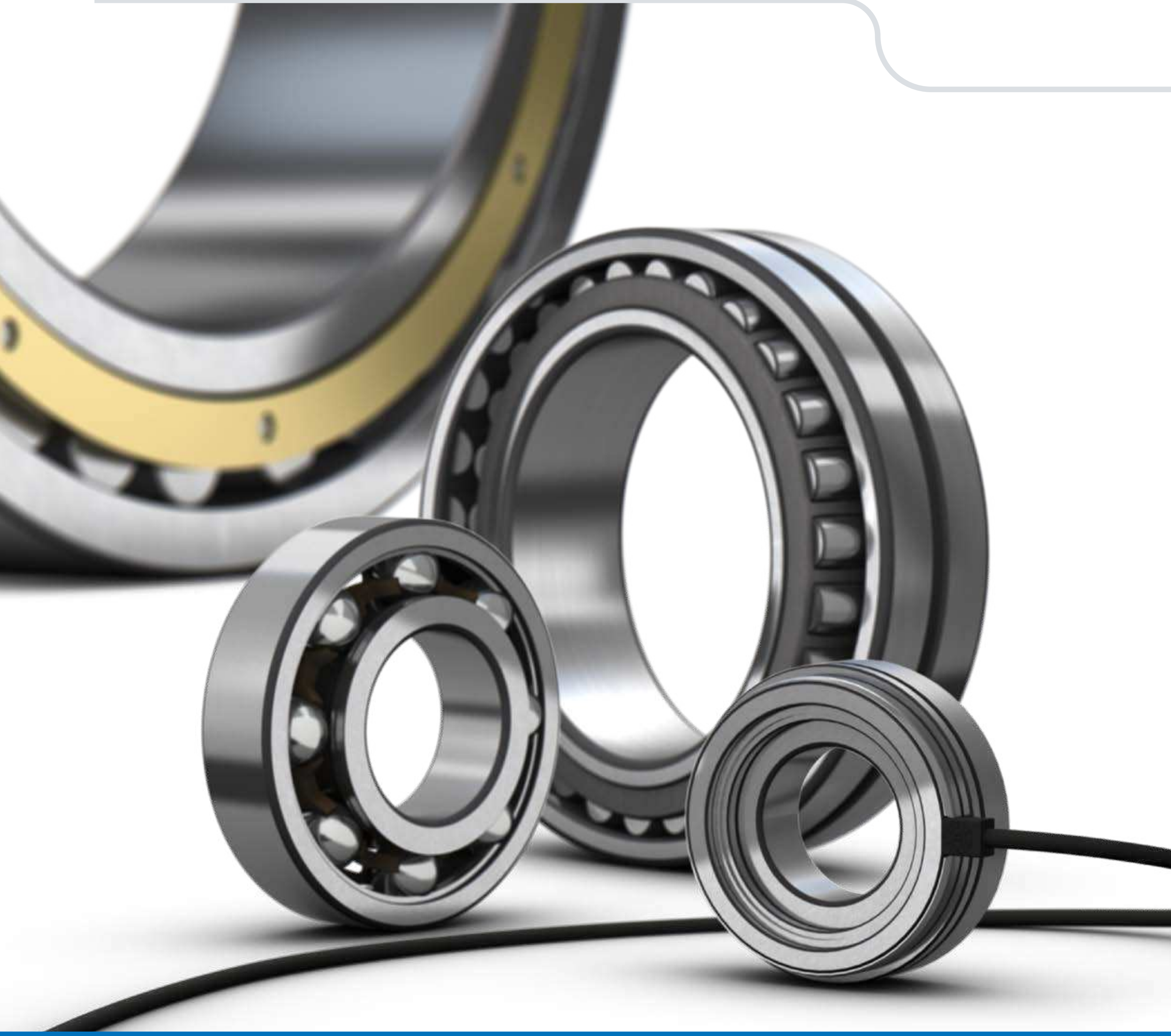
Rolling bearings



SKF mobile apps

SKF mobile apps are available from both Apple App Store and Google Play. These apps provide useful information and allow you to make critical calculations, providing SKF Knowledge Engineering at your fingertips.

Apple App Store

Google Play

To download a PDF document of this catalogue and for information about important updates, go to skf.com/go/17000. Please note product data in this printed catalogue was accurate on the day of printing. The latest and most accurate product data is always available for you on skf.com.

skf.com

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Rolling bearings

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Unit conversions

Quantity	Unit	Conversion			
Length	inch	1 mm	0.03937 in	1 in	25,4 mm
	foot	1 m	3.281 ft	1 ft	0,3048 m
	yard	1 m	1.094 yd	1 yd	0,9144 m
	mile	1 km	0.6214 mi	1 mi	1,609 km
Area	square inch	1 mm^2	0.00155 in^2	1 in^2	645,16 mm^2
	square foot	1 m^2	10.76 ft^2	1 ft^2	0,0929 m^2
Volume	cubic inch	1 cm^3	0.061 in^3	1 in^3	16,387 cm^3
	cubic foot	1 m^3	35 ft^3	1 ft^3	0,02832 m^3
	imperial gallon	1 l	0.22 gallon	1 gallon	4,5461 l
	US gallon	1 l	0.2642 US gallon	1 US gallon	3,7854 l
Speed, velocity	foot per second	1 m/s	3.28 ft/s	1 ft/s	0,3048 m/s
	mile per hour	1 km/h	0.6214 mph	1 mph	1,609 km/h
Mass	ounce	1 g	0.03527 oz	1 oz	28,35 g
	pound	1 kg	2.205 lb	1 lb	0,45359 kg
	short ton	1 tonne	1.1023 short ton	1 short ton	0,90719 tonne
	long ton	1 tonne	0.9842 long ton	1 long ton	1,0161 tonne
Density	pound per cubic inch	1 g/cm^3	0.0361 lb/in^3	1 lb/in^3	27,68 g/cm^3
Force	pound-force	1 N	0.225 lbf	1 lbf	4,4482 N
Pressure, stress	pounds per square inch	1 MPa	145 psi	1 psi	$6,8948 \times 10^3$ Pa
		1 N/mm^2	145 psi		
		1 bar	14.5 psi	1 psi	0,068948 bar
Moment	pound-force inch	1 Nm	8.85 lbf-in	1 lbf-in	0,113 Nm
Power	foot-pound per second	1 W	0.7376 ft-lb/s	1 ft-lb/s	1,3558 W
	horsepower	1 kW	1.36 hp	1 hp	0,736 kW
Temperature	degree	Celsius	$t_C = 0.555 (t_F - 32)$	Fahrenheit	$t_F = 1,8\, t_C + 32$

Foreword

This catalogue contains detailed information on SKF rolling bearings that are typically used in industrial applications. It also includes information on engineered products such as:

- motor encoder units, which measure rotation speed and direction
- rolling bearings designed to withstand extreme temperatures
- bearings with electrical insulation
- bearings with balls or rollers made from ceramic materials

Products presented in this catalogue are available worldwide through SKF sales channels. For information about lead times and deliveries, contact your local SKF representative or SKF Authorized Distributor.

The complete assortment of SKF rolling bearings is much larger than what is presented in this catalogue. Visit skf.com or contact SKF to learn more about rolling bearings, including:

- super-precision bearings
- ball and roller bearing units
- fixed section ball bearings
- large deep groove ball bearings with filling slots
- large angular contact thrust ball bearings
- tapered roller thrust bearings
- multi-row ball or roller bearings
- split roller bearings
- crossed tapered roller bearings
- slewing bearings
- linear ball bearings
- bearings for inline skates and skateboards
- backing bearings for cluster mills
- indexing roller units for continuous furnaces of sintering plants
- application specific bearings for railway rolling stock
- application specific bearings for cars and trucks
- triple ring bearings for the pulp and paper industry
- bearings for printing press rollers
- bearings for critical aerospace applications

The information in this catalogue reflects SKF's state-of-the-art technology and production capabilities as of 2018. The information herein may differ from that shown in earlier catalogues because of redesign, technological developments, or revised calculation methods. SKF reserves the right to continually improve its products with respect to materials, design and manufacturing methods, some of which are driven by technological developments.

SKF Explorer bearings

SKF Explorer rolling bearings accommodate higher load levels and provide extended service life. Optimized internal geometry reduces friction, wear and heat generation, allowing heavier loads to be accommodated. Their advanced surface finish reduces friction and enhances lubricating conditions.

Benefits of using SKF Explorer bearings include:

- significantly extended service life
- increased uptime and productivity
- extended lubricant life
- reduced sensitivity to misalignment
- reduced noise and vibration
- the prospect of downsizing applications

SKF Explorer bearings are shown coloured blue in the product tables.

What is new in this edition

The four main differences in this edition of the SKF catalogue *Rolling bearings*, compared to the previous, are described below.

1. The bearing selection process

When selecting bearings for any purpose, ultimately you want to be certain of achieving the required level of equipment performance – and at the lowest possible cost. In addition to the bearing rating life, there are other key factors you must consider when putting together the bearing specifications for an application. The bearing selection process helps to evaluate these key factors.

Go to section B, **page 60**, to learn more about bearing selection.

- Performance and operationg conditions
- Bearing type and arrangement
- Bearing size
- Lubrication
- Operating temperature and speed
- Bearing interfaces
- Bearing execution
- Sealing, mounting and dismounting

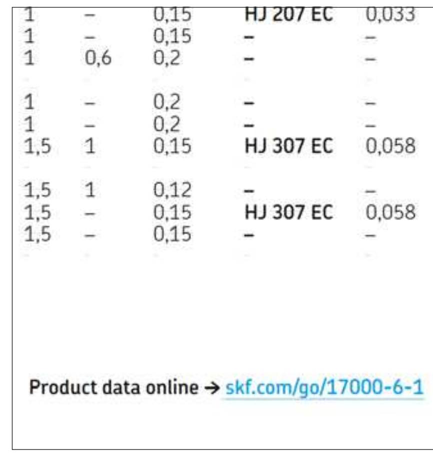
2. Popular items

Popular items are marked in the product tables with the symbol ►. Bearings marked as popular items are of sizes that SKF produces for many customers and are usually in stock. They have a high level of availability and generally provide a cost-effective solution.

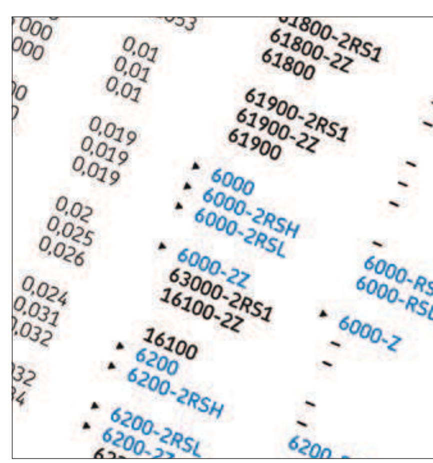
3. Streamlined content and easy online access

This catalogue contains information on rolling bearings commonly used in industrial applications. To reduce the volume of the book and make it more manageable, we have excluded less common bearing types and sizes, though you can readily find these in our online product information.

Short URLs in the product sections provide direct access to related online information.



Short URLs in the product sections provide direct access to related online information.



A triangle indicates popular items. They have a high level of availability and generally provide a costeffective solution.

4. Important product updates

Tapered roller bearings

Tapered roller bearings with an outside diameter up to 600 mm have been redesigned. These new bearings have an increased dynamic load rating, and most of the range is available as SKF Explorer bearings. A consolidated catalogue assortment and a simplified designation system provide a clear view of what is available.

Angular contact ball bearings with 25° contact angle

These new bearings have a raceway geometry optimized for high speeds and reduced sensitivity to axial loading and misalignment. They can increase robustness when used as the backup bearing in sets that are predominantly loaded in one direction.

Upgraded INSOCOAT bearings

INSOCOAT bearings feature electrical insulation on either the inner or outer ring. The upgraded coating provides higher Ohmic resistance, including high Ohmic resistance even in a humid environment, and higher breakdown voltage.

Spherical roller bearings for wind energy applications

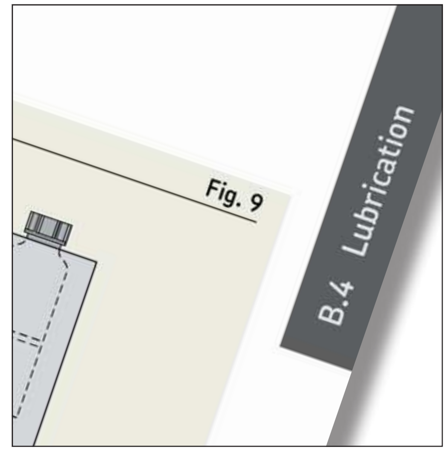
Spherical roller bearings for wind energy applications are designed explicitly for wind turbine main shafts. They have an optimized internal geometry with large diameter rollers and increased contact angle for increased axial load carrying capacity.

Catalogue information and how to use it

This catalogue is divided into three parts:

Principles of rolling bearing selection

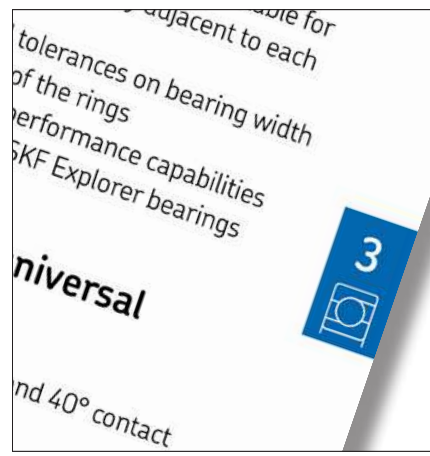
This part is marked by grey bars at the page edge. It provides general information about rolling bearings (section A), explains the bearing selection process (section B), and presents three examples on how to apply the bearing selection process for various applications (section C).



Grey bars mark the three sections under Principles of rolling bearing selection.

Product data

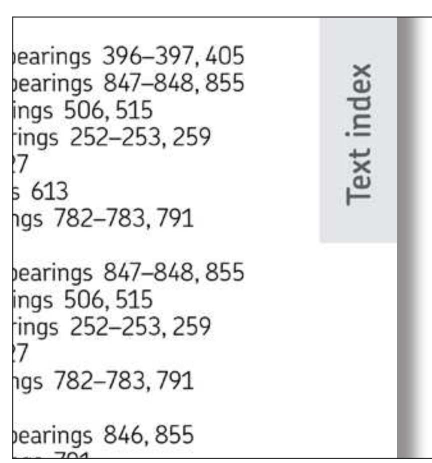
The part is divided into sections per bearing type. Each product section is marked by blue tabs containing the section number and a descriptive icon.



Sections by bearing type are marked with blue tabs including section number and an icon.

Indexes

The product index and text index are marked with grey bars. The product index lists series designations, relates them to the bearing type, and guides you to the relevant product section and product table. The text index lists entries in alphabetical order, including designation suffixes, and helps you locate specific information quickly.



Indexes are marked with grey bars.

Use case: Select a bearing for an application

If you are unsure whether you have adequate knowledge or experience to select a bearing that best suits your application requirements, you will probably find the *Bearing selection process*, page 60, helpful.

If you are an experienced bearing expert, go directly to the section for the relevant bearing type, browse the product tables for the required size, and then look at additional details and information on more specific variants in the text part preceding the product tables.

Use case: Find details of a known bearing

The easiest way to find detailed information about a bearing for which you have the designation is to use the product index, page 1136. Compare the initial characters in a bearing designation with the entries in the product index; each entry specifies the related bearing type, and the relevant product section and product table.

To understand the suffixes used in a bearing designation, go to the text index, page 1120, locate the entry for the suffix and follow the reference to the relevant product sectioin, where you can find detailed information.

Units of measurement

This catalogue is for global use. Therefore, the predominant units of measurement are in accordance with ISO 80000-1. Imperial units are used wherever necessary. Unit conversions can be made using the conversion table, page 6.

For easier use, temperature values are provided in both °C and °F. Specified temperature values are typically rounded. Therefore, values obtained using conversion formulae may not exactly match those specified.

Rotating equipment performance

Every customer is different, with their own drivers and needs, and we have engineered a broad range of products and services to better serve all our customers. So whether you have a problem that needs solving, you want to digitalize your operations, or you want access to design advice, SKF has the right solution to help you get the most out of your rotating equipment.

What does it mean to you?

Performance looks different for every business. We are here to help our customers make choices that deliver against what performance means to them:

- **Improve output**
  By working with SKF to optimise the performance of your rotating equipment you can increase availability, application speed and quality – all driving greater overall equipment effectiveness, and boosting output for your business.

- **Trim your total cost of ownership**
  Poor performance doesn't just affect your output, it can cost you in energy, maintenance, spare parts, labour and more – all adding up to a greater TCO. SKF can help you achieve more reliable rotation, so you can reduce your total cost of ownership.

- **Realise your digital ambitions**
  Make immediate and tangible progress towards your digitalization ambitions. SKF has the digital products, software, services and analytics capabilities to help you gain visibility of the health of your equipment and to turn data into performance-driving insight. Allowing your business to be more agile, deliver greater output, or optimise safety and sustainability.

- **Reduce reliance on scarce talent**
  Work with us to bring rotating equipment expertise into your business, and you can reduce the time and cost burden of recruiting and retaining increasingly scarce and expensive maintenance and diagnostic skillsets.

- **Operate more safely**
  Whether you want to ensure maximum operational safety, reduce hygiene incidents or navigate the minefieldof EHSS regulations, SKF can help you drive operational safety, and a reduced incident rate will feed into your productivity too.

- **Be more sustainable**
  SKF can work with you to reduce energy usage, waste output, spare parts consumption and more, helping you to deliver against your sustainability agenda, as well as saving on costs.

The way that works for you

It is not all about the technologies, services and solutions to meet your business needs. Every customer can have different commercial needs. As a result we have created innovative business models for delivering our rotating equipment performance solutions, which in themselves can contribute towards the performance that matters to your business.

Delivered through our distribution partners

Many of our distribution partners are now delivering greater value to their customers through maintenance, reliability and operations services powered by SKF digitalization capabilities. Find out how SKF Authorized

Distributors and SKF Certified Maintenance Partners could support you on this journey via our support network and services tailored for distributor enablement.

SKF Care

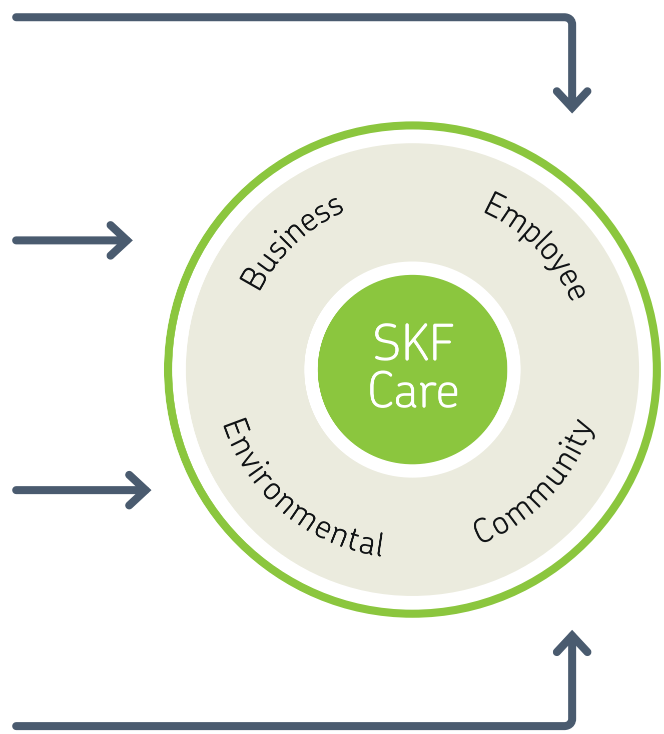
SKF Care is our definition of sustainability. The framework comprises four main perspectives that help us to create value for business partners, the environment, our employees, and the communities around us.

The employee perspective is about ensuring a safe working environment and promote health, education and well-being of employees at SKF and in the supply chain.

The business perspective is about customer focus, financial performance and returns for shareholders – with the highest standards of ethical behaviour.

The environmental perspective is about continually reducing the environmental impact from the Group's operations, as well as actions to significantly improve customers' environmental performance through the products, solutions and services that SKF supplies.

The community perspective is about making positive contributions to the communities in which we operate and guides us to run our business in a way that supports positive development.



SKF BeyondZero

SKF BeyondZero is our mindset to integrate environmental concern into the way we do business. It includes actions to reduce the environmental impact resulting from SKF's operations and those of our suppliers, while at the same time providing customers with solutions to reduce the impact of their products or operations.

Principles of rolling bearing selection

A

General bearing knowledge

General bearing knowledge

This section provides general information that is valid for rolling bearings.

