubricating oil and have a significant influence on the operating viscosity (*Natural and sour gas compressors*, **page 88**).

Lubrication, oil-injected compressors

The oil flow rate is determined by the differential pressure at steady state operation. This flow rate is usually more than needed to lubricate the bearings, while the flow rate at start up may be marginal. To ensure oil availability at start-up, the drain holes in the bearing housing should be located between the 7 and 8 o'clock positions, so that a pool of oil is left in the housing at shutdown. The drain holes should be adequately sized to eliminate the risk of flooding the housing and bearings during operation.

The required oil flow can be calculated from the balance of heat input by friction and heat removed by convection and oil flow. This can be done with SKF computer programs. Please contact SKF application engineering service for information. The minimum flow rate required can be estimated from the formula:

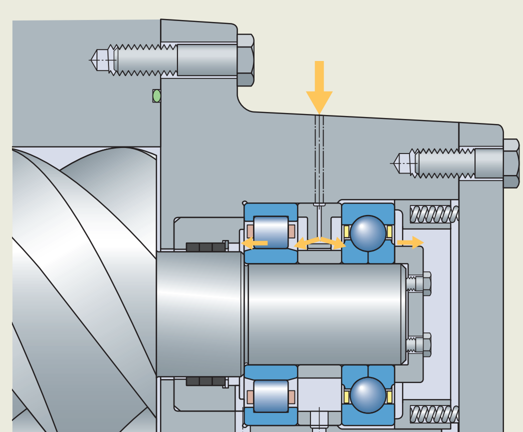
$$Q_{min} = 0,00005\ B\ D$$

where

Q_{min} = minimum required oil flow [l/min]
B = the total bearing width [mm]
D = bearing outer diameter [mm]

Fig. 30

Oil flow path for oil jet lubrication at nd_m greater than 800 000



Lubrication by oil jet

In dry running (oil free) air compressors operating at ndm speeds in excess of 800 000, oil is supplied to the bearings by oil jet (**fig. 30**). The compression cavity is sealed from the bearing housing.

The oil jet must be directed axially between the bearing inner ring and cage (**fig. 30**). The oil viscosity and flow rate must be suitable to lubricate and cool the bearings. The jet speed should be 15 to 20 m/s to avoid deflection from the air curtain of the bearing. The shaft and housing temperature of dry air compressors is high because the heat of compression is not removed by oil injection. Depending on speed and temperature, the bearings are lubricated by a synthetic oil with an ISO VG 32 to ISO VG 68 viscosity. The lubrication system should include fine filtration. The oil flow to the bearings should not be too great, as that would cause excessive friction and temperature rise. In order to drain off excessive oil, SKF recommends drainage on both sides of the bearings. The oil flow required should be determined by testing, whereby the flow and jet speed are varied and the temperature rise of the oil is measured.

The oil flow path upstream of the jet hole, or nozzle, should be designed to minimize pressure drop before the nozzle. The pressure drop should take place in the nozzle. The jet velocity can be calculated by the Bernoulli equation:

$$v = \sqrt{\frac{2\,P}{\rho}}$$

where
v = jet velocity [m/s]
P = pressure drop across the nozzle [Pa]
ρ = Oil density [kg/m^3]

NOTE: This equation is valid only for short nozzles where the effect of viscosity is small. The oil flow can be calculated by the equation:

$$Q = 0{,}05\,\mu\,v\,d^2$$

where
Q = oil flow [l/min]
μ = nozzle coefficient, typically 0,6 to 0,8
v = jet velocity [m/s]
d = nozzle diameter [mm]

The nozzle should have a minimum 1 mm diameter to prevent the risk of being clogged by debris. As a rule of thumb, a 1,0 mm nozzle diameter and a pressure across the nozzle of 100 kPa will produce a jet velocity of 15 m/s and a flow of 0,5 l/min.

Bearing rating life

Three different life calculations can be used for selection of bearings and calculation of bearing life:

- basic rating life
- SKF rating life
- SKF advanced fatigue life

SKF basic rating life

$$L_{10h} = \frac{1\,000\,000}{60\,n}\left(\frac{C}{P}\right)^p$$

where
L_{10h} = basic rating life (at 90% reliability) [operating hours]
n = rotational speed [r/min]
C = basic dynamic load rating [kN]
P = equivalent dynamic bearing load [kN]
p = exponent of the life equation
– for ball bearings, p = 3
– for roller bearings, p = 10/3

This is the classic bearing life formula, which is still used by many engineers today. The only inputs are basic dynamic load rating, speed and (equivalent) bearing load.

It should be noted that if angular contact ball bearings in face to face or back to back are arrangement is used, the effect of centrifugal forces on the balls must be accounted for in the equivalent load. The axial component of centrifugal forces on the balls in one bearing becomes external load on the other, and vice versa. The effect of centrifugal force should be accounted for if the nd_m speed is greater than 450 000 and can be calculated with SKF computer programs.

Since the formula was first introduced in the 1940s, the basic dynamic load rating has been increased several times, due to improvements in bearing steel, manufacturing methods and optimization of internal geometry.

Care should also be taken when selecting the bearing load used in the calculation. If the compressor will be operating at different loads and speeds at different times, then load cycles should be used in the calculation. If instead, the maximum load and speed are used, the selected bearings may be too big, with too much friction and higher risk of failure from overheating and/or light load skidding.

For calculation of load cycle life, the loads and speeds should be represented as a histogram, where the load and speed are constant for each block. The life for each block is then calculated and the total life is the sum of the life in each block, using the formula on **page 37** in chapter *Bearing life at varying operating conditions*.

SKF rating life

In the SKF rating life method, not only are basic dynamic load rating, speed and (equivalent) load accounted for, but also lubrication, contamination and fatigue load limit (P_u). The fatigue load limit is the load at which a bearing life is infinite, if operating with full film lubrication and in a clean environment.

These influencing parameters are accounted for in the a_{SKF} factor, which can be found in **diagram 11**, **page 34** and **diagram 12**, **page 35**. This method of calculation has been standardized in ISO 281, and is much more precise in prediction of rating life, compared to the basic rating life formula.

The effects of lubrication and contamination on bearing life are very strong. Care should therefore be taken when selecting the lubrication factor κ and the contamination factor η_c, please refer to **diagram 1**, **page 27** and **diagram 3**, **page 28** for determination of the lubrication factor κ and **diagrams 7** to **10** and **table 3** for η_c. If actual lubrication and contamination conditions are not favourable, the actual life may be much shorter than calculated. Please note that if a_{SKF} = 1,0 the SKF rating life is the same as the basic rating life.

$$L_{10mh} = a_1\, a_{SKF}\, L_{10h} = \frac{1\,000\,000}{60\,n}\left(\frac{C}{P}\right)^p$$

where

- a_1 = life adjustment factor for reliability (→ **table 2**, values in accordance with ISO 281)
- a_{SKF} = SKF life modification factor (→ **diagrams 11** and **12**, **page 34** and **35**)
- L_{10h} = basic rating life (at 90% reliability) [operating hours]
- n = rotational speed [r/min]
- C = basic dynamic load rating [kN]
- P = equivalent dynamic bearing load [kN]
- p = exponent of the life equation
  - for ball bearings, p = 3
  - for roller bearings, p = 10/3

Table 2

Values for life adjustment factor a_1

Reliability	**Failure probability** n	**SKF rating life** L_{nm}	**Factor** a_1
%	%	million revolutions	–
90	10	L_{10m}	1
95	5	L_{5m}	0,64
96	4	L_{4m}	0,55
97	3	L_{3m}	0,47
98	2	L_{2m}	0,37
99	1	L_{1m}	0,25

Diagram 7

Contamination factor η_c for
- circulating oil lubrication with on line filters
- solid contamination level –/19/16 in accordance with ISO 4406
- filter rating $\beta_{40(c)}$ = 75

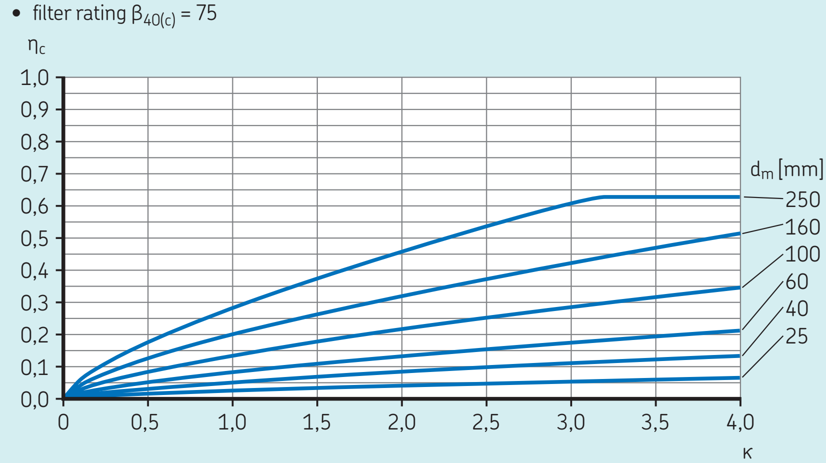
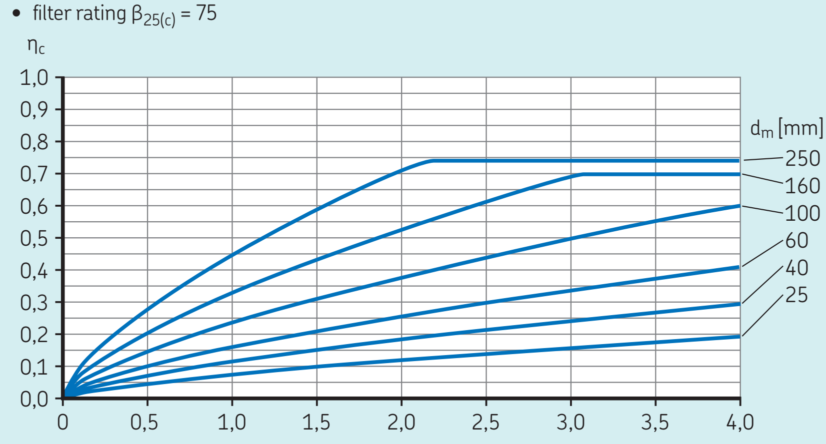


Diagram 8

Contamination factor η_c for
- circulating oil lubrication with on line filters
- solid contamination level –/17/14 in accordance with ISO 4406
- filter rating $\beta_{25(c)}$ = 75



SKF Advanced fatigue calculation

In the basic rating and SKF rating life method, the bearing load capacity is characterized by one constant – the basic dynamic load rating C. The external forces are used to calculate a equivalent load which is based on a series of assumptions, e.g. parallel bearing ring displacements, no misalignment, half of the rolling elements loaded. In the SKF Advanced fatigue calculation, the contact force and stress of each rolling element to raceway contact is calculated and the bearing is viewed as a system. The lives of the individual contacts are calculated and added statistically to yield the life of the bearing. In this life calculation, the effects of lubrication and contamination are also taken into account, whereby the η_c and a_{SKF} factors are different in each contact. For calculation of the contact loads and stresses, the effect of clearance, misalignment, speed, load distribution among the rolling elements and the centrifugal forces on the rolling elements are taken into consideration. This calculation requires computer analysis. The effect of centrifugal forces is especially important for angular contact ball bearings at higher speeds, typically 450 000 and higher, see **page 49**, and **table 1**, **page 79**. The SKF Advanced fatigue calculation is the most accurate calculation method that exists to predict rating life for SKF bearings.

Diagram 9

Contamination factor η_c for
- circulating oil lubrication with on line filters
- solid contamination level –/15/12 in accordance with ISO 4406
- filter rating $\beta_{12(c)} = 200$

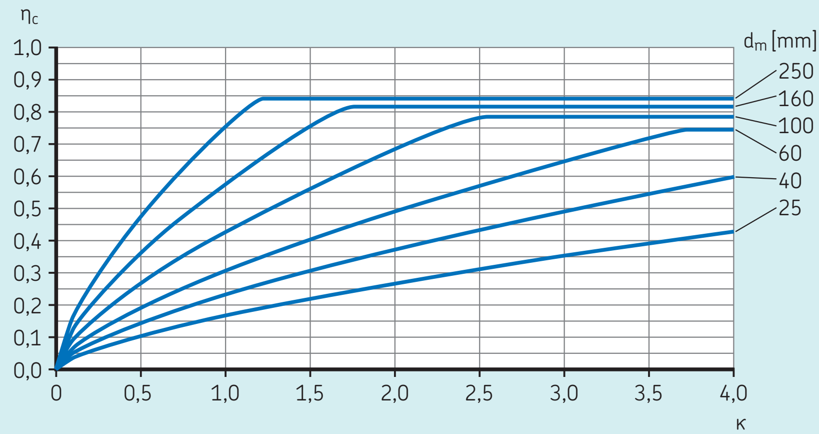


Diagram 10

Contamination factor η_c for
- circulating oil lubrication with on line filters
- solid contamination level –/13/10 in accordance with ISO 4406
- filter rating $\beta_{6(c)} = 200$

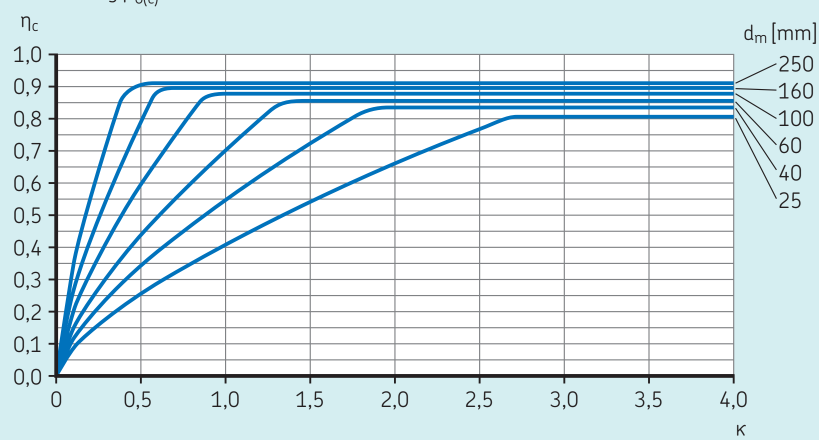


Table 3

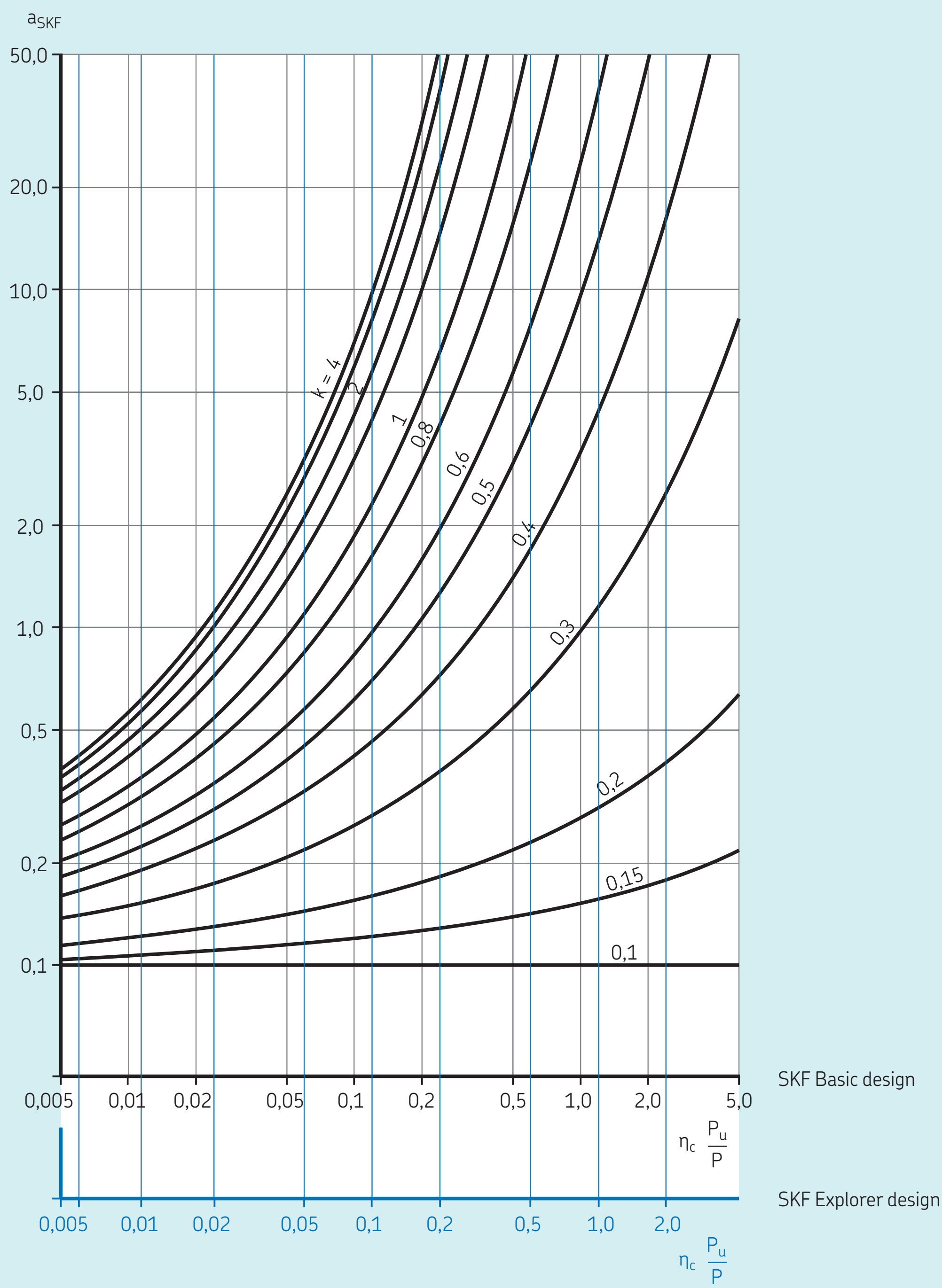
For filter rating $\beta_{3(c)} = 200$, it can be estimated that in-line filters will produce cleanliness class 11/8 and the following η_c values can be used if the contaminates are predominately hard metallic particles:

Contamination factor η_c values for:
- circulating oil lubrication with on line filters
- filter rating $\beta_3 = 200$
- $\kappa > 1,0$

d_m	η_c
mm	–
25 – 40	0,9 – 0,93
40 – 100	0,93 – 0,95
100 – 250	0,95 – 0,96

Diagram 11

Factor a_{SKF} for radial ball bearings



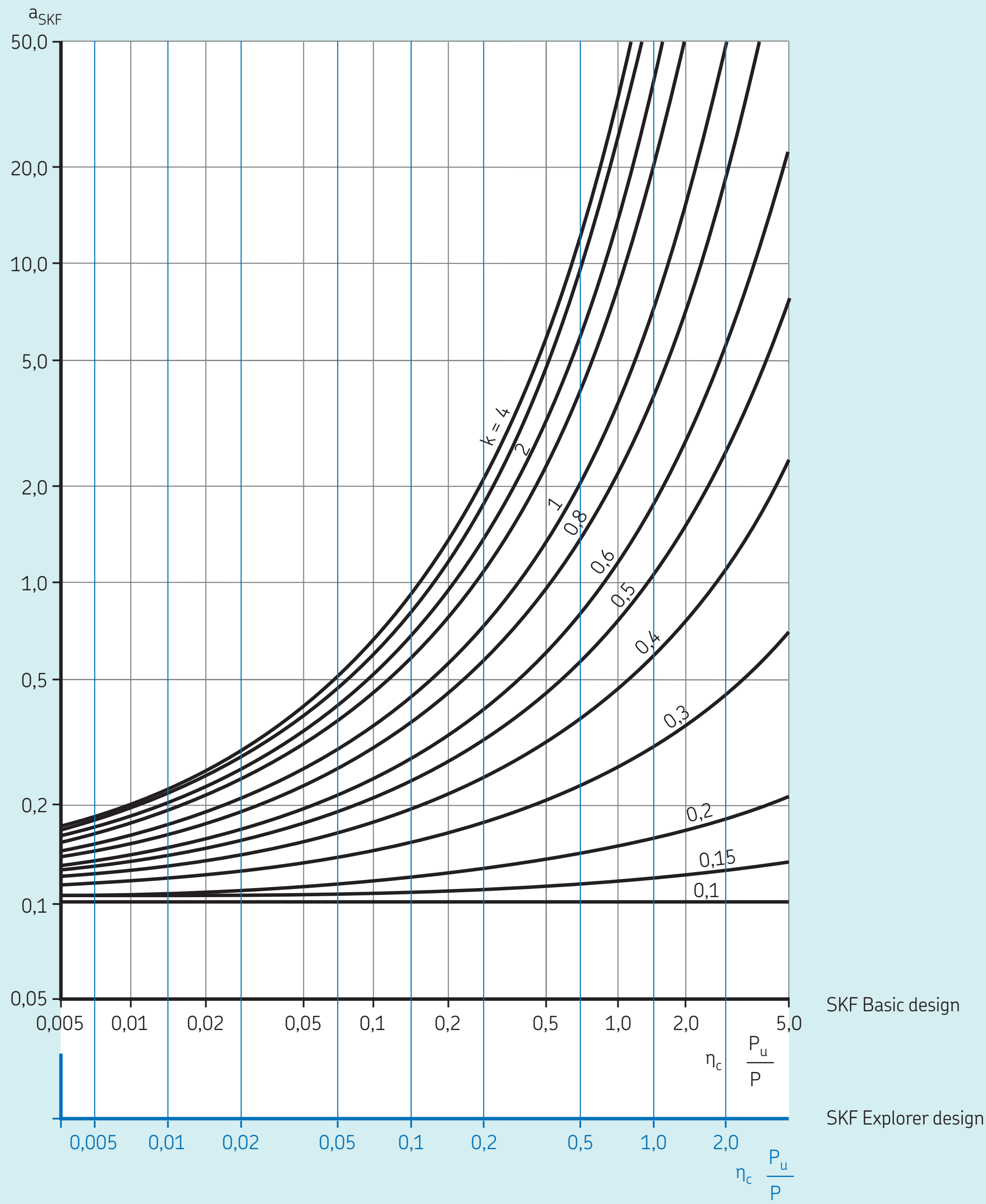
If κ > 4, use curve for κ = 4.

As the value of η_c (P_u/P) tends to zero, a_{SKF} tends to 0,1 for all values of κ.

2

Diagram 12

Factor a_{SKF} for radial roller bearings



If κ > 4, use curve for κ = 4.

As the value of η_c (P_u/P) tends to zero, a_{SKF} tends to 0,1 for all values of κ.

Examples of oil cleanliness and dilution of oil by refrigerant on SKF bearing rating life

Operating conditions	
Bearing	7306 BE
Speed [r/min]	2 950
nd_m	150 450
Axial load [kN]	3,5
Radial load [kN]	0,1
Fatigue load limit, P_u [kN]	0,9
Equivalent bearing load, P [kN]	2,03
P_u/P	0,44
Dynamic load capacity, C [kN]	35,5
Basic L_{10} bearing [h]	30 200

Air compressor lubricated with (pure) oil

Lubrication conditions	
Oil viscosity grade	ISO 46
Operating temperature [°C]	70
Operating viscosity	15 from **diagram 1**, **page 27**
Minimum adequate viscosity, ν_1	9,7 from **diagram 3**, **page 28**
Lubrication condition, κ	1,5

ISO cleanliness class	η_c	a_{SKF}	SKF L_{10} rating life	Filter size	
–	–	–	–	µm	β
–/19/16	0,09	1,4	42 300	40	75
–/17/14	0,19	3,1	93 600	25	75
–/15/12	0,63	9,7	293 000	12	200
–/13/10	0,84	50	> 1 000 000	6	200
–/11/8	0,93	50	> 1 000 000	3	200

Refrigerant compressor lubricated by a mixture of oil and refrigerant

Lubrication conditions	
POE oil viscosity grade	ISO 150
Operating temperature [°C]	80
Refrigerant	R134a
Oil dilution by refrigerant	10%
Oil/mixture viscosity [mm²/s]	17 from **diagram 4**, **page 29**
Correction factor	0,57 from **diagram 5**, **page 29**
Adjusted viscosity [mm²/s]	9,7
Minimum adequate viscosity, ν_1	9,7 from **diagram 3**, **page 28**
Adjusted κ	1,0

ISO cleanliness class	η_c	a_{SKF}	SKF L_{10nmh} rating life	Filter size	
–	–	–	–	µm	β
-/19/16	0,07	0,75	22 700	40	75
-/17/14	0,14	1,4	42 300	25	75
-/15/12	0,27	3,0	90 600	12	200
-/13/10	0,63	13	393 000	6	200
-/11/8	0,93[1)]	32	> 1 000 000	3	200

η_c from **diagrams 7** to **10**, **page 32** and **33**
a_{SKF} from **diagram 11**, **page 34**
1) From **table 3**, **page 33**

Bearing system life

It is sometimes of interest to calculate the system life for all bearings in the compressor. In bearing system life, the L_{10} life of all bearings is considered, such that if one bearing fails, then the system of bearings is considered to have failed.

System life, L_{10s} can be calculated as follows:

$$\frac{1}{L_{10s}\,e_s} = \sum_{i=1}^{N} \frac{1}{L_{10i}\,e_i}$$

where

L_{10s} = system life, in hours or revolutions

L_{10i} = life of an individual bearing, in hours or revolutions

N = number of bearings in the system

e_i = 10/9 for ball bearings; 1,35 for roller bearings

e_s = equivalent exponent for all bearings in a system of ball and roller bearings, this can be estimated with the following formula:

$$e_s = \frac{\sum_{i=1}^{N} e_i}{N}$$

Whenever possible, the loading used to evaluate the selection of the bearing should be based on the duty cycle in which the compressor will be operated. The duty cycle considers the period or percentage of time that the compressor will operate at a given load, speed, temperature, etc. condition.

Bearing life at varying operating conditions

If the operating conditions are continually changing, bearing life cannot be calculated without first reducing the duty cycle to a limited number of simpler load cases (**diagram 13**). In situations of continuously changing load, each different load level can be accumulated and the load spectrum represented as a histogram of constant load blocks. Each block should characterize a given percentage or time-fraction during operation. Note that heavy and normal loads consume bearing life at a faster rate than light loads. Therefore, it is important to have peak loads well represented in the load diagram, even if the occurrence of these loads is relatively rare and limited to a few revolutions. In case of peak loads, the static safety factor need to be verified (SKF catalogue *Rolling bearings*).

