Bearings in twin screw compressors

Application handbook
Bearings in twin screw compressors

Application handbook
This application handbook is one of a series of application handbooks designed to provide specific application recommendations for SKF customers to be used with the SKF General Catalog 4000.

It is not possible, in the limited space of this handbook, to present all the information necessary to cover every application in detail.

SKF applications engineers should be contacted for specific bearing recommendations. The higher the technical demands of an application and the more limited the available experience, the more advisable it is to make use of SKF's engineering service.

We hope you find this handbook interesting and useful.
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Introduction

The twin screw compressor was invented in the 1930's by Alf Lysholm, Chief Engineer at Svenska Rotor Maskiner (SRM) in Stockholm, Sweden. SRM acquired several key patents on the new compressor. The first application of the twin screw compressor was a supercharger for jet engines for airplanes. After further developments, an industrial air compressor was introduced in the mid 1940's. At that time, SRM also started to sell technology licenses. The first license was sold in the UK in 1946, followed by others in Europe, Japan and the USA. SRM is still active in compressor development today.

The first air 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 an injection of a fluid into the compression cavity.

A new development in the late 1950's was the oil flooded air compressor. This design did not use timing gears. Instead, the male rotor drives the female by contacts through the rotor flanks. Oil is injected into the compression cavity for purposes of lubrication, sealing and cooling.

The oil flooded design made it possible to operate at a lower speed while maintaining compressor efficiency and reducing the cost.

In the late 1960's, the oil flooded design was used in the development of industrial refrigeration compressors using both ammonia and halocarbons as refrigerants.

In the early 1980's, the industrial refrigeration compressors were followed by air conditioning compressors using primarily CFC-12 refrigerant and later HFC refrigerants.

The twin screw compressor design competed with the reciprocating piston compressor. The twin screw compressor design offered potential advantages in smaller physical size, lower vibration and noise level, and improved reliability, efficiency and cost. These benefits were sometimes not easily realized since the reciprocating piston compressor was easier to manufacture and could be made with less sophisticated equipment. Reciprocating piston compressors are still used today in many applications, at low volumes and high pressures.

From a rolling element bearing application standpoint, the twin screw compressor is very important since a large number of rolling element bearings are used in each compressor.
Twin screw compressor function

In a twin screw compressor, the two meshing rotors are turning in opposite directions inside the compressor housing. 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 in the housing. At this point the cavity begins to decrease in size as it continues to move forward in the compressor. As the cavity reaches the discharge side of the compressor, the compressed gas is discharged through the discharge opening in the housing (→ fig 1).

The position of the suction and discharge openings can be varied by means of a sliding valve. This makes it possible to control both the size and the ratio between the suction and discharge gas volume. The pressure increase in the compressor depends on the volume ratio, but for a given volume ratio, the pressure ratio depends on the thermodynamic properties of the gas.

Several compression cavities in various stages of compression are being compressed simultaneously. 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. By minimizing the clearance between the rotors and the housing, the leakage is minimized. Three different clearances must be considered, the clearances between the tips of the rotors and the cylindrical surface in the housing, the clearance between the end faces of the rotors and the housing ends, and the clearance between the rotors. The rotor end clearance is adjusted by axial positioning of the thrust bearing during compressor assembly, usually by grinding a shaft spacer to a width determined from measurements of compressor components.
1 General - twin screw compressor types

Twin screw compressor types

It is possible to classify screw compressor types in many different terms. The following distinctions of compressor types are important:
- Flooded compressors
- Dry running compressors

Flooded compressors

In flooded compressors, a fluid is injected into the rotor cavities during the compression process. The purpose of the fluid injection is:
- To seal the leakage gaps between the two rotors by filling them with fluid
- To absorb compression heat from the gas
- To lubricate the contacts between the two rotors

Oil flooded screw compressors

In oil flooded compressors, oil performs all of the three functions mentioned above. The sealing of leakage gaps and lubrication of the rotor contacts are very efficient.

The injected oil passes out of the rotor cavity with the discharge gas which passes 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 compression cavities and for bearing lubrication.

Oil flooded compressors operate with rotor tip speeds in the range 30-50 m/s, which means bearing \( nd_m \) values of 250,000-650,000. Typical shaft speeds are in the range of 3000 to 6000 r/min. Typical pressures for oil flooded screw compressors are 7 to 13 bar in one single compression step.

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

\( d_m = (d + D)/2 \)
**Water flooded compressors**

Water is also used for injection into the rotor cavity to absorb compression heat and to seal the leakage gaps. Since water has a high specific heat, water injection is more efficient at absorbing the heat and thereby reducing the discharge temperature.

Water is corrosive and is not an efficient lubricant, therefore the rotors have to be either made of stainless steel or coated with a polymer or similar material. An alternate design uses stainless steel rotors which do not touch and external timing gears are used to facilitate appropriate meshing.

Water injected compressors must have efficient seals between the rotors and the bearings to prevent leakage of water into the bearing lubricating oil.

**Liquid refrigerant injected compressors**

In refrigeration 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. This increases the adiabatic efficiency.

The compressor design can be either similar to the water injected compressor or a few percent of oil dissolved in the refrigerant can be used for bearing lubrication.

**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 tend to operate at higher temperatures than flooded compressors because no liquid is injected between the rotors. Because of the larger clearances and the lack of fluid for sealing the clearances, the leakage rate is higher for dry running compressors, therefore they are designed to run at high speed. By running at high speed, the compression is faster and there is less time for the leakage to occur.

Dry running compressors operate at rotor tip speeds over 60 m/s and bearing \( n d_m \) values in the range of 650,000-1,200,000. Typical shaft speeds are in the range of 10,000 to 25,000 r/min.

Since there is no contamination of the air by oil injection, dry running air compressors are used in applications such as medical and electronics and others where air contamination with oil is sensitive or environmentally prohibited.

Typically the pressure is 3 to 7 bar. To reach 7 bar, two compression steps are needed, with intermediate cooling of the air.

\[ *n d_m = n r/min \text{ multiplied by the bearing mean diameter } d_m \text{ in mm.} \]
\[ d_m = (D+D)/2 \]
Twin screw compressor bearing function and selection criteria

The function of bearings in twin screw compressors is to provide accurate radial and axial positioning of the rotors and to support the load on the rotors. These functions are to be performed reliably, with low friction and low noise generation.

With accurate positioning of the rotors, it is possible to design the compressor with small clearances for high efficiency. Radial positioning accuracy of the rotors is accomplished by using bearings having small operating clearances and high running accuracy (low run-out). Axial positioning accuracy is accomplished by small axial bearing clearance or preload. Axial positioning accuracy is also affected by the fit between the thrust bearing inner ring and the shaft and bearing deflection and displacement due to centrifugal forces. Interference fits will change the axial position of the outer ring after mounting. Axial positioning accuracy is also affected by the accuracy of the adjustment of the rotor end clearance during assembly. Thrust bearings mounted with interference fits make the adjustment more difficult.

In large industrial refrigeration compressors and in some air conditioning compressors, the available rotor center distance can be a primary criteria in the bearing selection. The rotor profile affects the center distance; therefore, the rotor design and bearing selection process is sometimes iterative. The center distance limits the size of the bearing outer diameters. 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.

This design option is however in conflict with the desire to minimize the number of different bearings in the compressor and 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. The main advantage with rolling bearings is their small operating clearances. 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 refrigerant flooding than hydrodynamic bearings.

Twin screw compressor bearing loads

Bearing loads in twin screw compressors are resultants of the following factors:

- 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 the inertia of the rotors at startup
- Electric motor rotor weight and forces for semi hermetic compressors
- Spring preload or balance piston forces

The gas pressure is low at the suction side and then increases towards the discharge side. The gas pressure along the length of the rotor cavity produces radial forces on the rotors. These forces are heavier towards the discharge side.

The gas also produces axial forces from the pressures 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 force on the rotor. This force is always directed towards the suction side and is heavier on the male rotor. In order to reduce the
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. 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 helps to balance out the net axial gas force on the rotor (→ fig 3).

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

If the compressor is gear driven, the forces from the input gear are also supported by the bearings. By varying the gear helix angle, it is possible to control both the magnitude and direction of the gear axial forces. Sometimes the axial gear force is used to balance the net axial gas force. This can cause the net axial gas force on the rotors to reverse which can cause rubbing between the ends of the rotors and the housing on the discharge side if the axial clearance of the selected bearing arrangement is larger than the rotor end clearance. Too low an axial force on the bearings can also be detrimental to the bearings 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 very complicated and should be performed through detailed analysis of compressor design parameters.
A stationary balance piston uses a bearing for transmission of the balancing force to the rotor.