Bearing basics contains information that all readers should know. When you have read that section you will:

- know what rolling bearings are
- know about their components
- have a basic understanding about materials used for rolling bearings
- be familiar with the terminology
- understand the system of standardized boundary dimension
- be able to determine information about a bearing from its designation (part number)

Tolerances provides information that enables you to identify and determine the tolerances of practically every bearing presented here. This is possible because bearing tolerances are standardized internationally, predominantly by ISO. The individual product sections refer to the information in this section, where needed.

Storage provides advice on how to deal with SKF bearings and how to administer them while in storage.

A.1 Bearing basics

A.1 Bearing basics

A.1 Bearing basics

Why rolling bearings?

Rolling bearings support and guide, with minimal friction (fig. 1), rotating or oscillating machine elements – such as shafts, axles or wheels – and transfer loads between machine components. Rolling bearings provide high precision and low friction and therefore enable high rotational speeds while reducing noise, heat, energy consumption and wear. They are cost-effective and exchangeable machine elements that typically follow national or international dimension standards.

Ball and roller bearings

The two basic types of rolling element distinguish the two basic types of rolling bearing:

- ball → ball bearing
- roller → roller bearing

Balls and rollers are different in how they make contact with the raceways.

Balls make point contact with the ring raceways (fig. 2). With increasing load acting on the bearing, the contact point becomes an elliptical area. The small contact area provides low rolling friction, which enables ball bearings to accommodate high speeds but also limits their load-carrying capacity.

Rollers make line contact with the ring raceways (fig. 3). With increasing load acting on the bearing, the contact line becomes somewhat rectangular in shape. Because of the larger contact area and the consequently higher friction, a roller bearing can accommodate heavier loads, but lower speeds, than a same-sized ball bearing.

Fig. 2

Point contact

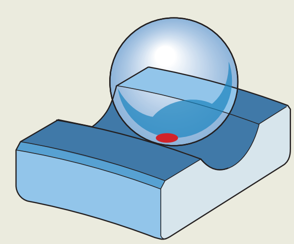


Fig. 1

Why rolling bearings?

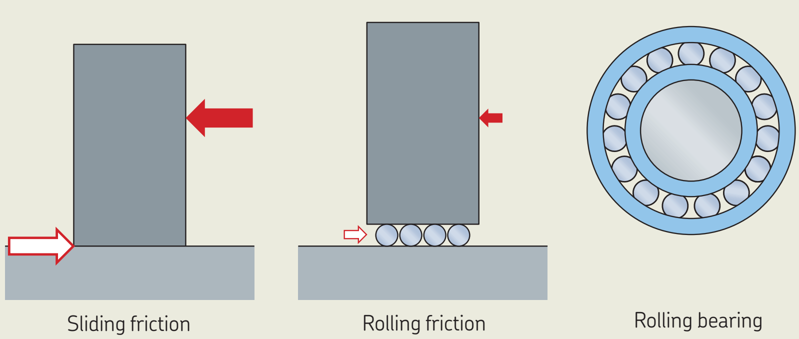
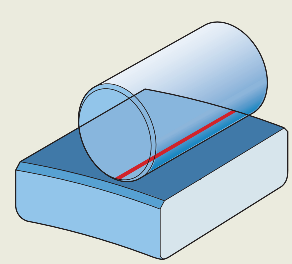


Fig. 3

Line contact



Radial and thrust bearings

Rolling bearings are classified into two groups based on the direction of the load they predominantly accommodate:

- **Radial bearings**
  Radial bearings accommodate loads that are predominantly perpendicular to the shaft. Some radial bearings can support only pure radial loads, while most can additionally accommodate some axial loads in one direction and, in some cases, both directions (**fig. 4**).

- **Thrust bearings**
  Thrust bearings accommodate loads that act predominantly along the axis of the shaft. Depending on their design, thrust bearings may support pure axial loads in one or both directions (**fig. 5**), and some can additionally accommodate radial loads (combined loads, **fig. 6**). Thrust bearings cannot accommodate speeds as high as same-sized radial bearings.

The contact angle (**fig. 7**) determines which group the bearing belongs to. Bearings with a contact angle ≤ 45° are radial bearings, the others are thrust bearings.

Fig. 4

Radial bearings

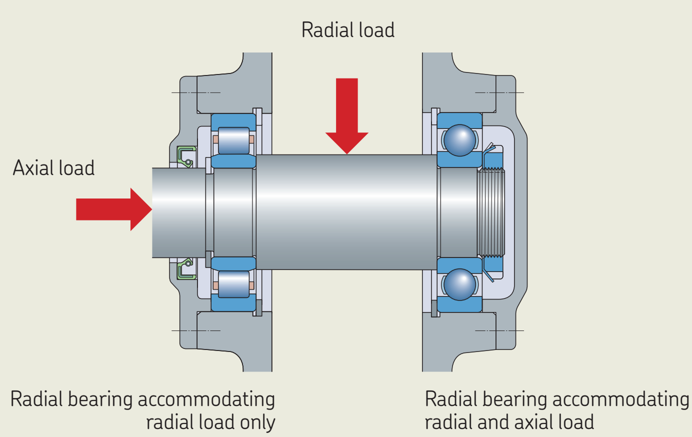


Fig. 5

Thrust bearing for pure axial load

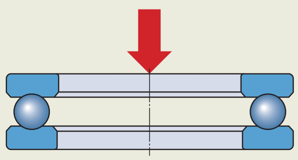


Fig. 6

Thrust bearing for combined load

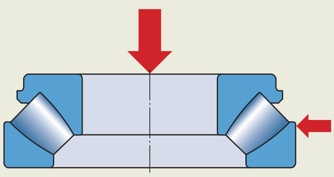
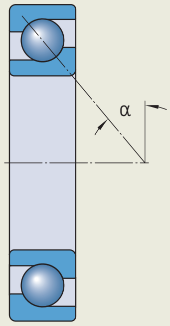


Fig. 7

Contact angle



Terminology

Some frequently used bearing terms are explained here. For a detailed collection of bearing-specific terms and definitions, refer to ISO 5593 *Rolling bearings – Vocabulary*. Symbols used in this catalogue are mainly in accordance with ISO standards. The most common symbols are (fig. 8 and fig. 9):

d Bore diameter
D Outside diameter
B Bearing width
H Bearing height
r Chamfer dimension
α Contact angle

Fig. 8

Symbols for boundary dimensions – radial bearings

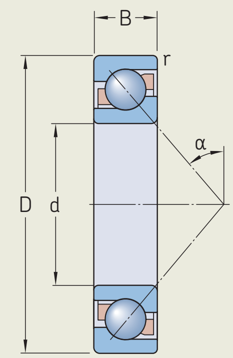
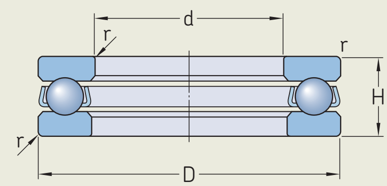


Fig. 9

Symbols for boundary dimensions – thrust bearings



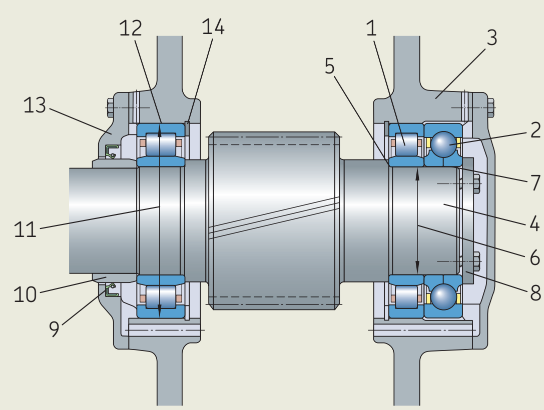
Shaft-bearing-housing system

(fig. 10)

1 Cylindrical roller bearing
2 Four-point contact ball bearing
3 Housing
4 Shaft
5 Shaft abutment shoulder
6 Shaft diameter
7 Shaft seat
8 End plate
9 Radial shaft seal
10 Seal wear ring
11 Housing bore diameter
12 Housing seat
13 Housing cover
14 Snap ring

Fig. 10

Terminology – Shaft-bearing-housing system



Radial bearings

(fig. 11 and fig. 12)

1 Inner ring
2 Outer ring
3 Rolling element: ball, cylindrical roller, needle roller, tapered roller, spherical roller, or toroidal roller
4 Cage
5 Capping device
Seal – made of elastomer
Shield – made of sheet steel
6 Outer ring outside surface
7 Inner ring bore
8 Inner ring shoulder surface
9 Outer ring shoulder surface
10 Snap ring groove
11 Snap ring
12 Outer ring side face
13 Recess for capping device
14 Outer ring raceway
15 Inner ring raceway
16 Recess for capping device
17 Inner ring side face
18 Chamfer
19 Bearing pitch circle diameter
20 Total bearing width
21 Guide flange
22 Retaining flange
23 Contact angle

Thrust bearings

(fig. 13)

24 Shaft washer
25 Rolling element and cage assembly
26 Housing washer
27 Housing washer with a sphered seat surface
28 Seat washer

Fig. 11

Terminology – Radial bearing

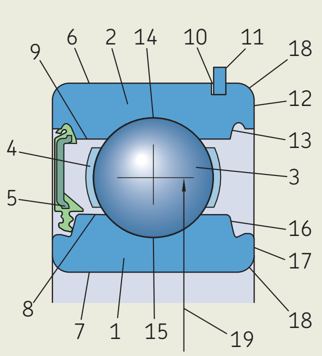


Fig. 12

Terminology – Radial bearing

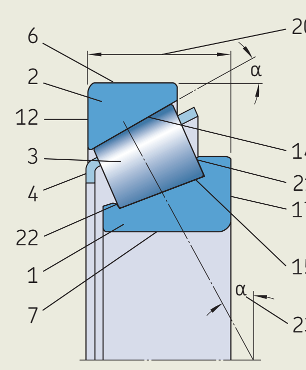
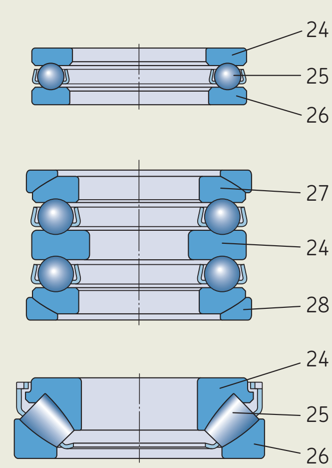


Fig. 13

Terminology – Thrust bearing



Components and materials

A typical rolling bearing consists of the following components (fig. 14):

- an inner ring
- an outer ring
- balls or rollers, as rolling elements
- a cage

SKF supplies several bearing types capped with a seal or shield on one or both sides. Bearings capped on both sides are factory-filled with grease. They provide an economic and space-saving solution compared to external sealing.

Bearing rings

The pressure at the rolling contact area and the cyclic overrolling creates fatigue in the bearing rings when the bearing is in operation. To cope with such fatigue, rings that are made of steel must be hardened.

The standard steel for bearing rings and washers is 100Cr6, a steel containing approximately 1% carbon and 1,5% chromium.

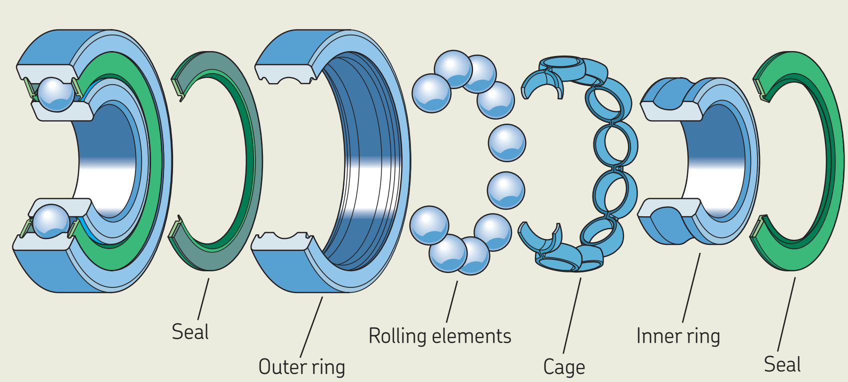
SKF bearing rings and washers are made of steel in accordance with SKF specifications. They cover all aspects that are relevant to providing a long service life for the bearing. Depending on specific requirements, SKF uses stainless steels or high-temperature steels.

Rolling elements

The rolling elements (balls or rollers) transfer the load between inner and outer rings. Typically, the same steel is used for rolling elements as for bearing rings and washers. When required, rolling elements can be made of ceramic material. Bearings containing ceramic rolling elements are considered hybrid bearings and are becoming more and more common.

Fig. 14

Bearing components



Cages

The primary purposes of a cage are:

- separating the rolling elements to reduce the frictional heat generated in the bearing
- keeping the rolling elements evenly spaced to optimize load distribution
- guiding the rolling elements in the unloaded zone of the bearing
- retaining the rolling elements of separable bearings when one bearing ring is removed during mounting or dismounting

Cages are radially centred (fig. 15) either on:

- the rolling elements
- the inner ring
- the outer ring

Cages centred on the rolling elements permit the lubricant to enter the bearing easily. Ring centred cages, which provide more precise guidance, are typically used when bearings must accommodate high speeds, high vibration levels or inertia forces stemming from movements of the whole bearing.

The main cage types are:

- **Stamped metal cages (fig. 16)**
  Stamped metal cages (sheet steel or sometimes sheet brass) are lightweight and withstand high temperatures.

- **Machined metal cages (fig. 17)**
  Machined metal cages are made of brass or sometimes steel or light alloy. They permit high speeds, temperatures, accelerations and vibrations.

- **Polymer cages (fig. 18)**
  Polymer cages are made of polyamide 66 (PA66), polyamide 46 (PA46) or sometimes polyetheretherketone (PEEK) or other polymer materials. The good sliding properties of polymer cages produce little friction and, therefore, permit high speeds. Under poor lubrication conditions, these cages reduce the risk of seizure and secondary damage because they can operate for some time with limited lubrication.

- **Pin-type cages (fig. 19)**
  Steel pin-type cages need pierced rollers and are only used together with large-sized roller bearings. These cages have relatively low weight and enable a large number of rollers to be incorporated.

Fig. 15

Options for cage centring

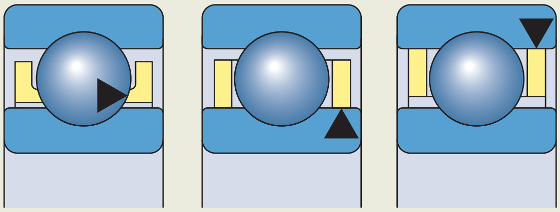


Fig. 16

Stamped metal cage

Fig. 17

Machined metal cage

Fig. 18

Polymer cage

Fig. 19

Pin-type cage

Integral sealing

Integral sealing can significantly prolong bearing service life because it keeps lubricant in the bearing and contaminants out of it. SKF bearings are available with various capping devices:

- **Shields**
  There is a small gap between the shield and inner ring. Bearings fitted with shields (**fig. 20**) are used where the operating conditions are relatively clean, or where low friction is important because of speed or operating temperature considerations.

- **Seals**
  Bearings with seals are preferred for arrangements where contamination is moderate. Where the presence of water or moisture cannot be ruled out, contact seals (**fig. 21**) are typically used. These seals make positive contact with the sliding surface on one of the bearing rings. Low-friction seals (**fig. 22**) and non-contact seals (**fig. 23**) can accommodate the same speeds as bearings with shields, but with improved sealing effectiveness.

Fig. 20

Shield

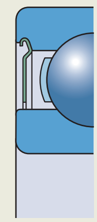


Fig. 21

Contact seal

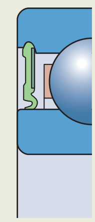


Fig. 22

Low-friction seal

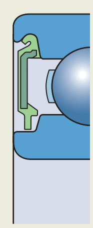
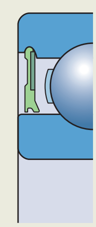


Fig. 23

Non-contact seal



Internal clearance

Bearing internal clearance (**fig. 24**) is defined as the total distance through which one bearing ring can be moved relative to the other in the radial direction (radial internal clearance) or in the axial direction (axial internal clearance).

In almost all applications, the initial clearance in a bearing is greater than its operating clearance. The difference is mainly caused by two effects:

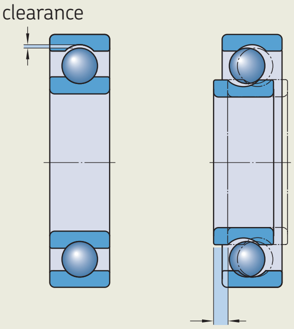
- Bearings are typically mounted with an interference fit on the shaft or in the housing. The expansion of the inner ring or the compression of the outer ring reduces the internal clearance.
- Bearings generate heat in operation. Differential thermal expansion of the bearing and mating components influences the internal clearance.

Sufficient internal clearance in a bearing during operation is important. Preload (clearance below zero) is possible for certain bearing types.

To enable selection of the appropriate initial internal clearance to achieve the desired operational internal clearance, bearings are available in different clearance classes. ISO has established five clearance classes for many bearing types. SKF uses designation suffixes to indicate when the bearing internal clearance differs from Normal (**table 1**).

Fig. 24

Bearing internal clearance



Heat and surface treatment

Rolling bearing rings and rolling elements must:

- be hard enough to cope with fatigue and plastic deformations
- be tough enough to cope with applied loads
- be sufficiently stable to experience only limited changes of dimensions over time

The required properties are achieved by heat and surface treatments.

Hardening

There are three typical hardening methods that may be applied to bearing components:

- **Through-hardening**
  This is the standard method for most bearings and provides good fatigue and wear-resistance, as hardening is applied over the full cross section.

- **Induction-hardening**
  Surface induction-hardening is used to selectively harden a component's raceway to limit rolling contact fatigue, leaving the remainder of the component unaffected to maintain structural strength.

- **Case-hardening**
  Case-hardening provides hardness to the surface. It is used, for example, where bearing rings are subjected to high shock loads causing structural deformations.

Dimensional stability

Heat treatment is used to limit dimensional changes caused by metallurgical effects at extreme temperatures. There is a standardized classification system for dimensional stability (**table 2**). The various SKF bearing types are stabilized to different classes as standard.

Surface treatment and coatings

Coating is a well-established method for providing bearings with additional functional benefits to accommodate specific application conditions. Widely used coatings are zinc chromate and black oxide.

Two other methods developed by SKF have proven successful in many applications:

- INSOCOAT bearings are standard bearings that have the external surfaces of their inner or outer ring coated with an aluminium oxide layer. This coating increases resistance to electric current through the bearing.
- NoWear enhances wear-resistance of the raceway or rolling element surfaces. It can help the bearing withstand long periods of operation under poor lubrication conditions and to reduce the risk for low load damage.

Table 1

Internal clearance classes

ISO clearance class	SKF designation suffix	Internal clearance
–	C1	Smaller than C2
Group 2	C2	Smaller than Normal
Group N	–	Normal
Group 3	C3	Greater than Normal
Group 4	C4	Greater than C3
Group 5	C5	Greater than C4

Table 2

Dimensional stability

Stabilization class	Stabilized up to	
–	°C	*°F*
SN	120	*250*
S0	150	*300*
S1	200	*390*
S2	250	*480*
S3	300	*570*
S4	350	*660*

Standardized boundary dimensions

Boundary dimensions are the main dimensions of a bearing (fig. 25 and fig. 26). They comprise:

- the bore diameter (d)
- the outside diameter (D)
- the width or height (B, C, T or H)
- the chamfer dimensions (r)

The boundary dimensions for metric bearings are standardized in the ISO (International Organization for Standardization) general plans:

- ISO 15 for radial rolling bearings, except insert bearings, some types of needle roller bearings and tapered roller bearings
- ISO 104 for thrust bearings
- ISO 355 for tapered roller bearings

Most rolling bearings follow ISO standard dimensions, which is a prerequisite to enable interchangeability.

The ISO general plan for radial bearings provides several series of standardized outside diameters for every standard bore diameter. They are called diameter series and are numbered 7, 8, 9, 0, 1, 2, 3 and 4 (in order of increasing outside diameter). Within each diameter series, different width series exist (width series 8, 0, 1, 2, 3, 4, 5 and 6 in order of increasing width). The diameter series 0, 2 and 3, combined with width series 0, 1, 2 and 3, are shown in fig. 27.

For thrust bearings, height series are used instead of width series. Height series are numbered 7, 9, 1 and 2.

Bearings to ISO general plans have the same boundary dimensions when they share the same bore diameter and dimension series (table 3). If not, they have different boundary dimensions.

Bearings with inch dimensions

In addition to the bearings in accordance with ISO dimensions, SKF has a comprehensive assortment of bearings with inch dimensions following American and British standards.