Within each duty interval, the bearing load and operating conditions can be averaged to some constant value. The number of operating hours or revolutions expected from each duty interval, showing the life fraction required by that particular load condition, should also be included. Therefore, if N_1 equals the number of revolutions required under the load condition P_1, and N is the expected number of revolutions for the completion of all variable loading cycles, then the cycle fraction $U_1 = N_1/N$ is used by the load condition P_1, which has a calculated life of L_{10m1}. Under variable operating conditions, bearing life can be rated using:

$$L_{10m} = \frac{1}{\frac{U_1}{L_{10m1}} + \frac{U_2}{L_{10m2}} + \frac{U_3}{L_{10m3}} + \dots}$$

where

L_{10m} = SKF rating life (at 90% reliability) [million revolutions]

$L_{10m1}, L_{10m2}, \dots$ = SKF rating lives (at 90% reliability) under constant conditions 1, 2, … [million revolutions]

$U_1, U_2, \dots$ = life cycle fraction under the conditions 1, 2, …
Note: $U_1 + U_2 + \dots U_n = 1$

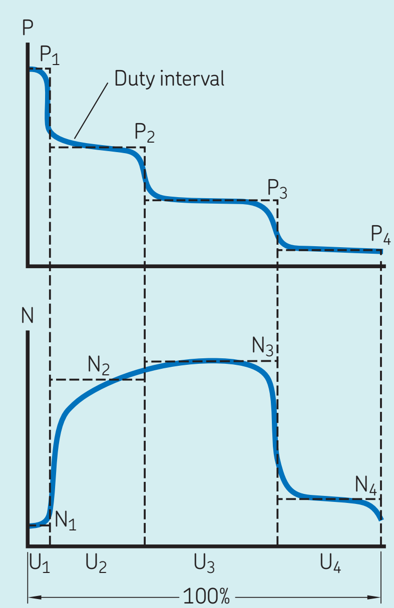
The use of this calculation method depends very much on the availability of representative load diagrams for the application.

Requisite bearing life

The requisite screw compressor bearing lives have traditionally been a basic rating life of 30 000 to 60 000 hours. This has worked well for the industry, but there are reasons to challenge this practice. To optimize the bearing size selection, a more detailed consideration of the lubrication condition and cleanliness is a must. The SKF rating life method allows this. The SKF Advanced fatigue life method allows to even consider more aspects like centrifugal loads on angular contact ball bearings. Care should be taken not to over dimension bearings to achieve long rating life, as this could result in higher friction and risk of light load skidding. It is recommended to calculate with both basic and state of the art life formula, such as the SKF Rating life and the SKF Advanced Fatigue calculation.

Diagram 13

Duty intervals with constant bearing load P and number of revolutions N



Operational limitations

Limiting speeds

In the SKF catalogue *Rolling bearings*, two speed ratings are defined; the reference speed and the limiting speed. The reference speed is based on the balance of frictional heat generated by the bearing and heat dissipated from the bearing housing. The reference speed listed in the SKF catalogue *Rolling bearings* is based on the SKF friction model and derived from thermal equilibrium in oil bath operating conditions under standardized operating and cooling conditions defined in ISO 15312. Since screw compressors operate with circulating oil lubrication, where the frictional heat is removed by oil flow, the reference speed is not of practical use.

The limiting speed is based on mechanical and kinematic considerations such as cage strength, mass and guidance. For high speed operations such as in dry air compressors, ring guided cages are used because with ring guidance the cage is better centered in the bearing than with rolling element guidance. For angular contact ball bearings, the contact angle is also important. The minimum load needed for angular contact ball bearings increase with speed and contact angle. If the speed is too high and the contact angle too large, the minimum load required becomes so high that the bearing cannot carry any external loads. By using ceramic balls (hybrid bearings) the minimum load required is reduced and the operating speed can be increased. (**page 57**). Running precision, or runout, is another speed limiting factor. At high speed the balance of the rotor is affected by the runout of the bearing and the shaft seat. This means that bearings with reduced runout tolerance should be used and the run out tolerance of the shaft seat should be reduced (**table 2**, **page 83**). The limiting speed is listed in the SKF catalogue *Rolling bearings*, but application parameters such as bearing clearance, shaft and housing fits and lubrication must also be considered. At very high speeds, oil jet lubrication should be used (**page 31**)

Because of the circulating oil and oil jet lubrication schemes used in screw compressors, and optimizing the application parameters and bearing design details, the limiting speed may be exceeded. Please contact SKF application engineering for advice. The recommended maximum speeds are often, as in this book, referred to in terms of ndm speed, which is the inner ring speed multiplied with the bearing mean diameter (**page 12**).

Table 4

Cage and seal material compatibility with gases
The table shows maximum continuous operating temperatures [°C]

Material	SKF suffix	Air	NH3	Sour gas	Sweet gas	FR
PA 66	P, TN, TN9	120	80	80	110	110
PEEK	PH, PHA	160	120	120	160	160
Brass, pressed	Y	+	–	–	+	+
Brass, machined	M, MA, ML	+	+	+	+	+
Steel, pressed	J	+	o	o	+	o
Steel, machined	F, FA	+	+	+	+	+
Nitrile (NBR)	RS1	100	o	o	80	o
FKM	RS2	230	–	–	200	o

Sour gas = gas containing H_2S
FR = fluorinated refrigerants
+ = possible
– = not possible
o = not recommended

Limiting temperatures

The maximum allowable bearing operating temperature is limited by either lubrication conditions or bearing materials. The higher the temperature, the lower the lubricant viscosity and the thinner the lubricant film separating the bearing raceway and the rolling elements. The thermal stability of the lubricant is also reduced at high temperature (**page 27**). Lubrication considerations limits the bearing temperature of most screw compressors to 100 to 120 °C.

The materials used in bearing components also have temperature limits. The limiting criteria is different for different material types. For polymer cages the criteria is ageing, which makes the cage brittle and weak. The presence of aggressive media can accelerate the ageing process and this lowers the temperature limit (**table 4**). The presence of aggressive media also limits the temperature for metal cages (**table 4**).

The steel in the bearing rings and rolling elements also have temperature limits, depending on the heat treatment. At elevated temperatures structural transformations occur in the steel, affecting the dimensions of the components. In general, the higher the hardness, the lower the temperature at which the transformations occur. To control dimensional change at elevated temperature, SKF bearings are heat stabilized to different temperatures depending on the bearing type (**table 6**).

Bearing compatibility with gases

Exposure of the bearings to gases within the compressor may require the selection of certain specific materials for the bearing seals and cages. The gases may adversely affect the materials, making them age or become ineffective. Experience and tests made at the SKF Engineering and Research Centre in the Netherlands have established the suitability of common bearing materials for use in the presence of certain gases, see **table 4**. The gaskets, paints, and other seals within the compressor may also be affected by the composition of the gases. In some cases, the possibility that the gas could contain liquid contaminates such as water, may also dictate the suitability of the material.

For more details, please also refer to the section on gas compressors in this handbook.

Table 5

Values of ISO standard tolerance grades

Nominal dimension		**Tolerance grades**		
		IT5	IT6	IT7
>	≤	max.		
mm		µm		
10	**18**	8	11	18
18	**30**	9	13	21
30	**50**	11	16	25
50	**80**	13	19	30
80	**120**	15	22	35
120	**180**	18	25	40

Table 6

Dimensional stability

Bearing type	**Temperature [°C]**
Deep groove ball bearings	120
BE and AC angular contact ball bearings	150
ACD and CE angular contact ball bearings	120
Four point contact ball bearings	150
Cylindrical roller bearings	150
Needle roller bearings	120
Tapered roller bearings	120

Shaft and housing tolerances

Recommended shaft tolerances

SKF recommendations for ball and roller bearing shaft tolerances in twin screw compressor applications appear in **table 7**.

Tolerance grades for bearing seats and values of ISO standard tolerance grades can be found in **table 5**, **page 39** and **table 8**.

Housing tolerances – bearings taking radial loads

ISO K6 is the generally recommended housing tolerance for cylindrical and needle roller bearings supporting radial loads. This tolerance results in a transition fit between the bearing outer ring and housing. This allows for easy assembly and prevents the outer ring from creeping in the housing bore. It also minimizes the total clearance between the rotor shaft and the housing.

This is important for precise positioning of the rotor. For easier mounting, it is also possible to use K7 or J7 tolerance but then the clearance between the outer ring and housing is greater and there is a greater risk of ring creep (**table 1**, **page 79** and **table 2**, **page 83**).

Table 7

Recommended shaft and housing tolerances

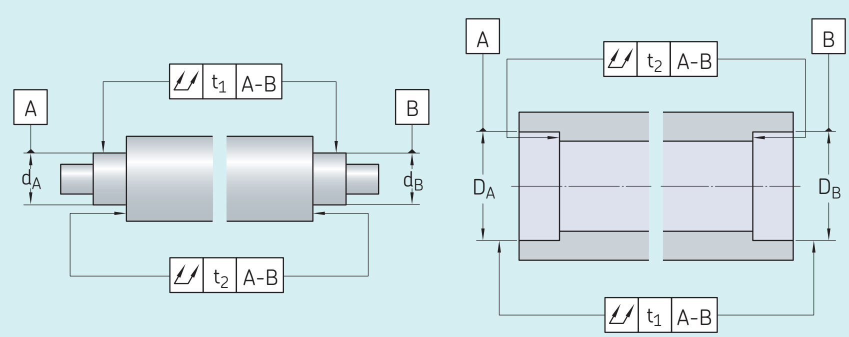
Bearing type	Bearing bore diameter, d 10 ≤ d ≤ 50	 50 < d ≤ 100	 100 < d ≤ 150
–	mm		
Ball bearings, taking axial load only	j5, j6 RF	j5, j6 RF	j5, j6 RF
Deep groove ball bearings, taking radial load only	k5, k6 H6, H7	k5, k6 H6, H7	m5, m6 H6, H7
Cylindrical roller bearings, taking radial load only	m5 K6, K7, J7	n5, n6 K6, K7, J7	p5, p6 K6, K7, J7
Tapered roller bearings taking combined loads	m5, m6 K6, K7, J7	n5, n6 K6, K7, J7	p6 K6, K7, J7
Tapered roller bearings taking axial load only	k5, k6 RF	k5, k6 RF	k5, k6 RF

Recommended bearing shaft seat surface finish Ra = 0,8 µm
Recommended bearing housing seat surface finish Ra = 1,6 µm
For cylindrical roller bearings, when using n5 and p6 tolerances, radial internal clearance greater than Normal is necessary.
RF = radially free
Envelope requirement (symbol Ⓔ from ISO 14405-1) not written but applies.

Table 8

Tolerance grades for bearing seats

Shaft seat



Dimensional tolerance grade	**Geometrical tolerance grades** Radial run-out t_1	 Axial run-out t_2
IT5	IT4/2	IT4
IT6	IT5/2	IT5
IT7	IT6/2	IT6

Housing tolerances – bearings taking axial loads only

Bearings taking axial loads only, in combination with a separate radial bearing, including reverse thrust bearings, should be radially free in the housing. The housing bore should be 1 or 2 mm larger in diameter than the bearing outer ring. If such bearings are constrained from moving radially, by too heavy clamping force, they will take a combination of radial and axial loads, and the radial bearing will take less or no radial load.

Bearings taking axial loads only should be clamped with a spring to ensure sufficient but not excessive clamping see **page 52**, *Clamping of outer rings*. The bearing rings can also be slotted (N_1 or N_2 suffix) and fitted with an anti-rotation pin, mounted in the housing or housing end cover. This is common with four point and single row angular contact ball bearings. The spring can function either as a clamping spring or as a preload spring, **fig. 12** and **fig. 13**, **page 22**.

Ball bearings in twin screw compressors

Ball bearings are used in twin screw compressors, either as main thrust bearings , or as reverse thrust bearings, or backup bearings to control axial clearance or preload in a set of bearings. Angular contact ball bearings with large contact angles are well suited as main thrust bearings because of their higher axial load capacity, compared to bearings with smaller contact angles. Angular contact ball bearings can also be used as backup bearings. Deep groove ball bearings are also a good choice as spring loaded reverse thrust bearings. If the loads are light, deep groove ball bearings can also be used to take radial loads.

Deep groove ball bearings

Deep groove ball bearings have deep uninterrupted raceway groves with close osculation (conformity) with the balls. This design enables the bearing to take radial and axial loads in both directions. In order to make bearing assembly possible, there are fewer balls compared to the same size angular contact ball bearing.

Deep groove ball bearings have limited radial and axial load capacity compared to angular contact ball bearings and cylindrical roller bearings. When high load capacity is not needed, deep groove ball bearings can be a good and economical alternative.

The main application is as reverse thrust bearings, but deep groove ball bearings can also be used as the suction end radial bearing, if the radial load is light. Bearings with steel or polyamide cages can be used in most cases. The bearing internal clearance depends on the application but is typically greater than Normal (C3 suffix). **Table 1, page 44** lists the values of unmounted radial clearance. The initial clearance in a bearing is reduced by interference fits and if the shaft and inner ring operate with a higher temperature than the outer ring and housing.

The bearing must support at least a minimum radial or axial load for satisfactory operation. The recommended minimum radial load can be determined according to the SKF catalogue *Rolling bearings*. An axial load can be provided by a spring.

Recommended minimum axial spring force

The recommended minimum axial spring force is determined as follows:

$$F = k\,0{,}01\,d$$

where
F = minimum spring force [kN]
d = bearing bore diameter [mm]

Cages

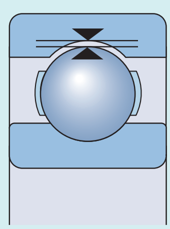
Depending on their design, series and size, SKF deep groove ball bearings are fitted with one of the cages shown in **table 2**. The standard stamped steel cage is not identified in the bearing designation. For compatibility of gases see **table 4**, **page 38**.

Allowable misalignment

Operation under misalignment reduces bearing life. The maximum allowable misalignment is 2 to 4 minutes, but limiting misalignment to less than 1,5 minute is recommended in screw compressors. For detailed recommendations see **table 1**, **page 79**. Misalignment is caused by bending of the rotor, positioning tolerances of the radial bearing housings and clearances.

Table 1

Radial internal clearance of deep groove ball bearings



Bore diameter		**Radial internal clearance**							
d		C2		Normal		C3		C4	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.
mm		µm							
18	**24**	0	10	5	20	13	28	20	36
24	**30**	1	11	5	20	13	28	23	41
30	**40**	1	11	6	20	15	33	28	46
40	**50**	1	11	6	23	18	36	30	51
50	**65**	1	15	8	28	23	43	38	61
65	**80**	1	15	10	30	25	51	46	71
80	**100**	1	18	12	36	30	58	53	84
100	**120**	2	20	15	41	36	66	61	97
120	**140**	2	23	18	48	41	81	71	114
140	**160**	2	23	18	53	46	91	81	130
260	**180**	2	25	20	61	53	102	91	147
180	**200**	2	30	25	71	63	117	107	163

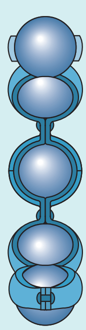
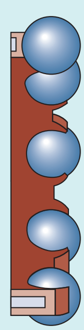
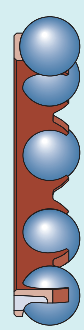
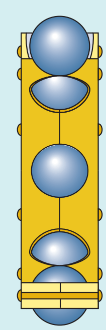
The axial clearance is 5 to 15 times the radial, depending on bearing series and radial clearance class

Hybrid deep groove ball bearings

SKF deep groove ball bearings are also available as hybrid bearings, with ceramic balls made of bearing grade silicon nitride (Si_3N_4). See chapter *Hybrid angular contact ball bearings*, **page 57**.

Table 2

Cages for deep groove ball bearings

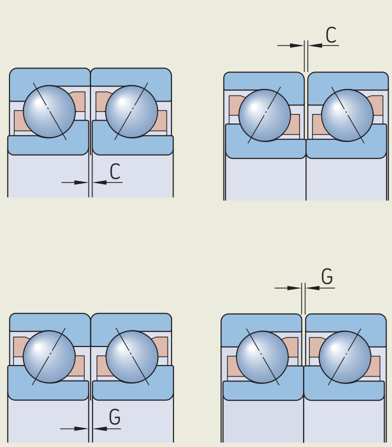
	Steel cages	Polymer cages		Brass cages
				
Cage type	Riveted, ball centered	Snap-type, ball centered		Riveted, ball, outer ring or inner ring centered
Material	Stamped steel	PA66, glass fibre reinforced	PEEK, glass fibre reinforced	Machined brass
Suffix	–	TN9	TNH	M, MA or MB

Single row angular contact ball bearings

Single-row angular contact ball bearings are the most commonly used ball bearings in twin screw compressors. They support pure axial loads or combined radial and axial loads. The most important features of this bearing type are its high axial load capacity combined with a high speed rating. Single-row angular contact ball bearings operating with a small clearance or a light preload provide good axial positioning accuracy of the rotor. For oil injected screw compressors, the most commonly used SKF single row angular contact ball bearings are of the 72 BE and 73 BE series which have 40° contact angle. Also used are series 72 AC and 73 AC with 25° contact angle.

Fig. 1

Clearance / Preload class



For dry air compressors, which operate at nd_m speeds over 800 000, bearings of 70 and 72 series with 15° and 25° contact angles are used, 70 ACD, 70 CD, 72 ACD, 72 CD. See **table 1**, **page 79** and **table 2**, **page 83**.

Universally matchable single row angular contact ball bearings

Bearings for universal matching are intended to be used in double and triple arrangements in screw compressors. Single row angular contact bearings are usually mounted in face-to-face arrangements to facilitate easy adjustment of the rotor end clearance and for easy dismounting.

The standoff between the rings is manufactured to close tolerances. When two bearings are mounted immediately adjacent to each other, a given specified internal clearance or preload, for example CB or GA, will be produced without shimming (**fig. 1**). Bearings mounted in tandem will distribute the load equally between the bearings (**fig. 2**).

Universally matchable bearings can also be used as single bearings. Most bearings belong to the SKF Explorer design and as such have smaller dimensional and geometrical tolerances, increased load carrying capacity and speed capability.

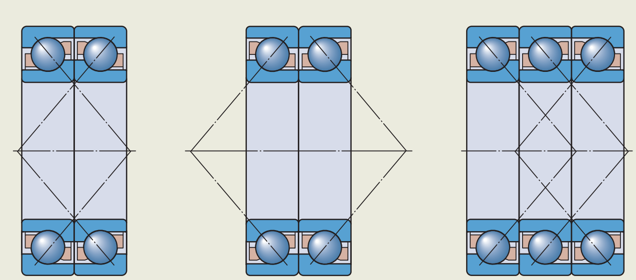
Universally matchable bearings in the 72 B(E) and 73 B(E) series are identified by the suffix CA, CB or CC for internal clearance or GA, GB or GC for preload.

Arrangements of universally matchable bearings used to support axial loads, provide accurate axial positioning of the rotor. Bearing sets made for preload (e.g.: GA suffix) provide axial positioning on both directions without clearance. Angular contact ball bearings combined with a spring loaded reverse thrust bearing also provide accurate axial positioning, but the reverse axial force capability is limited by the spring force. If the axial load of the compressor is heavy, two axial load carrying bearings can be arranged in tandem, with a third bearing as a backup bearing in **fig. 16**, **page 23**.

The standard SKF bearings available for universal matching have the CB or GA suffix, e.g. 7310 BECB or 7310 BEGA. 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 the arrangements shown in **fig. 2**. The GA suffix denotes that the bearing is universally matchable, but a pair of these bearings will have a light preload when mounted in any of the arrangements shown in **fig. 2**.

Tables 4 and **5**, **page 48** lists the values of unmounted axial clearance and preload for the universally matchable bearings. The initial bearing clearance or preload is only 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. SKF Explorer universally matchable bearings are produced with P6 precision class dimensional tolerances and P5 precision class geometrical tolerances as standard.

Fig. 2



Cages

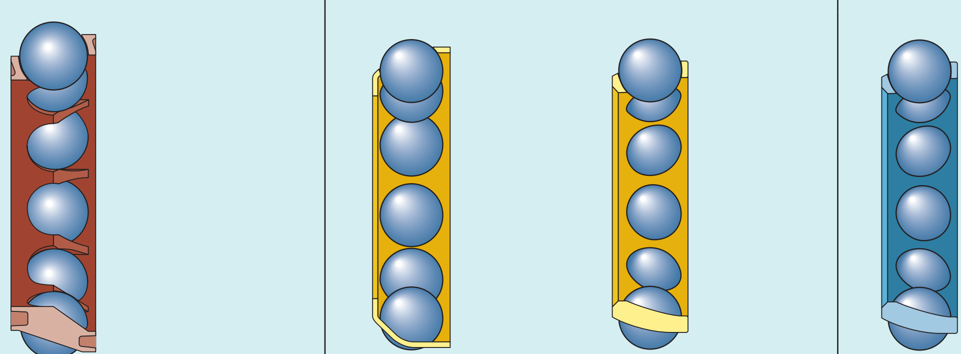
The SKF BE and AC design bearings are produced as standard with five optional cages (**table 3**); the glass fibre reinforced polyamide 6,6 cage (P suffix), the PEEK cage (PH suffix), the pressed brass (Y suffix), and the machined brass cage (M suffix). A limited number of bearings are also available with a machined steel cage (F suffix). For compatibility of gases see **table 4**, **page 38**.

Allowable misalignment

Operation under misalignment reduces bearing life. The maximum allowable misalignment is 2 to 4 minutes, but it is recommended to keep the misalignment lees than 1,5 minute in screw compressors. Misalignment is caused by bending of the rotors, positioning tolerances of the radial bearing housings and clearances.

Table 3

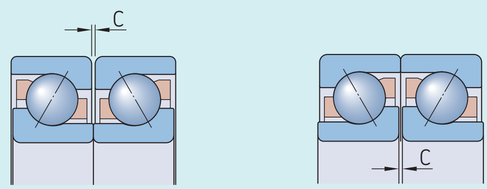
Cages for single row angular contact ball bearings



	Polymer cages		Brass cages		Steel cages
Cage type	Window-type, ball centred		Window-type, ball centred	Window-type, ball centred	
Material	PA66, glass fibre reinforced	PEEK, glass fibre reinforced	Stamped brass, stamped steel	Machined brass	Machined steel
Suffix	P	PH	Y	M	F

Table 4

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

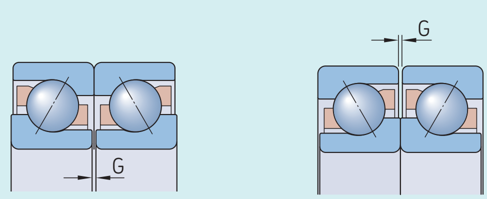


Bore diameter d over	incl.	Axial internal clearance class CA min.	max.	CB min.	max.	CC min.	max.	G min.	max.
mm		µm							
–	**18**	5	13	15	23	24	32	–	–
18	**30**	7	15	18	26	32	40	–	–
30	**50**	9	17	22	30	40	48	–	–
50	**80**	11	23	26	38	48	60	–	–
80	**120**	14	26	32	44	55	67	–	–
120	**160**	17	29	35	47	62	74	26	76
160	**180**	17	29	35	47	62	74	20	72
180	**250**	21	37	45	61	74	90	20	72
250	**280**	–	–	–	–	–	–	20	72

Axial clearance for BE and AC are the same.
Radial clearance for a pair of BE is ~0.85 axial clearance.
Radial clearance for a pair of AC is ~0.46 axial clearance.

Table 5

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



Bore diameter d over	incl.	Preload class GA min.	max.	GB min.	max.	GC min.	max.
mm		µm					
10	**18**	+4	–4	–2	–10	–8	–16
18	**30**	+4	–4	–2	–10	–8	–16
30	**50**	+4	–4	–2	–10	–8	–16
50	**80**	+6	–6	–3	–15	–12	–24
80	**120**	+6	–6	–3	–15	–12	–24
120	**180**	+6	–6	–3	–15	–12	–24
180	**250**	+8	–8	–4	–20	–16	–32

Bearing minimum axial load

For satisfactory operation, an angular contact ball bearing must carry a certain minimum axial load. At increased speed, centrifugal forces on the balls will cause a change in the contact angle between the inner and outer raceways, (**fig. 3**). The differences in contact angle will cause sliding, in particular at the inner ring, which increase friction and can cause adhesive wear. At increased speed, the gyroscopic moment acting on the balls will increase and cause spinning motion of the balls, causing additional sliding on the raceways. Adequate axial load minimizes the risk of sliding, both from centrifugal force and gyroscopic moment. With effective lubrication, the negative effect of insufficient axial load is reduced. At increased speed, axial displacement of the bearing inner ring relative to the outer ring will also occur (**diagram 1**). The axial displacement can cause the compressor rotor end clearance to decrease. If the axial force is insufficient, this may result in the rotor coming into contact with the housing. The magnitude of the ring displacement varies with rotor speed. Insufficient axial load also reduces traction of the balls on the raceways and can cause ball skidding and variation in ball orbital speed from ball to ball. This will result in increased loads on the cage and possibly cause cage damage. The minimum required axial load for satisfactory operation of angular contact ball bearings can be calculated from the following equation:

$$F_{a\,min} = A\left(\frac{n}{1\,000}\right)^2$$

where

$F_{a\,min}$ = minimum required axial load [kN]
A = minimum load factor
n = rotational speed [r/min]

Values of minimum load factor A are listed in **table 6**, **page 50**.

Bearings with small contact angles are better suited for high speed and light axial load applications because of their lower requirement for axial load.

During operation, the minimum required axial load can be internally maintained by limiting the internal axial clearance. This can be accomplished with a backup bearing. With small axial clearance, the balls are loaded by centrifugal force against the raceways with low variation of contact angles (**diagram 2**, **page 50**). As the axial clearance increases, so does the difference in the inner and outer ring contact angles. This allows increased internal sliding. The minimum axial load can also be maintained by spring preloading.