Gas Force
Bearing types

Figure 3 illustrates the rolling bearings used in twin screw compressors. The most commonly used bearing types are the single row angular contact ball bearing and the cylindrical roller bearing, however, deep groove ball bearings, four-point contact ball bearings, needle roller bearings, and taper roller bearings are also used.

Bearing life

It is recommended that the basic rating life $L_{10h}$ and the SKF Life Theory rating life $L_{10aah}$ be used to select the SKF bearings for screw compressor applications. The SKF Life Theory considers that a rolling bearing can have infinite fatigue life provided the applied loads are below the fatigue limit, the bearing operates in a sufficiently clean environment, and was manufactured to accurate tolerances and with high quality steel. The SKF Life Theory enables the optimum bearings to be selected based on the service life conditions.

<table>
<thead>
<tr>
<th>Approximate relative load, speed and misalignment capabilities</th>
<th>Radial load</th>
<th>Axial load</th>
<th>Speed</th>
<th>Misalignment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single row deep groove ball bearing</td>
<td>X</td>
<td>X</td>
<td>XXX</td>
<td>XX</td>
</tr>
<tr>
<td>BE design single row angular contact ball bearing pair</td>
<td>XX</td>
<td>XXXX</td>
<td>XXX</td>
<td>X</td>
</tr>
<tr>
<td>High speed single row angular contact ball bearing pair</td>
<td>XX</td>
<td>XXX one direction</td>
<td>XXXX</td>
<td>X</td>
</tr>
<tr>
<td>Cylindrical roller bearing</td>
<td>XXX</td>
<td>—</td>
<td>XXXX</td>
<td>X</td>
</tr>
<tr>
<td>Needle roller bearing</td>
<td>XXXX</td>
<td>—</td>
<td>XX</td>
<td>X</td>
</tr>
<tr>
<td>CARB™</td>
<td>XXXX</td>
<td>—</td>
<td>XX</td>
<td>XXX</td>
</tr>
<tr>
<td>Four-point contact ball bearing</td>
<td>—</td>
<td>XXX</td>
<td>XXX</td>
<td>X</td>
</tr>
<tr>
<td>Taper roller bearing set</td>
<td>XXXX</td>
<td>XXXX</td>
<td>XX</td>
<td>X</td>
</tr>
</tbody>
</table>

—: No Capacity  
X: Low  
XX: Moderate  
XXX: High  
XXXX: Very High
The equations to use when calculating the bearing rating life are as follows:

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

\[ L_{10aah} = a_{SKF} \, L_{10h} \]

where

- \( L_{10h} \) = basic rating life in operating hours
- \( n \) = rotational speed, r/min
- \( C \) = basic dynamic load rating, N
- \( P \) = equivalent dynamic bearing load, N
- \( p \) = exponent for the life equation
  - \( p = 3 \) for ball bearings
  - \( p = 10/3 \) for roller bearings
- \( L_{10aah} \) = adjusted rating life according to SKF Life Theory in operating hours
- \( a_{SKF} \) = life adjustment factor based on SKF Life Theory

Each individual bearing within the compressor is generally selected to provide a basic rating life, \( L_{10h} \), in the range of 20000-60000 hours. Care should be taken not to over-dimension bearings to achieve too long bearing life due to risks of higher friction and light load skidding. Because of the high reliability requirements and number of bearings used in compressors, the system life, \( L_{10s} \), adjusted with the SKF Life Theory is sometimes considered. System life can be used to compare entire bearing arrangements.

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

\[ L_{10s} = \left[ \frac{N}{\sum_{i=1}^{N} \left( \frac{1}{L_{10i} e_s} \right) } \right]^{-1/ e_s} \]

where

- \( L_{10i} \) = life of an individual bearing, hours or revolutions
- \( L_{10s} \) = system life, hours or revolutions
- \( N \) = number of bearings in the system
- \( e_i \) = 10/9 for ball bearings only; 1.35 for roller bearings only, if \( p = 10/3 \); \( e_s \) for ball and roller bearing systems
- \( e_s \) = exponent for all bearings in the system; for ball and roller bearing systems, this can be estimated with the formula:

\[ e_s = \left( \frac{\sum_{i=1}^{N} e_i}{N} \right) \]

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.

The adjustment factor \( a_{SKF} \) for application of the SKF Life Theory is dependent on the viscosity (\( \nu \)) of the lubricant at the operating conditions compared to the minimum required viscosity (\( \nu_1 \)), the fatigue load limit of the bearing (\( P_u \)), and the contamination level (\( \eta_c \)) in the application. To enable a systematic and consistent evaluation of the contamination level, an SKF computer program, CADalog® (available upon request) has been developed for application of the SKF Life Theory. Continued research on the quantification of the contamination level in applications and the use with the SKF Life Theory will lead to further refinement of this program.

Contact SKF Applications Engineering for assistance in selection of the parameters used in the computer analysis.
If $\kappa > 4$, use $\kappa = 4$ curve.
As the value of $\eta_C (P_u/P)$ tends to zero, $a_{SKF}$ tends to 0.1 for all values of $\kappa$. 
If \( \kappa > 4 \), use \( \kappa = 4 \) curve
As the value of \( \eta_C (Pu/P) \) tends to zero, \( a_{SKF} \) tends to 0.1 for all values of \( \kappa \).
Bearing lubrication

Bearings used in twin screw compressors are lubricated by a 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. The oil lubricating the bearings can be the same oil injected into the compressor to lubricate the rotors and remove the heat of compression. In dry running (oil free) compressors the oil is supplied directly to the bearings and sealed from the compression cavity. The principal parameter for the selection of a lubricant for the bearings is the operating viscosity, \( \nu \).

Lubricating oils are identified by an ISO Viscosity Grade (VG) Number. The VG Number is the viscosity of the oil at 40°C (104°F). The common mineral oil grades are shown (\( \rightarrow \) fig 3). This is for oils having a Viscosity Index of 95. The Viscosity Index (VI) is indicative of the change in oil viscosity with increase in temperature. From this chart, 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. 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 have a higher viscosity at elevated temperature (\( \rightarrow \) fig 3). The Viscosity Index of synthetic oils can be in the range of 130 to 200. Oils having a high Viscosity Index have less viscosity decrease with increase in temperature. Common synthetic oil

![Fig 3](image-url)

**NOTE** Viscosity classification numbers are according to International Standard ISO 3448-1975 for oils having a viscosity index of 95.
The actual lubricant selected for an application should ideally provide greater viscosity than the minimum required viscosity $\nu_1$ (i.e. $\kappa > 1.0$). For bearings operating in air compressors, the viscosity ratio, $\kappa$, should be the guideline for evaluation of satisfactory viscosity. $\kappa > 1.5$ is preferred. The lubricant viscosity should not be too great since this causes excessive bearing friction and heat.

Some synthetic lubricants have different effective viscosity in the rolling contact as compared to mineral oils due to a greater or lower viscosity increase in the pressures of the rolling contact. This difference is indicated by the pressure-viscosity coefficient ($\alpha$) of the lubricant. The adjusted viscosity, $\nu_{adj}$ for use in the evaluation of the bearing lubrication can be determined as follows:

![Graph](image)

The lubricant viscosity requirements for a rolling bearing depend on bearing size $d_m$ and operating speed $n$, but little on bearing load. The minimum required lubricant viscosity $\nu_1$ needed at the bearing operating temperature is obtained from (→ fig 9). The minimum required viscosity from figure 10 is for bearings operating in air.

Types of synthetic lubricants used in compressors are the polyalphaglycol (PAG), polyalphaolefin (PAO), and POE. Synthetic oils are more thermally stable than mineral oils and therefore have longer service life. Synthetic oils can reduce the bearing internal rolling friction for improved compressor efficiency.

Furthermore, it should be noted that for the same basic viscosity, synthetic oils have different oil film thickness formation capability compared to the standard mineral oils. This is because of their different pressure viscosity coefficients (see literature on EHL theory). This coefficient can be higher or lower compared to mineral oils. This point should also be considered when selecting an oil.