Fig. 25

Boundary dimensions, radial bearing

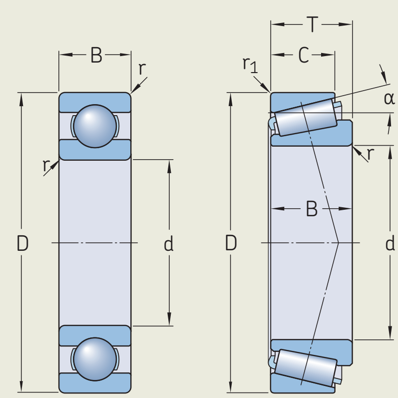
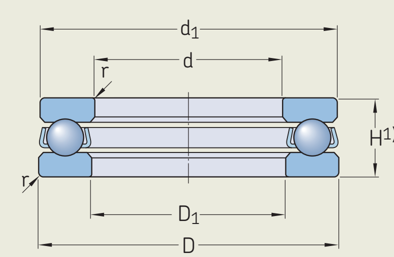


Fig. 26

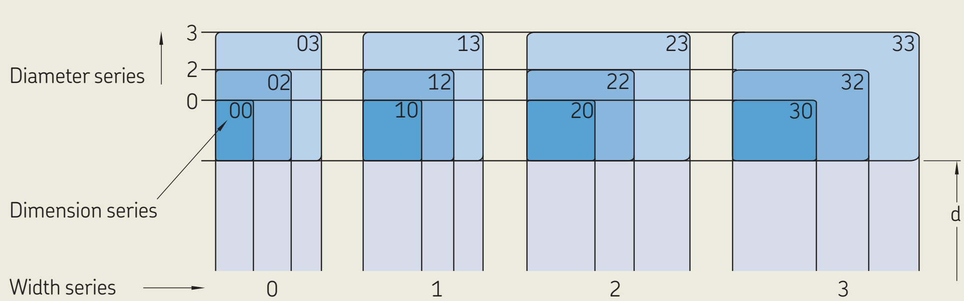
Boundary dimensions, thrust bearing



1) ISO uses T.

Fig. 27

System of ISO dimension series



Basic bearing designation system

The designations of most SKF rolling bearings follow a designation system. The complete bearing designation may consist of a basic designation with or without one or more supplementary prefixes and suffixes (**diagram 1**). The basic designation identifies:

- the bearing type
- the basic design
- the boundary dimensions

Prefixes and suffixes identify design features or bearing components.

Diagram 1

Designations for SKF rolling bearings

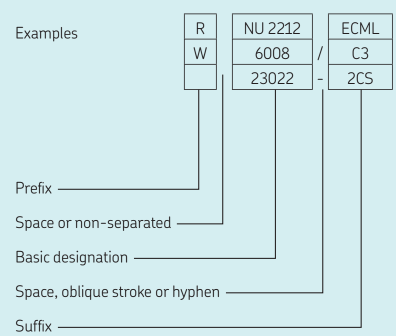


Table 3

Examples of boundary dimensions

Same bore diameter and dimension series		Same bore diameter, but different dimension series	
Deep groove ball bearing 6205	Cylindrical roller bearing NU 205	Cylindrical roller bearing NU 2205 ECP	Angular contact ball bearing 3205 A
Dimension series 02	**Dimension series 02**	**Dimension series 22**	**Dimension series 32**

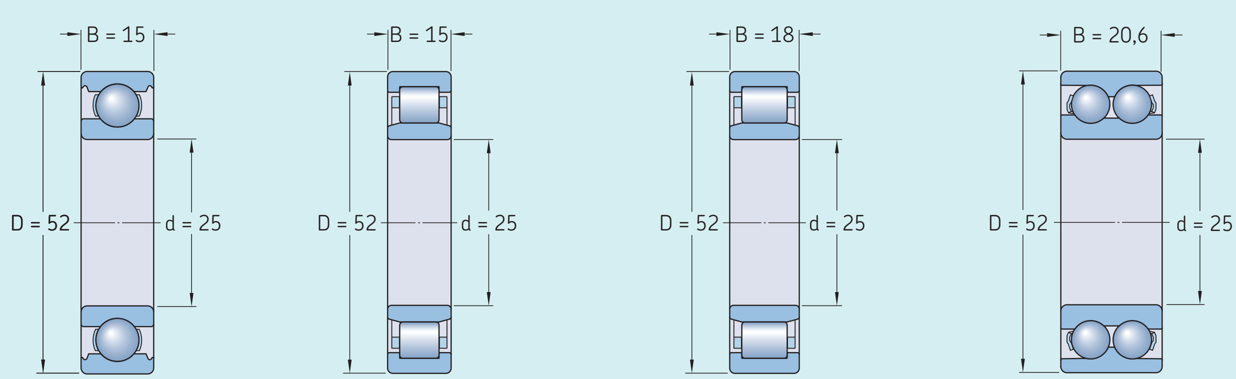
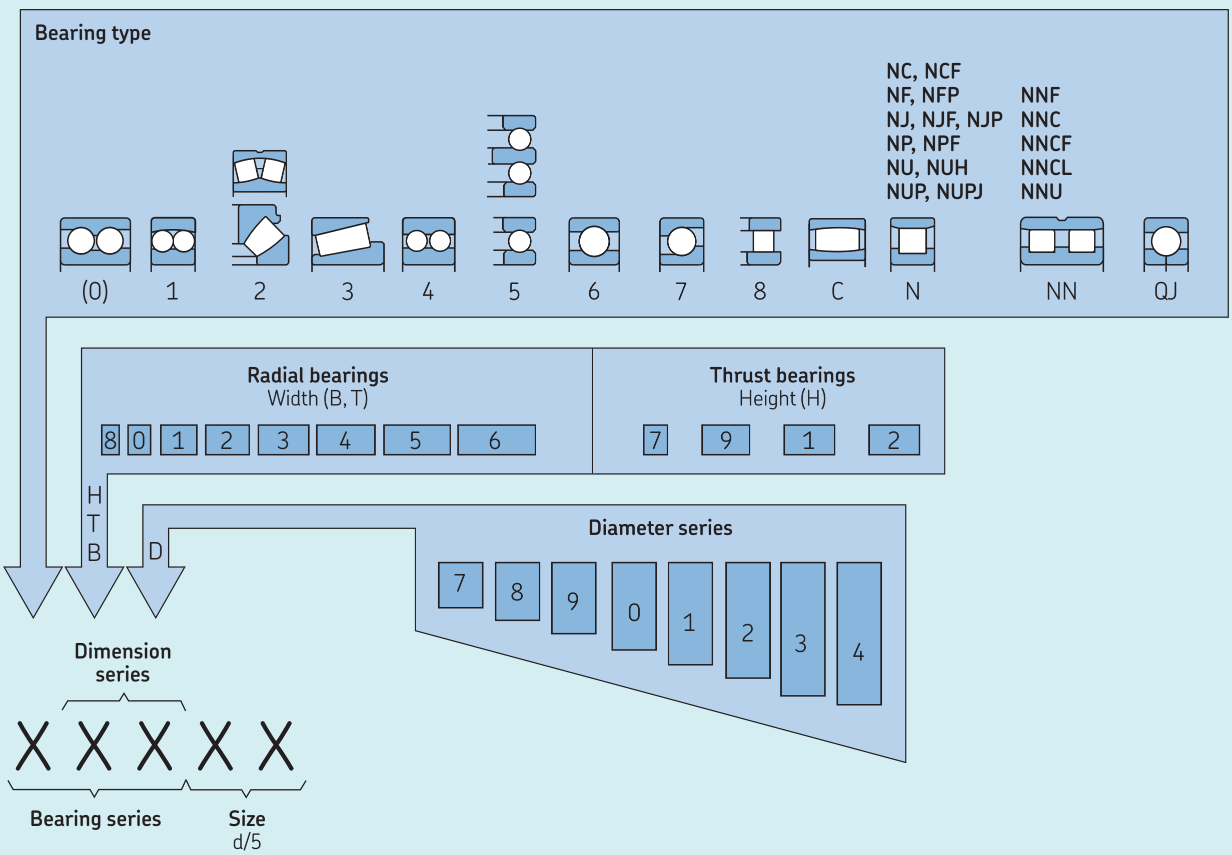


Table 4

Basic designation system for SKF standard metric ball and roller bearings

Bearing series

Bearing type	Bearing series
(0)	(0)33, (0)32
1	139, 130, (1)23, 1(0)3, (1)22, 1(0)2, 1(1)0
2	223, 213, 232, 222, 241, 231, 240, 230, 249, 239, 248, 238, 294, 293, 292
3	323, 313, 303, 332, 322, 302, 331, 330, 320, 329
4	4(2)3, 4(2)2
5	544, 524, 543, 523, 542, 522, 534, 514, 533, 513, 532, 512, 511, 510, 591, 590
6	6(0)4, 623, 6(0)3, 622, 6(0)2, 630, 6(1)0, 16(0)0, 639, 619, 609, 638, 628, 618, 608, 637, 627, 617
7	7(0)4, 7(0)3, 7(0)2, 7(1)0, 719, 718, 708
8	814, 894, 874, 813, 893, 812, 811
C	23, 32, 22, 41, 31, 60, 50, 40, 30, 69, 59, 49, 39
N	(0)4, 33, 23, (0)3, 22, 12, (0)2, 31, 30, 20, 10, 39, 29, 19, 38, 28, 18
NN	41, 31, 60, 50, 40, 30, 69, 49, 39, 48
QJ	23, (0)3, 12, (0)2, 10, 19



Code	Bearing type
0	Double row angular contact ball bearing
1	Self-aligning ball bearing
2	Spherical roller bearing, spherical roller thrust bearing
3	Tapered roller bearing
4	Double row deep groove ball bearing
5	Thrust ball bearing
6	Single row deep groove ball bearing
7	Single row angular contact ball bearing
8	Cylindrical roller thrust bearing
C	CARB toroidal roller bearing
N	Cylindrical roller bearing. Two or more letters are used to identify the number of the rows or the configuration of the flanges, e.g. NJ, NU, NUP, NN, NNU, NNCF
QJ	Four-point contact ball bearing
T	Tapered roller bearing in accordance with ISO 355

Basic designations

A basic designation typically contains three to five digits. The basic designation system is shown in **table 4**. The number and letter combinations have the following meaning:

- The first digit or letter or combination of letters identifies the bearing type and eventually a basic variant.
- The following two digits identify the ISO dimension series. The first digit indicates the width or height series (dimensions B, T or H). The second digit identifies the diameter series (dimension D).
- The last two digits of the basic designation identify the size code of the bearing bore. The size code multiplied by 5 gives the bore diameter (d) in mm.

The most important exceptions in the basic bearing designation system are:

1 In a few cases the digit for the bearing type or the first digit of the dimension series identification is omitted. These digits are shown in brackets in **table 4**.

2 Bearings with a bore diameter of 10, 12, 15 or 17 mm have the following size code identifications:
 00 = 10 mm
 01 = 12 mm
 02 = 15 mm
 03 = 17 mm

3 For bearings with a bore diameter < 10 mm, or ≥ 500 mm, the bore diameter is generally given in millimetres (uncoded). The size identification is separated from the rest of the bearing designation by an oblique stroke, e.g. 618/8 (d = 8 mm) or 511/530 (d = 530 mm). This is also true for standard bearings in accordance with ISO 15 that have a bore diameter of 22, 28 or 32 mm, e.g. 62/22 (d = 22 mm).

4 For some bearings with a bore diameter < 10 mm, such as deep groove, self-aligning and angular contact ball bearings, the bore diameter is also given in millimetres (uncoded) but is not separated from the series designation by an oblique stroke, e.g. 629 or 129 (d = 9 mm).

5 Bore diameters that deviate from the standard bore diameter of a bearing are uncoded and given in millimetres up to three decimal places. This bore diameter identification is part of the basic designation and is separated by an oblique stroke, e.g. 6202/15.875 (d = 15,875 mm = 5/8 in).

Bearing series

Bearing series designations consist of an identification for the bearing type and the dimension series. The most common series designations are shown in **table 4**. The digits in brackets belong to the system, but are not used in the series designation in practice.

Prefixes and suffixes

The designations of most SKF rolling bearings follow a system that consists of a basic designation with or without one or more prefixes and/or suffixes, as shown in **diagram 2**.

Prefixes and suffixes provide additional information about the bearing.

Prefixes are mainly used to identify components of a bearing. They can also identify bearing variants.

Suffixes identify designs or variants, which differ in some way from the original design or from the current basic design. The suffixes are divided into groups. When more than one special feature is to be identified, suffixes are provided in the order shown in **diagram 2**.

Details of the significance of specific prefixes and suffixes are given in the relevant product sections.

Bearing designations not covered by the basic system

Insert bearings

The designations for insert bearings differ somewhat from those described in the basic designation system and are described under *Insert bearings*, **page 339**.

Needle roller bearings

The designations for needle roller bearings do not fully follow the basic designation system and are described under *Needle roller bearings*, **page 581**.

Tapered roller bearings

The designations for metric tapered roller bearings follow either the basic designation system or a designation system, established by ISO in 1977, covered in ISO 355. Inch tapered roller bearings are designated in accordance with the relevant ANSI/ABMA standard. The designation system is explained under *Tapered roller bearings*, **page 665**.

Customized bearings

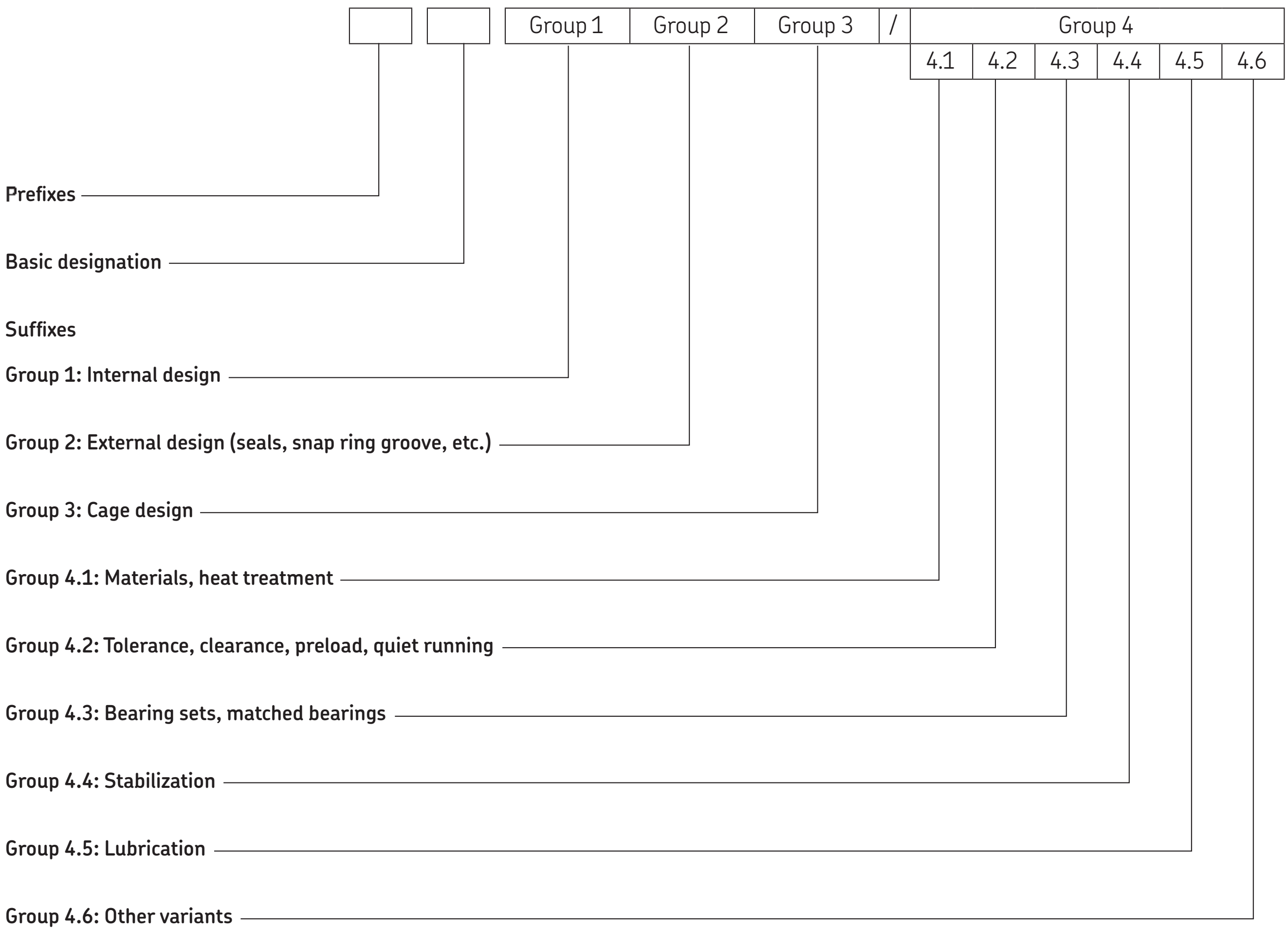
Bearings designed to meet a specific customer requirement are typically designated by a drawing number. The drawing number does not provide any information about the bearing.

Other rolling bearings

Rolling bearings not covered in the ball bearings and roller bearings sections, such as super-precision bearings, thin section bearings, slewing bearings or linear bearings, follow designation systems that can differ significantly from the basic designation system.

Diagram 2

Designation system



A.2 Tolerances

A.2 Tolerances

A.2 Tolerances

Tolerance classes and the corresponding values for certain tolerance characteristics are specified in ISO 492 (for radial bearings) and ISO 199 (for thrust bearings). In 2014 these standards were aligned with general ISO GPS (Geometrical Product Specification) standards such as ISO 1101 and ISO 5459. For additional information on ISO 492 and ISO 199, and the changes that have been made to their previous editions, refer to the SKF e-learning platform (skf.com/go/17000-learnGPS).

There are three common tolerance classes for SKF ball and roller bearings (table 1).

The product sections for the various bearing types provide information on compliance with applicable tolerance classes. The tolerance class of a bearing cannot always be determined from its designation suffixes. Where the tolerance class is standard for the bearing, it is not specified in the designation suffixes.

For information about SKF bearings that have a tolerance class better than class 5, refer to the SKF catalogue *Super-precision bearings* or skf.com/super-precision.

Tolerance values

Actual tolerance values are listed in the following tables.

Metric radial bearings, except tapered roller bearings:

- Normal tolerances (**table 2**, **page 38**)
- P6 class tolerances (**table 3**, **page 39**)
- P5 class tolerances (**table 4**, **page 40**)

Metric tapered roller bearings:

- Normal and CL7C class tolerances (**table 5**, **page 41**)
- CLN class tolerances (**table 6**, **page 42**)
- P5 class tolerances (**table 7**, **page 43**)

Inch radial bearings, except tapered roller bearings:

- Normal tolerances (**table 8**, **page 44**)

Inch tapered roller bearings:

- Normal, CL2, CL3 and CL0 class tolerances (**table 9**, **page 45**)

Thrust bearings:

- Normal, P6 and P5 class tolerances (**table 10**, **page 46**)

Tapered bore, taper 1:12:

- Normal, P6 and P5 class tolerances (**table 11**, **page 47**)

Tapered bore, taper 1:30:

- Normal tolerances (**table 12**, **page 48**)

Where standardized, the values are in accordance with ISO 492, ISO 199 and ANSI/ABMA Std. 19.2.

Tolerance symbols

The tolerance symbols that we use are in line with ISO 492 and ISO 199 and are explained in **table 13**, **page 49**. The symbols normally refer to dimensional tolerances, only Kia, Kea, Sd, SD, Sia and Sea refer to geometrical tolerances.

Table 1

Common tolerance classes for SKF ball and roller bearings

ISO tolerance class	SKF designation suffix	Description
Normal	–	Minimum standard for all SKF ball and roller bearings.
Class 6	P6	Tighter tolerances than Normal.
Class 5	P5	Tighter tolerances than class 6.

Diameter series identification

The bore and outside diameter variation tolerances t_{Vdsp} and t_{VDsp} for metric radial bearings (**table 2**, **page 38**, to **table 4**, **page 40**) vary depending on the diameter series to which the bearing belongs. To determine the diameter series, refer to **table 14**, **page 52**.

Chamfer dimensions

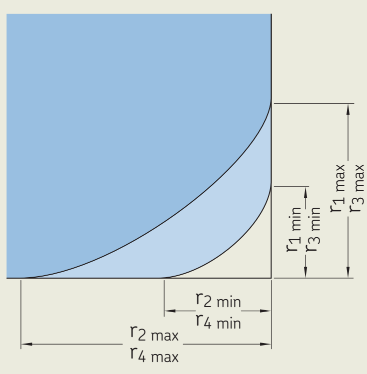
Minimum chamfer dimensions

Minimum chamfer dimensions (**fig. 1**) are listed in the product tables, for the radial (r_1, r_3) and axial (r_2, r_4) directions. For metric SKF bearings, these values are in accordance with the general plans listed in the following standards:

- ISO 15, ISO 12043 and ISO 12044 for radial bearings
- ISO 355 for radial tapered roller bearings
- ISO 104 for thrust bearings

Fig. 1

Minimum and maximum chamfer dimensions



Maximum chamfer dimensions

The maximum chamfer dimensions (**fig. 1**) for the radial (r_1, r_3) and axial (r_2, r_4) directions, appropriate to the respective minimum values and the bore or outside diameter, are listed in the following tables:

- Metric radial and thrust bearings, except radial tapered roller bearings (**table 15**, **page 53**)
- Metric radial tapered roller bearings (**table 16**, **page 53**)
- Inch tapered roller bearings (**table 17**, **page 54**)

The maximum chamfer dimensions for metric SKF bearings are in accordance with ISO 582.