Fig. 3

Contact angles

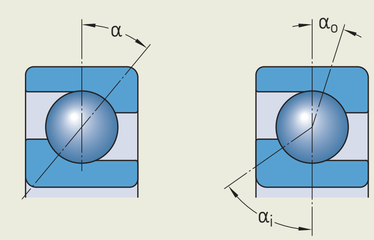


Diagram 1

Axial rotor displacement vs axial load
SKF 7312 BEGAP

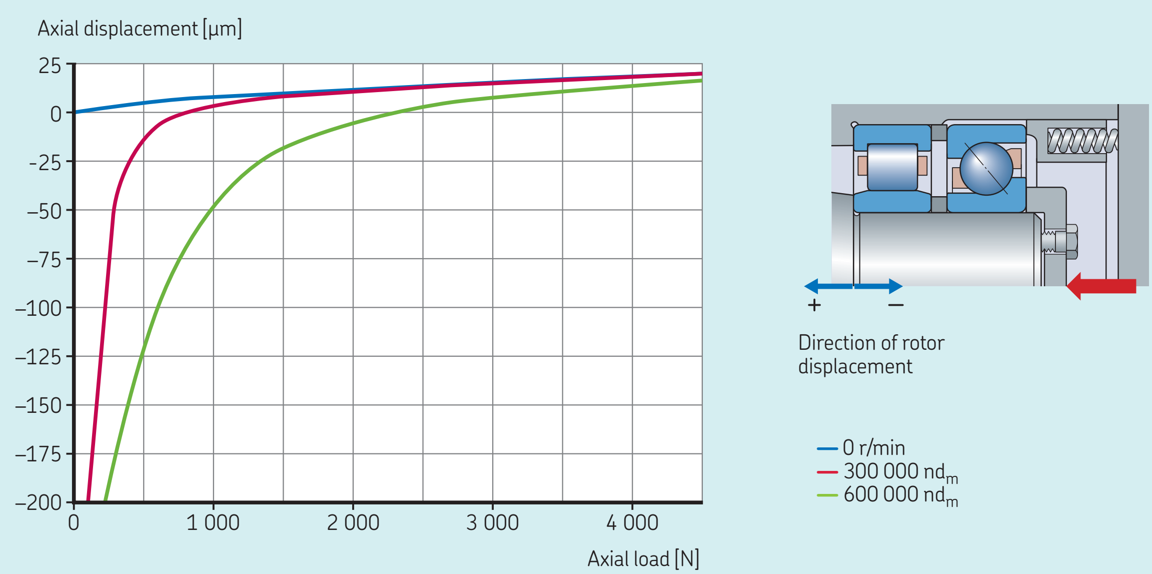


Table 6

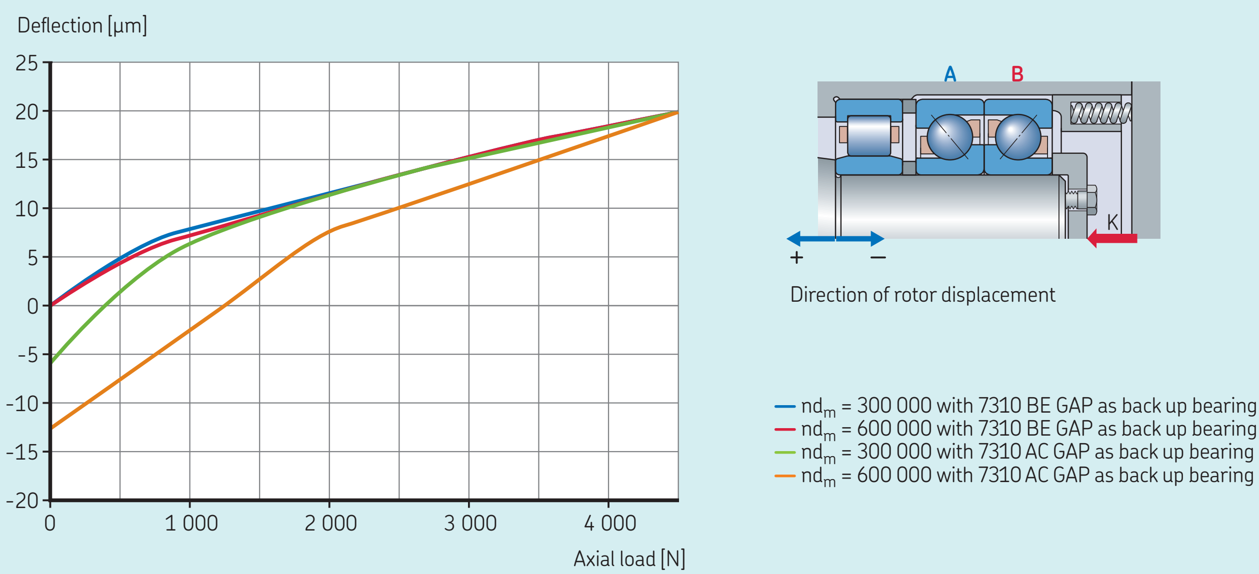
SKF single row angular contact ball bearing, minimum axial load factor

Size	Bore diameter	A factor 72 BE	73 BE	72 AC	73 AC	70 ACD	72 ACD	70 CD	72 CD
–	mm								
01	12	0,000283	0,000536	0,000114	0,000212	0,000054	0,000092	0,000020	0,000034
02	15	0,000383	0,000907	0,000156	0,000363	0,000080	0,000156	0,000029	0,000057
03	17	0,000625	0,00141	0,000254	0,000563	0,000138	0,000254	0,000051	0,000093
04	20	0,00113	0,00191	0,000461	0,000771	0,000250	0,000460	0,000092	0,000168
05	25	0,00159	0,00391	0,000656	0,00158	0,000352	0,000655	0,000130	0,000241
06	30	0,00377	0,0074	0,00155	0,003	0,000801	0,00135	0,000295	0,000496
07	35	0,00674	0,0111	0,00277	0,00453	0,001078	0,00249	0,000398	0,000915
08	40	0,0102	0,0173	0,00419	0,00707	0,00138	0,00311	0,000511	0,00115
09	45	0,012	0,0268	0,00496	0,0109	0,00246	0,00495	0,000911	0,00182
10	50	0,014	0,0418	0,00584	0,017	0,00287	0,00583	0,00106	0,00215
11	55	0,022	0,0574	0,00917	0,0234	0,00504	0,00915	0,00187	0,00337
12	60	0,0344	0,0846	0,0143	0,0345	0,00576	0,0110	0,00213	0,00405
13	65	0,0478	0,112	0,0199	0,0456	0,00653	0,0153	0,00242	0,00563
14	70	0,0529	0,145	0,022	0,0592	0,0100	0,0174	0,00371	0,00641
15	75	0,0599	0,171	0,025	0,0701	0,0112	0,0196	0,00417	0,00725
16	80	0,0801	0,216	0,0334	0,0887	0,0171	0,0278	0,00635	0,0103
17	85	0,114	0,27	0,0474	0,111	0,0190	0,0384	0,00706	0,0142
18	90	0,149	0,333	0,062	0,137	0,0264	0,0598	0,00979	0,0221
19	95	0,191	0,406	0,0796	0,167	0,0292	0,0729	0,0108	0,0269
20	100	0,239	0,63	0,0997	0,258	0,0320	0,0933	0,0119	0,0344
21	105	0,302	0,669	0,126	0,274	0,0429	0,118	0,0159	0,0434
22	110	0,353	0,906	0,147	0,371	0,0557	0,133	0,0207	0,0493

For hybrid bearing A factors, multiply the values in the table by 0,4.

Diagram 2

Axial rotor displacement vs axial load
Two universally matchable angular contact ball bearings in face-to-face arrangement
Active bearing A : 7310 BEGAP
Back up bearing B : 7310 BEGAP or 7310 ACGAP



Tandem-mounted single row angular contact ball bearings

Single row angular contact ball bearings arranged in tandem are commonly used in screw compressors to support high axial load on the male rotor. The tandem mounted bearings can be positioned adjacent to a cylindrical roller bearing, which supports the radial load (**figs. 12**, **page 21** and **16**, **page 23**).

The angular contact ball bearings are mounted radially free (RF) in the housing and have a radial clearance of 1 or 2 mm with the housing bore. With this arrangement, it is appropriate to determine the rating life of each angular contact bearing individually rather than as a set. The SKF Explorer universally matchable bearings are made to precise tolerances and can distribute the axial load equally between the bearings in tandem arrangements.

For tandem bearings supporting combined axial and radial load, the bearing rating life is determined according to the SKF catalogue *Rolling bearings*. For SKF bearings mounted in tandem and supporting only axial load, the bearing rating life is determined using the external axial load distributed equally among the bearings, and using the basic dynamic load rating C for one single row angular contact ball bearing.

Below is an example of the life calculation for bearings arranged in tandem, supporting only axial load:

2 SKF 7310 BEGAP arranged in tandem
F_r = 0 [kN]
F_a = 10 [kN]
n = 3 600 [r/min]
C = 75 kN for 1 bearing

$P = 0{,}35\ F_r + 0{,}57\ F_a$ for $F_a/F_r > 1{,}14$

Load per bearing
$F_a/2 = 5$ kN

$P = 0{,}57 \times 5\ 000 = 2{,}85$ kN

$$L_{10h} = \frac{10^6}{60\ n}\left(\frac{C}{P}\right)^P = \frac{10^6}{60 \times 3\ 600}\left(\frac{74\ 100}{2\ 850}\right)^3$$

= 84 400 h

Bearing preload

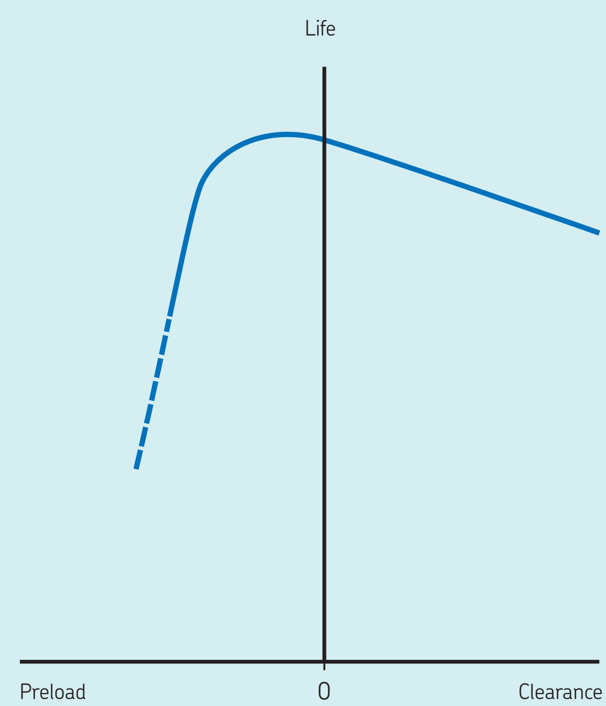
The purpose of bearing preload in angular contact ball bearings is to:

- prevent light load skidding and sliding
- control contact angles
- improve load distribution
- increase bearing stiffness
- improve shaft positioning accuracy

In twin screw compressor applications, bearing preload is used in angular contact ball bearings for all these reasons. Preload can increase the fatigue life of a bearing by improvement of internal load distribution (**diagram 3**), but too much preload can reduce bearing fatigue life. Preloaded bearings are more sensitive to misalignment and incorrect mounting than bearings with clearance.

Diagram 3

Relationship between bearing life and preload/clearance



NOTE: the dotted line indicates potentially unstable working conditions

Preload by axial displacement

Diagram 4 shows the static load deflection for two preloaded angular contact ball bearings. This diagram is typical of 40° bearings arranged either back-to-back or face-to-face. The preload force F_0 in this example is produced by the elastic deflection δ_0 of the bearings when clamped against one another. When an axial load K is applied to the shaft, the load in one bearing (**A**) increases and the load in the other (**B**) decreases. Bearing A is denoted the "active" bearing. The deflection in the active bearing increases, while the deflection in the inactive bearing decreases. The inactive bearing is the backup bearing.

Diagram 5, shows load-deflection for a pair of rotating preloaded bearings. Under rotation, the preload force increases due to centrifugal forces on the balls. The force of the backup bearing is never fully reduced to zero due to centrifugal forces and will always add force to the active bearing. At increased speeds, spinning of the balls due to gyroscopic moment will occur if the residual preload in the inactive bearing is less than the minimum required axial load, $F_{a\ min}$.

Preload by springs

It is also possible to preload bearings by using springs. In that case bearing **B** is loaded by a spring, acting on the outer ring (see **diagram 6**). There must be a clearance fit in the housing to allow the spring loaded bearing outer ring to displace axially. There must also be a spacer between the inner rings of bearing **A** and bearing **B** to assure that there is a gap between the outer rings, and that the spring force cannot be transmitted to the outer ring of bearing A. The spring force on bearing **B** will add force to bearing **A**. **Diagram 6** also shows the load deflection for an angular contact ball bearing, spring preloaded by a deep groove ball bearing.

An advantage with spring preloading is that the force on the preloaded backup bearing is always the same and equal to the spring load. When used as reverse thrust bearing the spring force limits the amount of reverse force that can be taken. If the reverse force exceeds the spring force, then the rotor will displace and rub against the housing.

Spring loading can also be used to reduce force on the main (fixed) thrust bearing, by applying the spring force on the suction side bearing in the opposite direction to the main (gas) load.

The criteria for selection of spring force are given in **table 7**.

Clamping of outer rings

To assure separation of radial and axial force between the radial and the thrust bearings, there should be a radial gap between the outer ring of the thrust bearing and the housing. This way the rotor shaft can displace radially in the radial bearing without adding radial force to the thrust bearings. In twin screw compressors, the preferred arrangement of angular contact bearings is face to face. This means that the outer rings must be clamped axially to control bearing clearance. The axial clamping force must not be too large, since that would prevent radial displacement of the thrust bearings. It must also not be too light. For this reason springs are used to provide a controlled clamping force. The criteria for selecting the clamping force are given in **table 8**.

Table 7

Criteria for selecting axial preload spring force

1. The spring preload should be greater than the maximum possible reverse axial load from the rotor.
2. The spring preload should be greater than the minimum load required for the spring preload bearing.
3. The spring preload should give an acceptable L_{10} life of the spring preload bearing.
4. The spring preload should be great enough to prevent any reverse displacement of the main thrust bearings due to centrifugal forces generated in the main thrust bearings.
5. The addition of the spring preload to the load of the main thrust bearings should be acceptable for the L_{10} life requirement of the main thrust bearings.

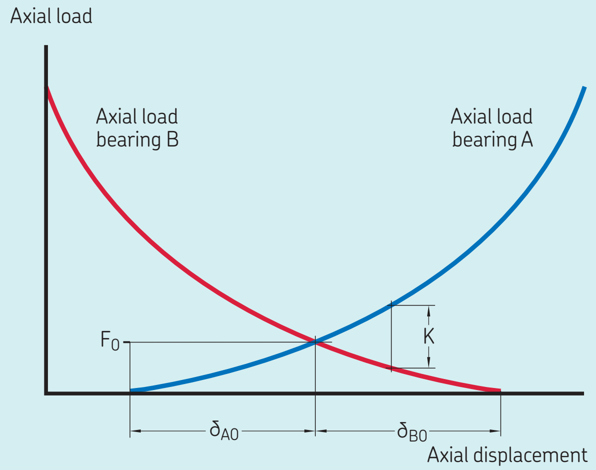
Table 8

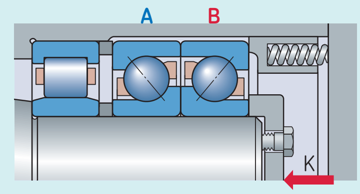
Criteria for selecting clamping spring force

1. The clamping spring force should be greater than the maximum possible reverse axial load from the rotor.
2. The clamping spring force should be greater than the induced axial load from centrifugal force in the thrust bearings.
3. For bearings that should not be radially free and for bearing inner rings, the clamping force should be less than $C_0/4$, where C_0 is the lowest static load capacity of a bearing which is part of a set of bearings. This criterion is based on deformation/distortion of bearing rings which can occur if the clamping force is too high.
4. The clamping force should not be larger than necessary to limit radial loading of thrust bearings.

Diagram 4

Preload diagram for two angular contact ball bearings in face to face arrangement

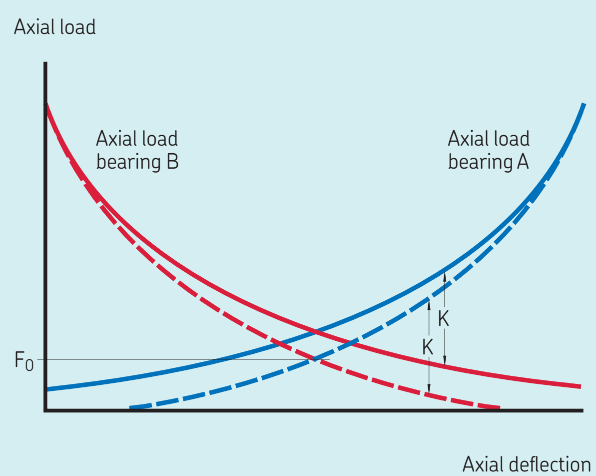


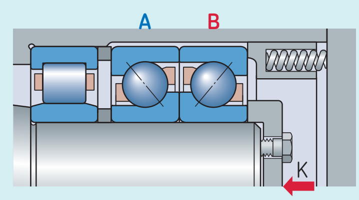


F_0 preload force
K external force
δ_{A0}, δ_{B0} initial preload displacement

Diagram 5

Preload diagram for two angular contact ball bearings in face to face arrangement

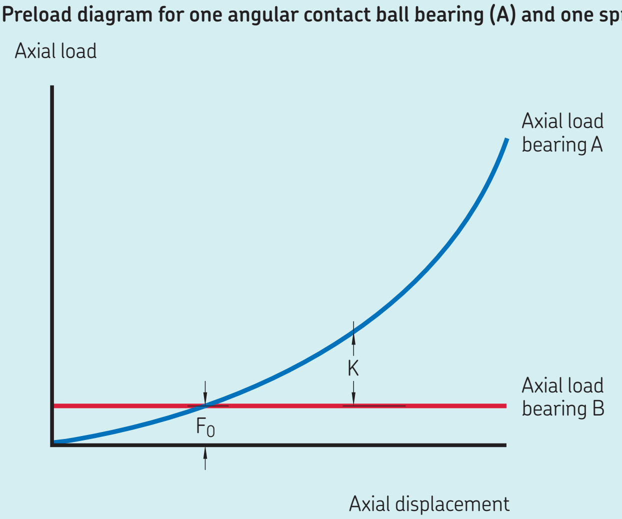


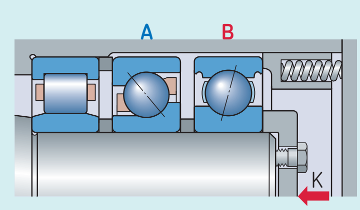


At speed: blue and red curve
At zero speed: blue and red dotted curve
F_0 preload force
K external force

Diagram 6

Preload diagram for one angular contact ball bearing (A) and one spring loaded deep groove ball bearing (B)





F_0 preload force = spring force
K external force

3

Four-point contact ball bearings

Four-point contact ball bearings (QJ prefix) are primarily used in oil flooded refrigeration and high speed dry air compressors to support axial load only. The bearings have a two piece inner ring and an outer ring with two raceways (**fig. 4**). This design makes possible a large contact angle, small axial and radial clearance and a large number of balls. The contact angle is 35°. The cage is outer-ring guided made of machined brass or PEEK (**table 9**). The above mentioned features make the bearing well suited for high speed applications.

If loaded with pure radial load, the bearing will have two contact points on the inner ring and two on the outer ring. This explains why it is called a four-point contact ball bearing, but it is not intended to be used that way. In screw compressors the bearing should be used for thrust loads only. The thrust load can act in both directions (**fig. 4b** and **4c**). Because it cannot be preloaded and must always operate with clearance, functionally it is similar to but not the same as a pair of single-row angular contact ball bearings. The inner rings must be clamped axially to ensure correct axial clearance and the side face of the inner ring is recessed to avoid warping if clamped too high up on the side face.

At high speed and light axial load, the balls are forced radially outwards by centrifugal force contacting both raceways of the outer ring and the load-carrying inner ring raceway.

If the axial load increases, the load on the outer ring increases on one raceway and decreases on the opposite one, but there is little change in contact angle. The contact motion is a combination of rolling and sliding at all three contacts. With increased load, the rolling motion increases at the loaded contacts. At high load the contact at the opposite side of the outer ring ceases (**fig. 5**, **page 55**).

If the speed increases, then the centrifugal force on the balls increases and the contact force on the outer ring increases, but again there is little change in contact angle. At high speed, the load at which three-point contact ceases is higher than at lower speed.

If the axial load reverses, the relative axial position of the outer and inner ring shifts and the previously unloaded inner ring becomes loaded.

By applying a spring loaded backup bearing, it is possible to shift the axial load at which the clearance shifts (**diagram 7**). With the contact angle more or less constant, the deflection curve of the four-point contact ball change very little with speed and load, compared to angular contact ball bearings (**diagram 1**, **page 49**).

Because of the partial rolling and sliding motion, it is important that four-point contact ball bearings operate with the lubrication parameter κ equal to or greater than 1,5.

Fig. 4

Loads and contact points

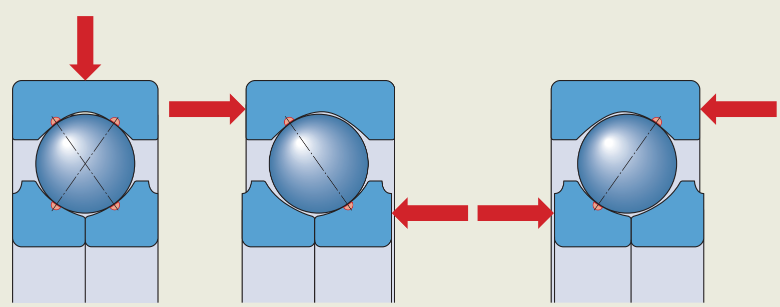
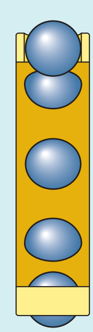
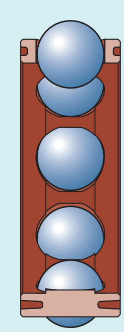


Table 9

Cages for four-point contact ball bearings

	Brass cages	**Polymer cages**
		
Cage type	Window-type, outer ring centred	Window-type, lubrication grooves in the guiding surface, outer ring centred
Material	Machined brass	PEEK, glass fibre reinforced
Suffix	MA	PHAS

Bearing minimum load and clearance

Four-point contact ball bearings should carry a minimum axial load for satisfactory operation. The criteria for determination of minimum axial load is gyroscopic spinning of the balls, as described for single row angular contact ball bearings.

The formula for calculating the minimum required axial load is the same as for single row angular contact ball bearings, but with the A factor adjusted for the difference in ball complement and contact angles of the two bearing types:

$$F_{a\,min} = A \left(\frac{n}{1\,000}\right)^2$$

where

$F_{a\,min}$ = minimum required axial load [kN]
A = minimum load factor
n = rotational speed [r/min]

Values of minimum load factor A for series QJ 2 and QJ 3 four-point contact ball bearings are listed in **table 10**, **page 56**. As mentioned above, four-point contact ball bearings cannot be preloaded like a pair of single row angular contact ball bearings. It is necessary to maintain an operating clearance at all times. Avoid excessive clearance since it negatively affects screw axial positioning accuracy. In oil flooded twin screw compressors, C2L clearance is a common choice for this reason. For dry air twin screw compressors, which operate at high speed and high temperature, it may be necessary to use larger clearances, such as C2H, CN, or C3L (**table 11**, **page 56**).

Allowable misalignment

Operation under misalignment reduces bearing life. The maximum allowable misalignment is 2 to 4 minutes. SKF recommends limiting misalignment to less than 1,5 minute in screw compressors. For detailed recommendation see **table 1**, **page 79**. Misalignment is caused by bending of the rotor, positioning tolerances of the radial bearing housings and clearances.

Bearing loads

Four-point contact ball bearings can theoretically carry combined loads, but radial loading should be avoided because it can lead to severe three-point contact on one side and cause bearing damage. Because it is difficult to predict the ratio of axial to radial force in screw compressors, use four-point contact ball bearings for thrust loads only and in combination with another bearing to take the radial load (**fig. 17**, **page 23**).

3

Fig. 5

The effect of centrifugal force in a four-point contact ball bearing
The centrifugal force F_c moves the ball radially outwards, but the movement is constrained by contact with the opposite side of the outer ring. This is causing three point contact.

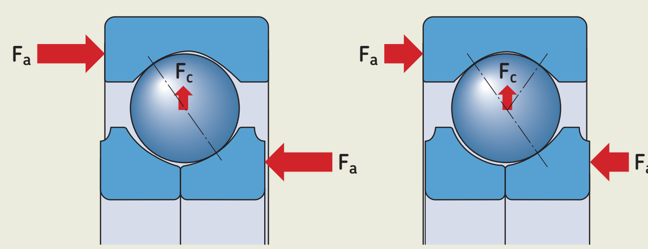
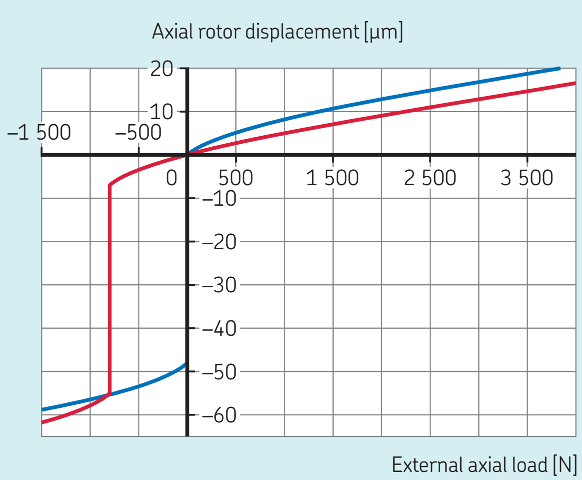
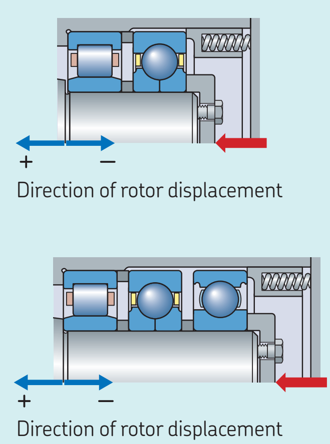


Diagram 7

Rotor axial displacement vs axial load





— One Four point contact ball bearing, SKF QJ 210 C2L
— One four point contact ball bearing SKF QJ 210 C2L and one spring loaded angular contact ball SKF 7210 CD

Fits and clamping

The recommended shaft tolerance is j5, see **table 7**, **page 40**, this allows the shaft to locate the two inner rings concentrically and minimize rotor run out. If a tighter fit is used, bearing clearance has to be analyzed accurately and possibly adjusted, especially if the bearing clearance is small (eg. C2L).

SKF recommends clamping the outer ring axially by means of a spring. This makes it possible to control the clamping force accurately. Excessive clamping can lead to outer ring distortion and radial loading. Too little clamping can lead to outer ring rotation. To determine the clamping force, please refer to **table 8**, **page 52**. The larger size QJ bearings have an anti-rotation slot (N2 suffix) to be used in combination with a locking pin.

Table 10

SKF four-point contact ball bearing, minimum axial load factors

Size	Bore diameter	A factor QJ2	QJ3
–	mm		
03	17	0,000427	0,00087
04	20	N/A	0,00143
05	25	0,00126	0,00278
06	30	0,00256	0,00508
07	35	0,00473	0,00744
08	40	0,0066	0,0118
09	45	0,00871	0,022
10	50	0,0103	0,029
11	55	0,0172	0,0404
12	60	0,0242	0,0549
13	65	0,033	0,0731
14	70	0,04	0,0954
15	75	0,0453	0,122
16	80	0,0629	0,155
17	85	0,0768	0,193
18	90	0,106	0,26
19	95	0,138	0,317
20	100	0,176	0,442
22	110	0,277	0,635
24	120	0,354	0,785
26	130	0,411	1,06
28	140	0,556	1,40
30	150	0,793	1,65
32	160	1,099	2,12
34	170	1,257	2,92
36	180	1,392	3,38

N/A: Please consult SKF for availability

Table 11

QJ axial clearance

Bore diameter		Axial internal clearance C2L		C2M		C2H		C2		CN		C3	
over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
mm		µm											
10	**18**	15	40	28	53	40	65	15	65	50	95	85	130
18	**40**	25	50	38	63	50	75	25	75	65	110	100	150
40	**60**	35	60	48	73	60	85	35	85	75	125	110	165
60	**80**	45	73	59	86	73	100	45	100	85	140	125	175
80	**100**	55	83	69	96	83	110	55	110	95	150	135	190
100	**140**	70	100	85	115	100	130	70	130	115	175	160	220
140	**180**	90	123	106	139	123	155	90	155	135	200	185	250
180	**220**	105	140	123	158	140	175	105	175	155	225	210	280

The radial clearance is 0,7 times the axial.

Hybrid angular contact ball bearings

Hybrid bearings have rings made of bearing steel and rolling elements made of bearing grade silicon nitride (Si_3N_4).