The lubricant viscosity requirements for a rolling bearing depend on bearing size $d_m$ and operating speed $n$, but little on bearing load. The minimum required lubricant viscosity $\nu_1$ needed at the bearing operating temperature is obtained from (→ fig 9). The minimum required viscosity from figure 10 is for bearings operating in air.
The gas in the compressor (refrigerant, air, natural gas, etc.) can have a significant influence on the operating viscosity of the lubricating oil (see the section on refrigeration compressors). Extreme pressure and anti-wear additives are generally not used in refrigeration compressors.

Compressor lubrication systems often include filtration to remove solid particle contamination. Filtration is needed to clean the system of debris that can clog the small clearances in the compressor, the orifices of the lubrication and refrigeration system, and damage the bearings. Bearing life is affected by the cleanliness of the compressor and lubricant. Filters are rated according to a $\beta$ ratio. The $\beta_x$ ratio defines the efficiency of the filter to remove particles and the size of the particle. The $\beta_x$ ratio is defined as follows:

$$\beta_x = \frac{\text{No. of Particles entering filter}}{\text{No. of particles leaving filter}}$$

where:
- $x = \text{particle size, microns (} \mu\text{m)}$

For example, a rating of $\beta_3 = 200$ 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. The filter specification in compressors is typically in the range of $\beta_3 = 200$ to $\beta_{12} = 75$. Finer filters also increase bearing life, but the degree of increase depends also on the viscosity ratio Kappa and the bearing load intensity $P_u/P$. If the Kappa value is high, a change to a finer filter can give significant improvement in bearing life. If Kappa is low, a finer filter cannot compensate for the poor lubrication 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 Applications Engineering can consult computer programs which take into account all of the above mentioned factors.
Bearing speed ratings

The permissible operating speed is generally dependent on the kinematics of the rolling elements, which is influenced by these bearing features. Consult SKF Applications Engineering for advice and availability of bearings suitable for high speed operation.

Table below shows the recommended executions for the different bearing types for ranges of $n_{dm}^*$ values. These recommendations are generally valid for twin screw compressor applications only.

* $n_{dm}$ is the bearing speed $n$ in r/min multiplied by the bearing mean diameter $d_m$ in mm.

\[ d_m = \frac{(d+D)}{2} \]

Table 1

<table>
<thead>
<tr>
<th>Speed Range $n_{dm}$</th>
<th>single row angular contact ball bearings BE design</th>
<th>four-point contact ball bearings</th>
<th>deep groove ball bearings</th>
<th>cylindrical roller bearings</th>
<th>taper roller bearings</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cage Precision</td>
<td>Cage Precision</td>
<td>Cage Precision</td>
<td>Cage Precision</td>
<td>Cage Precision</td>
</tr>
<tr>
<td>Up to 450,000</td>
<td>$P,Y$ N 1)</td>
<td>MA,FA N 1)</td>
<td>J N 1)</td>
<td>P,J N 1)</td>
<td>J N 1)</td>
</tr>
<tr>
<td>450,000 to 650,000</td>
<td>M,F P6</td>
<td>MA,FA N 1)</td>
<td>J N 1)</td>
<td>P,ML N 1)</td>
<td>J *</td>
</tr>
<tr>
<td>650,000 to 850,000</td>
<td></td>
<td>MA,FA P6</td>
<td>MA P6</td>
<td>ML N 2)</td>
<td></td>
</tr>
<tr>
<td>Please contact SKFF</td>
<td>MA,FA P6</td>
<td>MA P6</td>
<td>ML N 2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>SKF Appl. Eng.**</td>
<td>MA P6</td>
<td>MA P6</td>
<td>ML N 2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>850,000 to 1,000,000</td>
<td>MA P6</td>
<td>MA P6</td>
<td>ML N 2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1,000,000 to 1,200,000</td>
<td>LA P5</td>
<td>MA P5</td>
<td>ML P6 3)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

1) N = normal precision
2) P6 running accuracy required however SKF Cylindrical Roller Bearings have this as standard
3) Shaft roundness tolerance to IT3/2 is recommended

* CL7C design

** An execution for special high speed angular contact ball bearings has been developed by SKF; these bearings have different cage executions and contact angles than those of the BE design. See section on “Bearings for high speed compressors”.

Please contact SKF Applications Engineering for advice and availability of bearings suitable for high speed operation.
Operating temperature is increased when heat is transferred to the bearings from compression heat and/or an electric motor. In refrigeration compressors, evaporation of refrigerant flowing through the bearings carries away heat. A lubrication method patented by SKF makes it possible to enrich the oil concentration in the bearings by the use of bearing frictional heat to vaporize the refrigerant and thereby deposit oil in the bearings. SKF makes this method available royalty free to its customers. Contact SKF Applications Engineering for more information.

It is important to know the expected temperature differences between the bearing inner ring and outer ring so that sufficient initial or unmounted clearances are provided. This is also dependent on the selected shaft and housing fits. Contact SKF Applications Engineering for a computer evaluation of the bearing clearance and fitting.

Bearing mounting

Shaft fits

The recommended shaft tolerances for ball and roller bearings in twin screw compressor applications supporting radial load or combined axial and radial loads are given in Table 2.

<table>
<thead>
<tr>
<th>Shaft diameter, mm</th>
<th>Tolerance for cylindrical and needle roller bearings</th>
<th>Tolerance for1 single row angular contact and deep groove ball bearings taking axial load only</th>
<th>Tolerance for taper roller bearings taking axial load only</th>
<th>Tolerance for single row angular contact and deep groove ball bearings taking combined loads</th>
<th>Tolerance for taper roller bearings taking combined loads</th>
<th>Tolerance for four-point ball bearings taking axial loads only</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; or = 18</td>
<td>m5</td>
<td>h5</td>
<td>j5</td>
<td>j5</td>
<td>j5</td>
<td>j5</td>
</tr>
<tr>
<td>(18) to 100</td>
<td>m5</td>
<td>h5</td>
<td>j5</td>
<td>k5</td>
<td>k5</td>
<td>j5</td>
</tr>
<tr>
<td>(100) to 140</td>
<td>m5</td>
<td>h5</td>
<td>j5</td>
<td>m5</td>
<td>m5</td>
<td>j5</td>
</tr>
<tr>
<td>(140) to 200</td>
<td>m6</td>
<td>h5</td>
<td>j5</td>
<td>m6</td>
<td>m6</td>
<td>j5</td>
</tr>
</tbody>
</table>

1) for nd_m greater than or equal to 650,000 use ISO k5 tolerance.
Housing fits, bearings taking axial load only

It is common for bearings supporting only axial load to be mounted radially free in the compressor housing. The housing bore should be 1 or 2 mm larger in diameter than the bearing outer ring. The axial load can vary and be very low or momentarily reverse direction when the compressor is operating at reduced pressures and volumes or at compressor start-up. For compressors with balancing pistons, the risk for reverse loading is higher. In these cases, the applied axial load can be less than the friction force between the face of the bearing outer ring and the compressor housing, in which case the outer ring may creep or turn from its initial position. This can cause wear and damage to the housing and bearing. This wear can change the axial positioning of the screw rotors changing the performance of the compressor.

For the bearing outer ring not to move against the housing, the axial force on the outer ring must be greater than the internal friction in the bearing. The axial force can come from the applied load or additionally from a clamping spring force (→ fig 11).

The recommendations above are the same as those in the General Catalog with the following exceptions:

• For smaller cylindrical and needle roller bearings, an m5 tolerance is recommended instead of a k5 to avoid ring creep.

• Single row angular contact ball bearings taking only axial load can use an h5 or a k5 tolerance. The h5 tolerance makes mounting easier and the screw positioning more accurate since an interference fit will lead to axial displacement of the outer ring.

Housing fits, bearings taking radial loads

The generally recommended housing tolerance for cylindrical and needle roller bearings supporting radial load is ISO K6. This tolerance results in an interference between the bearing outer ring and housing. This allows for easy assembly and radial clearance for bearing expansion with increases in temperature and is recommended to avoid creeping of the outer ring in the housing bore.

For easier mounting, it is also possible to use J6 or J7 tolerance. See Table 13 on page 38.
The necessary axial force can be determined from the following equation: \( F_a > 20 \frac{M_0}{D'} \)

where
- \( F_a \) = necessary axial load to prevent outer ring rotation, N
- \( M_0 \) = load-independent frictional moment in the bearing, Nmm
- \( D' \) = mean diameter of bearing outer ring side face, mm
- \( D' = \frac{(D + D_1)}{2} \)

The load independent frictional moment \( (M_0) \) for the bearing should be evaluated at the compressor start-up condition of temperature and viscosity.