Example

What is the largest radial and axial value ($r_{1\,max}$ and $r_{2\,max}$) for the chamfer of a 6211 deep groove ball bearing?

From the relevant product table, $r_{1,2\,min}$ = 1,5 mm and d = 55 mm.

From **table 15**, with $r_{s\,min}$ = 1,5 mm and d < 120 mm, the largest radial value $r_{1\,max}$ = 2,3 mm and the largest axial value $r_{2\,max}$ = 4 mm.

A.2 Tolerances

Table 2

Normal tolerances for radial bearings, except tapered roller bearings

Inner ring

d		$t_{\Delta dmp}$ [1]		t_{Vdsp} [1] Diameter series			t_{Vdmp}	$t_{\Delta Bs}$ All	Normal	Modified [3]	t_{VBs}	t_{Kia}
>	≤	U	L	7, 8, 9 [2]	0, 1	2, 3, 4		U	L	L		
mm		µm		µm			µm	µm			µm	µm
–	**2,5**	0	–8	10	8	6	6	0	–40	–	12	10
2,5	**10**	0	–8	10	8	6	6	0	–120	–250	15	10
10	**18**	0	–8	10	8	6	6	0	–120	–250	20	10
18	**30**	0	–10	13	10	8	8	0	–120	–250	20	13
30	**50**	0	–12	15	12	9	9	0	–120	–250	20	15
50	**80**	0	–15	19	19	11	11	0	–150	–380	25	20
80	**120**	0	–20	25	25	15	15	0	–200	–380	25	25
120	**180**	0	–25	31	31	19	19	0	–250	–500	30	30
180	**250**	0	–30	38	38	23	23	0	–300	–500	30	40
250	**315**	0	–35	44	44	26	26	0	–350	–500	35	50
315	**400**	0	–40	50	50	30	30	0	–400	–630	40	60
400	**500**	0	–45	56	56	34	34	0	–450	–	50	65
500	**630**	0	–50	63	63	38	38	0	–500	–	60	70
630	**800**	0	–75	–	–	–	–	0	–750	–	70	80
800	**1 000**	0	–100	–	–	–	–	0	–1 000	–	80	90
1 000	**1 250**	0	–125	–	–	–	–	0	–1 250	–	100	100
1 250	**1 600**	0	–160	–	–	–	–	0	–1 600	–	120	120
1 600	**2 000**	0	–200	–	–	–	–	0	–2 000	–	140	140

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp} [4] Open bearings Diameter series			Capped bearings [5]	t_{VDmp} [4]	$t_{\Delta Cs}$, t_{VCs}	t_{Kea}
>	≤	U	L	7, 8, 9 [2]	0, 1	2, 3, 4	2, 3, 4			
mm		µm		µm				µm	µm	µm
2,5	**18**	0	–8	10	8	6	10	6	Identical to $t_{\Delta Bs}$ and t_{VBs} of an inner ring of the same bearing as the outer ring	15
18	**30**	0	–9	12	9	7	12	7		15
30	**50**	0	–11	14	11	8	16	8		20
50	**80**	0	–13	16	13	10	20	10		25
80	**120**	0	–15	19	19	11	26	11		35
120	**150**	0	–18	23	23	14	30	14		40
150	**180**	0	–25	31	31	19	38	19		45
180	**250**	0	–30	38	38	23	–	23		50
250	**315**	0	–35	44	44	26	–	26		60
315	**400**	0	–40	50	50	30	–	30		70
400	**500**	0	–45	56	56	34	–	34		80
500	**630**	0	–50	63	63	38	–	38		100
630	**800**	0	–75	94	94	55	–	55		120
800	**1 000**	0	–100	125	125	75	–	75		140
1 000	**1 250**	0	–125	–	–	–	–	–		160
1 250	**1 600**	0	–160	–	–	–	–	–		190
1 600	**2 000**	0	–200	–	–	–	–	–		220
2 000	**2 500**	0	–250	–	–	–	–	–		250

1) Tolerances for tapered bores (table 11, page 47 and table 12, page 48).
2) Diameter series 7 and 8 not covered by ISO 492.
3) Applies to inner rings and outer rings of bearings of matched bearing sets consisting of two or more bearings. Does not apply to universally matchable angular contact ball bearings.
4) Applies to bearings prior to mounting and after removal of internal or external snap ring.
5) Capped bearings are sealed or shielded bearings.

Table 3

P6 class tolerances for radial bearings, except tapered roller bearings

Inner ring

d		$t_{\Delta dmp}$ [1]		t_{Vdsp} [1] Diameter series			t_{Vdmp}	$t_{\Delta Bs}$			t_{VBs}	t_{Kia}
>	≤	U	L	7, 8, 9[2]	0, 1	2, 3, 4		All	Normal	Modified[3]		
mm		µm		µm			µm	µm			µm	µm
–	**2,5**	0	–7	9	7	5	5	0	–40	–	12	5
2,5	**10**	0	–7	9	7	5	5	0	–120	–250	15	6
10	**18**	0	–7	9	7	5	5	0	–120	–250	20	7
18	**30**	0	–8	10	8	6	6	0	–120	–250	20	8
30	**50**	0	–10	13	10	8	8	0	–120	–250	20	10
50	**80**	0	–12	15	15	9	9	0	–150	–380	25	10
80	**120**	0	–15	19	19	11	11	0	–200	–380	25	13
120	**180**	0	–18	23	23	14	14	0	–250	–500	30	18
180	**250**	0	–22	28	28	17	17	0	–300	–500	30	20
250	**315**	0	–25	31	31	19	19	0	–350	–500	35	25
315	**400**	0	–30	38	38	23	23	0	–400	–630	40	30
400	**500**	0	–35	44	44	26	26	0	–450	–	45	35
500	**630**	0	–40	50	50	30	30	0	–500	–	50	40
630	**800**	0	–50	–	–	–	–	0	–750	–	60	45
800	**1 000**	0	–60	–	–	–	–	0	–1 000	–	60	50
1 000	**1 250**	0	–75	–	–	–	–	0	–1 250	–	70	60
1 250	**1 600**	0	–90	–	–	–	–	0	–1 600	–	70	70
1 600	**2 000**	0	–115	–	–	–	–	0	–2 000	–	80	80

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp} [4] Open bearings Diameter series			Capped bearings[5]	t_{VDmp} [4]	$t_{\Delta Cs}$, t_{VCs}	t_{Kea}
>	≤	U	L	7, 8, 9[2]	0, 1	2, 3, 4	0, 1, 2, 3, 4			
mm		µm		µm				µm		µm
2,5	**18**	0	–7	9	7	5	9	5	Identical to $t_{\Delta Bs}$ and t_{VBs} of an inner ring of the same bearing as the outer ring	8
18	**30**	0	–8	10	8	6	10	6		9
30	**50**	0	–9	11	9	7	13	7		10
50	**80**	0	–11	14	11	8	16	8		13
80	**120**	0	–13	16	16	10	20	10		18
120	**150**	0	–15	19	19	11	25	11		20
150	**180**	0	–18	23	23	14	30	14		23
180	**250**	0	–20	25	25	15	–	15		25
250	**315**	0	–25	31	31	19	–	19		30
315	**400**	0	–28	35	35	21	–	21		35
400	**500**	0	–33	41	41	25	–	25		40
500	**630**	0	–38	48	48	29	–	29		50
630	**800**	0	–45	56	56	34	–	34		60
800	**1 000**	0	–60	75	75	45	–	45		75
1 000	**1 250**	0	–75	–	–	–	–	–		85
1 250	**1 600**	0	–90	–	–	–	–	–		100
1 600	**2 000**	0	–115	–	–	–	–	–		100
2 000	**2 500**	0	–135	–	–	–	–	–		120

1) Tolerances for tapered bores (table 11, page 47).
2) Diameter series 7 and 8 not covered by ISO 492.
3) Applies to inner rings and outer rings of bearings of matched bearing sets consisting of two or more bearings. Does not apply to universally matchable angular contact ball bearings.
4) Applies to bearings prior to mounting and after removal of internal or external snap ring.
5) Capped bearings are sealed or shielded bearings.

Table 4

P5 class tolerances for radial bearings, except tapered roller bearings

Inner ring

d >	d ≤	$t_{\Delta dmp}$[1] U	$t_{\Delta dmp}$[1] L	t_{Vdsp}[1] Diameter series 7, 8, 9[2]	t_{Vdsp}[1] Diameter series 0, 1, 2, 3, 4	t_{Vdmp}	$t_{\Delta Bs}$ All U	$t_{\Delta Bs}$ Normal L	$t_{\Delta Bs}$ Modified[4] L	t_{VBs}	t_{Kia}	t_{Sd}	t_{Sia}[3]
mm		µm		µm		µm	µm			µm	µm	µm	µm
–	2,5	0	–5	5	4	3	0	–40	–250	5	4	7	7
2,5	10	0	–5	5	4	3	0	–40	–250	5	4	7	7
10	18	0	–5	5	4	3	0	–80	–250	5	4	7	7
18	30	0	–6	6	5	3	0	–120	–250	5	4	8	8
30	50	0	–8	8	6	4	0	–120	–250	5	5	8	8
50	80	0	–9	9	7	5	0	–150	–250	6	5	8	8
80	120	0	–10	10	8	5	0	–200	–380	7	6	9	9
120	180	0	–13	13	10	7	0	–250	–380	8	8	10	10
180	250	0	–15	15	12	8	0	–300	–500	10	10	11	13
250	315	0	–18	18	14	9	0	–350	–500	13	13	13	15
315	400	0	–23	23	18	12	0	–400	–630	15	15	15	20
400	500	0	–28	28	21	14	0	–450	–	18	17	18	23
500	630	0	–35	35	26	18	0	–500	–	20	19	20	25
630	800	0	–45	–	–	–	0	–750	–	26	22	26	30
800	1 000	0	–60	–	–	–	0	–1 000	–	32	26	32	30
1 000	1 250	0	–75	–	–	–	0	–1 250	–	38	30	38	30
1 250	1 600	0	–90	–	–	–	0	–1 600	–	45	35	45	30
1 600	2 000	0	–115	–	–	–	0	–2 000	–	55	40	55	30

Outer ring

D >	D ≤	$t_{\Delta Dmp}$ U	$t_{\Delta Dmp}$ L	t_{VDsp}[5] Diameter series 7, 8, 9[2]	t_{VDsp}[5] Diameter series 0, 1, 2, 3, 4	t_{VDmp}	$t_{\Delta Cs}$	t_{VCs}	t_{Kea}	t_{SD}[6]	t_{Sea}[3]
mm		µm		µm		µm		µm	µm	µm	µm
2,5	18	0	–5	5	4	3	Identical to $t_{\Delta Bs}$ of an inner ring of the same bearing as the outer ring	5	5	4	8
18	30	0	–6	6	5	3		5	6	4	8
30	50	0	–7	7	5	4		5	7	4	8
50	80	0	–9	9	7	5		6	8	4	10
80	120	0	–10	10	8	5		8	10	4,5	11
120	150	0	–11	11	8	6		8	11	5	13
150	180	0	–13	13	10	7		8	13	5	14
180	250	0	–15	15	11	8		10	15	5,5	15
250	315	0	–18	18	14	9		11	18	6,5	18
315	400	0	–20	20	15	10		13	20	6,5	20
400	500	0	–23	23	17	12		15	23	7,5	23
500	630	0	–28	28	21	14		18	25	9	25
630	800	0	–35	35	26	18		20	30	10	30
800	1 000	0	–50	50	29	25		25	35	12,5	–
1 000	1 250	0	–63	–	–	–		30	40	15	–
1 250	1 600	0	–80	–	–	–		35	45	17,5	–
1 600	2 000	0	–100	–	–	–		38	55	20	–
2 000	2 500	0	–125	–	–	–		45	65	25	–

[1] Tolerances for tapered bores (table 11, page 47).
[2] Diameter series 7 and 8 not covered by ISO 492.
[3] Applies to groove ball bearings only, except for self-aligning ball bearings.
[4] Applies to inner rings and outer rings of bearings of matched bearing sets consisting of two or more bearings. Does not apply to universally matchable angular contact ball bearings.
[5] No values have been established for capped (sealed or shielded) bearings.
[6] Tolerance values have become half the values in accordance with the revised ISO standard because SD is defined as perpendicularity of outer ring outside surface axis with respect to datum established from the outer ring face.

Table 5

Normal and CL7C class tolerances for metric tapered roller bearings

Inner ring, bearing width and ring widths

d		$t_{\Delta dmp}$		t_{Vdsp}	t_{Vdmp}	$t_{\Delta Bs}$		t_{Kia} Tolerance classes		$t_{\Delta Ts}$		$t_{\Delta T1s}$		$t_{\Delta T2s}$	
>	≤	U	L			U	L	Normal	CL7C[1)]	U	L	U	L	U	L
mm		µm		µm	µm	µm		µm		µm		µm		µm	
10	**18**	0	–12	12	9	0	–120	15	7	200	0	100	0	100	0
18	**30**	0	–12	12	9	0	–120	18	8	200	0	100	0	100	0
30	**50**	0	–12	12	9	0	–120	20	10	200	0	100	0	100	0
50	**80**	0	–15	15	11	0	–150	25	10	200	0	100	0	100	0
80	**120**	0	–20	20	15	0	–200	30	13	200	–200	100	–100	100	–100
120	**180**	0	–25	25	19	0	–250	35	–	350	–250	150	–150	200	–100
180	**250**	0	–30	30	23	0	–300	50	–	350	–250	150	–150	200	–100
250	**315**	0	–35	35	26	0	–350	60	–	350	–250	150	–150	200	–100
315	**400**	0	–40	40	30	0	–400	70	–	400	–400	200	–200	200	–200

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp}	t_{VDmp}	$t_{\Delta Cs}$		t_{Kea} Tolerance classes	
>	≤	U	L			U	L	Normal	CL7C[1)]
mm		µm		µm	µm	µm		µm	
18	**30**	0	–12	12	9	0	–120	18	9
30	**50**	0	–14	14	11	0	–120	20	10
50	**80**	0	–16	16	12	0	–150	25	13
80	**120**	0	–18	18	14	0	–200	35	18
120	**150**	0	–20	20	15	0	–250	40	20
150	**180**	0	–25	25	19	0	–250	45	23
180	**250**	0	–30	30	23	0	–300	50	–
250	**315**	0	–35	35	26	0	–350	60	–
315	**400**	0	–40	40	30	0	–400	70	–
400	**500**	0	–45	45	34	0	–450	80	–
500	**630**	0	–50	60	38	0	–500	100	–
630	**800**	0	–75	80	55	0	–750	120	–

[1)] Tolerances are not in accordance with any ISO tolerance class and are for high-performance design tapered roller bearings.

Table 6

CLN class tolerances[1] for metric tapered roller bearings

Inner ring, bearing width and ring widths

d		$t_{\Delta dmp}$		t_{Vdsp}	t_{Vdmp}	$t_{\Delta Bs}$		t_{Kia}	$t_{\Delta Ts}$		$t_{\Delta T1s}$		$t_{\Delta T2s}$	
>	≤	U	L			U	L		U	L	U	L	U	L
mm		µm		µm	µm	µm		µm	µm		µm		µm	
10	**18**	0	–12	12	9	0	–50	15	100	0	50	0	50	0
18	**30**	0	–12	12	9	0	–50	18	100	0	50	0	50	0
30	**50**	0	–12	12	9	0	–50	20	100	0	50	0	50	0
50	**80**	0	–15	15	11	0	–50	25	100	0	50	0	50	0
80	**120**	0	–20	20	15	0	–50	30	100	0	50	0	50	0
120	**180**	0	–25	25	19	0	–50	35	150	0	50	0	100	0
180	**250**	0	–30	30	23	0	–50	50	150	0	50	0	100	0
250	**315**	0	–35	35	26	0	–50	60	200	0	100	0	100	0
315	**400**	0	–40	40	30	0	–50	70	200	0	100	0	100	0

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp}	t_{VDmp}	$t_{\Delta Cs}$		t_{Kea}
>	≤	U	L			U	L	
mm		µm		µm	µm	µm		µm
18	**30**	0	–12	12	9	0	–100	18
30	**50**	0	–14	14	11	0	–100	20
50	**80**	0	–16	16	12	0	–100	25
80	**120**	0	–18	18	14	0	–100	35
120	**150**	0	–20	20	15	0	–100	40
150	**180**	0	–25	25	19	0	–100	45
180	**250**	0	–30	30	23	0	–100	50
250	**315**	0	–35	35	26	0	–100	60
315	**400**	0	–40	40	30	0	–100	70
400	**500**	0	–45	45	34	0	–100	80
500	**630**	0	–50	60	38	0	–100	100

[1] Tolerance class CLN is in accordance with ISO tolerance class 6X.

Table 7

P5 class tolerances for metric tapered roller bearings

Inner ring and bearing width

d		$t_{\Delta dmp}$		t_{Vdsp}	t_{Vdmp}	$t_{\Delta Bs}$		t_{Kia}	t_{Sd}	$t_{\Delta Ts}$		$t_{\Delta T1s}$		$t_{\Delta T2s}$	
>	≤	U	L			U	L			U	L	U	L	U	L
mm		µm		µm	µm	µm		µm	µm	µm		µm		µm	
10	**18**	0	–7	5	5	0	–200	5	7	+200	–200	+100	–100	+100	–100
18	**30**	0	–8	6	5	0	–200	5	8	+200	–200	+100	–100	+100	–100
30	**50**	0	–10	8	5	0	–240	6	8	+200	–200	+100	–100	+100	–100
50	**80**	0	–12	9	6	0	–300	7	8	+200	–200	+100	–100	+100	–100
80	**120**	0	–15	11	8	0	–400	8	9	+200	–200	+100	–100	+100	–100
120	**180**	0	–18	14	9	0	–500	11	10	+350	–250	+150	–150	+200	–100
180	**250**	0	–22	17	11	0	–600	13	11	+350	–250	+150	–150	+200	–100
250	**315**	0	–25	19	13	0	–700	13	13	+350	–250	+150	–150	+200	–100
315	**400**	0	–30	23	15	0	–800	15	15	+400	–400	+200	–200	+200	–200
400	**500**	0	–35	28	17	0	–900	20	17	+450	–450	+225	–225	+225	–225
500	**630**	0	–40	35	20	0	–1 100	25	20	+500	–500	–	–	–	–
630	**800**	0	–50	45	25	0	–1 600	30	25	+600	–600	–	–	–	–
800	**1 000**	0	–60	60	30	0	–2 000	37	30	+750	–750	–	–	–	–
1 000	**1 250**	0	–75	75	37	0	–2 000	45	40	+750	–750	–	–	–	–
1 250	**1 600**	0	–90	90	45	0	–2 000	55	50	+900	–900	–	–	–	–

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp}	t_{VDmp}	$t_{\Delta Cs}$	t_{Kea}	t_{SD}[1]
>	≤	U	L					
mm		µm		µm	µm		µm	µm
18	**30**	0	–8	6	5	Identical to $t_{\Delta Bs}$ of an inner ring of the same bearing as the outer ring	6	4
30	**50**	0	–9	7	5		7	4
50	**80**	0	–11	8	6		8	4
80	**120**	0	–13	10	7		10	4,5
120	**150**	0	–15	11	8		11	5
150	**180**	0	–18	14	9		13	5
180	**250**	0	–20	15	10		15	5,5
250	**315**	0	–25	19	13		18	6,5
315	**400**	0	–28	22	14		20	6,5
400	**500**	0	–33	26	17		24	8,5
500	**630**	0	–38	30	20		30	10
630	**800**	0	–45	38	25		36	12,5
800	**1 000**	0	–60	50	30		43	15
1 000	**1 250**	0	–80	65	38		52	19
1250	**1 600**	0	–100	90	50		62	25
1 600	**2 000**	0	–125	120	65		73	32,5

[1] Tolerance values have become half the values in accordance with the revised ISO standard (2014) because SD is defined as perpendicularity of the outer ring outside surface axis with respect to datum established from the outer ring face.