The following hybrid bearing types are available:

- Deep groove ball bearings
- Angular contact ball bearings
- Four point contact ball bearings
- Cylindrical roller bearings
- Sour gas bearings, see **page 88**

The advantages of hybrid bearings in twin screw compressors are:

- Silicon nitride has lower density (60%) than steel. Because of this, the effect of gyroscopic moments and centrifugal forces, as discussed for angular contact ball bearings and four point contact ball bearings, is much less. For bearing performance, this means higher speed capability compared to bearings with steel balls. Cylindrical roller bearings with silicon nitride roller can run faster compared to bearings with steel rollers.
  CAUTION: Be sure to select an appropriate cage in order to take advantage of higher speed capabilities.
- Higher hardness and lower coefficient of friction than steel balls and rollers. The advantages are longer fatigue life, lower operating temperature and less wear in marginal lubrication conditions.
- For bearing life calculation purposes, $\kappa = 1{,}0$ can be used, even if the actual κ value is lower
- Another advantage, in cylindrical roller bearings with roller made of silicon nitride is that the risk for light load skidding is lower.
- Insulating properties. As a non-conductive material, silicon nitride protects the bearing rings from conducting electric current, thereby preventing damage caused by electrical erosion. This can occur if there is a voltage difference between the rotor and the housing.
- Silicon nitride has lower coefficient of thermal expansion than steel. This means less sensitivity to temperature gradients within the bearing and more accurate control of preload/clearance. Temperature gradients typically result from high speed operation.
- Silicon nitride has higher modulus of elasticity than steel, ensuring less deflection under load.

Roller bearings in twin screw compressors

Roller bearings, primarily NU type cylindrical roller bearings and tapered roller bearings, are used in twin screw compressors as radial bearings. Cylindrical roller bearings can operate at high speed, have high load capacity and are axially compliant, facilitating easy separation of axial and radial load when used in combination with angular contact ball bearings. The inner and outer rings can be mounted separately in the compressor. Tapered roller bearings are also used, taking combined radial and axial loads.

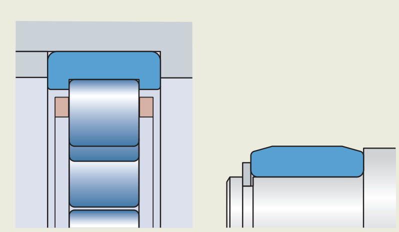
4

Cylindrical roller bearings

Bearing design and features

SKF cylindrical roller bearings of EC design are used in twin screw compressors for their high speed and high radial load capability. They have a large number and size of rollers, logarithmic roller profile, and optimized flange geometry. The bearings are available with ranges of internal radial clearance for optimization of rotor position accuracy. The SKF EC cylindrical roller bearing is produced standard with ISO P6 radial runout. All SKF cylindrical roller bearings are made to Explorer specifications. The NU type cylindrical roller bearing is commonly used in twin screw compressors since it allows separate assembly of the inner ring and outer ring/roller assemblies onto the shaft and into the housing respectively (**fig. 1**). The NU type bearing accommodates axial displacement

Fig. 1



Interchangeable components

due to thermal expansion of the shaft. NU type bearings allow both the inner and outer rings to be mounted with interference fits for more precise positioning of the rotors.

The SKF NU cylindrical roller bearings have two integral flanges on the outer ring to guide the rollers. The bearings have "open" flanges, i.e. the inward face of the flange is inclined by a defined angle (**fig. 2**). The flange design, together with the roller end design and surface finish, promote the formation of a lubricant film to reduce friction and frictional heat. The roller logarithmic profile (**fig. 3**) optimizes the stress distribution in the roller/raceway. The logarithmic profile also reduces sensitivity to misalignment and shaft deflection (**fig. 3**).

The optimized finish of the contact surfaces of the rollers and raceways maximizes the formation of a hydrodynamic lubricant film and minimizes friction.

Minimum load

Cylindrical roller bearings (like all bearings) must be subjected to a certain minimum load in order to create enough traction (friction) between the rollers and the inner ring. Without traction, the rollers will slide (skid) on the inner ring. If there is some but not enough traction, the rollers will partially roll and skid. Traction is the main driving force. There are also braking forces, viscous friction drag from lubricant on the roller and cage assembly and rolling resistance in the raceway contacts, flange contacts, cage to outer ring contacts as well as inertial forces from rapid acceleration.

Skidding can lead to metal smearing between the bearing rings and the rollers. The greatest risk is smearing on the inner ring of a NU type bearing, which is commonly used in screw compressors.

The following factors maximizes traction and minimizes braking;

- sufficient applied load
- low oil flow
- effective drain of oil from the housing
- minimized bearing clearance
- slow acceleration
- Use of bearings with small rollers such as the NU 2 series instead of NU 22, NU 3 or NU 23 series.
- Use of roller guided instead of outer ring guided cage if speeds permit.

The following factors reduce the risk of smearing damage from skidding:

- high oil viscosity (but low flow)
- use of AW and EP oil additives
- use of NoWear coated rollers
- black oxidized rollers
- ceramic rollers (hybrid bearing)

The minimum load required to avoid skidding and skid smearing can be calculated from the formula below

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

where

F_{rm} = minimum radial load [kN]

k_r = minimum load factor; see bearing tables in the SKF catalogue *Rolling bearings*

n = operating speed [r/min]

n_r = reference speed [r/min]; see bearing tables in the SKF catalogue *Rolling bearings*

d_m = mean diameter of bearing = 0,5 (d + D) [mm]

Fig. 2

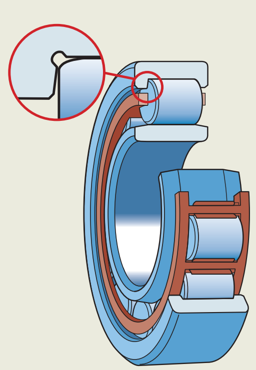
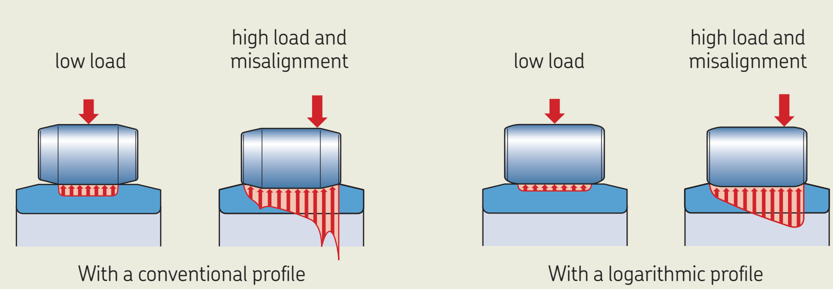


Fig. 3



Cages

The SKF EC design cylindrical roller bearings are produced with four basic cages (**table 1**):

- glass fibre reinforced polyamide 6,6 cage (P suffix)
- PEEK cage (PH and PHA suffix)
- pressed steel cage (J suffix)
- machined brass cage (M and ML suffix)

The bearings are optionally available with a machined steel cage (F suffix) for gas compressors.

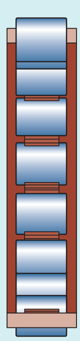
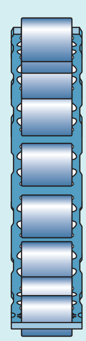
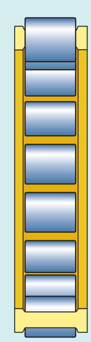
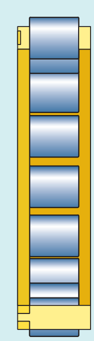
Allowable misalignment

Operation under misalignment reduces bearing life. The maximum allowable misalignment is 3 to 4 minutes, depending on the bearing series, but limiting the misalignment to less than 1,5 minutes in screw compressors is recommended to optimize performance and reliability. For detailed recommendations, see **table 1**, **page 79**. Misalignment is caused by bending of the rotor, positioning tolerances of the radial bearing housings and clearance in the radial bearings.

In order to reduce the sensitivity to misalignment, the rollers have logarithmic profile and the raceways are slightly crowned.

Table 1

Cages for single row cylindrical roller bearings

	Polymer cages	**Steel cages**	**Brass cages**	
				
Cage type	Window-type, roller or outer ring centred		Window-type, depending on bearing design, inner or outer ring centred	Riveted • roller centred • outer ring centred • inner ring centred
Material	• PA66, glass fibre reinforced • PEEK, glass fibre reinforced	Stamped steel	Machined brass	
Suffix	• P or PA • PH or PHA	• – • J	• ML	• M • MA • MB

Selection of fits and radial internal clearance

In order to achieve maximum positioning accuracy of the rotors and to maximize bearing life, cylindrical roller bearings should operate with a small radial internal clearance. However, the initial bearing clearance must be selected to avoid risk of the bearings becoming preloaded in operation. The operating radial internal bearing clearance depends on the initial radial internal bearing clearance, shaft and housing tolerances, and the temperature gradient from the shaft through the bearings and into the housings. In a screw compressor, it is important to keep the operating radial internal clearance of a bearing small for the following reasons:

- The operating radial internal clearance affects the positioning accuracy of the rotor and therefore also the compressor efficiency. The ability to operate with a small radial internal clearance is a major advantage of a cylindrical roller bearing over a hydrodynamic bearing.
- A smaller operating radial internal clearance leads to low noise and vibration levels in the compressor.
- A smaller operating radial internal clearance results in longer bearing life (**diagram 1**), increased stiffness, and reduced bearing deflection under load.

However, the radial internal clearance should not be too small, for the following reasons:

- There is risk of radial preload, higher temperatures and premature bearing failure.
- It can cause difficulties in assembling the compressor and risks damaging the bearing.
- If an interference housing fit is used the housing seat may be ovalized due to the effect of the press fit on the unsymmetrical housing. This may increase the possibility of preloading the bearings and cause difficulties with the assembly.

Radial internal clearance ranges

The internal clearance specifications in cylindrical roller bearings are according to ISO 5753 -1. The standard ranges of clearance are the normal range and greater than normal range (C3 suffix).

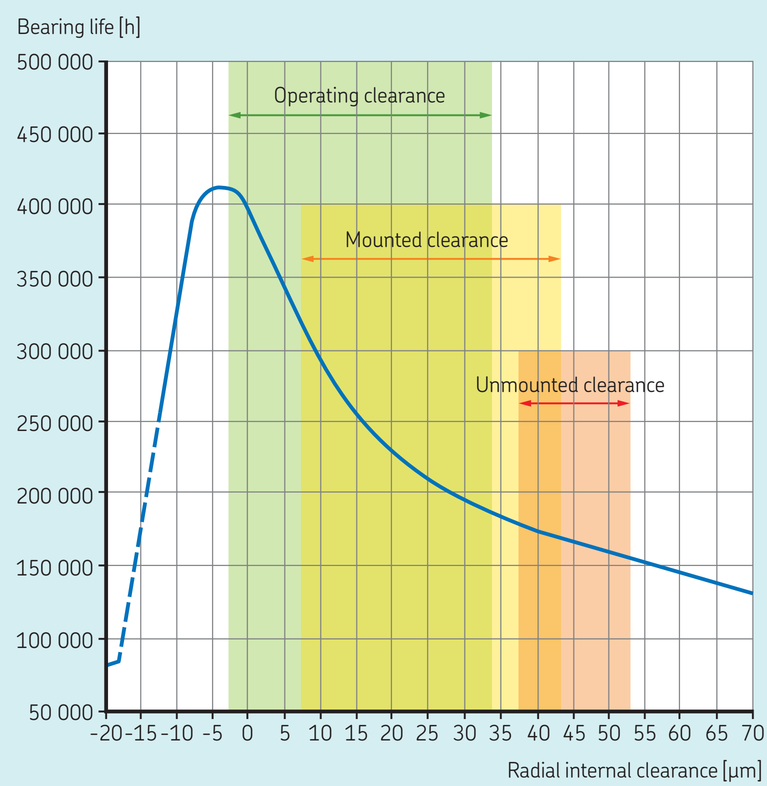
Additional clearances with a reduced range are also available as non-standards. The reduced clearance ranges use the lower, middle, and upper half of the standard ranges. For compressor applications, the common reduced clearance ranges are CNM and C3L. The CNM suffix denotes that the half clearance range is centred on the mean of the normal range. The C3L suffix denotes that the clearance is the lower half of C3 range. Bearings with reduced clearance range should be used for optimum rotor positioning accuracy. The radial internal clearance ranges in the SKF catalogue *Rolling bearings* and the reduced clearance ranges (CNM, C3L) are maintained even if rings between bearings are interchanged from different bearings. This is a unique feature of SKF EC design cylindrical roller bearings.

Not all cylindrical roller bearing variants (CNM, CNH, C3L, etc.) are currently in production. Please contact SKF sales for details on availability.

In the withdrawn clearance standard DIN 620-4:1887-08 the clearance values were slightly different and were shown as “interchangeable” and “matched assembly” values. Bearings having the old matched assembly clearance are marked by the suffix R, for example C3R. **Table 2** lists the old and current clearances.

Diagram 1

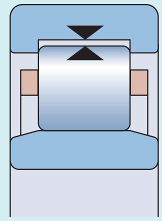
Bearing life as a function of radial clearance



The dotted line indicates potentially unstable working conditions

Table 2

Comparison of cylindrical roller bearing clearances according to withdrawn standard DIN 620-4:1987-08 and current ISO 5753-1



d		ISO 5753-1				ISO 5753-1 – common half ranges			
over	incl	C2	CN	C3	C4	CNM	CNH	C3L	C3M
µm		–				–			
–	**24**	0...25	20...45	45...60	50...75	26...39	33...45	45...53	49...56
24	**30**	0...25	20...45	45...60	50...75	26...39	33...45	45...53	49...56
30	**40**	5...30	25...50	45...70	60...85	31...44	38...50	45...58	51...64
40	**50**	5...35	30...60	50...80	70...100	38...53	45...60	50...65	58...73
50	**65**	10...40	40...70	60...90	80...11	48...63	55...70	60...75	68...83
65	**80**	10...45	40...75	65...100	90...125	49...66	58...75	65...83	74...91
80	**100**	15...50	50...85	75...110	105...140	59...76	68...85	75...93	84...101
100	**120**	15...55	50...90	85...125	125...165	60...80	70...90	85...105	95...115
120	**140**	15...60	60...105	100...145	145...190	71...94	83...105	100...123	111...134
140	**160**	20...70	70...120	115...165	165...215	83...108	95...120	115...140	128...153
160	**180**	25...75	75...125	120...170	170...220	88...113	100...125	120...145	133...158
180	**200**	35...90	90...145	140...195	195...250	104...131	118...145	140...168	154...181

d		DIN 620-4:1987-08 – interchangeable assembly				DIN 620-4:1987-08 – matched assembly			
over	incl	C2R	CNR	C3R	C4R	C2R	CNR	C3R	C4R
µm		–				–			
–	**24**	0...30	10...40	25...55	35...65	10...20	20...30	35...45	45...55
24	**30**	0...30	10...45	30...65	40...70	10...25	25...35	40...50	50...60
30	**40**	0...35	15...50	35...70	45...80	12...25	25...40	45...55	55...70
40	**50**	5...40	20...55	40...75	55...90	15...30	30...45	50...65	65...80
50	**65**	5...45	20...65	45...90	65...105	15...35	35...50	55...75	75...90
65	**80**	5...55	25...75	55...105	75...125	20...40	40...60	70...90	90...110
80	**100**	10...60	30...80	65...115	90...140	25...45	45...70	80...105	105...125
100	**120**	10...65	35...90	80...135	105...160	25...50	50...80	95...120	120...145
120	**140**	10...75	40...105	90...155	115...180	30...60	60...90	105...135	135...160
140	**160**	15...80	50...115	100...165	130...195	35...65	65...100	115...150	150...180
160	**180**	20...85	60...125	110...175	150...215	35...75	75...110	125...165	165...200
180	**200**	25...95	65...135	125...195	165...235	40...80	80...120	140...180	180...220

Needle roller bearings

Needle roller bearings are used in oil-injected air and refrigerant compressors because of their high radial load capability and compact size. This can be an advantage in small compressors, where space is limited. Needle roller bearings are more sensitive to misalignment then cylindrical roller bearings. Compared to cylindrical roller bearings the rollers (needles) are guided by the cage instead of being guided by the flanges. This limits the speed and can lead to higher noise level. Needle roller bearings have separable rings like the cylindrical roller bearings, but can also be used without inner rings if the raceways are made as per the specifications below.

Needle roller bearings with special clearances, cages or with higher precision have limited availability. To avoid significantly reduced bearing life, the angular misalignment should be less than 1,0 minute.

Specifications for raceways on shafts

- Hardness 58 to 64 HRC.
- Surface roughness Ra < 0,2 µm or Rz < 1 µm.
- Roundness tolerance IT5/2.
- Material: bearing steel per AISI 52100 or ISO 683-17. Other hardenable steels can also be used, but will result in reduced bearing life.
- Raceway diameter: contact SKF application engineering service.

Tapered roller bearings

Tapered roller bearings are used in oil flooded air screw compressors at the male and female discharge positions, and accommodate both axial and the radial load. (**figs. 18** to **23**, **pages 24** to **25**) . They can also be used to support the input gear shaft in both oil flooded compressors and high speed dry air compressors. High axial and radial load capabilities are the main features of tapered roller bearings.

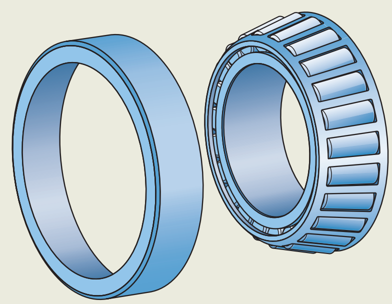
Designs and features

Tapered roller bearings have tapered inner and outer ring raceways and tapered rollers. They are designed to accommodate combined loads, i.e. simultaneously acting radial and axial loads. The projection lines of the raceways meet at a common point on the bearing axis (**fig. 4**) to provide true rolling and low friction between the raceways and the rollers. The axial load carrying capacity of tapered roller bearings increases with increasing outer ring contact angle (**fig. 5**). The size of the angle is related to the factor Y.

$Y = 0{,}4 \cot \alpha$

The value of Y can be found in the SKF catalogue *Rolling bearings*.

Separable bearings
Single row tapered roller bearings are separable, i.e. the inner ring with roller and cage assembly (cone) can be mounted separately from the outer ring (cup).



4

Fig. 4

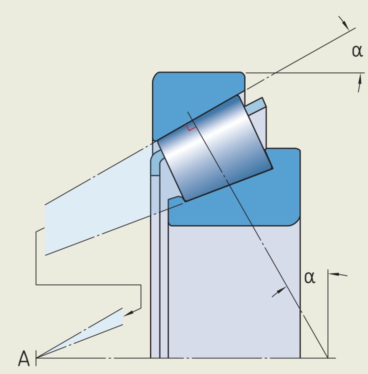
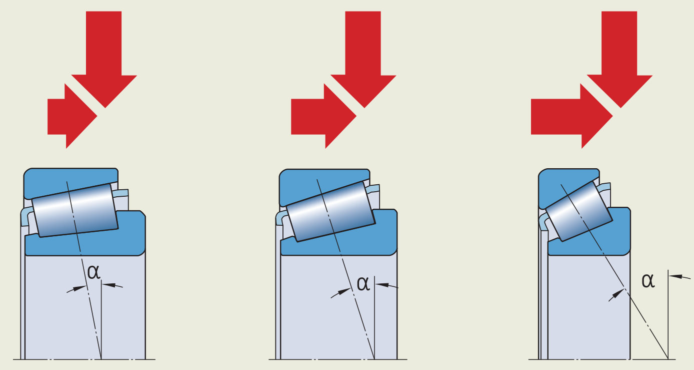


Fig. 5

The choice of a contact angle depends on the ratio axial to radial load
The larger the contact angle α, the higher the axial load carrying capacity.



Angle: Low Medium Steep

Because of the tapered shape of the rollers, the contact angles of the outer and inner rings are not the same, resulting in a force pushing the rollers against the inner ring flange (**fig. 6**). The contact motion between the roller end and the inner ring flange is a combination of rolling and sliding. The geometry of the roller end and the flange is critical for the formation of a lubricant film that reduces friction and wear. The friction between the roller end and the flange causes heat and limits the speed capability of tapered roller bearings.

Because of the difficulty in lubricating the roller/flange contact, SKF does not recommend tapered roller bearings in refrigerant compressors. This is especially difficult in ammonia compressors but also in other refrigerants, such as R-134a.

The SKF Explorer design of tapered roller bearing is well suited for compressor applications. This design of tapered roller bearing features logarithmic roller profile, low friction steel cage design, and optimized roller end and flange profiles The SKF tapered roller bearing is also available in the CL7C execution. This bearing has optimized roller end and flange geometry to reduce running-in wear. The SKF tapered roller bearings are available in the common ISO series. The 313 series, having high contact angle is well suited for the male thrust position in compressor applications.

Allowable Misalignment

Similar to cylindrical roller bearings, tapered roller bearings are somewhat sensitive to misalignment. Operation under misalignment reduces bearing life. The maximum allowable misalignment is 2 to 4 minutes, but limiting misalignment to less than 1,5 minute in screw compressors is recommended. For detailed recommendations see **table 1**, **page 79**. Misalignment is caused by bending of the rotor, positioning tolerances of the radial bearing housings and clearances.

In order to reduce sensitivity to misalignment, inner rings have logarithmic profile and the outer ring raceways are crowned.

Minimum load

A tapered roller bearing must operate with a certain minimum axial load, which may be applied by the compressor or produced by the bearing mounted adjacent to it. The axial load must be greater than

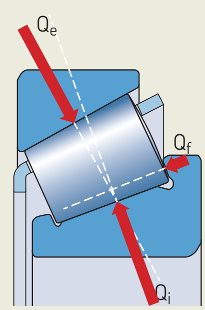
$$F_a > \frac{0{,}5\, F_r}{Y}$$

where
F_a = total axial load on bearing [N]
F_r = applied radial load [N]
Y = bearing axial load factor, according to the SKF catalogue *Rolling bearings*

See also the SKF catalogue *Rolling bearings* for additional details. If the axial force on the bearing ring is too low, it may be necessary to apply an additional force to the bearing outer ring to prevent rotation relative to the housing face. See the section *Housing tolerances – bearings taking axial loads*, **page 41**. The face of the bearing can be slotted to mate with an anti-rotation pin mounted in the housing. This is a non-standard feature.

Fig. 6

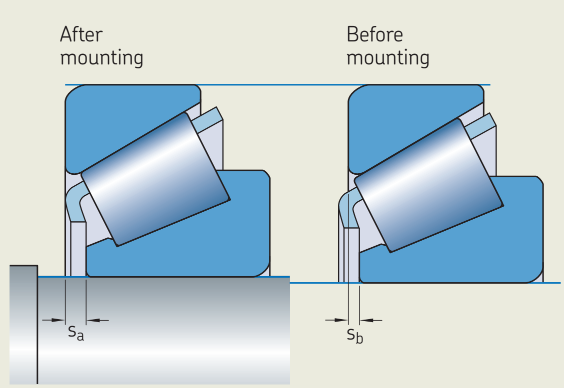
Design and function of tapered roller bearings



Q_e = Outer ring contact force
Q_i = Inner ring contact force
Q_f = Flange contact force

Fig. 7

Axial shift in the relative position of the inner and outer rings of tapered roller bearings after mounting on a shaft with interference fit



$s = s_a - s_b$

s = Shift in standoff
s_a = Standoff after mounting
s_b = Standoff before mounting

Mounting considerations

Because tapered roller bearings do not come with a specified standoff between the inner and outer rings, screw compressors fitted with tapered roller bearings are relatively complicated to assemble (**fig. 7**). Also, tapered roller bearings have to be mounted with an interference fit to the shaft. The interference causes an axial shift of the outer ring relative to the inner ring (**fig. 7**). This shift can be considered when the rotor end clearance is adjusted. The shift can be calculated with the formula below.

$$s = Y\, i$$

where

s = axial shift [µm]

Y = bearing axial load factor, according to the SKF catalogue *Rolling bearings*

i = interference between shaft and bearing inner ring [µm]

When a reverse thrust tapered roller bearing is used, then the clearance between the two tapered roller bearings must be adjusted by means of shimming. Alternatively the reverse thrust bearing can be spring loaded (**figs. 18** and **19**, **page 24** and **21**, **page 25**). Tapered roller bearings cannot operate with shimmed preload in screw compressors because of the sensitivity to the preload by shimming.

For the reasons mentioned above, it is difficult to set a small rotor end clearance with tapered roller bearings, therefore screw compressors fitted with tapered roller bearings typically have larger rotor end clearance.

PTFE seals in twin screw compressors

Seals in twin screw compressors are designed to prevent fluids from leaking out of the compressor, protect against the entry of contaminants and reduce possible leakage inside the compressor. PTFE lip seals are well suited to perform these functions at a reasonable cost. Elastomeric seals, on the other hand, do not offer the service life and chemical compatibility required for the application, while mechanical face seals need more space than is normally available and cannot perform all the required functions.

PTFE seal material and manufacturing

Material properties

PTFE (polytetrafluoroethylene) is a semi crystalline thermoplastic with a melting point of 327 °C. The properties of importance for seals are:

- **Favourable**
  - Low coefficient of friction
  - High againg and chemical resistance
  - Broad operating temperature range
- **Unfavorable**
  - High wear rate in pure form
  - High viscosity as melted makes injection molding difficult
  - Low creep resistance

Manufacturing

In order to reduce wear, increase the allowable operating sliding speed and reduce creep, a second material is usually added to the PTFE base material. Common secondary materials are polyimide, PPS, Ekonol, carbon fibre, graphite, bronze powder, glass fibre and MoS_2. The addition of a second material is made possible by mixing the two materials in the form of fine powders. The powders are mixed thoroughly and then compressed to produce a void free tube or rod (**fig. 1**). The material is then heated so that the powder grains are sintered together. Other methods include extrusion and isostatic moulding.

The final step involves machining (turning) the tube. After machining, the seal lip has the shape of a washer. When the seal is mounted on the shaft, the washer bends into a J shape (**fig. 2** and **3**).

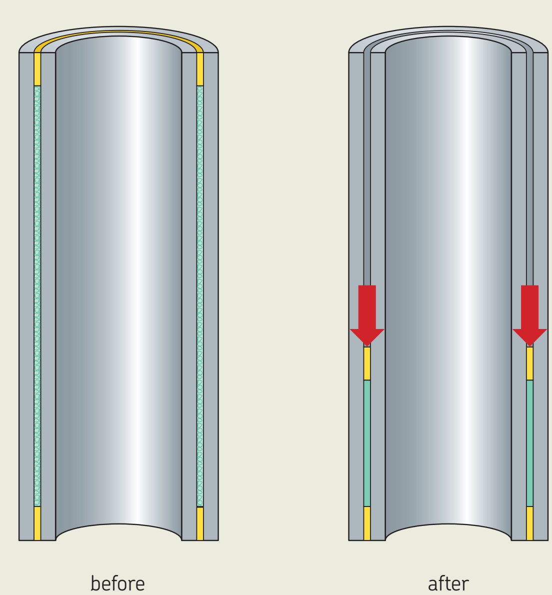
PTFE seal designs

PTFE seals can be either metal-cased or nonmetal-cased. A metal-cased seal provides more rigidity and resists change of dimensions with temperature. They are preferred for larger dimension seals. Metal-cased seals are pressed into the housing and are self-retained, requiring support only in higher pressure applications. Nonmetal-cased seals can be an economical alternative for smaller seals.