Another criteria when determining the spring force is the magnitude of reverse axial load. The spring force selected should be the greater of these two forces. The spring force should be applied against the outer ring of the reverse axial bearing (→ fig 2).

The bearing rings can also be slotted (N1 or N2 suffix) and fitted with an anti-rotation pin mounted in the housing. This is common with four-point and single row angular contact ball bearings. The bearings can also be fitted in a floating housing which is pinned to prevent rotation.

**Miscellaneous**

The diameter of the abutment shoulders of the shafts (da) against which the bearing inner rings are seated should be designed toward the smaller diameter recommended in the SKF General Catalog. The use of small abutment shoulder diameters minimizes the possibility that form errors, which might occur in the manufacture of the shafts will distort or cock the position and form of the bearing rings.

The shafts and housings in which the bearings are seated should be manufactured within the recommended limits of dimensional form and running accuracy according to the SKF General Catalog.

The rings of the bearing should not be excessively clamped by the mating components. The use of too high a clamp force can distort and deflect the geometry and reduce the internal clearance in the bearings. The clamp force preferably should not exceed one quarter of the bearing basic static load rating (e.g. \( C_0/4 \)).
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 & Research Centre in the Netherlands have established the suitability of the common bearing materials for use in some gases. The gaskets, paints, and other seals within the compressor may also be affected by the composition of the gases. In some cases, the eventuality that the gases could contain contaminants such as moisture may also dictate the suitability of the material. Below is a list of maximum recommended temperatures and suitability for cage and seal materials for use in the various gases commonly used in compressor applications.

<table>
<thead>
<tr>
<th>Cage/seal material</th>
<th>SKF Suffix</th>
<th>Air</th>
<th>NH₃</th>
<th>HCFC-22</th>
<th>HFC-134a</th>
<th>Natural gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polyamide 6.6</td>
<td>P, TN, TN9</td>
<td>100</td>
<td>70</td>
<td>100</td>
<td>110</td>
<td>(2)</td>
</tr>
<tr>
<td>Brass, pressed</td>
<td>Y</td>
<td>p</td>
<td>np</td>
<td>p</td>
<td>np</td>
<td>p</td>
</tr>
<tr>
<td>Brass, machined</td>
<td>M, MA, ML</td>
<td>p</td>
<td>p</td>
<td>p</td>
<td>np</td>
<td>p</td>
</tr>
<tr>
<td>Steel, pressed</td>
<td>J</td>
<td>p</td>
<td>np</td>
<td>p</td>
<td>np</td>
<td>p</td>
</tr>
<tr>
<td>Steel, machined</td>
<td>F, FA</td>
<td>p</td>
<td>p</td>
<td>p</td>
<td>np</td>
<td>p</td>
</tr>
<tr>
<td>Nitrile (NBR)</td>
<td>RS1</td>
<td>95</td>
<td>60</td>
<td>np</td>
<td>np</td>
<td>p</td>
</tr>
<tr>
<td>Viton® (FKM)</td>
<td>RS2</td>
<td>190</td>
<td>np</td>
<td>p</td>
<td>(3)</td>
<td>p</td>
</tr>
</tbody>
</table>

All temperatures are in °C
p = Possible
np = Not Possible

1. Because of the impaired lubrication, J cages have not been recommended in ammonia compressors. 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, but several compressor manufacturers who originally used J cages in angular contact ball bearings and the old generation of cylindrical roller bearings have redesigned to other types. However, other compressor manufacturers have successfully used ECJ cylindrical roller bearings. Based on this experience, the J cage may be used in cylindrical roller bearings, but verification testing by the user is recommended.
2. Natural gas with high concentration of hydrogen sulfide may be too acidic for polyamide materials.
3. Viton® is resistant to this gas but is not compatible with PAG oils.
© VITON is a registered trademark for DuPont Dow Elastomer's fluoroelastomers.
Ball bearings in twin screw compressors

Deep groove ball bearings

Single row deep groove ball bearings are sometimes used to support the electro-motor in hermetically sealed refrigeration compressors, the gearing in high speed air compressors, and as the reverse axial load bearings in screw compressors. Deep groove ball bearings are also used in scroll compressors and reciprocating compressors.

Bearings having steel and polyamide cages can be used in most cases. The bearing internal clearance depends on the application but is typically greater than Normal (C3 suffix).

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 General Catalog. An axial load can be provided by an axial wave spring. The recommended minimum axial spring force is determined as follows:

\[ F = k \cdot d \]

where

F = minimum spring force, N
k = factor between 5 and 10
d = bearing bore diameter, mm

Deep groove ball bearings having seals or shields can be used to maintain an oil reservoir for the mechanical seal of hermetically sealed compressors. The mechanical seal requires a minimum oil submergence for adequate cooling and satisfactory life. The closure of the deep groove ball bearing can perform this function since the bearing is mounted adjacent to the mechanical seal. The material of the bearing seal should be checked for compatibility with the refrigerant. Nitrile, Viton®, and other materials can be used in the seals. Each material should be examined to determine its compatibility with the refrigerant and oil used. The standard bearing shields can also perform this function satisfactorily.

The angular misalignment capability is 2 to 10 minutes, depending on the clearance and loading.

Single row angular contact ball bearings

Single row angular contact ball bearings are the most commonly used ball bearings in twin screw compressors. They are used to 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 positioning accuracy of the shaft.

The most commonly used SKF single row angular contact ball bearings are of the 72 or 73 series.
Universally matchable single row angular contact ball bearings

Single row angular contact bearings are usually mounted in face-to-face arrangements to facilitate easy adjustment of the rotor end clearance. They must be manufactured for universal matching. SKF universally matchable bearings are designated with one of the following suffixes: CA, CB, CC, GA, GB, GC. The first letter denotes a clearance (C) or preload (G) and the second letter denotes the magnitude of clearance or preload.

Arrangements of universally matchable bearings usually support axial loads and assure accurate positioning of the compressor shaft owing to the small internal clearance or bearing preload. If the axial load of the compressor is heavy, the bearings can be arranged with a third bearing mounted in tandem, such as shown (→ fig 13).

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 any of the three arrangements shown (→ fig 13).

The GA suffix also denotes that the bearing is universally matchable, but a pair of these bearings will have a light preload when mounted in any of the three arrangements shown (→ fig 13). The GA preload is generally recommended in screw compressors.

Table 4 lists the values of unmounted axial clearance and preload for the universally matchable bearings. The initial bearing clearance or preload is assured when the bearing rings are axially clamped together. The initial clearance in a bearing pair is reduced or initial preload is increased by interference fits and if the shaft and inner ring operate with a higher temperature than the outer ring and housing.

SKF universally matchable bearings are produced with P6 precision class tolerances (ANSI/ABMA Class ABEC 3) as standard.

Caution: Single bearings are not to be used where only radial loads are present.

---

*nd<sub>m</sub> is the bearing speed n in r/min multiplied by the bearing mean diameter d<sub>m</sub> in mm.

d<sub>m</sub> = (d + D)/2
Clearance / Preload Class

Axial internal clearance of angular contact ball bearings of series 72 BE and 73 BE for universal pairing back-to-back of face-to-face (unmounted).

<table>
<thead>
<tr>
<th>Bore diameter (d mm)</th>
<th>Axial internal clearance Class</th>
<th>Preload Class</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>over incl.</td>
<td>CA</td>
</tr>
<tr>
<td></td>
<td>mm</td>
<td>µm</td>
</tr>
<tr>
<td>10</td>
<td>4</td>
<td>12</td>
</tr>
<tr>
<td>18</td>
<td>5</td>
<td>13</td>
</tr>
<tr>
<td>30</td>
<td>7</td>
<td>15</td>
</tr>
<tr>
<td>50</td>
<td>9</td>
<td>17</td>
</tr>
<tr>
<td>80</td>
<td>11</td>
<td>23</td>
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<tr>
<td>120</td>
<td>14</td>
<td>26</td>
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<tr>
<td>180</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td>250</td>
<td>26</td>
<td>42</td>
</tr>
</tbody>
</table>

Radial Clearance = 0.85 Axial Clearance
(0.0010 in. = 25.4 µm)

Preload of angular contact ball bearings of series 72 BE and 73 BE for universal pairing back-to-back of face-to-face (unmounted).
increases friction. This in turn increases the bearing temperature, reducing the effectiveness of the lubricant and the bearing life. Adequate axial load minimizes the risk of sliding.