Table 8

Normal tolerances for inch radial bearings, except tapered roller bearings

Inner ring

d		$t_{\Delta dmp}$		t_{Vdsp}	$t_{\Delta Bs}$		t_{VBs}	t_{Kia}	t_{Sia}
>	≤	U	L		U	L			
mm		µm		µm	µm		µm	µm	µm
–	**25,4**	+5	–5	10	0	–127	13	10	15
25,4	**50,8**	+5	–8	10	0	–127	13	10	20
50,8	**76,2**	+5	–8	13	0	–127	13	15	30
76,2	**152,4**	+5	–8	18	0	–127	15	20	38
152,4	**203,2**	+5	–13	33	0	–127	15	25	51
203,2	**304,8**	+5	–13	33	0	–254	20	30	51
304,8	**381**	+5	–20	51	0	–406	25	38	64

Outer ring

D		$t_{\Delta Dmp}$		t_{VDsp}	$t_{\Delta Cs}$	t_{VCs}	t_{Kea}	t_{Sea}
>	≤	U	L					
mm		µm		µm		µm	µm	µm
–	**25,4**	–8	–18	10	Identical to $t_{\Delta Bs}$ of an inner ring of the same bearing as the outer ring	13	10	15
25,4	**50,8**	–8	–20	10		13	13	15
50,8	**76,2**	–13	–25	13		13	15	20
76,2	**127**	–20	–33	18		15	18	30
127	**203,2**	–33	–46	33		15	20	38
203,2	**304,8**	–33	–46	33		20	25	51
304,8	**381**	–33	–58	51		25	30	51
381	**508**	–33	–58	51		30	38	64

Table 9

Tolerances for inch tapered roller bearings

Inner ring

d		$t_{\Delta dmp}$ Tolerance classes				t_{Kia}, t_{Sia}
		Normal, CL2		CL3, CL0		
>	≤	U	L	U	L	
mm		µm		µm		
–	**76,2**	+13	0	+13	0	Values are given in outer ring table
76,2	**101,6**	+25	0	+13	0	
101,6	**266,7**	+25	0	+13	0	
266,7	**304,8**	+25	0	+13	0	
304,8	**609,6**	+51	0	+25	0	
609,6	**914,4**	+76	0	+38	0	

Outer ring

D		$t_{\Delta Dmp}$ Tolerance classes				t_{Kia}, t_{Kea}, t_{Sia}, t_{Sea} Tolerance classes				t_{Kea} Tolerance class
		Normal, CL2		CL3, CL0		Normal	CL2	CL3	CL0	CL7C
>	≤	H	L	H	L					
mm		µm				µm				µm
–	**304,8**	+25	0	+13	0	51	38	8	4	→ **table 5, page 41**
304,8	**609,6**	+51	0	+25	0	51	38	18	9	
609,6	**914,4**	+76	0	+38	0	76	51	51	26	

Abutment width of single row bearings

d		D		$t_{\Delta Ts}$ Tolerance classes					
				Normal		CL2		CL3, CL0	
>	≤	>	≤	U	L	U	L	U	L
mm		mm		µm					
–	**101,6**	**–**	**–**	+203	0	+203	0	+203	–203
101,6	**266,7**	**–**	**–**	+356	–254	+203	0	+203	–203
266,7	**304,8**	**–**	**–**	+356	–254	+203	0	+203	–203
304,8	**609,6**	**–**	**508**	+381	–381	+381	–381	+203	–203
304,8	**609,6**	**508**	**–**	+381	–381	+381	–381	+381	–381
609,6	**–**	**–**	**–**	+381	–381	–	–	+381	–381

Table 10

Tolerances for thrust bearings

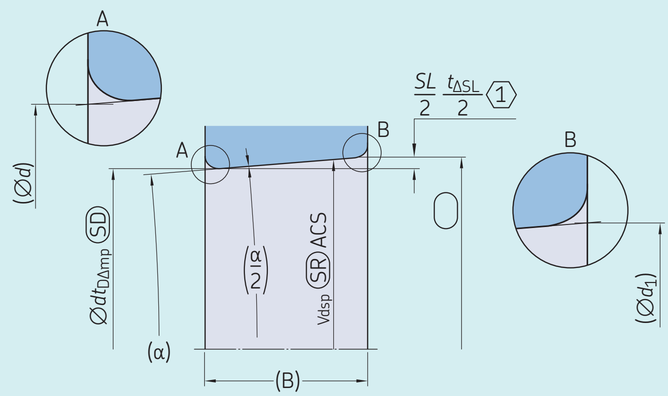
Nominal diameter d, d_2, D[1)] >	≤	Shaft washer $t_{\Delta dmp}, t_{\Delta d2mp}$ Tolerance classes Normal, P6, P5 U	L	t_{Vdsp}, t_{Vd2sp}	t_{Si}[2)3)] Tolerance classes Normal	t_{Si}[2)3)] P6	t_{Si}[2)3)] P5	Housing washer $t_{\Delta Dmp}$ Tolerance classes Normal, P6, P5 U	L	t_{VDsp}	t_{Se}[2)]
mm		µm						µm			
–	18	0	–8	6	10	5	3	0	–11	8	Identical to t_{Si} of shaft washer of same bearing
18	30	0	–10	8	10	5	3	0	–13	10	
30	50	0	–12	9	10	6	3	0	–16	12	
50	80	0	–15	11	10	7	4	0	–19	14	
80	120	0	–20	15	15	8	4	0	–22	17	
120	180	0	–25	19	15	9	5	0	–25	19	
180	250	0	–30	23	20	10	5	0	–30	23	
250	315	0	–35	26	25	13	7	0	–35	26	
315	400	0	–40	30	30	15	7	0	–40	30	
400	500	0	–45	34	30	18	9	0	–45	34	
500	630	0	–50	38	35	21	11	0	–50	38	
630	800	0	–75	55	40	25	13	0	–75	55	
800	1 000	0	–100	75	45	30	15	0	–100	75	
1 000	1 250	0	–125	95	50	35	18	0	–125	95	
1 250	1 600	0	–160	120	60	40	25	0	–160	120	
1 600	2 000	0	–200	150	75	45	30	0	–200	150	
2 000	2 500	0	–250	190	90	50	40	0	–250	190	

Bearing height d, d_2[1)] >	≤	$t_{\Delta Ts}$ Single direction bearings without seat washer U	L	$t_{\Delta T1s}$[4)] Single direction bearings with seat washer U	L	$t_{\Delta T1s}$ Double direction bearings without seat washers U	L	$t_{\Delta T3s}$[4)] Double direction bearings with seat washers U	L	$t_{\Delta T4s}$[4)5)] Spherical roller thrust bearings SKF U	L	SKF Explorer U	L
mm		µm		µm		µm		µm		µm			
–	30	20	–250	100	–250	150	–400	300	–400	–	–	–	–
30	50	20	–250	100	–250	150	–400	300	–400	–	–	–	–
50	80	20	–300	100	–300	150	–500	300	–500	0	–125	0	–100
80	120	25	–300	150	–300	200	–500	400	–500	0	–150	0	–100
120	180	25	–400	150	–400	200	–600	400	–600	0	–175	0	–125
180	250	30	–400	150	–400	250	–600	500	–600	0	–200	0	–125
250	315	40	–400	–	–	–	–	–	–	0	–225	0	–150
315	400	40	–500	–	–	–	–	–	–	0	–300	0	–200
400	500	50	–500	–	–	–	–	–	–	0	–400	–	–
500	630	60	–600	–	–	–	–	–	–	0	–500	–	–
630	800	70	–750	–	–	–	–	–	–	0	–630	–	–
800	1 000	80	–1 000	–	–	–	–	–	–	0	–800	–	–
1 000	1 250	100	–1 400	–	–	–	–	–	–	0	–1 000	–	–
1 250	1 600	120	–1 600	–	–	–	–	–	–	0	–1 200	–	–
1 600	2 000	140	–1 900	–	–	–	–	–	–	–	–	–	–
2 000	2 500	160	–2 300	–	–	–	–	–	–	–	–	–	–

1) For double direction bearings, the values apply only for $d_2 \leq 190$ mm and $D \leq 360$ mm.
2) Applies only to thrust ball bearings and thrust cylindrical roller bearings, each with 90° contact angle.
3) Not applicable for central shaft washers.
4) Not included in ISO 199.
5) ISO 199 uses symbol T.

Table 11

Normal, P6 and P5 class tolerances for tapered bores, taper 1:12



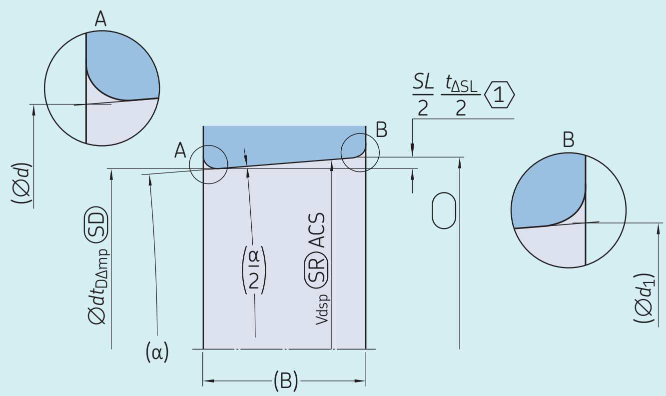
1 = *SL* is a calculated nominal size from *d* and d_1, i.e. $SL = (d_1 - d) = 2B \tan(\alpha/2)$;
ΔSL is a calculated characteristic, i.e. ΔSL = Δd1mp – Δdmp

Bore diameter		**Tolerance classes**									
		Normal[1], P6					P5				
d		**$t_{\Delta dmp}$**		t_{Vdsp}[2]	**$t_{\Delta SL}$**		**$t_{\Delta dmp}$**		**t_{Vdsp}**[2]	**$t_{\Delta SL}$**	
>	≤	U	L		U	L	U	L		U	L
mm		µm		µm	µm		µm		µm	µm	
18	**30**	+21	0	13	+21	0	+13	0	13	+13	0
30	**50**	+25	0	15	+25	0	+16	0	15	+16	0
50	**80**	+30	0	19	+30	0	+19	0	19	+19	0
80	**120**	+35	0	22	+35	0	+22	0	22	+22	0
120	**180**	+40	0	31	+40	0	+25	0	25	+25	0
180	**250**	+46	0	38	+46	0	+29	0	29	+29	0
250	**315**	+52	0	44	+52	0	+32	0	32	+32	0
315	**400**	+57	0	50	+57	0	+36	0	36	+36	0
400	**500**	+63	0	56	+63	0	+40	0	–	+40	0
500	**630**	+70	0	70	+70	0	+44	0	–	+44	0
630	**800**	+80	0	–	+80	0	+50	0	–	+50	0
800	**1 000**	+90	0	–	+90	0	+56	0	–	+56	0
1 000	**1 250**	+105	0	–	+105	0	+66	0	–	+66	0
1 250	**1 600**	+125	0	–	+125	0	+78	0	–	+78	0
1 600	**2 000**	+150	0	–	+150	0	+92	0	–	+92	0

1) Smaller tolerance zones than ISO 492.
2) Applies in any cross section of the bore.

Table 12

Normal tolerances for tapered bores, taper 1:30



1 = SL is a calculated nominal size from d and d_1, i.e. $SL = (d_1 - d) = 2B \tan(\alpha/2)$;
ΔSL is a calculated characteristic, i.e. $\Delta SL = \Delta d1mp - \Delta dmp$

Bore diameter		**Tolerance class** Normal				
d		$t_{\Delta dmp}$		t_{Vdsp}[1)]	$t_{\Delta SL}$	
>	≤	U	L		U	L
mm		µm		µm	µm	
–	**80**	+15	0	19	+30	0
80	**120**	+20	0	22	+35	0
120	**180**	+25	0	40	+40	0
180	**250**	+30	0	46	+46	0
250	**315**	+35	0	52	+52	0
315	**400**	+40	0	57	+57	0
400	**500**	+45	0	63	+63	0
500	**630**	+50	0	70	+70	0
630	**800**	+75	0	–	+100	0
800	**1 000**	+100	0	–	+100	0
1 000	**1 250**	+125	0	–	+115	0
1 250	**1 600**	+160	0	–	+125	0
1 600	**2 000**	+200	0	–	+150	0

1) Applies in any cross section of the bore.

Table 13

Tolerance symbols

Tolerance symbol	Definition
	Radial bearings inner ring – cylindrical and tapered bore
d	**1** Cylindrical bore: Nominal bore diameter **2** Tapered bore: Nominal bore diameter at the theoretical small end
Δdmp	**1** Cylindrical bore: Deviation of a mid-range size (out of two-point sizes) of bore diameter in any cross section from its nominal size **2** Tapered bore: Deviation of a mid-range size (out of two-point sizes) of bore diameter at the theoretical small end from its nominal size
Δds	Deviation of a two-point size of bore diameter of a cylindrical bore from its nominal size
Vdsp	Range of two-point sizes of bore diameter in any cross section of a cylindrical or tapered bore
Vdmp	Range of mid-range sizes (out of two-point sizes) of bore diameter obtained from any cross section of a cylindrical bore
B	Nominal inner ring width
ΔBs Nomal, Modified[1)]	**1** Symmetrical rings: Deviation of a two-point size of inner ring width from its nominal size **2** Asymmetrical rings, upper limit: Deviation of a minimum circumscribed size of inner ring width, between two opposite lines, in any longitudinal section which includes the inner ring bore axis, from its nominal size **3** Asymmetrical rings, lower limit: Deviation of a two-point size of inner ring width from its nominal size
VBs	**1** Symmetrical rings: Range of two-point sizes of inner ring width **2** Asymmetrical rings: Range of minimum circumscribed sizes of inner ring width, between two opposite lines, obtained from any longitudinal section which includes the inner ring bore axis
Kia[2)]	Circular radial run-out of inner ring bore surface of assembled bearing with respect to datum, i.e. axis, established from the outer ring outside surface
Sd[2)]	Circular axial run-out of inner ring face with respect to datum, i.e. axis, established from the inner ring bore surface
Sia[2)]	Circular axial run-out of inner ring face of assembled bearing with respect to datum, i.e. axis, established from the outer ring outside surface
	Radial bearings inner ring – tapered bore only
d_1	Nominal bore diameter at the theoretical large end of a tapered bore
Δd1mp	Deviation of a mid-range size (out of two-point sizes) of bore diameter at the theoretical large end from its nominal size
SL	Taper slope, the difference between nominal diameters at the theoretical large end and small end of a tapered bore (d_1 – d)
ΔSL	Deviation of taper slope of a tapered inner ring bore from its nominal size

1) Modified applies to inner rings and outer rings of bearings of matched bearing sets consisting of two or more bearings. Does not apply to universally matchable angular contact ball bearings.
2) Geometrical tolerances

cont. table 13

Tolerance symbols

Tolerance symbol	Definition
	Radial bearings outer ring
D	Nominal outside diameter
ΔDmp	Deviation of a mid-range size (out of two-point sizes) of outside diameter in any cross section from its nominal size
ΔDs	Deviation of a two-point size of outside diameter from its nominal size
VDsp	Range of two-point sizes of outside diameter in any cross section
VDmp	Range of mid-range sizes (out of two-point sizes) of outside diameter obtained from any cross section
C	Nominal outer ring width
ΔCs Nomal, Modified[1]	**1** Symmetrical rings: Deviation of a two-point size of outer ring width from its nominal size **2** Asymmetrical rings, upper limit: Deviation of a minimum circumscribed size of outer ring width, between two opposite lines, in any longitudinal section which includes the outer ring outside surface axis, from its nominal size **3** Asymmetrical rings, lower limit: Deviation of a two-point size of outer ring width from its nominal size
VCs	**1** Symmetrical rings: Range of two-point sizes of outer ring width **2** Asymmetrical rings: Range of minimum circumscribed sizes of outer ring width, between two opposite lines, obtained from any longitudinal section which includes the outer ring outside surface axis
Kea[2]	Circular radial run-out of outer ring outside surface of assembled bearing with respect to datum, i.e. axis, established from the inner ring bore surface
SD[2]	Perpendicularity of outer ring outside surface axis with respect to datum established from the outer ring face
Sea[2]	Circular axial run-out of outer ring face of assembled bearing with respect to datum, i.e. axis, established from the inner ring bore surface
	Chamfer limits
r_s	Single chamfer dimension
$r_{s\ min}$	Smallest single chamfer dimension of r_s, r_1, r_2, r_3, r_4...
r_1, r_3	Radial direction chamfer dimensions
r_2, r_4	Axial direction chamfer dimensions
	Tapered roller bearings
T	Nominal assembled bearing width
ΔTs	Deviation of minimum circumscribed size of assembled bearing width from its nominal size
T_1	Nominal effective width of cone (inner ring, with roller and cage assembly) assembled with a master cup (outer ring)
T_2	Nominal effective width of cup assembled with a master cone
ΔT1s	Deviation of minimum circumscribed size of effective width (cone assembled with a master cup) from its nominal size
ΔT2s	Deviation of minimum circumscribed size of effective width (cup assembled with a master cone) from its nominal size

[1] Modified applies to inner rings and outer rings of bearings of matched bearing sets consisting of two or more bearings. Does not apply to universally matchable angular contact ball bearings.
[2] Geometrical tolerances

cont. table 13

Tolerance symbols

Tolerance symbol	Definition
	Thrust bearings shaft washer
d	Nominal bore diameter of shaft washer, single direction bearing
Δds	Deviation of a two-point size of shaft washer bore diameter from its nominal size
Δdmp	Deviation of a mid-range size (out of two-point sizes) of shaft washer bore diameter in any cross section from its nominal size
Vdsp	Range of two-point sizes of shaft washer bore diameter in any cross section
$\mathbf{d_2}$	Nominal bore diameter of central shaft washer, double direction bearing
Δd2mp	Deviation of a mid-range size (out of two-point sizes) of central shaft washer bore diameter in any cross section from its nominal size
Vd2sp	Range of two-point sizes of central shaft washer bore diameter in any cross section
Si	**1** Range of two-point sizes of thickness between shaft washer raceway and the back face, cylindrical roller thrust bearing **2** Range of minimum spherical sizes between the raceway and the opposite back face of the shaft washer, obtained from any longitudinal section which includes the shaft washer bore axis, thrust ball bearing
	Thrust bearings housing washer
D	Nominal outside diameter of housing washer
ΔDs	Deviation of a two-point size of housing washer outside diameter from its nominal size
ΔDmp	Deviation of a mid-range size (out of two-point sizes) of housing washer outside diameter in any cross section from its nominal size
VDsp	Range of two-point sizes of housing washer outside diameter in any cross section
Se	**1** Range of two-point sizes of thickness between housing washer raceway and the back face, cylindrical roller thrust bearing **2** Range of minimum spherical sizes between the raceway and the opposite back face of the housing washer, obtained from any longitudinal section which includes the housing washer outside surface axis, thrust ball bearing
	Thrust bearings assembled bearing height
T	Nominal assembled bearing height, single direction thrust bearing (except spherical roller thrust bearing → T_4)
ΔTs	Deviation of minimum circumscribed size of assembled bearing height from its nominal size, single direction thrust bearing (except spherical roller thrust bearing → ΔT4s)
$\mathbf{T_1}$	**1** Nominal assembled bearing height, double direction thrust bearing **2** Nominal assembled bearing height, single direction thrust bearing with a seat washer
ΔT1s	**1** Deviation of minimum circumscribed size of assembled bearing height from its nominal size, double direction thrust bearing **2** Deviation of minimum circumscribed size of assembled bearing height from its nominal size, single direction thrust bearing with a seat washer
$\mathbf{T_3}$[3)]	Nominal assembled bearing height, double direction thrust bearing with seat washers
ΔT3s[3)]	Deviation of minimum circumscribed size of assembled bearing height from its nominal size, double direction thrust bearing with seat washers
$\mathbf{T_4}$[4)]	Nominal assembled bearing height, spherical roller thrust bearing
ΔT4s[4)]	Deviation of minimum circumscribed size of assembled bearing height from its nominal size, spherical roller thrust bearing

3) Not included in ISO 199.
4) In ISO 199, the symbol T is used.

Table 14

Diameter series (radial bearings)

Bearing type	Diameter series 7, 8, 9	0, 1	2, 3, 4
Deep groove ball bearings[1)]	617, 618, 619 627, 628 637, 638, 639	60 160, 161 630	2, 3 42, 43 62, 63, 64, 622, 623
Angular contact ball bearings		70	32, 33 72, 73 QJ 2, QJ 3
Self-aligning ball bearings[2)]	139	10, 130	12, 13, 112 22, 23
Cylindrical roller bearings		NU 10, 20 NJ 10	NU 2, 3, 4, 12, 22, 23 NJ 2, 3, 4, 22, 23 NUP 2, 3, 22, 23 N 2, 3
Full complement cylindrical roller bearings	NCF 18, 19, 28, 29 NNC 48, 49 NNCF 48, 49 NNCL 48, 49	NCF 30 NNF 50 NNCF 50	NCF 22 NJG 23
Needle roller bearings	NA 48, 49, 69		
Spherical roller bearings	238, 239 248, 249	230, 231 240, 241	222, 232 213, 223
CARB toroidal roller bearings	C 39, 49, 59, 69	C 30, 31 C 40, 41	C 22, 23 C 32

1) Bearings 604, 607, 608, 609 belong to diameter series 0,
bearings 623, 624, 625, 626, 627, 628 and 629 to diameter series 2,
bearings 634, 635 and 638 to diameter series 3, bearing 607/8 to diameter series 9.
2) Bearing 108 belongs to diameter series 0,
bearings 126, 127 and 129 to diameter series 2,
bearing 135 to diameter series 3.