Metal-cased seals offer the possibility of using different materials in the lips of a multi-lip seal, optimizing the seal's performance. The lips can have spiral grooves to provide a pumping action to assist the sealing function at lower pressures. A number of different lip configurations are possible, depending on the application requirements (**figs. 2** and **3**).

Fig. 1

Compression of mixed powders to make a tube for sintering



PTFE seal applications in twin screw compressors

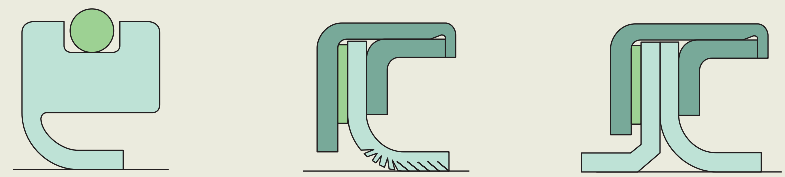
Air compressor input shaft seals

During normal operation, seal pressure is below atmospheric pressure, and the input shaft seal prevents air and contaminants from entering the compressor. At shutdown, there is a sudden increase in pressure at the seal location and the seal prevents oil from leaking out. The input shaft seal can be used in combination with a wear sleeve to avoid wear of the shaft and to enable easier field repairs (**fig. 4**). A front lip with a spiral groove is available as an option. A typical design for shop air compressors involves a primary seal lip with a thread, a secondary seal lip without a thread and a wiper seal lip. The threaded primary seal lip seals when the compressor is running, but leaks at higher pressure during shutdown. The non threaded secondary seal lip prevents leakage at shutdown. When the compressor is restarted, the threaded primary seal pumps the oil collected between the primary and secondary seal lips back into the bearings.

The wiper seal lip prevent contamination from entering the compressor.

Fig. 2

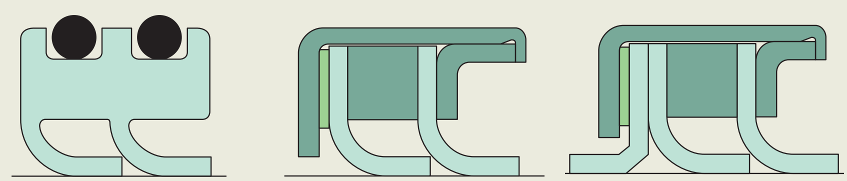
Single lip seal geometries
Single lip seals are used for high speed, low pressure, 0,5 bar max. Surface speeds up to 30 m/s.



Single hydrodynamic lip all PTFE | Single hydrodynamic lip metal cased | Single hydrodynamic lip with wiper metal cased

Fig. 3

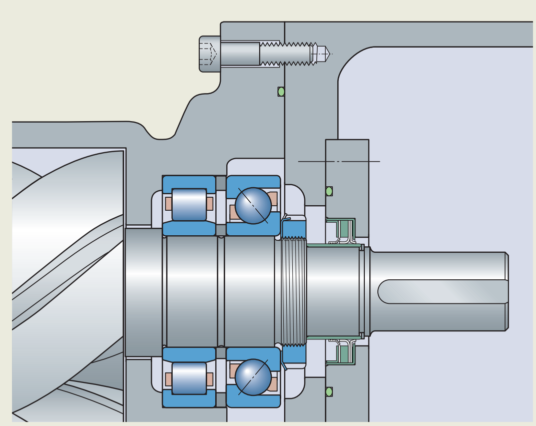
Double lip seal geometries
A secondary lip enables the seal to be used at pressures up to 13 bar with spikes to 20 bar.



Double lip all PTFE | Double lip metal cased | Double lip metal cased with wiper

Fig. 4

Air compressor input shaft seals



Internal rotor seals

Applied at the discharge end of air-conditioning screw compressors, internal rotor seals have several functions:

- To prevent hot gas and contaminants from entering the bearing housing
- To minimize the amount of internal gas leakage and thereby increasing compressor efficiency.
- To reduce bearing housing pressure which reduces axial bearing load
- To reduce bearing housing temperature and improve bearing lubrication conditions.

Internal rotor seals improve compressor reliability and efficiency, and are typically metal-cased, double-lip designs (**fig. 5**).

Mechanical seal replacement PTFE seals

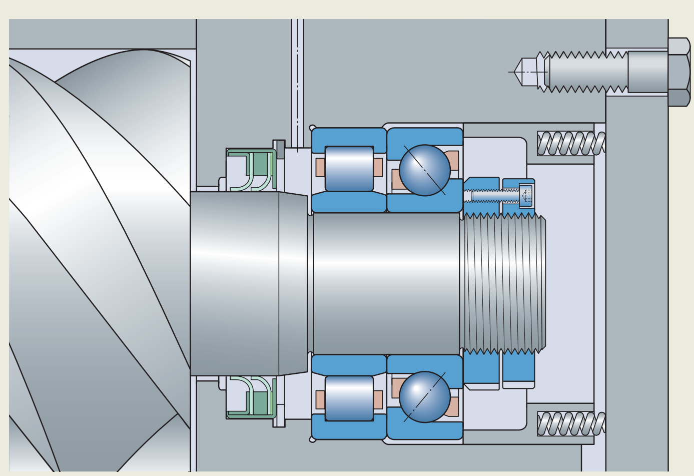
For higher pressure applications, up to 13 bar, a triple lip PTFE seal can be used as an economical and less complicated alternative to a mechanical seal. A dust lip prevents air and contaminants from entering the compressor, double seal lips prevent refrigerant and oil from leaking out. The lips can be with or without spiral grooves. The inner and outer lips are of different materials for optimum performance.

Mechanical seal oil retention PTFE seals

In refrigerant open drive compressors, mechanical seals prevent refrigerants and oil from leaking to the outside and air from leaking into the compressor. Additionally a PTFE dust seal is applied to prevent contaminants to enter. A PTFE oil-retention seal prevents oil from escaping the mechanical seal housing during shutoff and standstill and ensures lubrication of mechanical seal faces. Oil-retention seals have a single lip and can be metal cases or non metal cased, depending on size. (**fig. 6**)

Fig. 5

Bearing arrangement with internal rotor PTFE lip seal



Slide valve PTFE piston seals

Compressors with slide valves for capacity or pressure control commonly have a hydraulic piston to set the position of the slide valve. The piston has a piston ring, which prevents leakage across the piston and a glide ring which reduces friction and wear between the piston and the cylinder wall. The glide ring also locates and guides the piston. The piston ring usually has an overlap joint to minimize leakage. The guide ring has either a butt joint or an angled joint and is not sealing against the piston pressure (**fig. 7**).

Poppet valve PTFE seals

Poppet valves are sometimes used as an alternative to slide valves in screw compressors. Poppet valve seals are spring energized to minimize leakage.

Fig. 7

Different piston ring joints

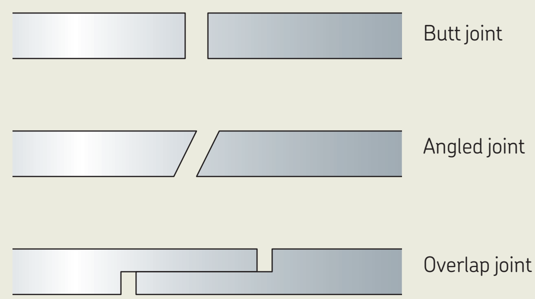
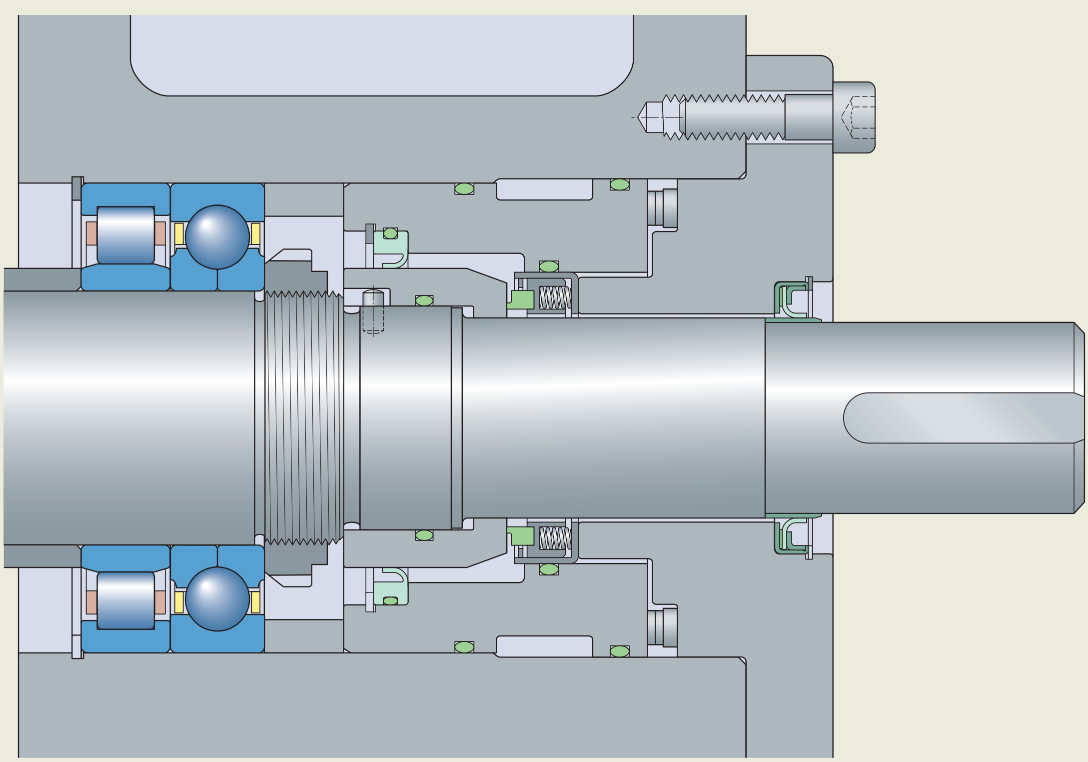


Fig. 6

Mechanical seal with PTFE oil retention seal



Wear sleeves

Compared to elastomeric lip seals, PTFE seal lips are harder and the lip pressures are higher. As a result, shaft wear is generally greater. To prevent wear, the rotor shaft mating surface should be hard and have a fine surface finish. An alternative way to prevent shaft wear is to use a wear sleeve. SKF wear sleeves are hardened to 58 to 60 HRC and have the same dimensions as needle roller bearing inner rings. The surface is plunge ground to prevent leakage that could occur due to a grind lead that is common on centerless ground surfaces (**fig. 8**).

Fig. 8

Diagram 1

Allowable application pressure differentials and surface speeds

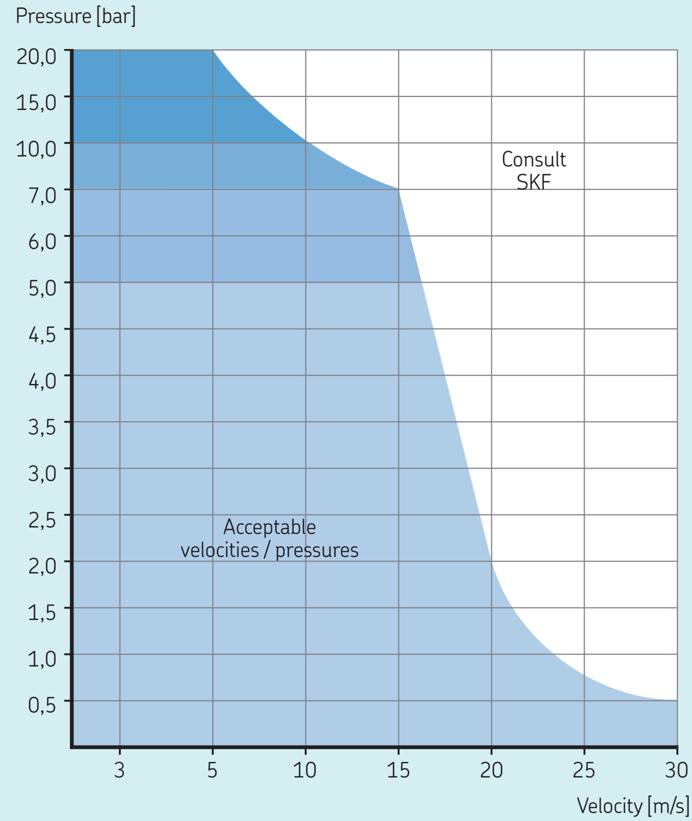
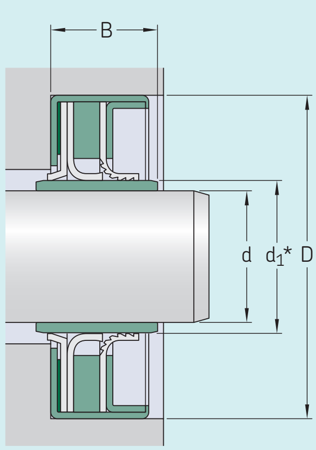


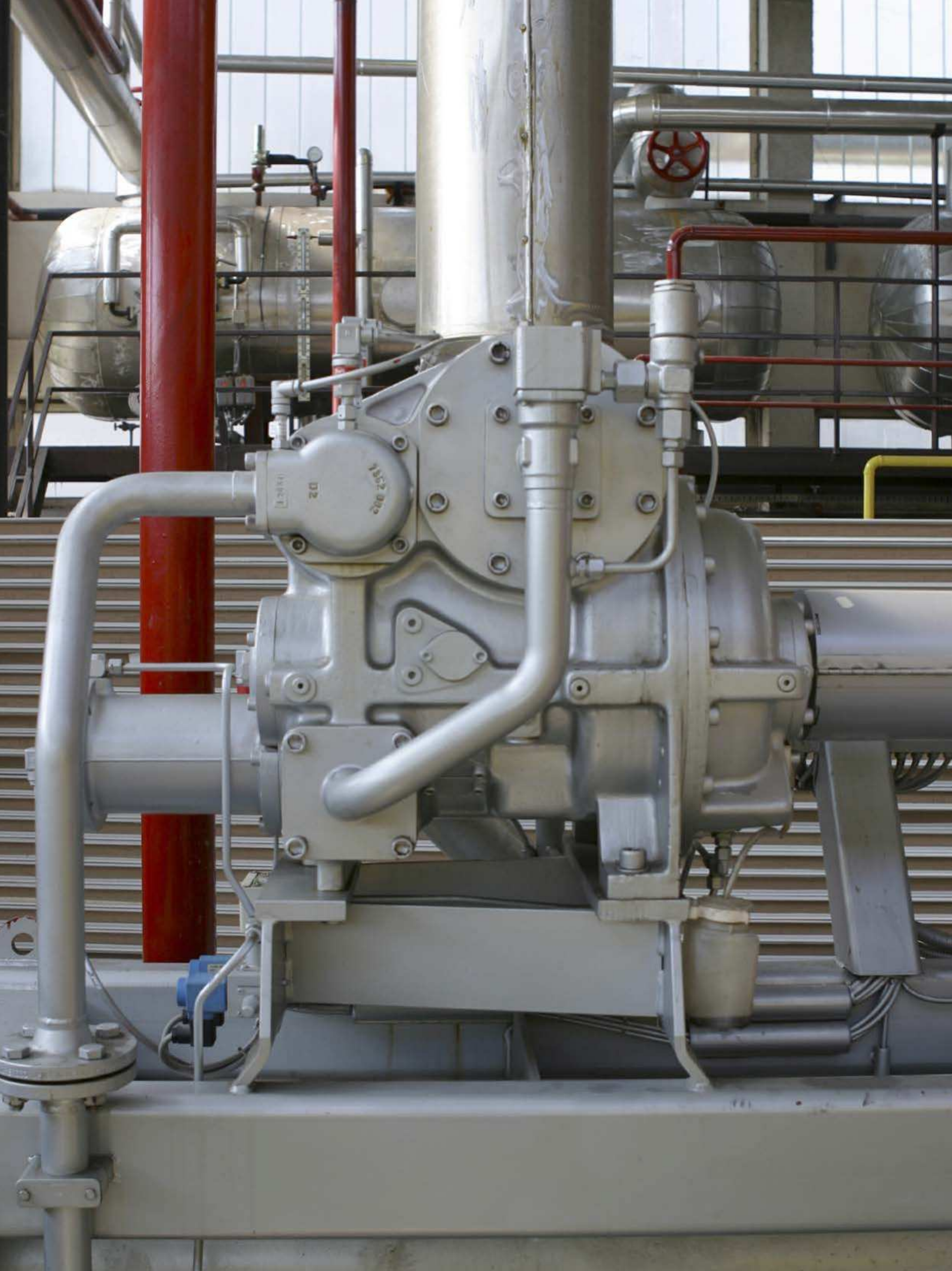
Table 1

Dimensions for commonly used PTFE seals and wear sleeves [mm]



Wear sleeve			Seal	
Shaft diameter d	Sleeve outside diameter d_1*	Nominal width B	Sleeve outside diameter d_1*	Bore diameter D
mm			mm	
17	22	13	**22**	35 / 40 / 47
20	25	17	**25**	40 / 47 / 52
25	30	18	**30**	42 / 47 / 52
30	35	20	**35**	50 / 52 / 55
40	45	20	**45**	62 / 65
45	55	22	**55**	72 / 80
50	55	25	**55**	72 / 80
60	70	25	**70**	90 / 95
65	75	28	**75**	95 / 100
70	80	25	**80**	100 / 110
75	85	30	**85**	110 / 120
80	90	25	**90**	110 / 120
90	100	36	**100**	120 / 125 / 130

* Mating PTFE lip diameter



Screw compressor types

Twin screw compressors are used for air compression, refrigeration, air conditioning and compression of various gases. These compressor types and associated bearing application considerations will be described in this chapter.

6

Air compressors

The first air screw compressors were high speed machines using external timing gears to synchronize the rotors. The rotors were designed to rotate without contact with the compressor housing or with each other. This compressor type was called “dry running” because the rotors operate without injection of a fluid into the compression cavity. The first oil-injected air compressor was introduced in the late 1950s. This design did not use timing gears. Instead, the male rotor would drive the female rotor by contacts through the rotor lobes. This opened up a larger market for air screw compressors, making it the leading design in medium-size air compressors.

Oil-injected air compressors

In oil-injected screw compressors, oil is injected into the compression cavity to lubricate the rotor-to-rotor contact, seal leakage gaps between the rootors and between the rotors and the housing, and cool the compressor. An oil separator on the discharge side filters out the oil, but the air is not 100% oil free. Oil injection makes it possible to generate higher pressures, operate at lower speeds and with higher efficiency, compared to dry running compressors. With oil injection, manufacturing is also simplified and less expensive.

Compression can be in one or two stages, with discharge pressures of 10 to 13 bar for single stage and 20 to 35 bar for two-stage compressors. The high pressure ratios are made possible by the oil injection, since the oil is cooling the air during compression.. The efficiency is reduced as the pressure ratio is increased because more oil is needed and the discharge temperature is increased. An example of a gear-driven oil-injected air compressor is shown in **fig. 1**, **page 78**.

Operating conditions

Speeds

The rotor tip speed is typically 30 to 40 m/s. The speeds are adjusted to the compressor capacity needed by exchange of gear sets or drive belts. The bearing nd_m speeds less than 450 000.

Radial bearing loads

The radial loads are highest at the female discharge position, lighter at the discharge male position and less on the suction side. With gear drive, the radial loads are higher at the drive position, which is usually on the suction side.

Axial bearing loads

The axial force is usually higher on the male rotor and lighter on the female rotor. The net thrust bearing force is a combination of the gas force, axial gear forces, balance piston forces, backup bearing or reverse thrust bearing forces and, at startup, inertial forces.

Overall, the bearing loads are moderate and several bearing arrangements are possible.

Bearing arrangements

Discharge side

The bearing arrangements are either angular contact ball reverse thrust bearings only or a combination of cylindrical roller and angular contact ball bearings (**figs. 9**, **10** and **11**, **page 21**) or cylindrical roller and four-point ball bearings (**fig. 17**, **page 23**). If the axial load on the male rotor is high, angular contact ball bearings in tandem is a possible solution (**figs. 12**, **13**, **14** and **16**, **pages 22** to **23**).

If there is a risk of reversing axial loads on either the male or female rotor, reverse thrust bearings can be used (**figs. 9**, **12** to **16**, **18** to **22**, pages **21** to **25**). To minimize internal loads generated by centrifugal forces, SKF recommends the use of angular contact ball bearings with a 25° contact angle, AC design for nd_m for speeds greater than 450 000.

It is also possible to use tapered roller bearings at the discharge end to take both radial and axial loads (**figs. 18** to **23**, **pages 24** and **25**).

Suction side

The most common arrangement is a single cylindrical roller bearing. If the radial loads are light it may be possible to use deep groove ball bearings (**fig. 26**, **page 26**) or angular contact ball bearings AC design (**fig. 27**, **page 26**).

Gear shaft

For gear shaft bearing arrangements see **figs. 27** and **28**, **page 26**.

Bearing and application recommendations

For detailed recommendations see **table 1**.

Lubrication

Bearing lubrication is by circulating oil, with lubricant flow driven by the pressure difference between the discharge and suction side of the compressor. The oil can be cooled by a cooler, usually integrated with the after-cooler for the compressed air. The oil viscosity grade is typically ISO 46, but should be high enough to produce a κ value greater than 1,0 and preferably greater than 1,5.

Drive arrangements

The compressor is typically driven by an induction motor coupled to a gear shaft in the compressor, with speed-increasing gears for smaller compressors. Compressor speed and capacity can be customized by changing the gear set. Bearing arrangements for the gears are shown in **figs. 28** and **29**, **page 26**.

Another arrangement involves driving the compressor by belt drive, although direct drive with the motor rotor mounted on the extended male shaft is increasingly being used. The speed is then controlled with an inverter-type variable speed drive. This allows for capacity control by speed and the elimination of valves for that purpose.

Fig. 1

Gear driven oil injected air compressor

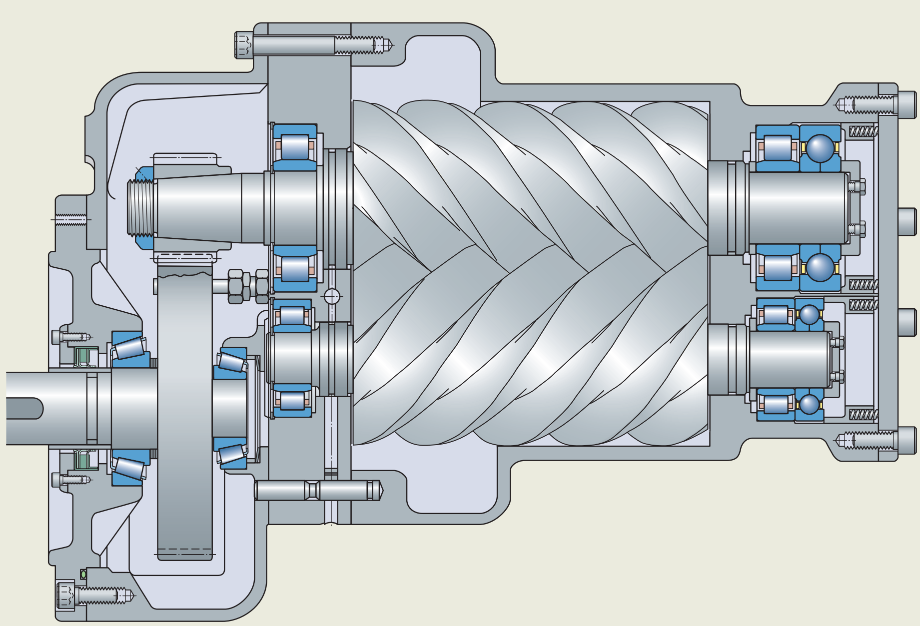


Table 1

Bearing final variant specification and application recommendations for oil injected screw compressors

Bearing type	Bearing series	Compressor design	Bearing bore diameter, d			Maximum misalignment
			10 ≤ d ≤ 50	50 < d ≤ 100	100 < d ≤ 150	
–	–	–	mm			minutes
Angular contact ball bearings for main axial load	72 BE, 73 BE 72 AC,73 AC	A	BEGAP j5/RF	BEGAP j5/RF	BEGAP j5/RF	1,5
		B	BECBP j6/RF	BECBP j6/RF	BECBP j6/RF	3,0
Angular contact backup bearings[1)]	72 AC,73 AC	A	ACGAP j5/RF	ACGAP j5/RF	ACGAP j5/RF	1,5
Angular contact backup bearings[2)]	72 BE, 73 BE	A	BEGAP j5/RF	BEGAP j5/RF	BEGAP j5/RF	1,5
		B	BECBP j6/RF	BECBP j6/RF	BECBP j6/RF	3,0
Four point ball bearings for main axial load	QJ 2, QJ 3	A	C2L j5/RF	C2L j5/RF	C2L j5/RF	1,5
		B	CN j6/RF	CN j6/RF	CN j6/RF	2,0
Deep groove ball back up bearing	60, 62, 63	–	C3 j6/RF	C3 j6/RF	C3 j6/RF	4,0
Cylindrical roller bearings	NU 2,NU22, NU 3,NU23	A	ECP/CNM m5/K6	ECP/CNH n5/K6	ECP/C3L p5/K6	1,5
		B	ECP m6/J7	ECP/C3 n6/J7	ECP/C3 p6/J7	3,0
Tapered roller bearings	All and CL7C	A	m5/K6	n5/K6	p5/K6	1,5
	All	B	m6/K7	n6/K7	p6/K7	3,0

Compressor design A: For maximum rotor positioning accuracy.
Compressor design B: For best bearing availability.

1) For nd_m speed higher than 450 000
2) For nd_m speed lower than 450 000

RF = radially free

The table is valid for the following conditions:
- Solid steel shafts/steel or cast iron housings.
- The inner ring is no more than 10 °C warmer than outer ring. If the temperature difference between the inner and outer ring is more than 10 °C, and/or if the bearing temperature is greater than 90 °C, a bearing with increased clearance may be required. The bearing operating clearance or preload should be verified by calculation.
- Bearing temperature is controlled by circulating oil lubrication.

Additional recommendations
- For angular contact ball bearings and four point contact ball bearings taking axial loads only, it is possible to use the same shaft tolerance as for the adjacent cylindrical roller bearing. The influence on clearance and/or stand-out between the rings shall be checked.
- Total run-out, surface finish and housing tolerance recommendations refer to **table 5** to **7** on **page 39** to **40**.
- Tapered roller bearings are not recommended in refrigerant compressors.
- Maximum temperature for cages, see **table 4**, **page 38**.
- For more details on these recommendations, contact SKF application engineering service.

Dry air compressors

In dry air compressors, there is no oil injection into the compression cavities and no contact between the rotors or between the rotors and the housing. The rotors are synchronized by timing gears, located outboard of the rotor bearings on the extended shafts, with seals between the compression cavity and the bearing and gear casing. To compensate for the absence of the sealing effect of injected oil (the compressor being dry), the speed is much higher, so that the time for leakage per revolution is minimized. Dry air compressors are usually two stage compressors with a pressure ratio up to 3 bar in the first stage and a discharge pressure of 7 bar. There is an intercooler between the two stages and an aftercooler after the second. The pressure ratios are low because there is no cooling by oil during compression. The efficiency is reduced with increased pressure ratio. The prime advantage of dry air compressors is that the air is 100% oil free.