Axial displacement of the bearing inner ring relative to the outer ring will occur depending on the magnitude of the axial load and speed (→ fig 15). The axial displacement can cause the compressor rotor end clearance to decrease. The rotor can contact the housing if the axial force on the bearing is insufficient. The magnitude of the force varies with the rotor speed. As bearing speed increases (nd_m* values 250 000 and greater) the axial load to minimize gyratory motion of the balls should be applied. Gyratory motion is ball spinning due to a gyroscopic moment. Gyratory motion will increase ball sliding and bearing friction. The value of nd_m* where greater axial load is needed in a particular situation is influenced by the magnitude of the applied loads, lubrication conditions, and construction of the bearing cage. Insufficient load can also cause variation in the orbital speed of the balls. This will result in increased loads on the cage and possibly cause damage.

Cages

The SKF BE design bearings are produced standard with three optional cages (→ fig 15): the glass fiber reinforced polyamide 6,6 cage (P suffix), the pressed brass (Y suffix), and the machined brass cage (M suffix). The bearings are also available with a machined steel cage (F suffix).

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 15). These differences in contact angle will cause sliding which damages the raceways, balls and cage, and

\[ nd_m^* = \text{bearing speed } n \text{ in r/min multiplied by the bearing mean diameter } d_m \text{ in mm.} \]

\[ d_m = \frac{(d+D)}{2} \]

Fig 14

BEP | BEM | BEY

Cages

Similar contact angles, inner and outer ring.

Variation in contact angles, inner and outer ring.

Fig 15

Axial displacement vs. axial load

SKF 7310 BEP   Speed = 3600 rpm

Fig 16

Axial displacement (µm)

0 2000 4000 6000 8000 10000

-300 -250 -200 -150 -100 -50 0 50

axial load (N)
Bearings having small contact angles are better suited for high speed, lightly loaded applications because of their lower requirement for axial load.

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

**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. The tandem mounted bearings can be positioned adjacent to a cylindrical roller bearing which supports the radial load (→ fig 5).

<table>
<thead>
<tr>
<th>SKF single row angular contact ball bearing minimum axial load factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>size</td>
</tr>
<tr>
<td>01</td>
</tr>
<tr>
<td>02</td>
</tr>
<tr>
<td>03</td>
</tr>
<tr>
<td>04</td>
</tr>
<tr>
<td>05</td>
</tr>
<tr>
<td>06</td>
</tr>
<tr>
<td>07</td>
</tr>
<tr>
<td>08</td>
</tr>
<tr>
<td>09</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>11</td>
</tr>
<tr>
<td>12</td>
</tr>
<tr>
<td>13</td>
</tr>
<tr>
<td>14</td>
</tr>
<tr>
<td>15</td>
</tr>
<tr>
<td>16</td>
</tr>
<tr>
<td>17</td>
</tr>
<tr>
<td>18</td>
</tr>
<tr>
<td>19</td>
</tr>
<tr>
<td>20</td>
</tr>
<tr>
<td>21</td>
</tr>
<tr>
<td>22</td>
</tr>
</tbody>
</table>
The angular contact ball bearings are mounted radially free (RF) in the housing, in which case they have a radial clearance of 1 or 2 mm with the housing bore. In this case, it is appropriate to determine the rating life of each angular contact bearing individually rather than as a set. The SKF 72 and 73 BE series universally matchable bearings have satisfactory tolerancing to distribute the axial load equally in tandem mounted arrangements. For tandem bearings supporting combined axial and radial load the bearing rating life is determined according to the SKF General Catalog.

For SKF bearings mounted in tandem supporting only axial load, the bearing rating life is determined using the applied axial load distributed amongst the bearings and using the basic dynamic load rating, C for one single row angular contact ball bearing.

An example of the life calculation for bearings arranged in tandem supporting only axial load is as follows:

(2) SKF 7310 BEGAP arranged in tandem
Fr = 0, Fa = 10000 N
n = 3600 r/min
C (1 brg.) = 74100 N

P = 0.35 Fr + 0.57 Fa
for Fa/Fr > 1.14

Load per bearing = Fa/2 = 5000 N
P = 0.57 (5000 N) = 2850 N

\[ L_{10h} = \frac{(C/P)^p}{(1000000/60 \times n)} = \frac{[74100 \text{ N}/2850 \text{ N}]^3}{[1000 \text{ s} / (60)(3600 \text{ r/min})]} = 81370 \text{ hours} \]

This calculation procedure does not apply to single row angular contact ball bearings supporting both axial and radial loads. In these cases, follow the procedures of the SKF General Catalog or contact SKF Applications Engineering to perform a computer analysis.

**Bearing preload**

The purposes of using preload in angular contact ball bearings are: avoidance of light load skidding, control of contact angles, improvement to internal load distribution, increased bearing stiffness, and improved shaft positioning accuracy. Bearing preload can increase the fatigue life of a bearing by improving internal distribution of the applied external loads (→ fig 18). Too great a preload can reduce bearing fatigue life. In twin screw compressor applications, bearing preload is used in angular contact ball bearings for all these reasons. Preloaded bearings are more sensitive to misalignment and incorrect mounting than are bearings with clearance.
achieved by the elastic deflection of the bearings against one another. The initial deflection of the bearings due to the preloading is $\delta_0$.

When an axial load is applied to the shaft, only one bearing supports this load. This bearing is denoted the "active" bearing. The deflection, $\delta$, of the active bearing reduces the load (i.e. preload) in the adjacent "inactive" bearing also called the back-up bearing.

The load-deflection diagram for a pair of preloaded bearings rotating at 3600 r/min is shown (→ fig 19).

Under rotation, the preload force is increased, and the force in the inactive bearing does not fully reduce to zero due to centrifugal forces. At increased speeds ($nd_m$ * values 250000 and greater), gyroscopic spinning of the balls will occur if the residual preload in the inactive bearing is less than the minimum required axial load, $F_a \text{min}$.

\[ nd_m = n \times \text{bearing mean diameter } d_m \text{ in mm.} \]

\[ d_m = \frac{d+D}{2} \]

Figure 19 shows the static load-deflection diagram for two preloaded angular contact ball bearings. This diagram is typical of 40° bearings arranged either back-to-back or face-to-face. Preload $P'$ in this example is achieved by the elastic deflection of the bearings against one another. The initial deflection of the bearings due to the preloading is $\delta_0$.

Under rotation, the preload force is increased, and the force in the inactive bearing does not fully reduce to zero due to centrifugal forces. At increased speeds ($nd_m$ * values 250000 and greater), gyroscopic spinning of the balls will occur if the residual preload in the inactive bearing is less than the minimum required axial load, $F_a \text{min}$.
Recommend bearing executions

Table below can be used as a guide to select a specific bearing execution, based on a given shaft and housing fit and an \( nd_m \) range.

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

To avoid significant bearing life reduction, the angular misalignment of the bearings should be limited to 4 minutes. This applies to preloaded single row angular contact ball bearings in the face-to-face arrangement, mounted radially free in the housing (→ fig 21).

---

### Table 6

<table>
<thead>
<tr>
<th>Shaft tolerance</th>
<th>( k_5 )</th>
<th>( k_5 )</th>
<th>( j_5 )</th>
<th>( j_5 )</th>
<th>( h_5 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Housing tolerance</td>
<td>J6</td>
<td>H6,RF</td>
<td>J6</td>
<td>H6,RF</td>
<td>RF</td>
</tr>
<tr>
<td>( nd_m ) value up to 250,000</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td>BECBY</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td>BECBP</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>BECBM</td>
</tr>
<tr>
<td>250,000 to 450,000</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td>BECBY</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td>BECBP</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>BECBM</td>
<td>BECBM</td>
</tr>
<tr>
<td>450,000 to 650,000</td>
<td>BECBM</td>
<td>BEGAM</td>
<td>BEGAM</td>
<td>BEGAM</td>
<td>BEGAM</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>not recommended area</td>
</tr>
</tbody>
</table>

RF = Radially free. This means that there is a 1-2 mm radial gap between the bearing outer ring and the housing.
For \( nd_m \) values lower than 450,000, housing tolerances J6, H6 may be replaced with J7, H7 respectively.
Applies to solid steel shafts/steel or cast iron housings. Applies to bearing bore size 20 to 100 mm (including 100 mm).
Applies to applications with inner ring temperature no more than 10 degrees C warmer than outer ring temperature.
Circulating oil lubrication or other means for improved cooling may be necessary for control of the bearing operating temperature, in particular, at high speeds.
Contact SKF Applications Engineering for details of the recommendations.