Table 15

Chamfer dimension limits for metric radial and thrust bearings, except tapered roller bearings

Minimum single chamfer dimension	Nominal bearing bore diameter		Maximum chamfer dimensions		
			Radial bearings		Thrust bearings
$r_{s\,min}$	d >	≤	$r_{1,3}$	$r_{2,4}$	$r_{1,2,3,4}$
mm	mm		mm		
0,05	–	–	0,1	0,2	0,1
0,08	–	–	0,16	0,3	0,16
0,1	–	–	0,2	0,4	0,2
0,15	–	–	0,3	0,6	0,3
0,2	–	–	0,5	0,8	0,5
0,3	–	40	0,6	1	0,8
	40	–	0,8	1	0,8
0,6	–	40	1	2	1,5
	40	–	1,3	2	1,5
1	–	50	1,5	3	2,2
	50	–	1,9	3	2,2
1,1	–	120	2	3,5	2,7
	120	–	2,5	4	2,7
1,5	–	120	2,3	4	3,5
	120	–	3	5	3,5
2	–	80	3	4,5	4
	80	220	3,5	5	4
	220	–	3,8	6	4
2,1	–	280	4	6,5	4,5
	280	–	4,5	7	4,5
2,5	–	100	3,8	6	–
	100	280	4,5	6	–
	280	–	5	7	–
3	–	280	5	8	5,5
	280	–	5,5	8	5,5
4	–	–	6,5	9	6,5
5	–	–	8	10	8
6	–	–	10	13	10
7,5	–	–	12,5	17	12,5
9,5	–	–	15	19	15
12	–	–	18	24	18

Table 16

Chamfer dimension limits for metric radial tapered roller bearings

Minimum single chamfer dimension	Nominal bearing bore/ outside diameter		Maximum chamfer dimensions	
$r_{s\,min}$	d, D >	≤	$r_{1,3}$	$r_{2,4}$
mm	mm		mm	
0,3	–	40	0,7	1,4
	40	–	0,9	1,6
0,5	–	40	1,1	1,7
	40	–	1,2	1,9
0,6	–	40	1,1	1,7
	40	–	1,3	2
1	–	50	1,6	2,5
	50	–	1,9	3
1,5	–	120	2,3	3
	120	250	2,8	3,5
	250	–	3,5	4
2	–	120	2,8	4
	120	250	3,5	4,5
	250	–	4	5
2,5	–	120	3,5	5
	120	250	4	5,5
	250	–	4,5	6
3	–	120	4	5,5
	120	250	4,5	6,5
	250	400	5	7
	400	–	5,5	7,5
4	–	120	5	7
	120	250	5,5	7,5
	250	400	6	8
	400	–	6,5	8,5
5	–	180	6,5	8
	180	–	7,5	9
6	–	180	7,5	10
	180	–	9	11

Table 17

Chamfer dimension limits for inch tapered roller bearings

Minimum single chamfer dimension		Inner ring Nominal bearing bore diameter		Maximum chamfer dimensions		Outer ring Nominal bearing outside diameter		Maximum chamfer dimensions	
$r_{s\,min}$ >	≤	d >	≤	r_1	r_2	D >	≤	r_3	r_4
mm		mm		mm		mm		mm	
0,6	**1,4**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 0,9	$r_{2\,min}$ + 2	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 0,9	$r_{4\,min}$ + 2
1,4	**2,5**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 2	$r_{2\,min}$ + 3	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 2	$r_{4\,min}$ + 3
2,5	**4,0**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	400	$r_{1\,min}$ + 2	$r_{2\,min}$ + 4	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
		400	–	$r_{1\,min}$ + 2,5	$r_{2\,min}$ + 4,5	355,6	400	$r_{3\,min}$ + 2	$r_{4\,min}$ + 4
						400	–	$r_{3\,min}$ + 2,5	$r_{4\,min}$ + 4,5
4,0	**5,0**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 2,5	$r_{2\,min}$ + 4	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 2,5	$r_{4\,min}$ + 4
5,0	**6,0**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 3	$r_{2\,min}$ + 5	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 3	$r_{4\,min}$ + 5
6,0	**7,5**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 4,5	$r_{2\,min}$ + 6,5	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 4,5	$r_{4\,min}$ + 6,5
7,5	**9,5**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 6,5	$r_{2\,min}$ + 9,5	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 6,5	$r_{4\,min}$ + 9,5
9,5	**12**	–	101,6	$r_{1\,min}$ + 0,5	$r_{2\,min}$ + 1,3	–	168,3	$r_{3\,min}$ + 0,6	$r_{4\,min}$ + 1,2
		101,6	254	$r_{1\,min}$ + 0,6	$r_{2\,min}$ + 1,8	168,3	266,7	$r_{3\,min}$ + 0,8	$r_{4\,min}$ + 1,4
		254	–	$r_{1\,min}$ + 8	$r_{2\,min}$ + 11	266,7	355,6	$r_{3\,min}$ + 1,7	$r_{4\,min}$ + 1,7
						355,6	–	$r_{3\,min}$ + 8	$r_{4\,min}$ + 11

Rounding values

Shoulder diameters

The dimensions for the shoulder diameters of radial bearings are rounded up or down to a level that is suitable for general machinery applications. Diameter dimensions of the inner ring are rounded down, whereas those of the outer ring are rounded up.

Load and speed ratings and fatigue load limits

The values of these parameters are rounded to a level that fits the accuracy of the calculations they are intended to be used in.

Masses

Masses are rounded to approximately ±5% of the actual value. They do not include the weight of any packaging.

Temperatures

Temperatures are typically rounded to 5 °C and are presented in both units (°C and °F). Because of the rounding, temperature values may not match when using unit conversion formulae.

A.3 Storage

A.3 Storage

Storage time is the period that a bearing can remain in storage in order to avoid adverse effects on operational performance of the bearing. SKF bearings are coated with a high-quality preservative oil to protect them from corrosion. Long storage times can be attained by storing bearings in their original, unopened and undamaged, packaging. The storage time of bearings also depends on their storage environment conditions. To maintain the potential operating performance of a bearing, SKF recommends a "first in, first out" inventory policy.

Storage time for open bearings

Typical storage times for open (unsealed) bearings are listed in **table 1**.

Storage time for capped bearings

Capped bearings (bearings with seals or shields) should be stored for a maximum of three years to avoid deterioration of their grease fill.

Additional storage-related factors

To avoid deterioration of your bearings while in storage, consider these factors:

- Store indoors, in a frost- and condensation-free environment, at a maximum ambient temperature of 40 °C *(105 °F)*, avoiding air flow.
- Store in vibration-free conditions. Vibration can cause damage to raceways.
- Store horizontally, preferably, to avoid damage that could be caused by the bearing falling over.
- Do not open or damage the original packaging.

Table 1

Storage time for open (unsealed) bearings

Storage environment conditions Relative air humidity	Ambient temperature		**Storage time**
%	°C	°F	years
65	20 to 25	*70 to 75*	10
75	20 to 25	*70 to 75*	5
75	35 to 40	*95 to 105*	3
Uncontrolled tropical conditions[1]			1

[1] Contact SKF for advice on coping with extreme conditions or attaining a longer storage time.

B

Bearing selection process

Bearing selection process

Bearing selection process

When selecting bearings for any purpose, ultimately you want to be certain of achieving the required level of equipment performance – and at the lowest possible cost. Robustness also is very important because the conditions in which your equipment is assembled, operated and maintained may not be precisely known and may, in fact, vary over time.

In addition to the bearing rating life, there are key factors you must consider when putting together the bearing specifications for an application, including:

- lubricant and supply method
- shaft and housing fits
- bearing clearance class
- cage material and guidance
- dimensional stability
- precision requirements
- bearing sealing
- mounting method and maintenance

To help evaluate these key factors, we recommend following the selection process shown on the right.

The process provides a straightforward step-by-step approach that shows the general relationship between each step. By clearly defining and naming the steps in this way, it should be easier to find information on a specific topic. In reality, however, you will find interdependencies that require you to loop back and forth between the steps.

Bearing selection process

Performance and operating conditions

Bearing type and arrangement

Bearing size

Lubrication

Operating temperature and speed

Bearing interfaces

Bearing execution

Sealing, mounting and dismounting

SKF support

SKF application engineering service

The SKF application engineering service provides expertise to help and support you with your technical needs.

Drawing on a wealth of experience, and supported by a global network of experts in a wide range of industries, local SKF application engineers work with original equipment manufacturers and end users to help and support them with their challenges.

Following a step-by-step application analysis process, and using SKF online and proprietary calculation tools, SKF application engineers can evaluate correct bearing type and size, and other requirements such as lubrication, fits and sealing, in order to obtain the right application solution and achieve reliable rotating equipment performance.

Contact the SKF application engineering service through your local SKF representative if you have any questions, or require any assistance, when using the bearing selection process guidelines or information in the product sections.

Supporting calculation tools

In the early stages of the application analysis and design process, bearing selection is initially made using various assumptions and, as the process progresses, additional input is included to fine tune results.

SKF can support you throughout this process with our engineering software tools (*Engineering software tools*, **page 63**), ranging from easy-to-use online tools, based on formulae provided in this catalogue, to our most sophisticated simulation systems incorporating the latest theories.

SKF is constantly developing its engineering software tools for SKF engineers and customers to support them in obtaining solutions that are technically, commercially and environmentally optimal.

Online tools

The SKF online engineering tools (*Engineering software tools*, **page 63**) provide functionality to:

- search for bearing data based on designation or dimensions
- calculate many useful bearing and application related parameters, including bearing basic rating life, SKF rating life, minimum load limit, shaft/housing tolerances and fits, relubrication intervals
- evaluate simple bearing arrangements
- generate drawings of bearings and housings that can be used in most commercially available CAD programs

SKF SimPro Quick

SKF SimPro Quick (*Engineering software tools*) is bearing simulation software that provides functionality to rapidly evaluate the design of bearing arrangements, and their field performance, based on relevant application requirements and conditions. In addition to the basic analysis provided by the online tools, it enables you to determine bearing load distribution and the effects of bearing stiffness and bearing clearance.

SKF SimPro Quick is intuitive, quick to learn, follows the SKF process for application analysis and bearing selection, and enables you to take greater advantage of SKF engineering know-how. It is fully compatible with the SKF SimPro platform, thus allowing you to easily exchange and discuss results with your SKF representative.

SKF SimPro Expert

SKF SimPro Expert (*Engineering software tools*) is the mainstream bearing application program used within the SKF application engineering community. It is a sophisticated bearing simulation system that enables analysis of muti-shaft systems at a deeper level than SKF SimPro Quick. It provides a wealth of functionality including:

- most of the needed modelling functionality for rotational analysis in general industry applications
- extensive analysis options for system behaviour, such as clearance effects, detailed rolling contact stress distribution
- design of experiments (DOE)

SKF SimPro Expert has also the option to add advanced modules for further analysis, as for example impact of bearing performance with a flexible support.

For additional information regarding SKF SimPro Expert and how it could help you, contact your local SKF representative.

SKF BEAST

SKF BEAST (Bearing Simulation Tool) (*Engineering software tools*) is a software simulation tool that enables SKF engineers to study the detailed dynamic behaviour within a mechanical sub-system, such as a bearing, under virtually any load condition.

It is a multibody system with special focus on transient conditions and detailed geometry and contacts, thus enabling detailed analysis, for example, of bearing cage behaviour and its wear mechanisms.

This enables the "testing" of new concepts and designs in a shorter time and with more information gained compared with traditional physical testing.

For additional information regarding SKF Sim Pro Expert and how it could help you, contact your local SKF representative.

Engineering software tools

Level of complexity

User needs	SKF tool	Software capabilities	
• Bearing design verification • Detailed, dynamic bearing and system evaluation • Evaluation of surface and contact behaviours	SKF BEAST	**Advanced analysis, bearing dynamics** Examples: • advanced contact models • dynamic behaviour of bearing components • structural fatigue	SKF internal use
• Bearing performance verification • Detailed bearing and system evaluation on complex models or multi-shafts	SKF SimPro Expert	**Advanced analysis, complex systems** Examples: • clearance optimization • flexible systems • detailed contact pressure distribution • influence on gear meshing	Customer accessible
• Bearing performance verification • Detailed bearing and system evaluation on single shaft	SKF SimPro Quick	**Advanced analysis, single shaft** Examples: • modified rating life according to ISO/TS16281 • bearing load distribution • bearing stiffness impact • clearance effect	
• Initial selection • Basic performance evaluation	Online tools • SKF Bearing Calculator • SKF Bearing Select • SKF LubeSelect	**Standard analysis, single bearing, single shaft** Examples: • SKF rating life • basic rating life • grease life • minimum load limit	

B.1

Performance and operating conditions

B.1 Performance and operating conditions

The first step in the bearing selection process is to understand and document:

- the required performance
- the operating conditions and assumptions of them
- any other application prerequisites

An application can set various requirements on the bearing solution. Common factors include:

- bearing life
- speed capability and ability to withstand applied acceleration levels
- precision of the radial and axial position of the shaft
- ability to cope with low or high temperatures or temperature gradients
- generated noise and vibration levels

The relative importance of these performance factors can influence the nature of the path you take through the steps of the bearing selection and application analysis process.

You should evaluate the operating conditions in as much detail as possible. The most important operating parameters are:

- load
- speed
- temperature
- lubricant and lubricant cleanliness

Usually these can be determined from physical and mechanical analysis of the application, or from experience with similar applications. Make sure that any assumptions made are clearly documented.

Operating conditions typically vary over time, e.g. in variable speed applications or because of seasonal temperature changes or increased output power. The range of the variation is important. In some cases, both limits of the range may be important, whereas in others, only the lower or the upper limit may be.

In order to optimize a design, you may need to loop through various steps of the bearing selection process. To minimize these, review and prioritize any application prerequisites, such as:

- available radial or axial space
- shaft diameters defined by shaft strength requirements
- lubricant choice determined by other components in the application

The shows the relationship between principal operating conditions, application requirements and various aspects of a bearing arrangement's design are shown in *Factors to consider when translating operating conditions and application requirements into a bearing solution*, **page 66**. The lists are not comprehensive and you may have to consider other factors and interrelationships, like cost and availability, when striving to obtain a robust and cost-effective solution.

Use the *Application data sheet*, at the end of this catalogue, to help when contacting the SKF application engineering service.

Factors to consider when translating operating conditions and application

Operating conditions and application requirements

Bearing type

- Space
- Load
- Speed
- Peak load
- Vibration
- Misalignment
- Noise
- Rigidity
- Friction
- Easy mounting

Precision

- Position control
- Run-out
- Speed
- Friction

Clearance

- Fits
- Component temperature
- Speed
- Preload
- Dimensional stability

Bearing size

- Shaft and housing deformation
- Cleanliness
- Viscosity
- Operating temperature
- Minimum load
- Permissible axial load
- Static load
- Dynamic load
- Required life

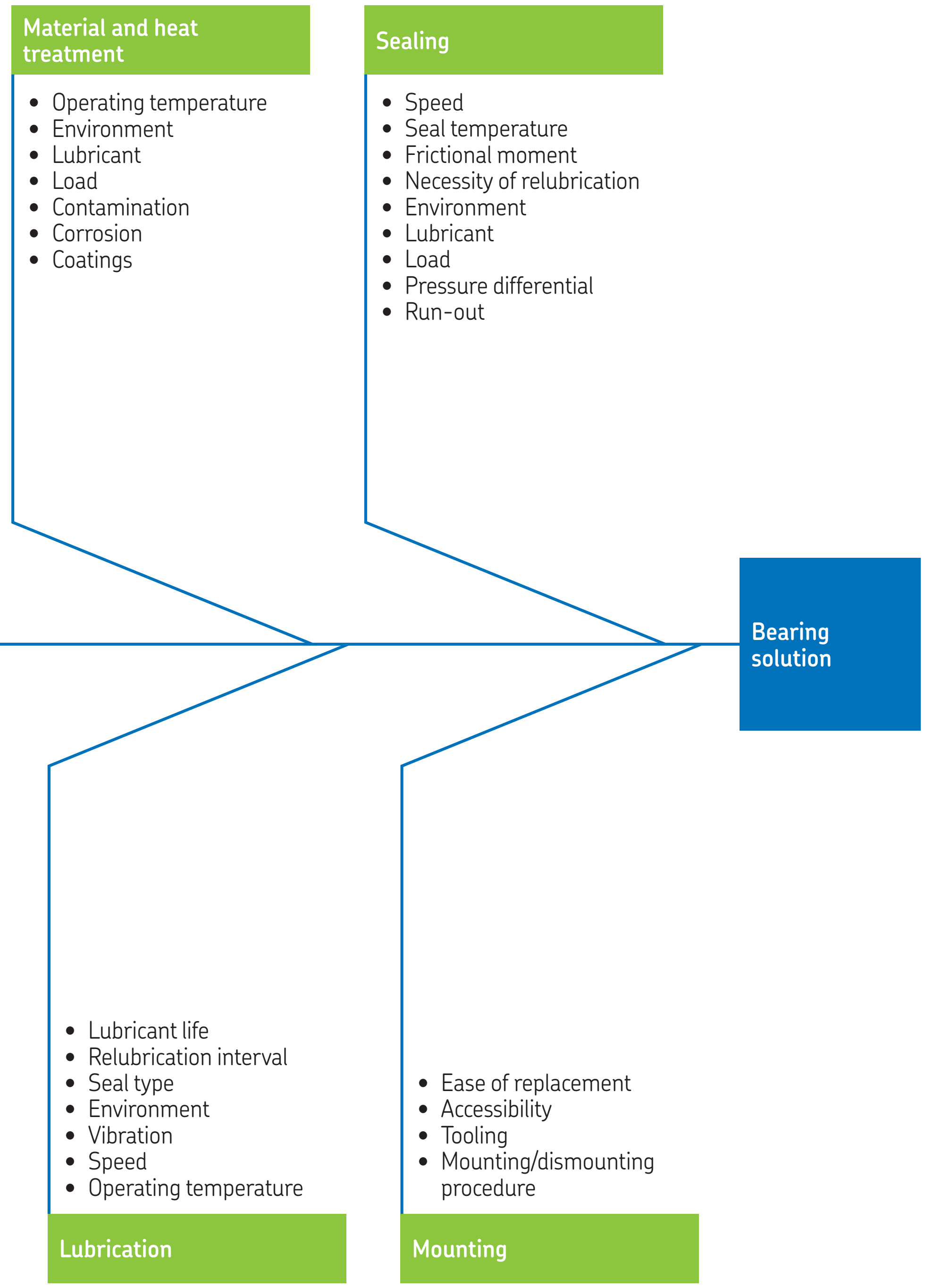
Fits

- Ease of mounting and replacement
- Speed
- Operating temperature
- Precision
- Shaft/housing material
- Load direction
- Load

Cage

- Lubricant
- Vibration
- Speed
- Operating temperature

requirements into a bearing solution



B.2 Bearing type and arrangement

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Each bearing type has characteristic properties that make it more or less suitable for use in a given application. An overview is provided in *Suitability of rolling bearings for industrial applications*, **page 72**, of the main bearing types (including their major features and design variants) and their degree of suitability for certain aspects of use.

This section provides information on what to consider when selecting a bearing arrangement and the types of bearing to use with it. It also provides guidelines on choosing bearing types to satisfy specific demands of an application, such as accommodating available space, loads, misalignment, and more.

Arrangements and their bearing types

A bearing arrangement supports and locates a shaft, radially and axially, relative to other components such as housings. Typically, two bearing supports are required to position a shaft. Depending on certain requirements, such as stiffness or load directions, a bearing support may consist of one or more bearings.

Bearing arrangements comprising two bearing supports are:

- locating/non-locating bearing arrangements
- adjusted bearing arrangements
- floating bearing arrangements

An overview is provided in *Suitability of rolling bearings for industrial applications*, **page 72**, of the suitability of various bearing types for different bearing arrangements.

A single bearing arrangement consists of just one bearing that supports radial, axial and moment loads.

Locating/non-locating bearing arrangements

In locating/non-locating bearing arrangements (**fig. 1**):

- The locating support provides axial location of the shaft relative to the housing.
- The non-locating support accommodates axial displacements that occur when thermal expansion of the shaft relative to the housing changes the distance between the two bearings. Additionally, it compensates for the accumulation of tolerances of the components, which affects the distance between the two bearings.