Operating conditions and bearing arrangements

Rotor tip speeds are typically 50 to 70 m/s with bearing nd_m speeds of 800 000 to 1 200 000. The loads are moderate, but because of the high speeds, the bearing arrangements are challenging. Due to the high speeds, the radial bearings are lighter series cylindrical roller bearings. The heavier 23 series, with large rollers, is not suitable for the highest speeds. The thrust bearings are high-speed angular contact ball bearings with 15 or 25° contact angles or four-point angular contact ball bearings with 35° contact angles (**fig. 2** and **fig. 3**) With angular contact ball bearings in arrangements with a backup bearing, the internal clearance is minimized and the axial positioning accuracy of the rotor is improved. However, forces develop internally between the bearings because of centrifugal forces acting on the balls. This reduces bearing life (**diagram 3**, **page 82**).

For a pair of two bearings, internal forces can be minimized by using a 15° backup bearing, CD design, which can also be a hybrid bearing, with ceramic balls (**fig. 4**). In an arrangement of three bearings, two thrust bearings in tandem and one backup bearing, the backup bearing can have either a 15 or 25° contact angle. With a 25° backup bearing the reverse displacement is minimized but the internal loads are higher, compared to that of a 15° bearing (**fig. 5**).

The backup bearing can also be a hybrid bearing to minimize induced axial force on the main thrust bearings and increase bearing life.

In comparison to the above-mentioned bearing arrangements of angular contact ball bearings, a single four-point contact ball bearing has larger internal clearance and requires a larger rotor end clearance to avoid rotor rub.

An advantage with a single four-point ball bearing is that there is no additional load from a backup bearing and the 35° contact angle means a higher axial load capacity, compared to the 25° angular contact ball bearing.

If a spring-loaded backup bearing is used in combination with a four-point contact ball bearing (**fig. 3**), the step in the deflection curve (**diagram 1** and **2**, **page 82**), occurs at a negative axial force. This may be sufficient to avoid rotor rub if the axial force reverses.

Bearings with ceramic rolling elements (hybrid bearings) can be used both as main thrust and backup bearings. The reduced mass of the ceramic rolling elements compared to steel elements allows hybrid bearings to operate at high speeds while inducing lower internal forces, compared to a bearing with steel balls. The minimum axial load required to avoid gyroscopic spinning of the balls is also lower. For detailed bearing recommendations see **table 2**, **page 83**.

Fig. 2

Four point angular contact ball bearing arrangement for dry air compressor

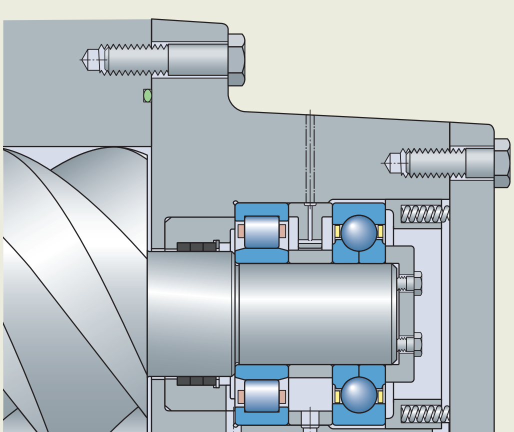
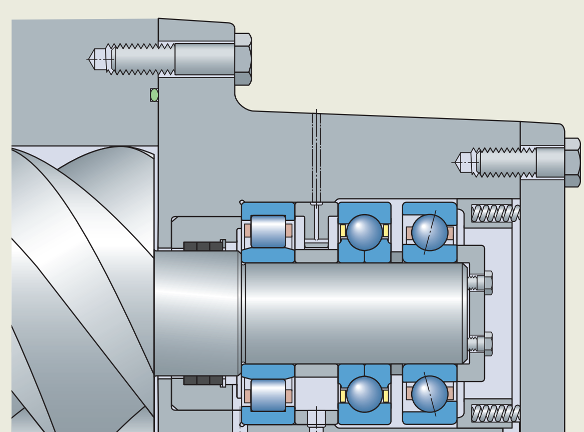


Fig. 3

Four point angular contact ball bearing arrangement for dry air compressor, with a spring loaded 15° angular contact backup bearing



Lubrication

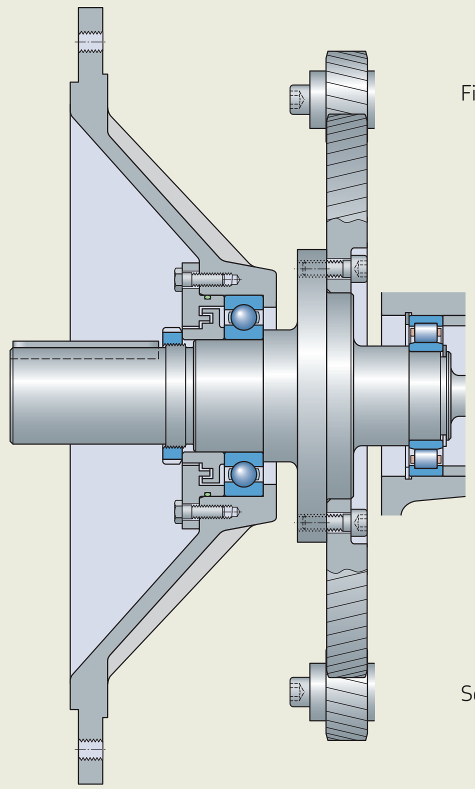
Lubrication is by oil jet – see chapter *Lubrication by oil jet*, **page 31** for details. A separate oil pump is used to circulate the oil and an oil cooler is used for cooling. The operating viscosity should be greater than ν_1, this means κ should be greater than 1,0 and preferably greater than 1,5.

Drive arrangements

The two compressors – first and second stage, are gear-driven by a common large diameter bull gear. The compressors and the bull gear are mounted together in a common housing. The bull gear is driven by a direct coupled motor and the oil pump is driven by the same motor. The bull gear is fitted on spherical roller bearing or cylindrical roller bearing with a deep groove ball bearing or a four-point contact ball bearing for axial loads (**fig. 6**). Alternatively, if the loads are light, it may be possible to use a deep groove ball bearing to take both radial and axial loads (**fig. 6**).

Fig. 6

Drive gear bearing arrangement for dry air compressor.
The large bull gear drives the pinions of the first and second stage compressors



6

Fig. 4

Angular contact ball bearing arrangement for dry air compressor with one 25° thrust bearings and one 15° backup bearing

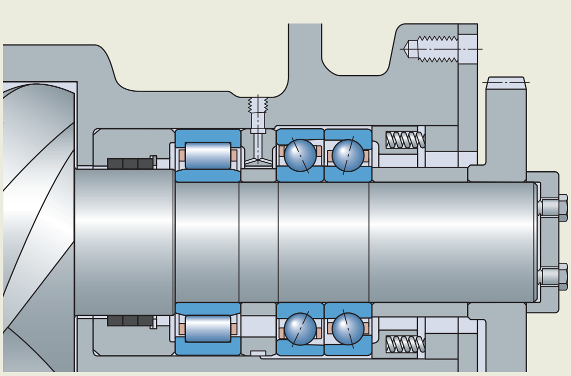


Fig. 5

Angular contact ball bearing arrangement for dry air compressor with two 25° thrust bearings in tandem and one 25° backup bearing

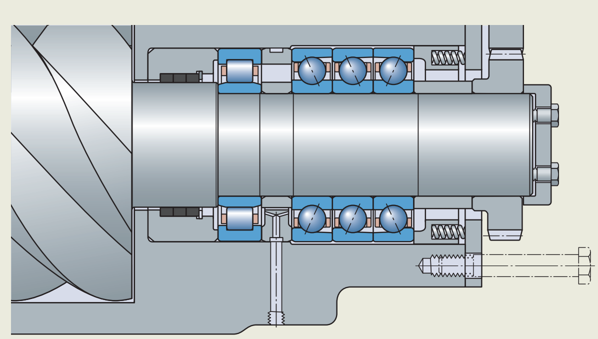


Diagram 1

Axial rotor displacement vs axial load
at zero speed

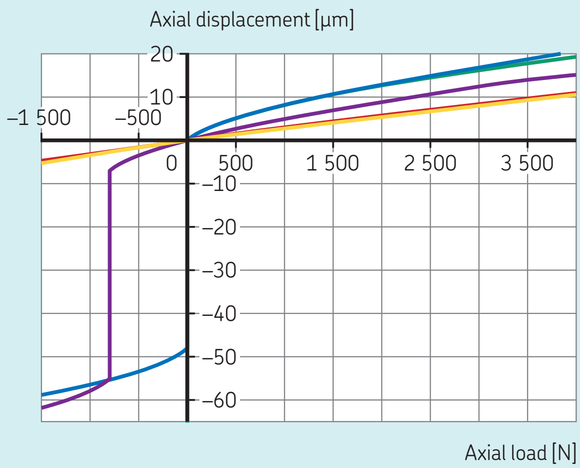


Diagram 2

Axial rotor displacement vs axial load
at nd_m = 1 000 000

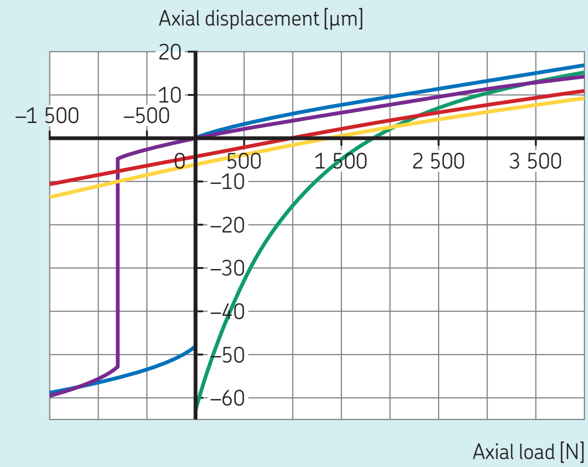
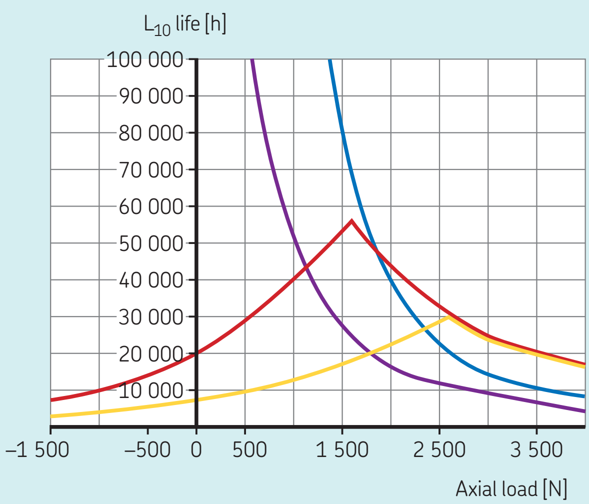
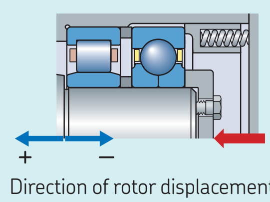


Diagram 3

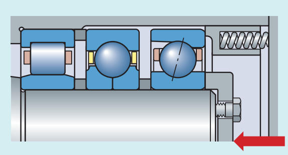
L_{10} life vs axial load.
The life shown is the shortest among the axial bearings of the arrangement at nd_m = 1 000 000



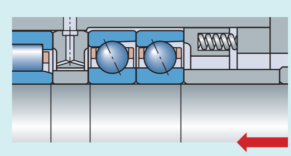
Legends for diagram 1 to 3



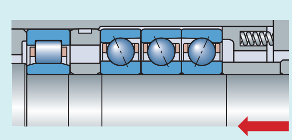
— One four point contact ball bearing, SKF QJ 210 C2L.



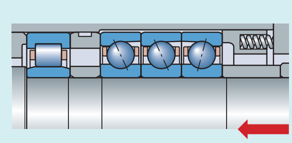
— One four point contact ball bearing SKF QJ 210 C2L and one spring loaded angular contact ball SKF 7210 CD.



— Two 25° angular contact ball bearings, SKF 7210 ACD, in tandem arrangement.
NOTE: This arrangement is not recommended because of the large negative displacement at high speed and low axial load (see **diagram 2**)



— Two 25° angular contact ball bearings, SKF 7210 ACD, in tandem arrangement and one 25° angular contact ball bearings, SKF 7210 ACD as back up bearing.



— Two 25° angular contact ball bearings, SKF 7210 ACD, in tandem arrangement and one 15° angular contact ball bearings, SKF 7210 CD as back up bearing.

Table 2

Bearing final variant specification and application recommendations for dry air screw compressors

Bearing type	Bearing series	Compressor design	Bearing bore diameter, d			Maximum misalignment
			$10 \le d \le 50$	$50 < d \le 100$	$100 < d \le 150$	
–	–	–	mm			minutes
nd_m range 800 000 to 1 000 000						
Angular contact for main load and backup bearings[1]	70 ACD, 72 ACD 72 AC, 73 AC	A	CB[2] k5/RF	CB[2] k5/RF	CB[2] m5/RF	1,5
		B	CB[2] k6/RF	CB[2] k6/RF	CB[2] m6/RF	3,0
	70 CD, 72 CD	A	CB[2] k5/RF	CB[2] k5/RF	CB[2] m5/RF	1,5
		B	CB[2] k6/RF	CB[2] k6/RF	CB[2] m6/RF	3,0
Four point contact ball bearings	QJ2 N2MA	A	C2L j5/RF	C2L j5/RF	C2L k5/RF	1,5
		B	CN k6/RF	CN k6/RF	CN m6/RF	2,0
Cylindrical roller bearings	NU 10, NU 2, NU 22 NU 3, NU 23	A	ECML/C3L m5/K6	ECML/C3L n5/K6	ECML/C3L p5/K6	1,5
		B	ECML/C3 m6/K7	ECML/C3 n6/K7	ECML/C3N p6/K7	3,0
nd_m range 1 000 000 to 1 200 000						
Angular contact for main load and backup bearings	70 ACD, 72 ACD	A	CB[2] k5/RF	CB[2] k5/RF	CB[2] m5/RF	1,5
	70 CD, 72 CD	A	CB[2] k5/RF	CB[2] k5/RF	CB[2] m5/RF	1,5
Four point contact ball bearings	QJ2 N2MA/P5	A	C2H k5/RF	C2H k5/RF	C2H m5/RF	1,5
Cylindrical roller bearings	NU 10, NU 2, NU 22	A	ECML/C3L m5/K6	ECML/C3L n5/K6	ECML/C3L p5/K6	1,5

Compressor design A: For maximum screw positioning accuracy.
Compressor design B: Best bearing availability.

[1] Single or tandem main bearings, single backup bearings
[2] CB clearance for DF and TFT sets (**table 4**, **page 48**)

Additional recommendations
Use hybrid bearings (HC suffix) for highest speeds and/or to minimize reverse axial displacement.
For angular contact ball bearings and four point contact ball bearings taking axial loads only, it is possible to use the same shaft tolerance as for the adjacent cylindrical roller bearing. The influence on clearance and/or stand-out between the rings shall be checked.
Total run-out and surface finish tolerance recommendations refer to **tables 5**, **7** and **8** on **pages 39** to **40**.
Maximum temperature for cages, see **table 4**, **page 38**.
Use ACD or AC backup bearings to minimize reverse axial displacement.
Use CD backup bearings to minimize internal axial loads induced by centrifugal force.
Use hybrid CD backup bearings (HC suffix) to further minimize internal axial loads induced by centrifugal force.
For advice on how to select bearings and bearing arrangement, please contact SKF application engineering service.

Refrigerant compressors

In the late 1960s, oil-injected screw compressors for refrigeration were developed, using both ammonia and fluorocarbons as refrigerants. Some compressors were new designs, while others were based on a similar size air compressor. This development was followed in the 1980s by air conditioning compressors using first R22 and after the Montreal Protocol in 1989, R134a as refrigerant. With refrigerant compressors came new challenges with material compatibility, dilution of oil by refrigerants and heavier bearing loads. The refrigerants changed again in 1997, after the Kyoto protocol, since the new hydro-chloro-fluoro-carbon (HCFC) refrigerants were found to have high Global Warming Potential (GWP).

A new phase down schedule for high GWP refrigerants was introduced at a conference in Kigali in 2016.

Hydrofluoroolefins (HFOs) such as R123yf, R1234ze have zero ozone depletion potential (ODP) and very low GWP and are increasingly used, following the Kigali phase down schedule, which varies from region to region. Various blends are also introduced such as R513 which is a blend of R134a and R1234yf with lower GWP (630) than R134a (1300) and zero flammbilty and toxcicity.

Air conditioning and refrigeration compressors using fluorocarbon refrigerants

As with air compressors, oil is injected into the compression cavity for lubrication of rotor-to-rotor contact, for sealing of leakage gaps between the rotors and between rotors and housing, and for cooling. An oil separator on the discharge side filters out oil, but the oil is diluted by the refrigerant. It is important to control the amount of dilution, since it reduces the effectiveness of lubrication, see **page 29**.

Operating conditions

The rotor tip speed is typically 30 to 60 m/s with bearing nd_m speeds of 300 000 to 650 000. The bearing loads are heavy and the choice of bearing arrangements may be limited by the space available radially between the rotors.

Speed

The motors have traditionally been induction motor two pole speeds, meaning 2 950 r/min for 50 Hz and 3 550 r/min for 60 Hz grids. This has limited the tip speeds for smaller compressors. With variable speed drives it is now possible to drive the motors at higher nominal speeds and to control compressor capacity by varying the speed.

Radial loads

The radial loads are highest at the female side discharge position, lighter at the discharge male position and less on the suction side.

If there is a gear drive, the radial loads are higher at the drive position, usually on the suction side.

Axial loads

The gas force is higher on the male rotor and lighter on the female. The net thrust bearing forces are a combination of the gas forces, backup or reverse thrust bearing forces, and at startup, inertial forces. Under certain conditions, the axial force may reverse. The risk is usually highest on the female rotor.

Bearing arrangements

Discharge side

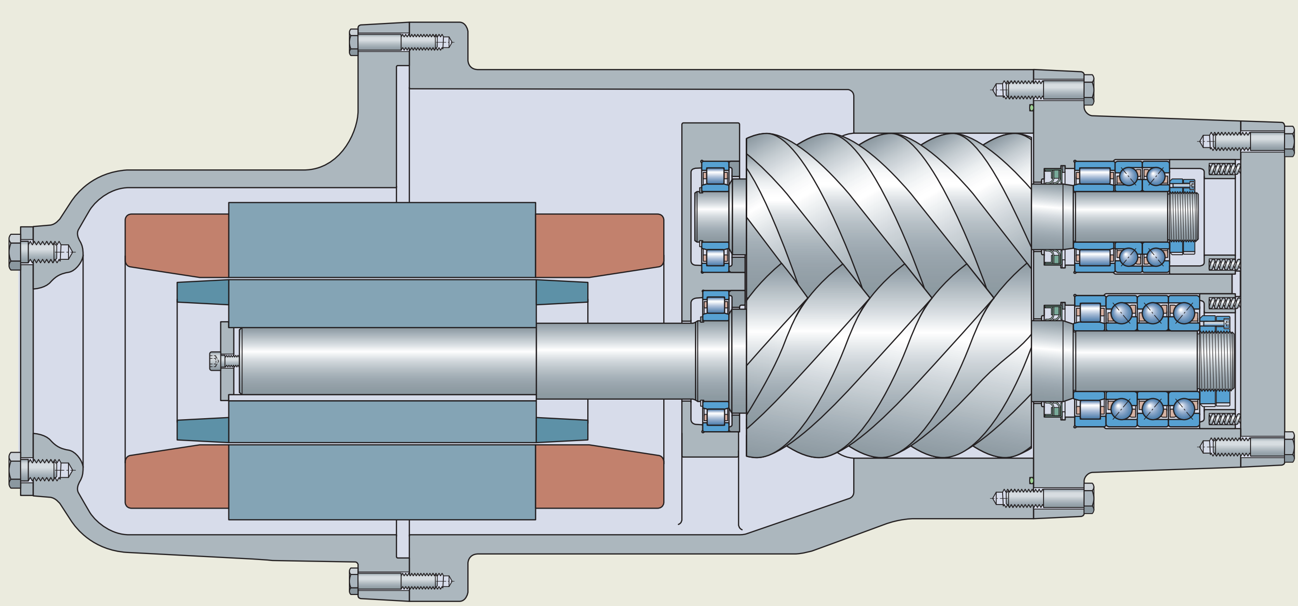
The bearing arrangements are a combination of cylindrical roller and angular contact ball bearing (**figs. 10** to **16**, **pages 21** to **23**) or cylindrical roller and four-point ball bearings.

The axial load on the male rotor is usually higher than the axial load on the female rotor and angular contact ball bearings in tandem is a common solution (**figs. 12** to **14**, **page 22** and **fig. 16**, **page 23**).

If there is a risk of reversing axial loads on either the male or the female rotor, reverse thrust bearings can be used (**figs. 13** and

Fig 7

Direct drive refrigerant screw compressor



14, **page 22** and **16** on **page 23**). For nd_m speeds greater than 450 000, angular contact ball bearings with 25° contact angle, AC design, are recommended as backup bearings to minimize internal loads generated by centrifugal forces.

Suction side

The common arrangement is a cylindrical roller bearing, (**figs. 24** and **25**, **page 26**). For semi hermetic compressors (*Drive arrangements*, **page 85**), the male suction bearing is selected for (large) bore size, the female discharge bearing for high load capacity.

Tapered roller bearings are not recommended, especially in ammonia compressors, due to the difficulty in lubricating the inner ring flange to roller end contact with oil diluted by refrigerant.

Bearings with ceramic rolling elements (hybrid bearings) can also be used in refrigerant applications. The properties of the ceramic rolling elements make them suitable for use in very high concentrations of refrigerant. Hybrid bearings also have higher speed ratings and are suitable in high speed applications.

Cages

The bearings can be fitted with polymer cages, P for polyamide and PH for PEEK, or brass cages M, MA, ML suffix. Tests at the SKF Engineering and Research Centre have shown polyamide cages to be suitable for use in fluorocarbon-based refrigerants up to bearing operating temperatures of 110 °C, while PEEK cages can be used in continuous temperatures up to 160 °C. For compatibility with gases see **table 4**, **page 38**.

Polymer cages are the optimum cages in low viscosity applications, provided the speed and temperature are within the permissible limits.

For detailed bearing recommendations see **table 1**, **page 79**.

Lubrication

Bearing lubrication is by circulating oil, with lubricant flow driven by the pressure difference between the discharge and suction side of the compressor. The oil can be cooled, but not all compressors use oil coolers.

Proper lubrication requires consideration of the effect of the refrigerant on oil selection and oil properties. Fluorocarbon-based refrigerants do not mix freely with mineral oils. When used with compatible oils, the refrigerants dissolve and reduce the viscosity of the oil, also reducing the property of the oil to increase in viscosity with pressure in the rolling contact. This diminishes the capability of the oil to develop a lubricant film at the point of rolling contact. The oil-diluting refrigerant also diminishes the ability of the oil to support sliding friction (cage and roller/flange contacts).

For proper bearing operation in refrigerant compressors, it is necessary to adjust the viscosity of the lubricant to account for the dilution of the oil by the refrigerant and also for the reduced pressure-viscosity relationship (see section on Oil viscosity, **page 27**). It is also necessary to adjust (increase) the minimum rated viscosity (ν_1) from **diagram 4**, **page 28**. The oil viscosity grade is typically ISO 46 or 68 for air conditioning compressors, but can be as high as ISO 120–370 for natural gas compressors to compensate for dilution of the oil by the gas. The operating viscosity should be greater than ν_1, this means κ should be greater than 1,0 and preferably greater than 1,5. Consult SKF application engineering service for more details.

Drive arrangements

Air conditioning compressors are almost all of the semi-hermitic design, meaning that the motor rotor is mounted on the extended rotor shaft and the stator is mounted inside the pressurized shell (**fig. 7**). With this design, there is no need for a coupling and seal between the motor and the rotor shaft. With variable frequency drives, the compressor capacity can be controlled by speed and the speed can be optimized for the compressor size. Other advantages with use of variable speed drives are less space requirement (footprint) and fewer parts needed.

Variable speed drives allow for drive frequencies both higher and lower than the grid frequency (50/60 Hz). Drive frequencies, and corresponding speeds higher than the grid frequency usually do not present problems, but the reduction in lubricant film thickness at low speeds is a limitation as to how low the speed can be. When calculating the κ value, the reduction in oil viscosity by refrigerant dilution must be considered.

Refrigeration compressors sometimes have a separate drive motor outside the pressurized shell, driving the compressor through a coupling and a mechanical face seal. A gearbox is sometimes used. For compressors with ammonia as refrigerant, such a driveline is necessary since the copper windings in the motor are not compatible with ammonia. One advantage with an open drive is that a defective motor or gearbox can easily be replaced.

Rust-preventative coatings

In refrigerant compressors, the rust preventative coatings used for rolling bearings must be selected carefully to avoid problems with chemical compatibility. The coating can act as a catalyst that accelerates reactions between the oil and the refrigerant, producing undesirable by-products. New refrigerants with low Global Warming Potential (GWP) are particularly sensitive to these reactions, because such refrigerants have been designed to have low chemical stability to achieve low GWP. Another potential problem is that the rust preventative can be partially dissolved by the refrigerant, only to coagulate at critical locations such as in pressure reduction valves. SKF has tested coatings for compatibility with refrigerants and compressor oils. Contact SKF application engineering service for more information.

Ammonia compressors

Ammonia compressors are similar to compressors using fluorocarbon-based refrigerants, with the exception that the ammonia refrigerant is not compatible with some materials, and some bearing types are not recommended.

Operating conditions

Screw tip speed is typically 40 to 60 m/s with bearing nd_m speeds of 300 000 to 500 000. The bearing loads are heavy and the choice of bearing arrangements may be limited by the space available radially between the rotors.

Radial loads

Radial loads are highest at the female discharge position, lighter at the discharge male position and less on the suction side. If there is gear or drive, the radial loads are higher at the drive position, usually on the suction side.

Axial loads

Axial gas force is usually higher on the male rotor and lighter on the female rotor. The net thrust bearing force is a combination of the gas force, axial gear forces, balance piston forces, backup bearing or reverse thrust bearing forces and, at startup, inertial forces.

Bearing arrangements

Discharge side

Bearing arrangements are a combination of cylindrical roller and angular contact ball bearings (**figs. 10** to **16**, **pages 21** to **23**) or cylindrical roller and four-point ball bearings (**fig. 17**, **page 23**). If the axial load on the male rotor is high, angular contact ball bearings in tandem is a possible solution. If there is a risk of reversing axial loads on either the male or female rotor, reverse thrust bearings can be used (**figs. 13** to **16**, **pages 22** to **23**).