\( *nd_m \) is the bearing speed \( n \) in \( r/min \) multiplied by the bearing mean diameter \( d_m \) in mm.
\( d_m = (D + d)/2 \)
Four-point contact ball bearings

Four-point contact ball bearings (QJ prefix) are used in oil flooded and high speed dry air compressors to support axial load. The QJ bearing has a two piece inner ring allowing a high number of balls and 35° contact angle for support of high axial load. It has an outer-ring guided machined brass cage. The larger sizes of QJ bearings have an anti-rotation slot (N2 suffix) as standard. These features make the bearing well suited for high speed applications.

Four-point contact ball bearings should carry a minimum axial load for satisfactory operation. The formula for calculating the minimum required axial load to minimize gyratory motion of the balls is the same as for BE design single row angular contact ball bearings:

\[ F_{a_{min}} = A \left(\frac{n}{1000}\right)^2 \]

where

- \( F_{a_{min}} \) = minimum required axial load, N
- \( A \) = minimum load factor
- \( n \) = rotational speed, r/min

The equation above is more accurate than the corresponding equation given in the SKF General Catalog.

Values of minimum load factor \( A \) for series QJ 2 and QJ 3 four-point contact ball bearings are listed in Table 7.

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. Excessive clearance should be avoided since it negatively affects screw 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 very high speeds, it is necessary to use a larger clearance, for example C3.
For four-point contact ball bearings, a second criteria for minimum axial load must also be considered. The centrifugal force acting on the balls will tend to move the balls radially outward, similarly to that shown in figure 35. This movement is prevented by the contact with the outer ring at its primary rolling position. However, if the axial force is too small, the inner ring, balls, and shaft will displace axially towards the center of the outer ring ball groove. This can allow the ball to contact both sides of the outer ring raceway at the same time, while still contacting the inner ring raceway. This three point contact can result in severe sliding friction in the contacts, damaging the raceways, balls, and cage.

The axial load necessary to prevent the axial displacement of the ring and balls due to centrifugal forces has been calculated with an advanced computer program. With the use of regression analysis, it was found that the force increases with the speed raised to an exponent of approximately 1.45.

The minimum axial force to prevent three-point contact due to centrifugal force is calculated according to the following equation:

$$F_{a\min} = B \left(\frac{n}{1000}\right)^{1.45}$$

where

- $F_{a\min}$ = minimum required axial load, N
- $B$ = minimum load factor
- $n$ = rotational speed, $r/min$

Three-point ball contact is prevented when the load applied to the bearing by the compressor or additional forces from springs or balance pistons exceeds the force calculated by the above equation.

Values of the minimum load factor $B$ for series QJ 2 and QJ 3 four-point contact ball bearings are listed in Table 7.

### Table 7

<table>
<thead>
<tr>
<th>Size</th>
<th>A factor</th>
<th>B factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>QJ2</td>
<td>QJ3</td>
<td>QJ2</td>
</tr>
<tr>
<td>d (mm)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>03</td>
<td>17</td>
<td>0.427</td>
</tr>
<tr>
<td>04</td>
<td>20</td>
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<td>05</td>
<td>25</td>
<td>1.261</td>
</tr>
<tr>
<td>06</td>
<td>30</td>
<td>3.082</td>
</tr>
<tr>
<td>07</td>
<td>35</td>
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</tr>
<tr>
<td>08</td>
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<td>7.098</td>
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<td>12</td>
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<td>13</td>
<td>65</td>
<td>32.94</td>
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<tr>
<td>14</td>
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<td>39.93</td>
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N/A: Please consult SKF for availability
Roller bearings in screw compressors

Cylindrical roller bearings

SKF cylindrical roller bearings of EC design are used in twin screw compressors for their high speed and high radial load capability. The SKF EC cylindrical roller bearing has a large number and size of rollers, logarithmic roller profile, and optimized flange geometry.

The SKF EC design cylindrical roller bearings are produced with three basic cages (→ fig 23): the glass fiber reinforced polyamide 6,6 cage (P suffix), the pressed steel cage (J suffix), and the machined brass cage (M and ML suffix). The bearings are optionally available with a high speed light alloy cage (LP suffix) for high speed air compressors and a machined steel cage (F suffix) for gas compressors. The bearings are available with ranges of internal radial clearance for optimization of the bearing position accuracy. The SKF EC cylindrical roller bearing is produced standard with ISO P6 running accuracy.

The NU type cylindrical roller bearing is commonly used since it allows separate assembly of the inner rings and outer ring/roller assemblies onto the shaft and into the housing, respectively. The NU type bearing accommodates axial displacement due to thermal expansion of the shaft. The use of the NU type bearing allows both the inner and outer rings to be mounted with a transition or interference fit for more precise positioning of the bearings and rotors.

Cylindrical roller bearings are somewhat sensitive to misalignment. The maximum allowable misalignment is three to four minutes, depending on the bearing series.

**Important:** For satisfactory operation, cylindrical roller bearings should be subjected to a given minimum radial load. The required minimum radial load to be applied to cylindrical roller bearings is estimated from the equation provided in the General Catalog, which is:

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

- \( F_{rm} \) = minimum radial load, N
- \( k_r \) = minimum load factor
  - = 100 for bearings of series 10
  - = 150 for bearings of series 2, 3, or 4
  - = 200 for bearings of series 22
  - = 250 for bearings of series 23
- \( n \) = operating speed, r/min
- \( n_r \) = speed rating for oil lubrication, r/min, see bearing tables in the SKF General Catalog
- \( d_m \) = mean diameter of bearing
  - = 0.5 \((d + D)\), mm
The required minimum radial load for a cylindrical roller bearing in a given application is further dependent on the bearing cage type and lubrication conditions. Consult SKF Applications Engineering for details.

- for bearings having roller guided cages (P, J, M), use the value calculated for $F_{rm}$ above as the minimum radial load.

- for bearings having outer ring guided cages (ML), double the value calculated for $F_{rm}$ above as the minimum radial load.

If the minimum load is not maintained, the rollers will in part slide, in part roll on the bearing raceway. This is called skidding and can, but does not have to, lead to smearing. With good lubrication, smearing can be avoided even if the bearing is skidding. To minimize the risk for skidding, good oil drainage of the bearing cavity is important.

Selection of fits and radial internal clearance for cylindrical roller bearings in screw compressors

In order to achieve maximum positioning accuracy of the screws in a compressor and maximum bearing life, it is necessary for the bearings to have a minimum operating radial internal clearance. However, the initial bearing clearance and fitting of the bearing into the compressor must be designed to avoid the risk that the bearings become preloaded in operation.

The operating radial internal bearing clearance is a function of the initial radial internal bearing clearance, shaft and housing tolerances, and temperature of the bearing rings. The recommendations in Table 3 are designed to give optimum operating radial internal bearing clearance. 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 screw 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 results in low noise and vibration level in the compressor.

- A smaller operating radial internal clearance results in longer bearing life, increased stiffness, and reduced deflection.

It is important that the radial internal clearance is not too small for the following reasons:

- With too small a radial internal clearance, there is risk of radial preload and premature bearing failure.

- Too small a radial internal clearance can cause difficulties in assembling the compressor and damage to the bearing.

- If an interference fit is used in the fitting of the bearing into the housing, the housing seat may be ovalized due to the effect of the press fit on the unsymmetrical housing stiffness. This may increase the possibility of preloading the bearings and cause difficulties with the assembly.
Radial internal clearance ranges
The internal clearances in cylindrical roller bearings are according to DIN 620, part 4. The standard ranges of clearance are the Normal range and greater than Normal range (C3 suffix). Additional clearances having 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 the CNM and C3L ranges. The CNM suffix denotes the clearance is centered on the mean of the Normal range. The C3L suffix denotes the clearance is the lower half of the greater than Normal range. Bearings having reduced clearance ranges should be used for optimum screw positioning accuracy. The radial internal clearance ranges in the SKF General Catalog and the reduced clearances ranges (CNM, C3L) are valid even if rings from different bearings are interchanged. This is a unique feature of SKF EC design cylindrical roller bearings.