Bearings for the locating support

Radial bearings that can accommodate combined (radial and axial) loads are used for the locating bearing support. These include:

- deep groove ball bearings (**page 239**)
- two universally matchable single row angular contact ball bearings, arranged back-to-back or face-to-face (**page 386**)
- double row angular contact ball bearings (**page 386**)
- self-aligning ball bearings (**page 438**)
- spherical roller bearings (**page 774**)
- matched tapered roller bearings, arranged back-to-back or face-to-face (**page 670**)
- cylindrical roller bearings with flanges on both rings or cylindrical roller bearings mounted with an angle ring (thrust collar) (**page 494**)

Fig. 1

Locating/non-locating bearing arrangement

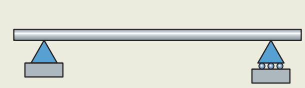
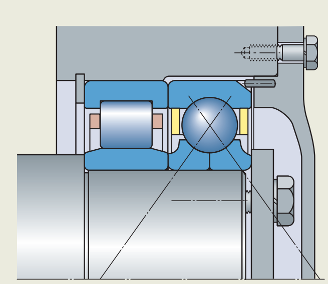


Fig. 2

Combination of cylindrical roller bearing and four-point contact ball bearing



Bearing combinations for the locating support

The locating bearing support can consist of a combination of bearings. For example (fig. 2):

- To accommodate the radial load, a cylindrical roller bearing that has one ring without flanges may be used.
- To provide the axial location, a deep groove ball bearing, a four-point contact ball bearing, or a pair of angular contact ball bearings may be used.

The outer ring of the axial locating bearing must be mounted radially free and should not be clamped. Otherwise, this bearing can be subjected to unintended radial loads.

Bearings for the non-locating support

There are two ways to accommodate axial displacements at the non-locating bearing support:

1 Use a bearing type that enables axial displacement within the bearing (fig. 3):
  - cylindrical roller bearings with flanges on one ring only (page 494)
  - needle roller bearings (page 582)
  - CARB toroidal roller bearings (page 842)

  When these bearings are rotating, they accommodate axial displacement and induce almost no axial load on the bearing arrangement. You should use this solution where an interference fit is required for both rings.

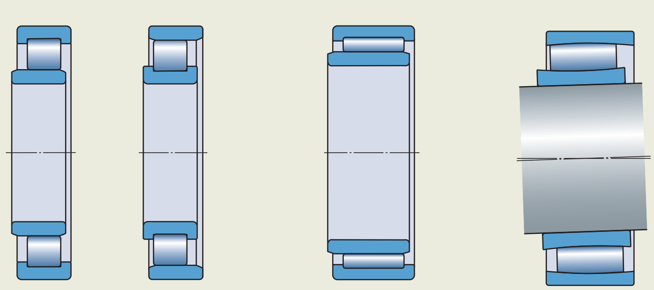
2 Use a loose fit between one bearing ring and its seat. Suitable bearing types include:
  - deep groove ball bearings (page 240)
  - self-aligning ball bearings (page 438)
  - spherical roller bearings (page 774)
  - pairs of angular contact ball bearings (page 385) or tapered roller bearings (page 670)

  Axial movements of a bearing on its seat cause axial loads, which might have an impact on the bearing service life.

When using other bearing types, you may need to take additional design considerations into account.

Fig. 3

Bearings that accommodate axial displacement



Cylindrical roller bearings (NU and N design)

Needle roller bearing

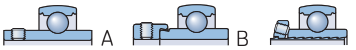
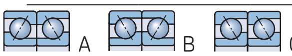
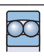
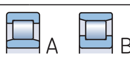
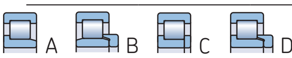
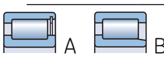
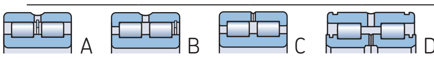
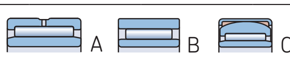
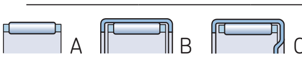
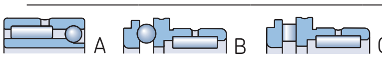
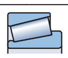
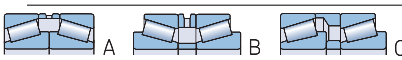
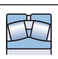
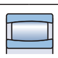
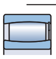
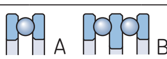
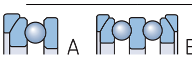
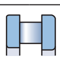
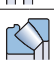
CARB toroidal roller bearing

Suitability of rolling bearings for industrial applications

Symbols

+++ excellent
++ good
+ fair
- poor
– – unsuitable

⇔ double direction
← single direction
□ non-locating displacement on the seat
■ non-locating displacement within the bearing
✓ yes
✗ no

Bearing type		Load carrying capability: Radial load	Load carrying capability: Axial load	Load carrying capability: Moment load	Misalignment: Static misalignment	Misalignment: Dynamic misalignment (few tenths of a degree)
Deep groove ball bearings	 A, B	+	+ ⇔	A –, B +	–	– –
Insert bearings	 A, B, C	+	+ ⇔	– –	++	– –
Angular contact ball bearings, single row		+[1]	++ ←	– –	–	– –
matched single row	 A, B, C	A, B ++ C ++[1]	A, B ++ ⇔ C ++ ←	A ++, B + C – –	A, C – –, B –	– –
double row	 A, B	++	++ ⇔	++	– –	– –
four-point contact		+[1]	++ ⇔	– –	– –	– –
Self-aligning ball bearings		+	–	– –	+++	+[2]
Cylindrical roller bearings, with cage	 A, B	++	– –	– –	–	– –
	 A, B, C, D	++	A, B + ← C, D + ⇔	– –	–	– –
full complement, single row	 A, B	+++	+ ←	– –	–	– –
full complement, double row	 A, B, C, D	+++	A – –, B + ← C + ⇔	– –	–	– –
Needle roller bearings, with steel rings	 A, B, C	++	– –	– –	A, B – C ++	– –
assemblies / drawn cups	 A, B, C	++	A, B – – C –	– –	–	– –
combined bearings	 A, B, C	++	A –, B + C ++	– –	– –	– –
Tapered roller bearings, single row		+++[1]	++ ←	– –	–	– –
matched single row	 A, B, C	A, B +++ C +++[1]	A, B ++ ⇔ C ++ ←	A +, B ++ C – –	A – B, C – –	– –
double row	 A, B	+++	++ ⇔	A + B ++	A –, B – –	– –
Spherical roller bearings		+++	+ ⇔	– –	+++	+[2]
CARB toroidal roller bearings, with cage		+++	– –	–	++	–
full complement		+++	– –	–	++	–
Thrust ball bearings	 A, B	– –	A + ← B + ⇔	– –	– –	– –
with sphered housing washer	 A, B	– –	A + ← B + ⇔	– –	++	– –
Cylindrical roller thrust bearings		– –	++ ←	– –	– –	– –
Needle roller thrust beairngs		– –	++ ←	– –	– –	– –
Spherical roller thrust bearings		+[1]	+++ ←	– –	+++	+[2]

[1] Provided the F_a/F_r ratio requirement is met [2] Reduced misalignment angle – contact SKF [3] Depending on cage and axial load level

Arrangement				Suitable for					Design features			
Locating	Non-locating	Adjusted	Floating	Long grease life	High speed	Low run-out	High stiffness	Low friction	Integral sealing	Separable ring mounting	Tapered bore	Standard housings and accessories available
⇔	◻	✗	✓	A +++ B ++	A +++ B +	A +++ B ++	+	+++	A ✓	✗	✗	✗
⇔	⇔	✗	✗	+++	++	A, B + C ++	+	++	✓	✗	✗	✓
✗	✗	✓	✗	++	++	+++	++	++	✓	✗	✗	✗
A, B ⇔ C ←	A, B ◻ C ✗	✗	✗	++	++	+++	++	++	✗	✗	✗	✗
↔	◻	✗	✗	++	++	++	++	++	A ✓	B ✓	✗	✗
⇔1)	– –	– –	– –	+	+++	++	++	++	✗	✓	✗	✗
⇔	◻	✗	✓	+++	++	++	+	+++	✓	✗	✓	✓
✗	■	✗	✗	++	+++	+++	++	+++	✗	✓	✗	✗
A, B ← C, D ⇔	A, B ■ ← C, D ✗	✗	A ✓ B, C, D ✗	++3)	+++	++	++	+++	✗	✓	✗	✗
←	A, B ←	✗	✓	–	+	+	+++	–	✗	A ✗ B ✓	✗	✗
B ← C, D ⇔	A ■ ⇔ B ■ ←	✗	✗	–	+	+	+++	–	D ✓	✗	✗	✗
✗	■ ⇔	✗	✗	++	++	+	++	+	A ✓	✓	✗	✗
A, B ✗ C ←	A, B ■ C ■ ←	✗	✗	++	++	+	++	+	B, C ✓	✓	✗	✗
←	✗	✓	✗	+	+	+	++	+	✗	✓	✗	✗
←	✗	✓	✗	+	++	+++	++	+	✗	✓	✗	✗
A, B ⇔ C ←	A, B ◻ C ✗	A, B ✗ C ✓	✗	+	+	++	+++	+	✗	✓	✗	✗
⇔	◻	✗	✗	+	+	++	+++	+	✓	✓	B ✓	✗
⇔	◻	✗	✓	+	++	+++	++	+	✓	✗	✓	✓
✗	■	✗	✗	+	++	+++	++	+	✗	✗	✓	✓
✗	■	✗	✗	–	+	+++	++	–	✓	✗	✓	✓
A ← B ⇔	✗	✗	✗	+	–	++	+	+	✗	✓	✗	✗
A ← B ⇔	✗	✗	✗	+	–	+	+	+	✗	✓	✗	✗
←	✗	✗	✗	–	–	+	+++	+	✗	✓	✗	✗
←	✗	✗	✗	–	–	+	+++	+	✗	✓	✗	✗
←	✗	✓	✗	–	+	+	+++	+	✗	✓	✗	✗

Typical combinations of bearing supports

From the large number of possible locating/non-locating bearing combinations, the following are the most popular.

For bearing arrangements where the axial displacement is accommodated within the bearing

Conventional bearing arrangements in which limited angular misalignment occurs include:

- deep groove ball bearing / cylindrical roller bearing (fig. 4)
- double row angular contact ball bearing / NU or N design cylindrical roller bearing (fig. 5)
- matched single row tapered roller bearings / NU or N design cylindrical roller bearing (fig. 6)
- NUP design cylindrical roller bearing / NU design cylindrical roller bearing (fig. 7)
- NU design cylindrical roller bearing and a four-point contact ball bearing / NU design cylindrical roller bearing (fig. 8)

SKF self-aligning bearing systems, which can compensate for more misalignment, are:

- spherical roller bearing / CARB toroidal roller bearing (fig. 9)
- self-aligning ball bearing / CARB toroidal roller bearing

For bearing arrangements where the axial displacement is accommodated between a bearing ring and its seat

- deep groove ball bearing / deep groove ball bearing (fig. 10)
- self-aligning ball bearings or spherical roller bearings (fig. 11) for both bearing positions
- matched single row angular contact ball bearings / deep groove ball bearing (fig. 12)

Fig. 4

Deep groove ball bearing / cylindrical roller bearing

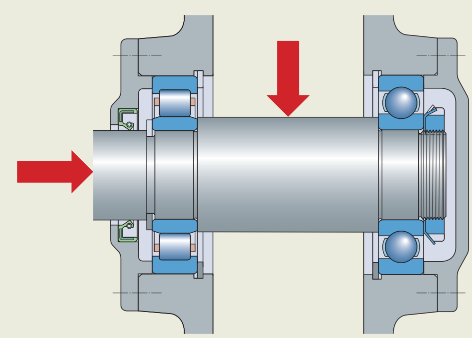


Fig. 5

Double row angular contact ball bearing / NU design cylindrical roller bearing

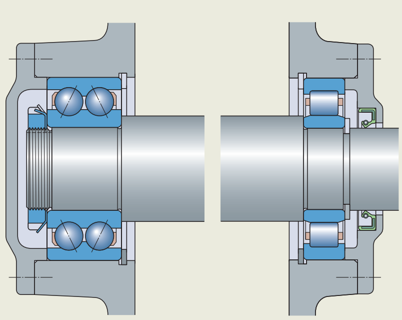


Fig. 6

Matched single row tapered roller bearings / NU design cylindrical roller bearing

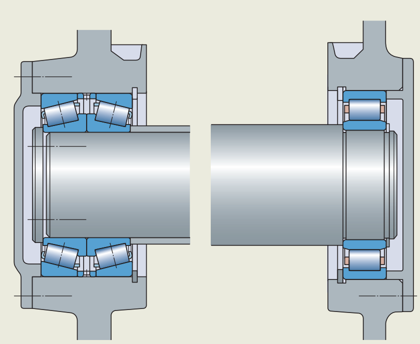


Fig. 7

NUP design cylindrical roller bearing / NU design cylindrical roller bearing

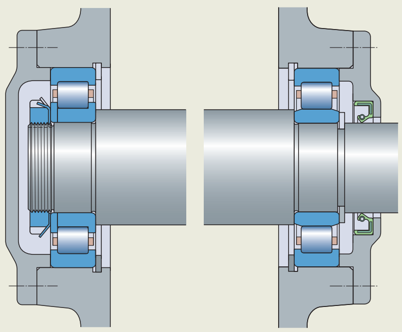


Fig. 8

NU design cylindrical roller bearing and a four-point contact ball bearing / NU design cylindrical roller bearing

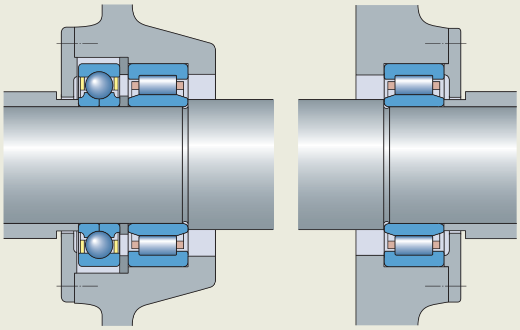


Fig. 9

Spherical roller bearing / CARB toroidal roller bearing

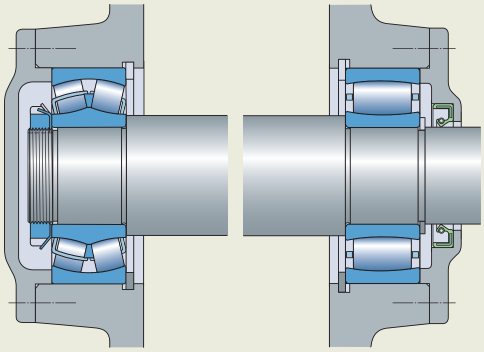


Fig. 10

Deep groove ball bearing / deep groove ball bearing

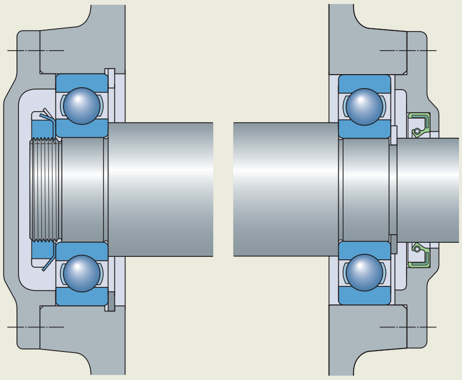


Fig. 11

Spherical roller bearing / spherical roller bearing

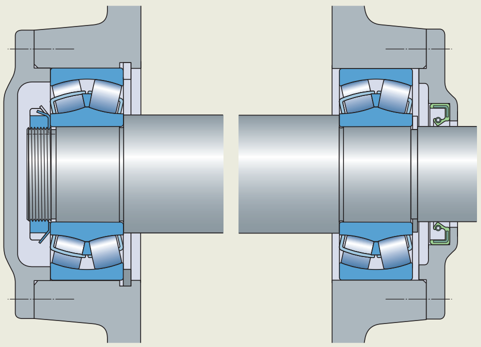
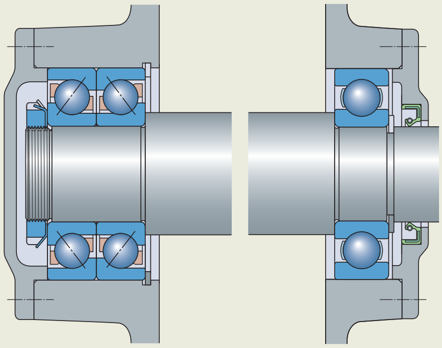


Fig. 12

Matched single row angular contact ball bearings / deep groove ball bearing



Adjusted bearing arrangements

In adjusted bearing arrangements, the shaft is located axially in one direction by one bearing support and in the opposite direction by the other (cross-located). Adjusted bearing arrangements require proper adjustment of clearance or preload during mounting.

These bearing arrangements are generally used for short shafts, where thermal expansion has only a little effect. The most suitable bearings are:

- angular contact ball bearings (fig. 13)
- tapered roller bearings (fig. 14)

Floating bearing arrangements

In floating bearing arrangements the shaft is cross-located, but is able to move axially a certain distance between the two end positions, i.e. “float”.

When determining the required “float” distance, consider thermal expansion of the shaft relative to the housing and tolerances of the components, which affect the distance between the two bearings.

With this arrangement, the shaft can also be axially located by other components on the shaft, e.g. a double helical gear. Most common bearings are:

- deep groove ball bearings (fig. 15)
- self-aligning ball bearings
- spherical roller bearings (fig. 16)
- NJ design cylindrical roller bearings, mirrored, with offset rings (fig. 17)

Fig. 13

Adjusted bearing arrangement, angular contact ball bearings arranged face-to-face

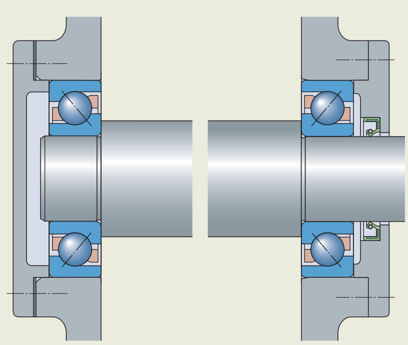


Fig. 14

Adjusted bearing arrangement, tapered roller bearings arranged back-to-back

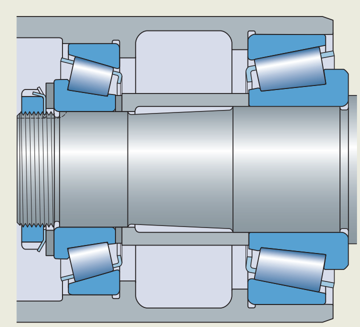


Fig. 15

Floating bearing arrangement, deep groove ball bearings

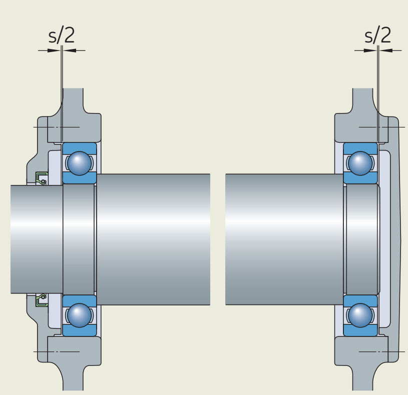
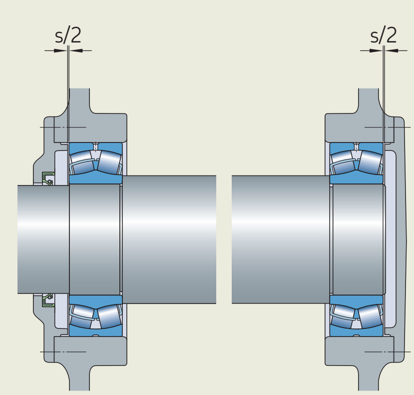


Fig. 16

Floating bearing arrangement, spherical roller bearings



Selection criteria

Available space

Often the boundary dimensions of a bearing are predetermined by the machine's design. Typically, the shaft diameter determines the bearing bore diameter. For the same bore diameter, different outside diameters and widths may be available (**fig. 18**). The availability of bearings in a certain ISO dimension series depends on bearing type and bore diameter.