One common arrangement is a single four point ball bearing as the (fixed) thrust bearing and a rotating balance piston on the suction side. The balance piston limits the axial force and makes the arrangement with a single four point ball bearing possible. The balance piston should not be too big, so that the axial gas force becomes overbalanced and causes frequent axial load reversals.

Suction side

The most common arrangement is a single cylindrical roller bearing. If the radial loads are light it may be possible to use deep groove ball bearings (**figs. 24**, **25** and **27**, **page 26**).

Gear shaft

For gear shaft bearing arrangements see **figs. 27** and **28**, **page 26**.

Preferred bearing types

Bearing types with low friction, such as angular contact ball bearings, four-point contact ball bearings and cylindrical roller bearings, are preferred. Customers have experienced severe wear in tapered roller bearings, between the roller end face and the inner ring flange. This wear phenomenon has been reproduced in laboratory testing. For this reason, SKF does not recommend tapered roller bearings and other bearing types with high sliding friction, such as cylindrical roller thrust bearings, spherical roller thrust bearings, and thrust ball bearings, in ammonia compressors.

For detailed bearing recommendations see **table 1**, **page 79**.

Fig. 8

GEA Screw compressor package SP2
Two-stage ammonia compressor package for refrigeration units

Cages

Pressed steel cage (J suffix)

SKF does not recommend pressed steel cages in ammonia compressors because of the impaired lubrication due to undissolved ammonia gas in the lubricating oil. The problems associated with J cages in ammonia compressors are cage wear and smearing between the cage and rolling elements. The experience is somewhat mixed, some customers have experienced such problems, others not.

Polyamide 6,6 cage (P and TN9 suffix)

The presence of ammonia accelerates the aging of polyamide cage material at elevated temperatures. Polyamide cages are successfully used in ammonia compressors, but the bearing operating temperature should be limited to 80 °C, see **table 4**, **page 38**. This limit has been established by experience and laboratory testing.

PEEK cages (PH and PHA suffix)

PEEK has excellent resistance to ageing and chemicals. The temperature limit for PEEK cages in ammonia compressors is 120 °C, a limit that has been established in laboratory testing. Bearings in ammonia compressors typically operate at lower temperatures than 120 °C, see **table 4**, **page 38**.

Pressed brass (Y suffix) and machined brass cages (M, MA, and ML suffixes).

In the ammonia refrigeration industry, specifications such as the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) Handbook do not recommend the use of copper alloys in ammonia systems, since these materials are subject to stress corrosion cracking. For example, as copper tubes have residual stresses from bending, steel tubing is used instead of copper tubing.

Tests using ammonia have been developed to determine the presence of residual stresses in copper and brass. One such test is DIN 50916. Because of the industry's general recommendations regarding copper and brass, ammonia compressor manufacturers have traditionally avoided pressed brass and machined brass cages in compressor bearings. This position is technically correct for pressed brass cages, since pressed brass cages contain residual stresses from the pressing operation. SKF solid, onepiece machined brass cages, however, are stress-free and are not subject to stress corrosion cracking. Testing at the SKF Engineering & Research Centre and long term experience have shown that solid, onepiece machined brass cages do not crack or corrode in a concentrated ammonia environment.

Machined steel cage (F and FA suffix)

Because of the residual stresses found in pressed and some machined brass cages, many ammonia compressor manufacturers hesitate to use them, and prefer machined steel cages. The machined steel cage offers good contact geometry between the rolling elements and the cage, and stress corrosion is not an issue with steel. However, limited production affects the availability of the machined steel cage.

Lubrication

Bearing lubrication is by circulating oil, with lubricant flow driven by the pressure difference between the discharge and suction sides of the compressor. The oil is separated from the ammonia in a large tank, with the compressor and drive motor mounted on top of the tank (**fig. 8**).

Drive arrangements

Ammonia compressors always have a separate drive motor outside the pressurized shell, driving the compressor through a coupling and a mechanical face seal. A gear box is sometimes used. Such a driveline is necessary since the copper windings in the motor are not compatible with ammonia. One advantage with an open drive is that a defective motor can easily be replaced.

Natural and sour gas compressors

A development in parallel with the refrigerant twin screw compressor was the process and natural gas screw compressor. These compressors can be used either to replace or boost reciprocating compressors. Boosting enables the use of a much smaller sized reciprocating compressor at low inlet pressures. This development, and the eventual depletion of natural gas fields, opened up a market for gas well screw compressors. With gas well pressure declining, it has become necessary to draw the gas out of the well with the compressor. Other applications boost the pressure between the well and the gas processing plant, or inject CO_2 into the wells to boost well pressure. There are also numerous applications within petrochemical processing plants. The use of such gases means more difficult operating conditions for bearings in terms of lubrication, chemical compatibility and temperature.

Gas compression applications

Some of the most common process gases within oil and gas applications are methane (CH_4) and higher hydrocarbons, carbon dioxide (CO_2), sour gas (H_2S) and hydrogen (H_2) (**table 4**, **page 91**). In most oil and gas applications industry process gas has high water content and in many cases being fully saturated at the inlet to the compressor. The general recommendation is to keep the temperature at all locations at more than 10 °C above the dew point of the gas to avoid water condensation. The dew point and other gas properties can be estimated by using GPSA estimation methods, or by calculation using equations of state programs.

In addition, the gases may contain contaminants such as brines (cations and anions) ranging from very low concentrations of 200 ppm, to saturation i.e. about 300 000 ppm (sea water has about 35 000 ppm). Other contaminants include mercaptans (thiols), mercury compounds and quartz sand, all of which may affect the running conditions of the compressors and associated bearings. Some of the effects of these contaminants are given in **table 4**.

Fig. 9

Gas well boosting compressor installation handling sour and acid gas.

Bearing arrangements

The bearings and bearing arrangements of twin screw oil-injected process gas compressors are similar to those of refrigerant compressors and in some cases, bearing arrangements that are similar to those of air compressors. Gas compressors are often modified versions of refrigerant or air compressors (**fig. 15** and **17**, **page 23**). For sweet gas applications, cage selection recommendations are the same as for refrigerant compressors, and for sour gas applications, the recommendations are the same as for ammonia compressors.

Sour gas bearings

In compressor applications containing sour gas (H_2S), carbon dioxide (CO_2) and/or gaseous or aqueous water, there are several hydrogen induced cracking (HIC) failure mechanisms that affect conventional bearing steel (reference ISO 15156):

- In environments with low PH and gaseous and aqueous water, hydrogen stress cracking (HSC) results from hydrogen atoms diffusing into the steel and combine to form molecular hydrogen. Cracking then results from the pressurization that occurs at the trap sites, in combination with stress from external loads.
- Sulfide stress cracking (SSC) is another form of hydrogen stress cracking that is caused by corrosion on a metal surface when the metal reacts with hydrogen sulfide, producing metal sulfides and atomic hydrogen as byproducts. Hydrogen atoms can then diffuse into the metal and combine to form molecular hydrogen.
- Stress corrosion cracking(SCC) is another mechanism that results from corrosion in combination with stress.

For such applications, SKF recommends utilizing the SKF sour gas bearings. SKF sour gas bearings have rings made of high nitrogen stainless steel, ceramic rolling elements and PEEK cages.

The high nitrogen stainless steel not only prevents common corrosion, it is also a very good bearing material with superior fatigue resistance and it is much less sensitive to hydrogen induced cracking than conventional bearing steel. The ceramic rolling elements make the sour gas bearings less sensitive to poor lubrication conditions

In such applications, SKF recommends purging the compressor at a standstill with sweet or nitrogen gas.

The in situ pH for the process gas containing H_2S and CO_2 is given for various gas compositions with additives in the ISO 15156 and NACE MR0175 standards. However, a worst case estimate can be calculated with the formula below:

$$\text{In situ } pH_{20C} = 4{,}9 - 0{,}5 \log (p_{H_2S} + p_{CO_2})$$

where
p_{H_2S} = Partial pressure of H_2S in the process gas [kPa]
p_{CO_2} = Partial pressure of CO_2 in the process gas [kPa]
pH_{20C} = pH at 20 °C

For more information, contact SKF application engineering service.

Lubrication

It is common for the oil in an oil-injected gas compressor to become diluted over time by light hydrocarbon liquids, if the compressor is handling hydrocarbon gases. The viscosity of such an oil and light hydrocarbon mixture can be estimated by using the Irving equation on a volume fraction basis:

$$\ln \nu_m = \sum x_i \ln \nu_i$$

where
ν_i = The viscosity of the i component in the blend/mixture [mm²/s]
x_i = The volume fraction of the i component
ν_m = The viscosity of the blend/mixture [mm²/s]

A good approximation of the viscosity properties of dilution by light hydrocarbon liquids is to assume it to be natural gasoline (0,68 relative density) with the approximate properties:

- Density, $\rho_{0,68}$ 680 kg/m³
- Viscosity, $\nu_{0,68}$ at 40 °C 0,19 mm²/s
- Viscosity, $\nu_{0,68}$ at 100 °C 0,12 mm²/s

The actual amount of dilution can be estimated by measuring the density of the diluted oil and comparing it to the density of the fresh oil, assuming no change of density from any other source.

Other more accurate methods exist, however. Please contact SKF applications engineering service for more information.

Dilution using the density method is given by assuming dilution by 0,68 natural gasoline:

$$x_{0,68} = \frac{\rho_{lube} - \rho_m}{\rho_{lube} - \rho_{0,68}}$$

where
ρ_{lube} = density of lubricating oil [kg/m³]
ρ_m = measured desity of mixture [kg/m³]
$\rho_{0,68}$ = desity of natural gasoline [kg/m³]

Or by using the Irving equation for viscosities for mixed blends, the volume fraction of the hydrocarbons becomes:

$$x_{0,68} = \frac{\ln \nu_m - \ln \nu_{oil}}{\ln \nu_{0,68} - \ln \nu_{oil}}$$

where
$x_{0,68}$ = Volume fraction of natrual gasoline [mm²/s]
ν_m = Viscosity of mixture [mm²/s]
ν_{oil} = Viscosity of lubricating mineral oil [mm²/s]
$\nu_{0,68}$ = Visocisty of natrual gasoline [mm²/s]

For calculation of κ, the viscosity of the mixture should be multiplied with a correction factor for the lower pressure coefficient of the light hydrocarbon liquid natural gasoline), see **diagram 5**, **page 29**, similar to the adjustment for refrigerant/oil mixtures, see refrigerating example **page 36**.

Table 3

Contaminants of process gases

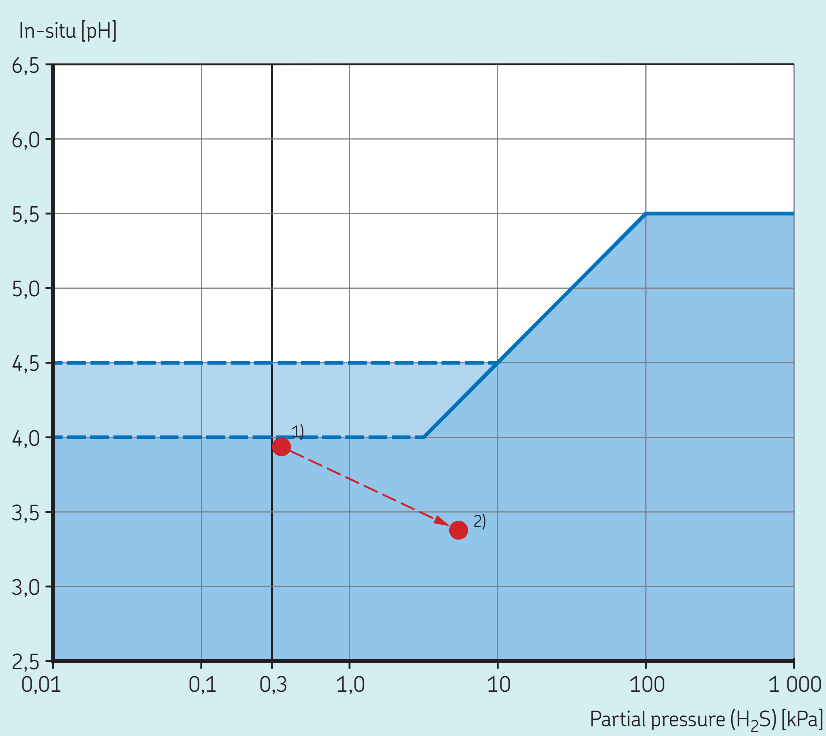
Contaminant	Comment
Brines / solids	They increase salinity of condensing water and therefore increase the electrical conductivity. Thus the ability to cause hydrogen charging, sulfide attacks, pitting corrosion and general corrosion increases. The solids may clog filters etc. at standstill and thus reduce oil-flow, especially at start-up by forming scale.
Mercaptans (Thiols)	Mercaptans are organosulfur compounds that often exhibit a strong odour and bond strongly with mercury compounds.
Mercury	Mercury (Hg) and mercury compounds such as HgS may be present in ppm levels in natural gases and refinery gases. Mercury and its compounds can be aggressive to metals by attack of grain boundaries.
Quartz sand	May cause wear and denting of rolling bearings if not separated or filtered from the gas stream.

6

Four point ball and cylindrical roller sour gas bearings with rings made of high nitrogen stainless steel and ceramic rolling elements and PEEK cages

Diagram 1

Recommended use of SKF sour gas bearings. (Ref ISO 15156)



Recommendations for use of SKF sour gas bearings:

- In applications defined by the blue area in the diagram
- When κ (adjusted for dilution of liquid gases) is less than 1.0
- At high risk of water condensation

In-situ pH = 4.9 – 0.5 log (p_{H2S} + p_{CO2})
p_{H2S} – Partial pressure H_2S (kPa)
p_{CO2} – Partial pressure CO_2 (kPa)
kPa = psi x 6.895

1) typical inlet conditions
2) typical outlet conditions

An example of the actual viscosity calculation for the blend /mixture of oil and natural gasoline is as follows:

Operating temperature:	T = 70 °C
Dilution rate is:	20 volume %
Viscosity at the actual operating temperature for natural gasoline:	0,14 mm²/s
Viscosity at the actual operating temperature for the oil:	20 mm²/s

$\ln \nu_m = x_{0,68} \ln \nu_{0,68} + (1 - x_{0,68}) \ln \nu_{oil}$
$\ln \nu_m = 0,2 \ln 14 + (1 - 0,2) \ln 20$
$\ln \nu_m = 2,00\ [mm^2/s]$
$\nu_m = e^{2,00}$
$\nu_m = 7,4\ [mm^2/s]$

The viscosity equation calculates the viscosity of natural gasoline at the appropriate temperature. This is done by using the viscosity vs. temperature, T [°C] equation for a 0,68 natural gasoline:

$$\nu_{0,68} = 0,0109\, e^{\left(\frac{894}{273 + T}\right)}$$

where
$\nu_{0,68}$ = Volume fraction of natural gasoline [mm²/s]
T = temperature [°C]

Please contact SKF application engineering service for more information.

Drive arrangements

Drive arrangements are similar to those of ammonia compressors (**fig. 8**, **page 86**), with the exception that in gas field applications, the driver is usually an engine fired by the gas being extracted. There are gears between the compressor and the engine to provide the correct compressor speed. In such drive arrangements, the same oil is used for compressor, gear and engine lubrication.

Table 4

Process gases, chemical formula and comments

Gas	Formula	Comments
Air	n/a	Air: 78% N_2 21% O_2 1% Ar
Ammonia	NH_3	Stress cracking gas by hydrogen ions in water.
Nitrogen	N_2	Inert if clean and dry. Used as purge gas at standstill of compressors.
Oxygen	O_2	Oxidizing and reactive gas. Requires special lubricants.
Water	H_2O	If condensed, creates conditions for stress cracking and general corrosion.
Carbon dioxide	CO_2	Acidic gas – Generally corrosive and may cause stress cracking in presence of liquid water.
Hydrogen sulfide	H_2S	Sour gas – Stress cracking gas either by sulfide stress cracking or by hydrogen ions. **NOTE:** Very poisonous, with a rotten smell, at low concentrations.
Hydrogen[1)]	H_2	Highly thermally conductive gas causing conditions for water condensation in hydrogen-rich gases. Reactive and may interact e.g. with oil additives causing detrimental conditions for conventional rolling bearings.
Methane	CH_4	Sweet gas – Used as purge gas at standstill, if clean and dry, of compressors.
Ethane	C_2H_4	Molecular weight higher than methane.
Ethene	C_2H_6	High molecular weight gas
Propene	C_3H_6	High molecular weight gas
Propane	C_3H_8	High molecular weight gas
Butane	C_4H_{10}	Very high molecular weight gas
Pentane	C_5H_{12}	Component in natural gasoline may dilute the lube oil over time.
Hexane plus	$\geq C_5H_{14}$	Component in natural gasoline, even small quantities may over time, dilute the lube oil

Hydrogen-rich gases, as often handled in refineries, frequently have a hydrogen content of about 60–80 mol%, with the remaining part comprising various hydrocarbons, acid and sour gases, and being fully saturated with water.
NOTE: Dry and pure hydrogen does not cause bearing issues by itself if not under extremely high pressures (tens of thousands kPa, hundreds of bars or several thousands psi). Ethene: Ethylene
Propene: Propylene

6

Bearing damage and failure

Bearings in screw compressors are very reliable, and most bearings outlive the screw compressors in which they are installed. Bearings are usually replaced when compressors are disassembled for reasons other than bearing failure. In some cases, however, bearing failure can occur. This chapter examines the underlying causes of bearing failures in screw compressors and suggests ways to prevent them.

Bearing damage and failure in screw compressors is usually the result of surface damage caused by:

- inadequate lubrication
- unforeseen contamination
- water condensation
- skidding of rolling elements
- the passage of electrical current

Bearings in screw compressors can fail prematurely for a number of different reasons. In addition to the causes mentioned, premature failure can also result from damage during transport or standstill, mounting problems or improper handling.

Each cause of failure produces its own particular type of damage and leaves its own special signature on the bearing.

Consequently, by examining a damaged bearing, it is possible in most cases to determine the root cause of the damage so that appropriate actions can be taken to prevent a recurrence.

Root cause analysis

It is difficult if not impossible to undertake root cause failure analysis on a damaged bearing, particularly if it has run to failure (**fig. 1**) without information on the operating conditions, a lubricant analysis, the bearing assembly drawing, and the operating speed and loads. Initial damage can give clues about the likely root causes. Often the initial damage is masked by secondary damage such as spalls, smearing or even ring cracks which ultimately lead to catastrophic bearing failure. For this reason, SKF recommends dismounting damaged bearings before they fail. This increases the chances of determining the root cause and allows appropriate corrective actions to be taken, reducing the risk of repeated bearing failures and collateral damage.

While some bearings can operate for months with easily-detectable minor surface damage, in others the damage develops more quickly. The period between the first detectable damage and failure can be less than the interval between two vibration analysis readings taken with a handheld device. As a result, continuous online temperature and vibration monitoring is often worth considering.

Whether bearing damage is light or severe, a thorough inspection can provide valuable information about the root cause of the problem. During an inspection, the key is to have a structured way of working. It is commonly understood that:

- A cause of damage (failure) results in a certain characteristic form of change.
- A certain failure mechanism results in a certain failure mode (pattern).
- From the damage observed, one can possibly determine the root cause of failure.

ISO has done much work defining and classifying different failure modes. This has resulted in the ISO 15243 standard (**diagram 1**). When looking at bearing failures, a total of six main failure modes can be observed, which can be further classified into a number of sub modes. The classification is based on three major factors:

- damage and changes that occurred during service (as soon as a bearing has left the factory)
- characteristic forms of change in appearance that can be attributed to a particular cause
- classified by visible features (including the use of non-destructive equipment for magnifying, such as microscopes)

Subsurface initiated fatigue

Material deterioration is caused by cyclic loading and the build-up of stresses just underneath the raceway surface, ultimately resulting in material decay.

Cracks are initiated and propagate underneath the surface, and when they come to the surface, spalling occurs (**figs. 2** and **3**).

Corrective actions

Considerations during operation

- Bearing steel manufacturing improvements have made pure material fatigue in high-quality rolling bearings rare.
- Fatigue spalling, where it does occur, is usually the result of an abnormal operating condition that leads to higher stress in the bearing,

Actions

- Only use high quality bearings.
- Check belt tension in a belt driven screw compressor.
- Check the bearing seating dimensional and geometrical tolerances.
- Make sure adjacent components are properly designed and manufactured.

Fig. 1

Failed bearings

Fig. 2

Spalling caused by fatigue

Fig. 3

Subsurface initiated fatigue on an angular contact ball bearing outer ring raceway

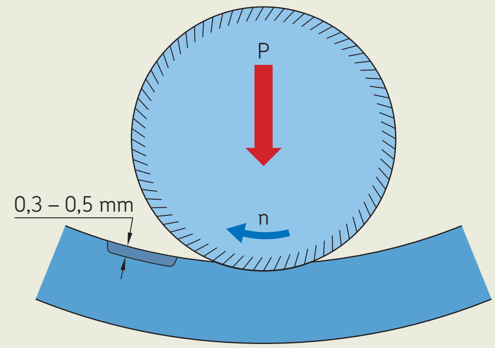


Diagram 1

Classification per ISO 15243:2017

- **5.1 Fatigue**
  - 5.1.2 Subsurface initated fatigue
  - 5.1.3 Surface initiated fatigue
- **5.2 Wear**
  - 5.2.2 Abrasive wear
  - 5.2.3 Adhesive wear
- **5.3 Corrosion**
  - 5.3.2 Moisture corrosion
  - 5.3.3 Frictional corrosion
    - 5.3.3.2 Fretting corrosion
    - 5.3.3.3 False brinelling
- **5.4 Electrical erosion**
  - 5.4.2 Excessive current erosion
  - 5.4.3 Current leakage erosion
- **5.5 Plastic deformation**
  - 5.5.2 Overload deformation
  - 5.5.3 Indentations from debris
- **5.6 Fracture and cracking**
  - 5.6.2 Forced fracture
  - 5.6.3 Fatigue fracture
  - 5.6.4 Thermal cracking

Numbering follows ISO 15243:2004 subclause numbers.

Surface initiated fatigue

Inadequate lubrication conditions cause surface-initiated fatigue. The role of the lubricant is to build up an oil film that separates the moving parts of the bearing. Under poor lubrication conditions, such as contamination or inadequate viscosity, surface asperities of the rolling element and the rings will contact, resulting in higher friction and stresses at asperity level. There is a risk of surface-initiated fatigue whenever the oil film does not fully separate the rolling contact surfaces.

Surface-initiated fatigue, in general, is the consequence of surface asperities coming in direct contact under mixed or boundary lubrication conditions (**fig. 4a**). When the loading and the frictional forces reach a given magnitude, small cracks form on the surface (**fig. 4b**). These cracks may then develop into microspalls (**fig. 4c**). Generally, the microspalls are only a few microns in size and the surface just looks dull and gray (**fig. 5**). Only under a microscope can cracks and spalls be detected (**fig. 6**).

Water in the lubricant can make lubrication inadequate. This can occur when condensate water has been present in the compressor. Another example: refrigeration compressors when the dilution by refrigerant is too high.

Corrective actions

Considerations during operation

- At high working temperatures there is a risk of inadequate lubrication conditions. For example: the cylindrical roller bearings on the female rotor discharge end see the highest load combined with the lowest speed.
- Viscosity can be inadequate at operating temperature due to variable speed drives, wear particles and the ingress of contaminants.

Actions

- Review the operating viscosity of the lubricant, taking actual operating conditions into consideration.
- Check the condition of the seals.

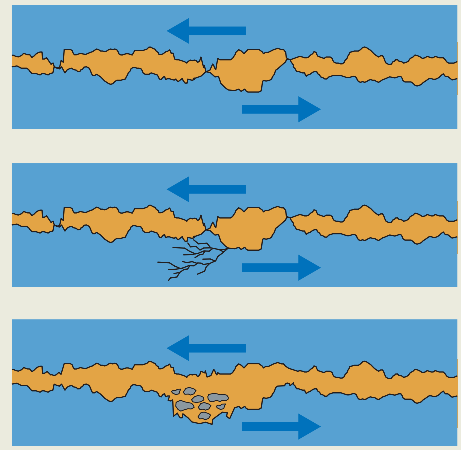
Fig. 4

Mechanism of surface initiated fatigue

a

b

c



Surface initiated fatigue on the inner ring raceway of a tapered roller bearing

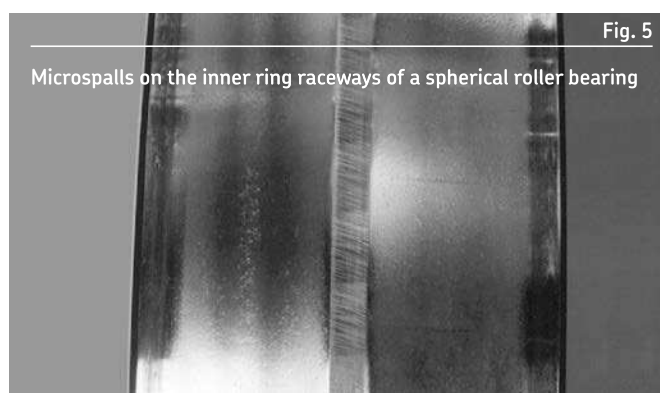


Fig. 5

Microspalls on the inner ring raceways of a spherical roller bearing

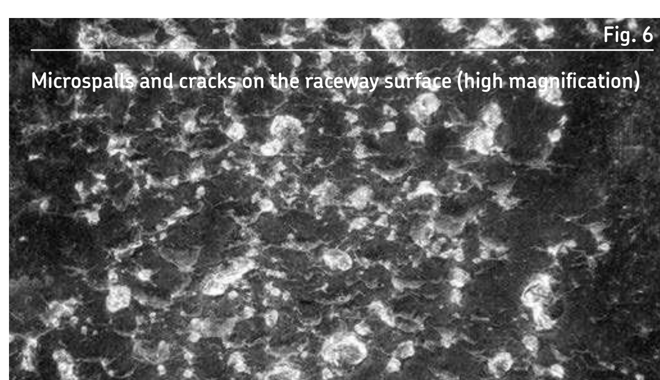


Fig. 6

Microspalls and cracks on the raceway surface (high magnification)

Hydrogen induced cracking[1)]

These are processes in which hydrogen atoms diffuse into bearing steel and weaken the material, causing cracks at low stress levels. The source of hydrogen can be either free water or hydrogen ions in acids. In sulfide stress cracking, hydrogen sulfide reacts with steel and releases hydrogen on the surface that can enter the material

A bearing subjected to hydrogen embrittlement or sulfide stress cracking can fail after a very short time, especially if the load is high. Damage can occur prematurely at lower loads also. Depending on the depth of the crack, the damage produces either surface flaking or local deep spalling. Both rings and rolling elements can be involved.

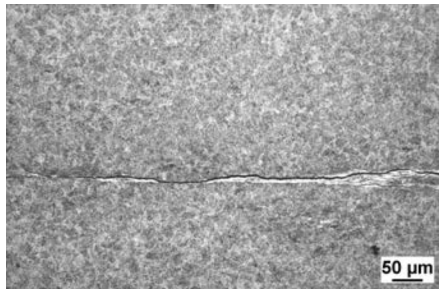
When viewed under a microscope, the sub-surface cracks in damaged rings appear to be decorated with white etching areas and are often referred to as white etching cracks (WEC). The white etching areas are harder than the surrounding steel matrix. They are produced when crack surfaces rub against each other, transforming the steel from martensite into nanocrystalline ferrite.