Bearing life as a function of radial internal clearance for an SKF NU 310 ECP/CNM (m5/K6 shaft and housing tolerance) C/P = 20; speed = 3600 rpm; Kappa = 1
Operating clearance based on Inner Ring temperature 10°C warmer than Outer Ring temperature
Application recommendations

Recommended shaft and housing tolerances and cylindrical roller bearing executions are defined in Table 8. The table includes the recommended cage execution, precision, bearing series, etc., since all these features have to be defined in the final bearing designation.

Availability

Not all cylindrical roller bearing variants (CNM, C3L, etc.) are currently in production. Availability may depend on usage by other customers, required volumes and delivery schedules. Please contact your local SKF sales office for details on availability.
Taper roller bearings

Taper roller bearings (TRB) are used in oil flooded air screw compressors at the male and female discharge positions, (→ fig 25). They can also be used to support the gearing in both oil flooded compressors and high speed dry air compressors. The taper roller bearing has high axial and radial load capability. Taper roller bearings arranged face-to-face (DF suffix) are used to limit the axial play of the rotors. A limiting factor is the speed capability of taper roller bearings. The speed rating is limited by the sliding friction between the rollers and the inner ring flange. Taper roller bearings are not recommended for refrigeration compressors because of the difficulty in lubricating the roller/flange contact and at the steel cage and roller contacts. This is especially difficult in ammonia compressors but also in other refrigerants, eg HCFC-22 and HFC-134a. The SKF Q line taper roller bearing is well suited for compressor applications. The Q line taper roller bearing features logarithmic roller profile, low friction steel cage design, and optimized roller end and flange profiles.

The SKF taper roller bearing is also available with the CL7C execution. This bearing has improved roller end and flange geometry to reduce running-in wear. The SKF taper roller bearing is available in the common ISO series. The 313 series, having high contact angle is well suited for the thrust position in compressor applications.

A taper roller bearing must operate with a certain minimum axial load for satisfactory operation. The axial load may be applied by the compressor or induced by the taper roller bearing mounted adjacent to it. The axial load must be greater than

\[ Fa > 0.5 \frac{Fr}{Y} \]

where

- \( Fa \) = total axial load on bearing, N
- \( Fr \) = applied radial load, N
- \( Y \) = bearing axial load factor, according to the SKF General Catalog

See also the SKF General Catalog for additional details.

It may be necessary to apply additional force to the bearing outer ring to prevent its rotation with the housing face if the force on the bearing ring is too low. See the section “Housing fits, axial loads only” on page 22. The face of the bearing can also be slotted to mate with an anti-rotation pin mounted in the housing. This is a non-standard feature.

Screw compressors fitted with taper roller bearings are relatively complicated to assemble, since the inner ring has to be mounted with an interference fit. The interference fit causes an axial displacement of the outer ring relative to the inner ring. This displacement may need to be considered when the rotor end clearance is adjusted.
When a reverse thrust taper roller bearing is used, then the clearance between the two taper roller bearings must be adjusted by means of shimming. A small axial clearance must be maintained since taper roller bearings cannot operate with preload in screw compressors.

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

**Needle roller bearings**

Needle roller bearings (→ fig 26) are used in oil flooded air and refrigeration compressors because of their high radial load capability and compact size. The needle roller bearing has separable rings similar to the cylindrical roller bearing. The needle roller bearing has limited availability with special clearances, cages or with higher precision.

To avoid significant bearing life reduction, the angular misalignment should be limited to 1 minute.

**CARB™ Bearing**

SKF has developed a new Toroidal Roller bearing type called CARB™ (→ fig 27) (Compact Aligning Roller Bearing). This is an entirely new bearing which is a combination of the best features of the spherical roller bearing, the cylindrical roller bearing, and the needle roller bearing.

The CARB bearing is capable of accommodating axial expansion and misalignment while providing high radial load carrying capabilities, low friction and compact cross section.

CARB bearings are produced in the same dimension series of many popular spherical roller, cylindrical roller, and needle roller bearings.

CARB bearings with cages are recommended for screw compressor applications.
Bearings for high speed compressors

High speed compressors, which are typically dry air twin screw compressors, operate at \( n \cdot d \) values greater than 750,000. Normal design and precision bearings are not suitable for operation at such high speeds since the centrifugal forces are too great and can lead to very high induced loads and thus low bearing lives.

SKF has developed special high speed bearings suitable for operation to \( n \cdot d \) values of 1,200,000. The bearings are SKF EC cylindrical roller and special sets of angular contact ball bearings which have high precision and running accuracy (\( \rightarrow \) fig 28). The roller bearing supports the radial load. The angular contact ball bearing set consists of a pair of bearings, mounted either face-to-face or back-to-back, with the thrust bearing having a contact angle between 30° and 40° and the back-up bearing having a contact angle between 15° and 20°. The difference between the contact angle of the thrust bearing and back-up bearing is between 10° and 20°. With this design, the internal force due to centrifugal force is small, so the induced axial force in the bearing system is minimized.

This arrangement of cylindrical roller and angular contact ball bearings for compressors is patented by SKF. The arrangement is available royalty free to customers. The bearings are made to order and inquiries should be directed through SKF Applications Engineering.

In high speed applications, it is also possible to use a four-point contact ball bearing as a thrust bearing. In such cases, the four-point contact ball bearing should have P5 running accuracy. The cage should be either a machined brass or machined light alloy outer ring guided cage.

In comparison with the above mentioned bearing sets, a single four-point contact ball bearing has larger axial clearance and thus requires a larger rotor end clearance since the bearing axial position is affected by the centrifugal loads of the balls.

Bearings having ceramic rolling elements (hybrid bearings) can also be used. The reduced mass of the ceramic rolling element compared to steel elements allows the hybrid bearing to be used at higher speeds. This will allow lower preloads since the risk of skidding due to gyroscopic ball movement is reduced.

*\( n \cdot d \) is the bearing speed \( n \) in r/min multiplied by the bearing mean diameter \( d \) in mm.

\[
d_m = (d+D)/2
\]
Air compressors

Oil flooded air compressors

Bearing arrangements

A typical oil flooded air compressor bearing arrangement is shown in (→ fig 29).
Bearsings

On the inlet side the most widely used bearing type is the cylindrical roller bearing in an NU execution. This type allows free axial movement of the rotor due to elongation under temperature influences and eliminates the risk for parasitic axial forces. For keeping tight control of the rotor end-play, the axial location of the rotor takes place on the outlet side.

Various bearing arrangements are used depending on speed, loads, mounting options, and rotor end-play setting procedure. Typical combinations are shown in (→ fig 30).

Dry air compressors

Bearing arrangements

A typical dry air compressor bearing arrangement is shown in (→ fig 31).
**Bearings**

Because of the high rotational speed of dry running air screw compressors, the bearings most commonly used are four-point contact ball bearings and cylindrical roller bearings. The patented high speed compressor bearing set is also used. These bearings can be fitted with the necessary features of internal clearance, cage construction, and tolerancing for suitable use in the compressor. Consult SKF Applications Engineering for selection of the bearing execution.

The lubricant must be jetted axially into the bearing between the bearing inner ring and cage (→ fig 32).

Orienting the jet this way will overcome the air resistance of the high speed bearing. The jet should have a minimum 1 mm diameter to prevent the risk of being clogged by debris in the oil. The oil viscosity and flow rate must be suitable to lubricate and cool the bearing. The jet speed should be at least 15 to 20 m/s to avoid deflection from the air resistance of the bearing. With a 1 mm jet diameter and a velocity of 15 m/s, the flow is usually more than enough to lubricate and cool the bearing.

The shaft and housing of the dry air compressor have high temperatures since the heat of compression is not removed by oil injection. Typically the bearings are lubricated by a synthetic lubricant having ISO VG 32 to 68 depending upon speed and temperature. The lubrication system should include fine filtration. The oil flow to the bearing should not be so great that it causes excessive bearing friction and temperature rise. Experimentation should be used to optimize the oil flow requirements. Drainage holes on both sides of the bearing may be required to drain off excessive oil.
Refrigeration compressors

Fluorocarbon based refrigerant compressors

Bearing arrangements

A typical fluorocarbon based refrigerant compressor bearing arrangement is shown in (→ fig 33).