Corrective actions

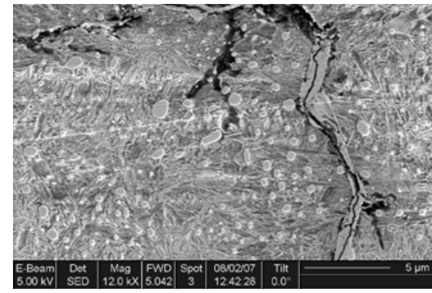
Avoid water condensation in the oil loop of the compressor, e.g. by ensuring that the discharge temperature is +10 °C above the dew point. In severe sulfide stress cracking regions (**diagram 1**, **page 90**) and for water saturated hydrogen-rich gases it is recommended to use so-called sour gas bearings.

1) Hydrogen embrittlement and sulfide stress cracking is not included in ISO 15243 but defined in ISO 15156.

Cylindrical roller bearing with a spalled area caused by hydrogen embrittlement



Example of white-etching crack in a tested deep groove ball bearing 6309 bearing inner ring that was tested with increased hydrogen content. Observed in an optical microscope, the white etching area appears lighter than the surrounding steel matrix.



White-etching areas (WEA) along the cracks in a bearing that was charged with hydrogen. Observation was made using a scanning electron microscope, in which the structure of the WEA is not white but can be clearly distinguished from the surrounding bearing steel matrix.

Some failure modes for conventional all-steel bearings in process gases

Spalling of steel raceways caused by stress corrosion cracking.

Splitting of steel balls caused by stress corrosion cracking

Calotte spalling of steel balls caused by hydrogen induced cracking from CO_2 and condensing water, causing general pitting corrosion.

Cracking of bearing shoulders in hydrogen-rich gases due to stress corrosion cracking caused by water condensation.

Wear

Wear is typical damage that occurs in the contact zones of moving parts. Wear most often is unavoidable. However, circumstances may cause wear to occur at an early stage of bearing operation. Two variants of wear can occur. These are abrasive (**fig. 7**) and adhesive (**fig. 8**) wear. They occur due to differences in the speed of the working contact surfaces. The cause of the speed differences can be kinematic slip, acceleration and/or deceleration.

Abrasive wear

This occurs due to abrasive particles in the lubricant. These can be contaminant particles coming from the outside or inside. The abrasive particles wear out the surfaces of the raceways, rolling elements and also metal cages. This normally results in dull surfaces. However, if the abrasive particles are very fine and hard, such as cement dust, a polishing effect might occur and mirror-like surfaces appear. Often, inadequate sealing arrangements result in contaminants entering the bearing cavity. An example is a compressor operating at high temperature and load. If particles and/or condensate water enter into the bearings the thin lubricant film under such conditions may not prevent abrasive wear.

Corrective action (against abrasive wear)

Check first whether the appropriate oil is being used. Verify lubricant selection (including viscosity, AW and EP additives).

If the lubricant contains contaminants or water, prevent condensation of water. Check the filter and improve filter ratings if needed. In high-temperature operating compressors, a lubricant with a higher viscosity may be needed to increase the viscosity ratio.

Adhesive wear

Adhesive wear occurs mainly in contact surfaces subjected to light loads, poor lubrication conditions and with important speed differences, resulting in sliding of the rolling elements. One example is the passage of rolling elements from the unloaded zone into the loaded zone. The rolling elements can lose speed in the unloaded zone and accelerate when returning to the loaded zone. This can result in break-through of the lubrication film, sliding, heat development and possibly material transfer from the rolling element to the raceway or vice versa. In an early stage, the appearance is shiny surfaces, but quickly it turns into a dull surface with (more or less) smeared material.

In angular contact ball bearings, the balls change their contact angle when they are outside the load zone but are forced back (with slip) to their correct contact angle as they enter the load zone.

Smearing can also occur between rolling elements and raceways when the load is too light relative to the speed of rotation (light load skidding). One example is on the discharge end, when clamping by springs is not used and when too high clamping force has been applied over the outer rings. The consequence will be that the cylindrical roller bearing is off loaded which causes sliding and smearing. Smearing can also occur in four point contact ball bearing if the axial load is light and lubrication poor. This can cause excessive three point contact and metal smearing (**fig. 9**). Three (or four) point contact can also occur if a four point contact ball bearing is loaded radially. In that case metal smearing and micro spalling can occur on both raceways in the radially loaded zone. Excessive amounts of ammonia make cylindrical roller bearings sensitive to light load skidding – typical failures are shown in **figs. 10** and **11**.

Fig. 7

Abrasive wear
Polishing wear causes mirror-like surfaces

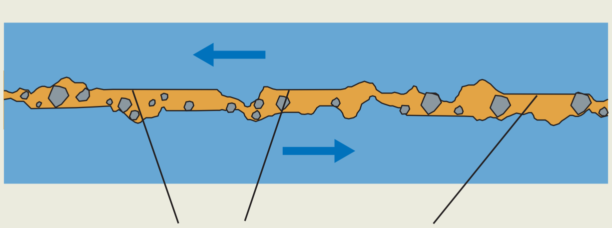
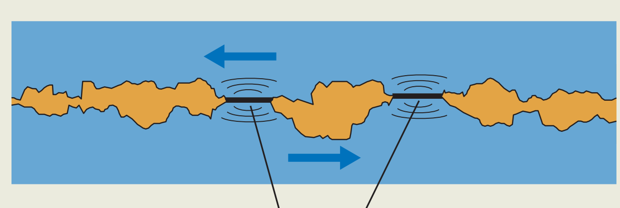


Fig. 8

Adhesive wear
Important speed differences and too light loads can cause sliding, resulting in very high temperatures and smearing



Corrective action (against adhesive wear)

In order to provide satisfactory operation and avoid smearing (adhesive wear), all ball and roller bearings must be subjected to a given minimum load.

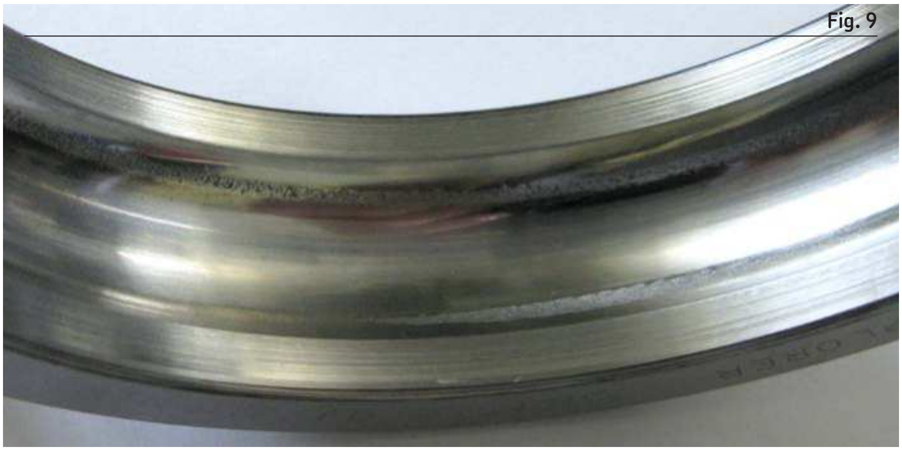
Select the right bearing type and bearing size to avoid conditions where the minimum load requirements for the bearings are not fulfilled.

Below are some of the methods to prevent or overcome smearing:

- Increase the load, which for ball bearings can be done by spring preloading
- Use smaller or lighter series bearings
- Use hybrid bearings (lighter rolling elements)
- Apply protective coatings
- Use different cage execution
- Select a smaller contact angle when using angular contact ball bearings.
- Verify lubricant selection (viscosity, AW and EP additives).

Both raceways in the outer raceways of a four point contact ball bearings are loaded due to that the bearing is loaded radially.

Fig. 9

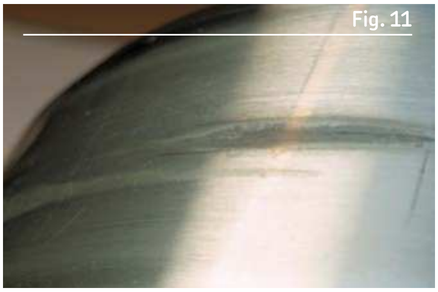


Light load in angular contact ball bearings can lead to spinning of the balls due to gyroscopic moment. Photo shows smearing marks on the ball of an angular contact ball bearing.

Fig. 10

Adhesive wear on the inner ring of a cylindrical roller bearing, caused by light load skidding.

Fig. 11



Corrosion

Moisture corrosion

In contrast to other damage processes, corrosion can occur quickly and penetrate deeply into the material. This can cause serious bearing damage. Corrosion occurs in the presence of water or moisture. Also, high humidity in the air and touching raceways with fingers can lead to this type of corrosion. It is therefore important to have good protection. Corrosion often happens during standstill and is then visible by corrosion marks at rolling element distance. Deep-seated rust leads to early bearing damage (**fig. 12**).

Corrective action (against moisture corrosion)

Store bearings in dry areas, where the risk of considerable temperature changes, which can cause considerable condensed water, is practically eliminated. For transporting into tropical areas – use tropical safe packages. Keep the preservative on new bearings and do not wipe it away during handling and mounting. Avoid contact of bearings with moisture, water, acid or other aggressive chemicals through appropriate sealing.

Fretting corrosion

Micro-movements between two loaded surfaces is the root cause of fretting corrosion. Mostly, this frictional corrosion occurs between the bearing outside diameter and housing and/or between the bearing bore and shaft. The micro-movements are mainly caused by the cyclic loads when rolling elements are passing by. Inadequate interference fit, shaft bending and/or imperfections in the contact surfaces can cause or accelerate corrosion. Air can contact the unprotected surfaces and accelerate the progression of corrosion. The formed iron oxide has a larger volume than pure steel. This can produce material growth and high stresses even to bearing raceways and can lead to premature fatigue. Fretting corrosion can easily lead to ring cracking.

Corrective action (against fretting corrosion)

Choose the correct shaft fit and housing fit (**table 7**, **page 40**).

Check the dimensional and geometrical tolerances and the surface finish.

Minimize high vibration levels and optimize rotor dynamics. Ensure sufficient stiffness of the bearing support structure and avoid overly high flexibility since this may lead to fretting.

Moisture corrosion schematic

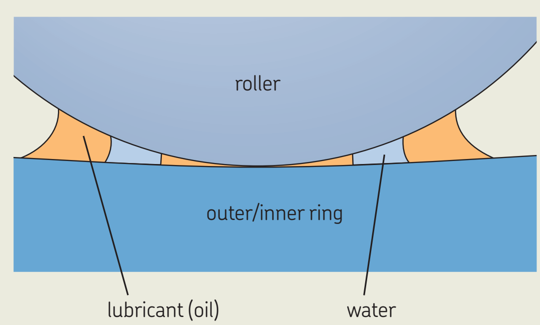


Fig. 12

Moisture corrosion on the inner ring of a cylindrical roller bearing

Fretting corrosion in the bore of bearings, resulting from inadequate shaft fit

False brinelling

False brinelling, also referred to as frictional corrosion damage, occurs in rolling element/ raceway contact areas due to micro movements and resilience of the elastic contact under cyclic vibrations. Since it occurs when the bearing is stationary and loaded, the damage appears at rolling element pitch. Depending on the vibration intensity, the lubrication condition and load, a combination of corrosion and wear occurs, forming shallow depressions in the raceways. Normally, the vibration results in a local breakthrough of the (protective) lubricant film, metal-to-metal contact, corrosion of the surfaces and abrasive wear. The appearance is therefore usually dull, often discoloured and sometimes reddish due to corrosion. Occasionally, the depressions can be shiny. False brinelling damage results in spherical cavities for ball bearings and lines for roller bearings.

Corrective action (against false brinelling)

False brinelling can be minimized by installing spring preloaded bearings (**page 52**).

Do not place non-rotating screw compressors close to vibrating machines or in an environment where significant vibrations occur.

Electrical erosion

Electrical erosion can occur when a current passes from one ring to the other through the rolling elements of a bearing. Excessive current (high current density) and current leakage (low current intensity) are two sub failure modes.

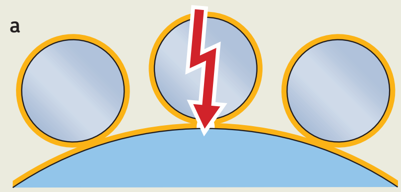
When an electric current (**fig. 13**) passes from one ring to the other via the rolling elements, damage will occur (**a**). At the contact surfaces, the process is similar to electric arc welding high current density over a small contact surface (**b**). The material is heated to temperatures ranging from tempering to melting levels. This leads to the appearance of discoloured areas, varying in size, where the material has been tempered, rehardened or melted. Craters also form where the material has melted and consequently broken away due to the rotation of the rolling element (**c**). The excess material on the rolling element wears away (**d**). Appearance: Craters in raceways and rolling elements. Sometimes zigzag burns can be seen in ball bearing raceways. Local burns are visible on the raceways and rolling elements.

In the initial stage of current leakage erosion damage, the surface is typically damaged by shallow craters that are closely positioned to one another and smaller in diameter compared to the damage from excessive current. This happens even if the current intensity is comparatively low. These are so small, they can hardly be seen without magnification.

False brinelling damage in the inner and outer ring of a cylinder roller bearing.
False brinelling damage occurs at standstill and is characterized by damage at rolling element distance.

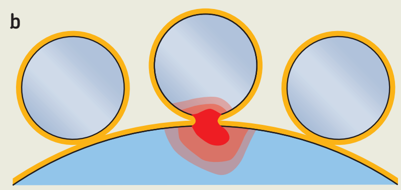
Fig. 13

a



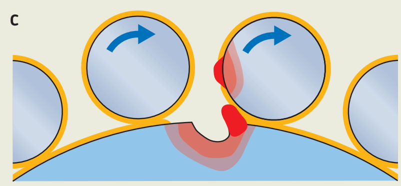
Electric current passes

b



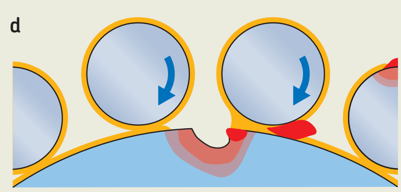
A kind of electric arc welding happens between the ring and rolling element

c



Melted material solidifies and breaks away

d



Excess material on the rolling element wears away

- Lubricant
- Molten material
- Rehardened material
- Annealed material

7

Fig. 14

Washboarding caused by current leakage erosion

A washboard pattern may develop from craters over time. The pattern appears on the raceways (**figs. 14** and **15**). In roller bearings, the washboard pattern also appears on the rollers (**fig. 15**). In ball bearings, the balls typically become discolored (dull, light to dark grey) over their entire surface (**fig. 16**).

The extent of the damage depends on a number of factors: current intensity, duration, bearing load, speed and lubricant (**fig. 17**).

Corrective action – a solution to electrical erosion

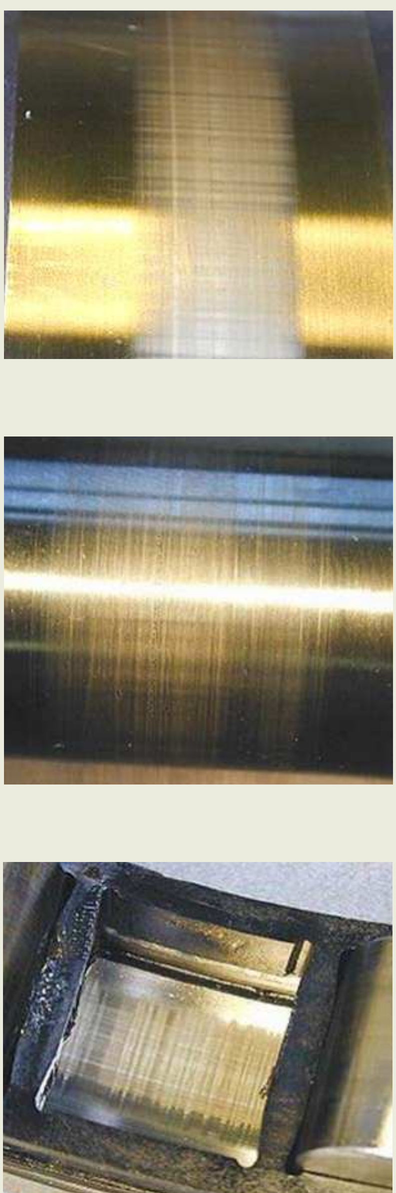
Currents traveling through the bearings can originate from grounding problems, frequency inverters, cabling and motors.

The very first stage of current leakage (electrical erosion) – current passing through the bearing is a grey dull surface in the raceway or on the rolling elements. The dull surfaces can easily be confused with other failure modes. It is therefore important to investigate the bearing with magnification.

Solving the problem of electrical erosion in compressor or motor bearings usually involves insulating the shaft from the housing so that stray currents are not grounded through the bearings. Although there is no single best way to do this, one solution is to use insulated bearings. SKF hybrid bearings with ceramic rolling elements provide sufficient insulation. An alternative way to insulate the shaft is to use bearings with rings that are coated with a ceramic material such as SKF Insocoat.

Fig. 15

Washboarding on inner and outer rings and roller raceways



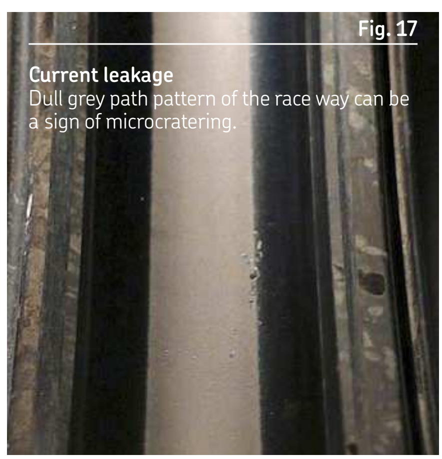


Fig. 17

Current leakage
Dull grey path pattern of the race way can be a sign of microcratering.

Fig. 16

Current leakage
Dull grey surface of the rolling elements can be a sign of microcratering.

Plastic deformation

Permanent deformation occurs whenever the yield strength of the material is exceeded.

Overload

Overload results from peak loads and leads to plastic deformation. This appears as depressions at rolling element distance in the bearing raceways. Often, wrong mounting procedures are the reason for the problem, i.e., applying the mounting force to the wrong ring thereby producing a peak load over the rolling elements. **Fig. 18** shows an example of poor assembly of a cylindrical roller bearing in the mounting stage. The rollers have made nicks on the inner ring raceway at roller pitch. If put into service, high noise and eventually failure will result.

Also, bearings must always be handled with care. Although made of highest quality steel, localized overloads, e.g., from dropping a bearing, might dent the surfaces and make the bearing unserviceable.

Indentation from debris

Indentations from debris (**fig. 19**) are caused by foreign particles (contaminants) that have gained entry into the bearing and are pressed into the raceways by the rolling elements. The dent size and shape depend on the particles' nature. The raceway geometry at the dent is destroyed and lubrication is impaired. Stresses arise at the edge of the dent and fatigue leads to premature spalling of the surface.

Fig. 18

Inner ring of a cylindrical roller bearing with nicks that occurred during mounting

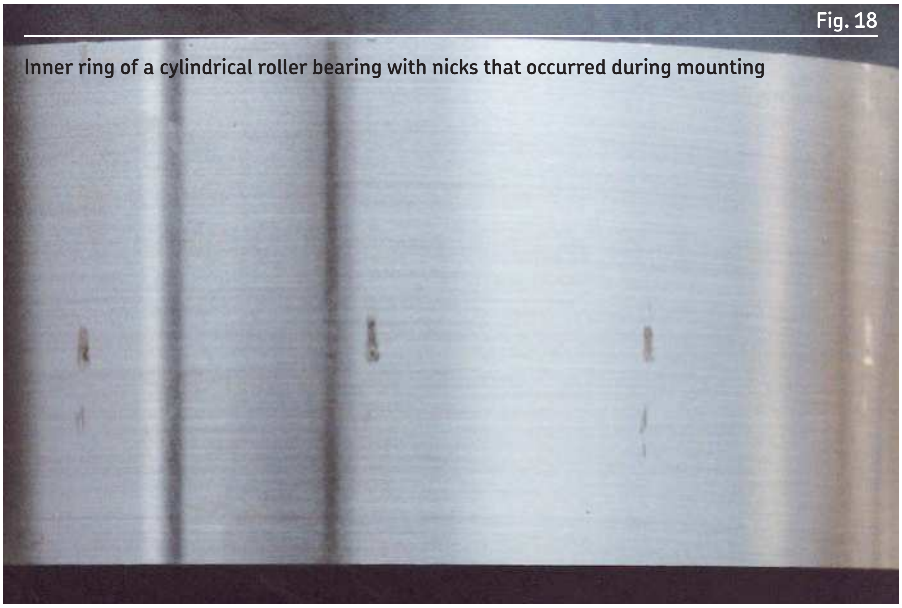
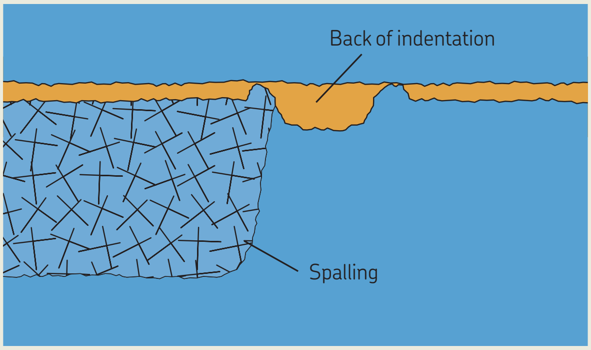


Fig. 19

Indentation from debris
Dents destroy the geometry and lead to surface initiated fatigue in the area behind the spall



7

Corrective action (overload)

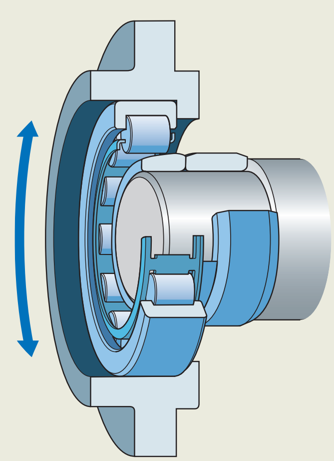
- Use appropriate mounting tools and methods.
- Vertical mounting is preferred to horizontal mounting in screw compressors
- Cylindrical roller bearings are separable radial bearings used in screw compressors. The outer ring is first mounted in the housing. The inner ring is mounted on the rotor shaft and then lowered into the housing. In order to prevent any damage during mounting of cylindrical roller bearings do the following:
- Apply lubricant to the cage and roller assembly. Make sure the lubricant also reaches the raceway. Also apply a thin layer of lubricant to the raceway of the other ring.
- When assembling, make sure that the roller assembly is not at an angle to the other ring. If either part of the bearing is assembled at an angle, it will be easy to damage a ring or rollers, especially if the rollers or raceways are not lubricated. To avoid this, use a guide sleeve when possible. Make sure the guide sleeve can be retrieved after assembly (**fig. 20**). To help prevent the rollers from scratching the raceway, turn the rotors during assembly.

Corrective action (against indentations from debris)

- Do not unpack the bearing until immediately before mounting.
- Keep the workshop and tools clean.
- Avoid using compressed air to clean the compressor parts.
- Flush the housing and the adjacent components before assembly.

Fig. 20

Mounting of separable bearings



Oil the raceway and rollers. Use a guiding sleeve. During assembly, rotate the rings relative to each other.

Fracture and cracking

Fracture (or cracking) occurs when the ultimate tensile strength of the material is exceeded.

Forced fracture

Forced fracture is caused by a stress con-cerntraction in excess of the material tensile strength by local impact or over-stressing. One common cause is rough treatment (impact) when a bearing is being mounted or dismounted. Hammer blows applied to a hardened chisel directly against the ring may cause the formation of fine cracks. The result is that pieces of the ring break off when the bearing is put into service.

Forced fracture
Forced fracture of a cylinder roller bearing inner ring.

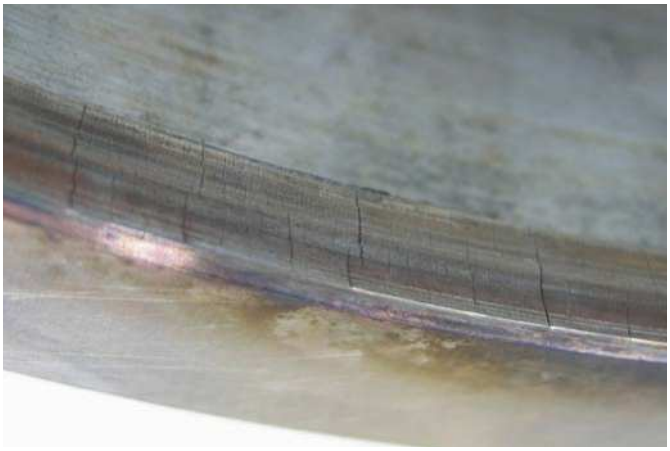
Fatigue fracture

This starts when the fatigue strength of the bearing material is exceeded. A crack is initiated and will then propagate. Finally, the whole ring or cage cracks through. A fatigue fracture starts when the fatigue strength of a material is exceeded under cyclic bending. Repeated bending causes a hairline crack which propagates until the ring or cage develops a through crack.

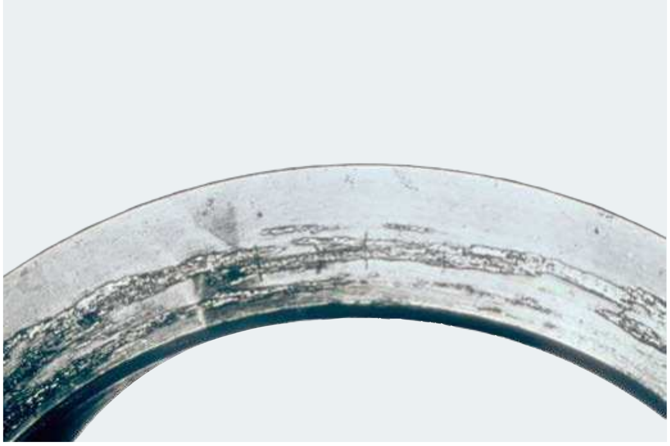
Fatigue cracking
fatigue cracking due to high dynamic cage pocket loads, also observable in the cage pocket guiding surfaces.

Thermal cracking

Thermal cracking can occur when two surfaces slide heavily against each other. One example is an outer ring centred cage that has seen poor lubrication conditions. Cracking can also occur if a ring is turning and rubbing against an abutment or nut. The frictional heat that is developed causes cracks, generally at right angles to the sliding direction.



Thermal cracking
Thermal cracking of a cylinder roller bearing outer ring shoulder.



Thermal cracking
Metal smearing and transverse thermal cracks on the side face of an inner ring caused by rubbing against a nut or abutment.

Corrective actions (against fracture and cracking)

- Apply fits according to **table 6**, **page 40** and use appropriate mounting tools and methods.
- Handle bearings with care.
- Ensure that minimum load requirement is fulfilled in high speed applications
- Ensure that an appropriate cages design have been is selected for the operating speed.

skf.com

PUB BU/P7 14510 EN · March 2018