Bearings

The selection of bearings for fluorocarbon based refrigeration (HCFC, HFC) applications requires consideration of the effects of the refrigerant on the oil selection and oil properties. All fluorocarbon based refrigerants do not mix freely with mineral oils. The refrigerants, when used with compatible oils, are dissolved in and dilute the viscosity of the oil and in some cases reduce their property to increase in viscosity with pressure in the rolling contact. This diminishes the oils capability to develop the elastohydrodynamic (EHD) film in the rolling contact. The capability of the oil to support sliding friction (cage and roller/flange contacts) are also diminished.
In normal applications of bearings operating in an air environment, an oxide layer is developed on the surfaces in the rolling contacts. This layer acts as an anti-wear protective coating to the steel surfaces. The normal selection of lubricants for rolling bearings, e.g. the SKF General Catalog, is based on the presence of the protective oxide layer. This lack of oxide layer in HCFC-22 applications is to some extent compensated for by the chlorine content in HCFC-22. HFC-134a has no chlorine content and therefore no anti-wear properties.

For the application of bearings in HCFC-22 and HFC-134a, 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. It is also necessary to adjust the minimum required viscosity from figure 10 to account for the lack of anti-wear protection in the rolling contact. Consult SKF Applications Engineering for more details.

The most commonly used bearings in refrigeration screw compressors are the single row angular contact ball bearing and the cylindrical roller bearings. These bearing types have smooth surface finishes and low sliding contact for best formation of the EHD film. Customers have had wear problems with bearings having high sliding speed such as taper roller bearings (roller end /flange contact).

The bearings can be fitted with polyamide (P, TN9 suffix) cages or brass cages (Y, M, MA, ML suffix). Tests at the SKF Engineering & Research Centre have shown the polyamide cages to be suitable for use in the fluorocarbon based refrigerants up to bearing operating temperatures of 100°C (212°F). The polyamide cage is the optimum cage in low viscosity applications provided the speed and temperature is within the allowable limits.

The preservation oil used on the rolling bearings must be compatible with the refrigerant and compressor. Incompatibility can cause chemical reactions between the refrigerant and the compressor oil. These reactions can destroy the compressor oil and the result is corrosion and deposits in the compressor. SKF has preservatives specially defined for refrigeration conditions within HFC-134a/POE and HCFC-22/mineral oil applications. Preservatives can be tested with other refrigerant and oil combinations upon request.

Bearings having ceramic rolling elements (hybrid bearings) can also be used in refrigeration applications. The smooth surface finish and high hardness of the ceramic rolling elements make them suitable for use in very high concentrations of refrigerant (e.g. low oil volume).

**Ammonia compressors**

*Typical bearing arrangement*

**Figure 34** shows the bearing arrangement of the locating bearing side for an industrial ammonia screw compressor. Plain bearings to support the radial loads are used in combination with rolling bearings for the axial guidance of the rotors. The small axial clearance of the rotors needed to obtain the high efficiency of these compressors can be achieved economically by using paired angular contact ball bearings. These bearings are mounted in a back-to-back arrangement in such a way that the bearings are able to move free in the radial direction. To prevent outer ring rotation and creeping, the bearings are fitted in special bushings with a slot, so that a pin stop can be used. The paired angular contact ball bearings have light
Bearings

Preferred bearing types

Bearing types with low friction, such as angular contact ball bearings and cylindrical roller bearings, are preferred. Customers have experienced severe wear in taper roller bearings, between the roller end face and the inner ring flange. This wear phenomena has been reproduced in laboratory testing. For this reason, taper preload and a machined cage. In order to prevent unloading of the inboard bearing at unexpected working conditions the outer rings are preloaded by springs so that the required minimum axial load is given. The bearings are lubricated by oil injection into the outboard bearing and oil leaves the bearing arrangement at the inboard bearing where the oil flow is supported by a flinger.
roller bearings, and other bearing types with high sliding friction such as cylindrical roller thrust bearings, spherical roller thrust bearings, and thrust ball bearings, are not recommended in ammonia compressors.

**Pressed steel cage (J suffix)**
Because of the impaired lubrication due to the undissolved ammonia gas in the lubricating oil, pressed steel cages have not been recommended in ammonia compressors. 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, but several customers who originally used J cages in angular contact ball bearings and the old generation of cylindrical roller bearings have redesigned to other cage types.

**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 70°C (158°F). This limit has been established by experience and laboratory testing.

**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 or brass alloys in ammonia systems, since these materials are subject to stress corrosion cracking. For example, since 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, one-piece machined brass cages, however, are stress free and are not subject to stress corrosion cracking. Testing at the SKF Engineering & Research Centre and longer term experience has shown that solid, one-piece 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, ammonia compressor manufacturers hesitate to use them, and they 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 limits the availability of the machined steel cage. A typical failure of a bearing due to excessive quantities of ammonia is shown (→ fig 35).
Semi hermetic compressors

Since refrigerant compressors operate in closed systems, it is important to minimize leakage of refrigerant from the compressor. The main source of leakage is the shaft seal which is necessary if an external drive motor is used. Semi-hermetic compressors enclose the compressor and the electro-motor in a common shell. This eliminates the need for a seal and improves the compressor reliability.

The motor rotor is mounted on the extended rotor shaft on the suction side. The motor stator is mounted in the extended compressor housing. The construction materials of the motor must be carefully selected to withstand the refrigerant environment.

Electro-motors in semi hermetic compressors can have very compact design since the refrigerant provides good cooling conditions.

An important consideration in the compressor design is the weight of the motor rotor mounted on the extended screw shaft. This design must be carefully analyzed from a rotor dynamics standpoint to avoid problems with vibration. To avoid such problems, it is desirable to use a large shaft diameter and to mount the rotor as closely to the screw as possible.

To allow a large shaft diameter, the suction side bearing should have a thin section height, since the available space is also limited. Cylindrical roller bearings of dimension series 10 or 2 are used commonly. Contact SKF for more information.
Natural and sour gas compressors

Sour gases are natural gases having high concentrations of hydrogen sulfide gas (H$_2$S). Hydrogen sulfide gas in the presence of the moisture (H$_2$O) often found in gas compressors can form sulfuric acid (H$_2$SO$_4$). This acid can corrode the bearing steel and damage the cages.

Brass components have historically not been used in natural or sour gas compressors. Bearings having pressed (J suffix) or machined steel (F or FA suffix) have been used. The bearing rings are protected by the lubricating oil. Tests at the SKF Engineering & Research Centre have established that bearings having polyamide 6.6 cages (P and TN9 suffix) can be used successfully in sour gas compressors to an operating temperature limit of 70°C (158°F).

Caution must be used since sour gas is not a well defined compound. Sour gas containing excessive acids can destroy the polyamide cage, however, the problems are not limited to the bearings. All compressor components must be designed to withstand the acid. The most reliable, but also the most expensive compressor designs, incorporate efficient mechanical seals for separating the compression cavity and the bearings.
Comparative viscosity classifications

Fig 37
10 Unit conversion

Unit conversion

Length
1 mm = 0.039 inch
1 inch = 25.4 mm
0.001 inch = 25.4 µm
1 m = 3.28 ft
1 ft = 0.305 m

Area
1 m² = 10.8 ft²
1 ft² = 0.093 m²

Volume
1 m³ = 35.3 ft³
1 ft³ = 0.028 m³
1 liter = 0.264 US Gallon
1 US Gallon = 3.79 liter
1 Imperial Gallon = 4.55 liter

Mass
1 kg = 2.20 lb

Force
1 N = 0.225 lbf
1 lbf = 4.45 N

Moment
1 Nmm = 8.85x10⁻³ in.lbf
1 in.lbf = 113 Nmm
1 Nm = 0.738 ft.lbf
1 ft.lbf = 1.36 Nm

Power
1 W = 1.36x10⁻³ HP
1 HP = 736 W

Pressure
1 MPa = 1 N/mm² = 145 lbf/in² (psi)
1 MPa = 10 bar
1 atm = 1.01 bar
1 psi = 6.89x10⁻³ N/mm² = 6.89x10⁻³ MPa

Kinematic viscosity
1 mm²/s = 1 cSt

Velocity
1 m/s = 3.28 ft/s
1 ft/s = 0.305 m/s

Flow rate
1 ft³/min = 4.72x10⁻⁴ m³/s
1 US Gallon/min(GPM) = 6.31x10⁻⁵ m³/s
1 m³/s = 15850 GPM
1 ft³/s = 449 GPM

Temperature
°F = (°C x 1.8) + 32
°C = (°F - 32) / 1.8
References


2] Svenningson, Kurt and Dr. Ulf Sjolin. "Svenska Rotor Maskiner AB; The History of SRM Screw Compressor Development."

3] SKF General Catalog 4000.