Other space-related criteria that influence the selection of bearing type include:

- shafts with small diameter (approx. d < 10 mm)
  - deep groove ball bearings
  - needle roller bearings
  - self-aligning ball bearings
  - thrust ball bearings

- shafts with normal diameter
  - all bearing types

- very limited radial space
  - needle roller bearings
  - deep groove ball bearings in the 618 or 619 series
  - CARB toroidal roller bearings in the C49, C59 or C69 series
  - bearings without inner or outer ring and raceways machined directly on the shaft or in the housing

Fig. 17

Floating bearing arrangement, NJ design cylindrical roller bearings, mirrored, with offset rings

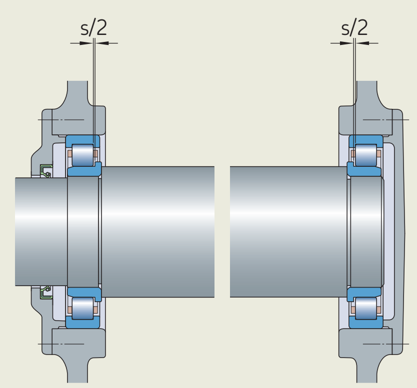
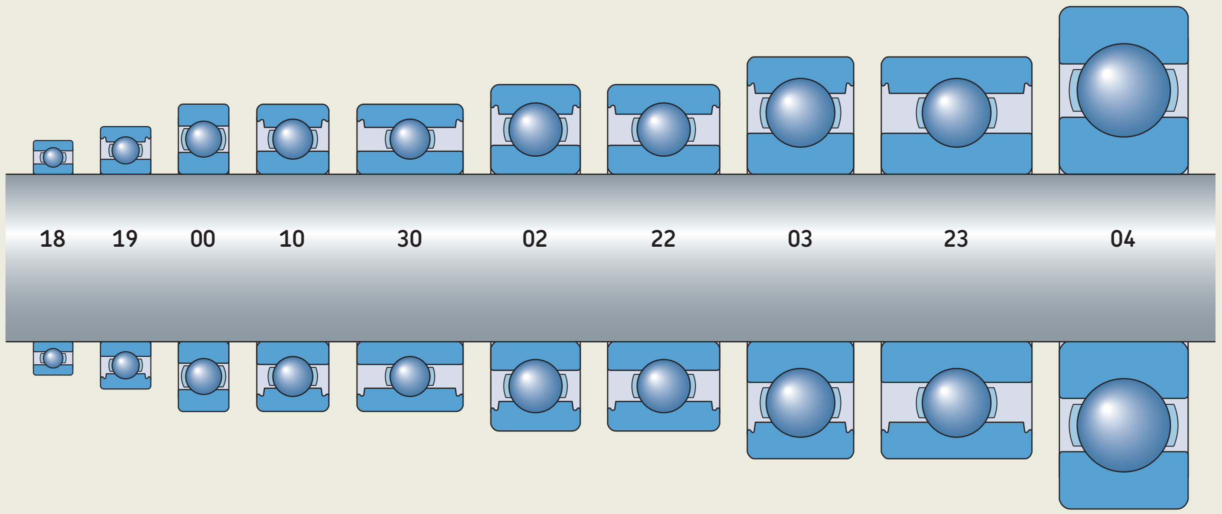


Fig. 18

ISO dimension series for the same bore diameter



Loads

When selecting bearing type based on load criteria, you should bear in mind that:

- Roller bearings accommodate heavier loads than same-sized ball bearings.
- Full complement bearings accommodate heavier loads than the corresponding bearing with a cage.

An overview is provided in *Suitability of rolling bearings for industrial applications*, page 72, the radial, axial and moment load capability of various bearing types.

Combined radial and axial loads

The direction of load is a primary factor in bearing type selection. Where the load on a bearing is a combination of radial and axial load, the ratio of the components determines the direction of the combined load (fig. 19).

The suitability of a bearing for a certain direction of load corresponds to its contact angle α (diagram 1) – the greater the contact angle, the higher the axial load carrying capacity of the bearing. You can see this indicated in the value of the calculation factor Y (refer to individual product sections), which decreases as the contact angle increases.

ISO defines bearings with contact angles ≤ 45° as radial bearings, and the others as thrust bearings, independent of their actual use.

To accommodate combined loads with a light axial component, bearings with a small contact angle can be used. Deep groove ball bearings are a common choice for light to moderate axial loads. With increasing axial load, a larger deep groove ball bearing (with higher axial load carrying capacity) can be used. For even higher axial load, bearings with a larger contact angle may be required, such as angular contact ball bearings or tapered roller bearings. These bearing types can be arranged in tandem to accommodate high axial loads.

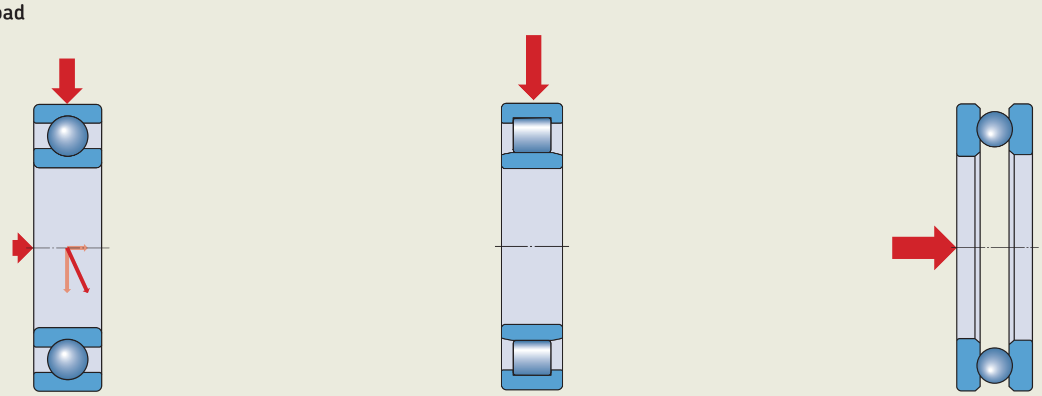
When combined loads have a large alternating axial load component, suitable solutions include:

- a pair of universally matchable angular contact ball bearings
- matched sets of tapered roller bearings
- double-row tapered roller bearings

Where a four-point contact ball bearing is used to accommodate the axial component of a combined load (fig. 2, page 70), the bearing outer ring must be mounted radially free and should not be clamped axially. Otherwise, the bearing may be subjected to unintended radial load.

Fig. 19

Direction of load



Combined load
The resulting load direction is determined by the ratio of radial to axial load.

Example: Deep groove ball bearing

Pure radial load
Load direction 0°

Example: NU design cylindrical roller bearing (accommodates radial load only)

Pure axial load
Load direction 90°

Example: Thrust ball bearing (accommodates axial load only)

Speed and friction

The permissible operating temperature of rolling bearings imposes limits on the speed at which they can be operated. The operating temperature is determined, to a great extent, on the frictional heat generated in the bearing, except in machines where process heat is dominant.

An overview is provided in *Suitability of rolling bearings for industrial applications*, page 72, of the speed capability of various bearing types.

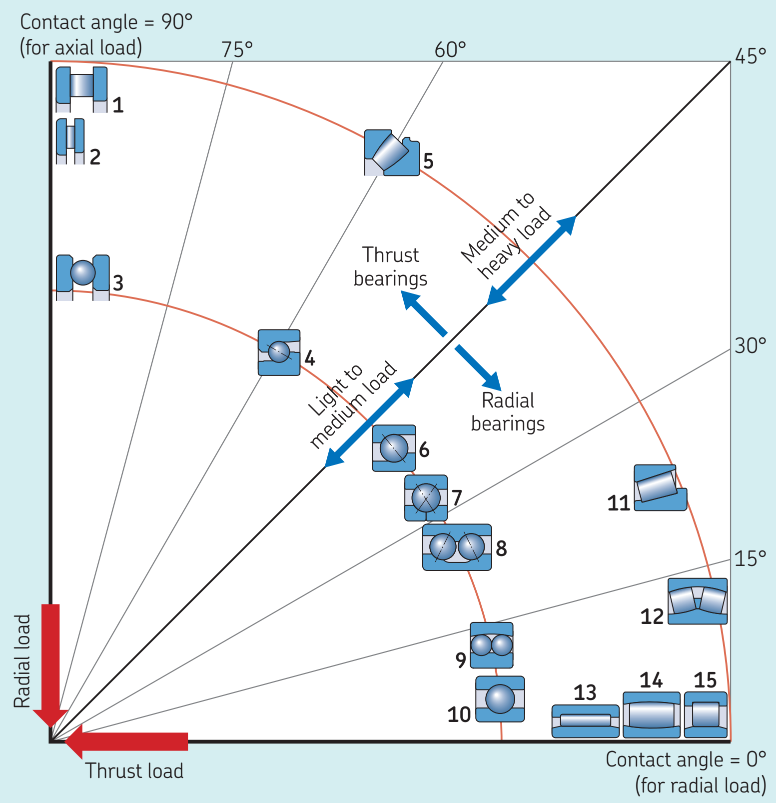
When selecting bearing type on the basis of operating speed, you should consider the following:

- Ball bearings have a lower frictional moment than same-sized roller bearings.
- Thrust bearings cannot accommodate speeds as high as same-sized radial bearings.
- Single row bearing types typically generate low frictional heat and are therefore more suitable for high-speed operation than double or multi-row bearings.
- Bearings with rolling elements made of ceramics (hybrid bearings) accommodate higher speeds than their all-steel equivalents.

Diagram 1

Contact angles of various bearing types

1 Cylindrical roller thrust bearing
2 Needle roller thrust bearing
3 Thrust ball bearing
4 Angular contact thrust ball bearing
5 Spherical roller thrust bearing
6 Single row angular contact ball bearing
7 Four-point contact ball bearing
8 Double row angular contact ball bearing
9 Self-aligning ball bearing
10 Deep groove ball bearing
The contact angle depends on load and clearance.
11 Tapered roller bearing
12 Spherical roller bearing
13 Needle roller bearing
14 CARB toroidal roller bearing
15 Cylindrical roller bearings



Misalignment

An overview is provided in *Suitability of rolling bearings for industrial applications*, page 72, of the capability of various bearing types to accommodate misalignment. The different types of misalignment are explained in table 1.

Bearing types vary in their ability to compensate for misalignment between the shaft and housing:

- **Self-aligning bearings (fig. 20)**
  Self-aligning bearings can compensate for misalignment within the bearing. Values for the permissible misalignment are listed in the relevant product section.

- **Alignment bearings (fig. 21)**
  Alignment bearings can accommodate initial static misalignment because of their sphered outside surface. Values for the permissible misalignment are listed in the relevant product section.

- **Rigid bearings**
  Rigid bearings (deep groove ball bearings, angular contact ball bearings, cylindrical, needle and tapered roller bearings) accommodate misalignment within the limits of their internal clearance. Values for the permissible misalignment are listed in the relevant product section. For rigid bearings, any misalignment may reduce service life.

Table 1

Types of misalignment

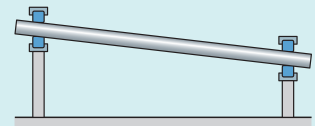
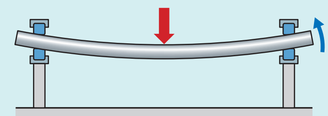
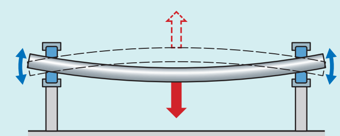
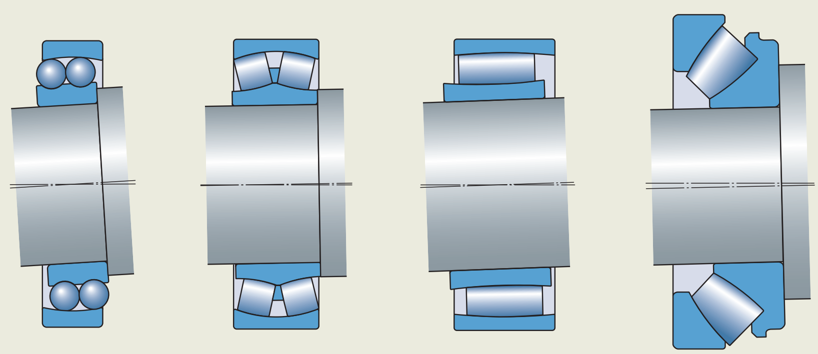
Static misalignment	There is an initial alignment error between the two supports of a shaft.	
	Shaft deflection creates misalignment between bearing inner and outer rings that is constant in magnitude and direction.	
Dynamic misalignment	Varying shaft deflection creates misalignment between bearing inner and outer rings that is continuously changing in magnitude or direction.	

Fig. 20

Self-aligning bearings



Self-aligning ball bearing | Spherical roller bearing | CARB toroidal roller bearing | Spherical roller thrust bearing

Temperature

The permissible operating temperature of rolling bearings can be limited by:

- the dimensional stability of the bearing rings and rolling elements (table 2, for details refer to the relevant product section)
- the cage (*Cages*, page 187)
- the seals (relevant product section)
- the lubricant (*Lubrication*, page 110)

Precision

Precision requirements typically do not influence bearing type selection. Most SKF bearings are available in various tolerance classes. Details are provided in the product sections.

For very high precision requirements, e.g. machine tool applications, use SKF super-precision bearings (SKF catalogue *Super-precision bearings* or available at skf.com/super-precision).

Stiffness

The stiffness of a rolling bearing is characterized by the magnitude of the elastic deformation in the bearing under load and depends not only on bearing type, but also on bearing size and operating clearance.

When selecting bearing type on the basis of stiffness requirements, you should consider, for bearings with the same size, that:

- stiffness is higher for roller than for ball bearings
- stiffness is higher for full complement bearings than for the corresponding bearing with a cage
- stiffness is higher for hybrid bearings than for the corresponding all-steel bearing
- stiffness can be enhanced by applying a preload (*Selecting preload*, **page 186**)

Table 2

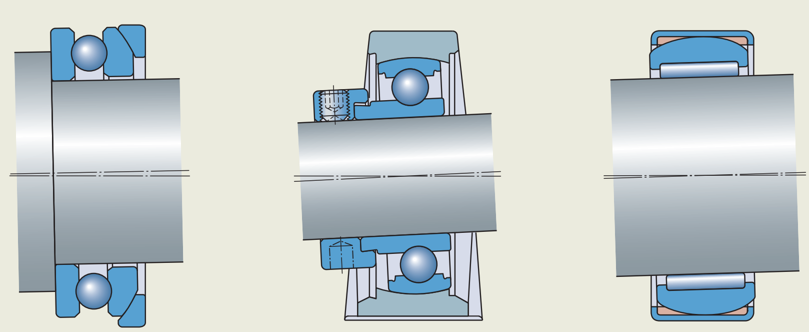
Stabilization of SKF rolling bearings

			Stabilized for temperatures ≤ 120 °C *(250 °F)*	≤ 150 °C *(300 °F)*	≤ 200 °C *(390 °F)*
Ball bearings	**Radial**	Deep groove ball bearings	•	–	–
		Angular contact ball bearings	•	•	–
		Four-point contact ball bearings	•	•	–
		Self-aligning ball bearings	•	◦	–
	Thrust	Thrust ball bearings	•	• [1]	–
Roller bearings	**Radial**	Cylindrical roller bearings	•	•	–
		Needle roller bearings	•	–	–
		Tapered roller bearings	•	•	–
		Spherical roller bearings	•	•	•
		CARB toroidal roller bearings	•	•	•
	Thrust	Cylindrical roller thrust bearings	•	–	–
		Needle roller thrust bearings	•	–	–
		Spherical roller thrust bearings	•	•	•

• Available as standard
◦ Check availability with SKF, check cage material
– Check with SKF
[1] Not for all sizes.

Fig. 21

Alignment bearings



Thrust ball bearing with sphered housing washer · Ball bearing unit · Alignment needle roller bearing

Mounting and dismounting

When selecting bearing type, you should consider the mounting and dismounting requirements:

- Is it required or beneficial to mount the inner and outer ring independently?
  - Select a separable bearing.
- Is it required or beneficial to mount the bearing on a tapered seat or with a tapered sleeve?
  - Select a bearing with a tapered bore.
  - Consider using SKF ConCentra ball or roller bearing units (skf.com/ball-bearing-units and skf.com/roller-bearing-units).

Separable bearings

Separable bearings are easier to mount and dismount, particularly if interference fits are required for both rings.

For separable bearing types, refer to *Suitability of rolling bearings for industrial applications*, page 72.

Tapered bore

Bearings with a tapered bore can be mounted on a tapered shaft seat or mounted on a cylindrical shaft seat using an adapter or withdrawal sleeve (fig. 22). For bearing types available with tapered bore, refer to *Suitability of rolling bearings for industrial applications*, page 72.

Integral sealing

There are two reasons for sealing bearings or bearing arrangements:

- keeping the lubricant in the bearing, and avoiding pollution of adjacent components
- protecting the bearing from contamination, and prolonging bearing service life

Capped bearings (sealed bearings or bearings with shields) can provide cost-effective and space-saving solutions for many applications. Bearing types, for which integral sealing is available, are indicated in *Suitability of rolling bearings for industrial applications*, page 72.

Cost and availability

Popular items

After determining your required bearing type, you may find it beneficial to select an appropriate bearing from our assortment of popular items, because they have a high level of availability and generally provide a cost-effective solution. Popular items are marked in the product tables with the symbol ►.

Large bearings

If a required bearing has an outside diameter D ≥ 420 mm, and is not marked as popular, then check its availability with SKF.

Capped bearings

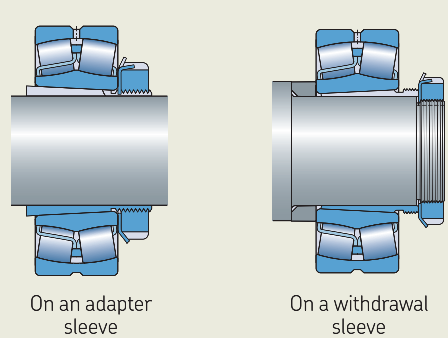
Capped (sealed bearings or bearings with shields) typically provide more cost-effective solutions than using external sealing. In addition to providing good sealing performance, these ready-greased bearings do not require initial grease fill.

Availability of standard housings and sleeves

Using standard housings and sleeves generally leads to more cost-effective bearing arrangements. Bearing types for which these standard components are available are indicated in *Suitability of rolling bearings for industrial applications*, page 72.

Fig. 22

Bearings with tapered bore on sleeves



B.3 Bearing size

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B.3 Bearing size

The size of a bearing must be sufficient to ensure that it is strong enough to deliver the required/expected life under defined operating conditions.

A bearing can be viewed as a system of components: raceways, rolling elements, cage, seals (if present) and lubricant (**fig. 1**). The performance of each component contributes to or determines the performance and life of the bearing (**diagram 1**). Consider these aspects:

- rolling contact fatigue (RCF) on the rolling elements and raceways – this is the primary aspect that dictates bearing life in most applications
- permanent deformation of rolling elements and raceways because of heavy loads acting on the bearing, while it is stationary or oscillating slowly, or high peak loads acting on the bearing while it is rotating
- cage type or cage material – these may limit the operating speed or the permissible acceleration or temperature[1)]
- speed limit of contacting seal lips – this can determine the maximum allowable speed, which affects operating temperature, thereby affecting life
- lubricant life – when the lubricant deteriorates, the resulting poor relubrication condition quickly reduces bearing life

The operating conditions of the application determine which of these factors most influence the performance and life of the bearing.

This section provides guidance on determining the required bearing size.

The effect of RCF or permanent deformation on rolling elements and raceways is directly related to bearing size. Effects of cage type and material are not related to bearing size. In capped bearings, the effects of the lubricant and integral seal are only indirectly related to bearing size.

Therefore, the two main criteria that can be used for determining appropriate bearing size are:

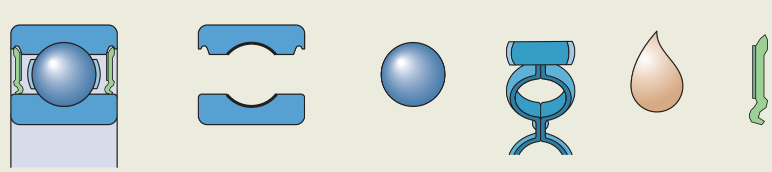
- **Size selection based on rating life,** **page 88**
  This is based on the required bearing life, taking into account the possible effects of rolling contact fatigue, and requires calculation of the basic rating life, L_{10}, or SKF rating life, L_{10m}, for the bearing.

- **Size selection based on static load,** **page 104**
  This is based on the static load that the bearing can accommodate, taking into account the possible effects of permanent deformation, and requires calculation of the static safety factor, s_0, for the bearing.

Fig. 1

Bearing system life

$L_{bearing} = f\,(L_{raceways}, L_{rolling\ elements}, L_{cage}, L_{lubricant}, L_{seals})$



1) Special cage executions are often available for bearing types that are commonly used in applications where such challenging conditions are present.

These selection criteria and the related bearing ratings and static safety factor are shown in **diagram 2** and are described in detail in the relevant subsections.

Which selection criteria you should use depends on the operating conditions of the bearing:

- For applications where bearings are running in typical operating conditions – i.e. normal speeds, good lubrication conditions and not highly or peak loaded – use *Size selection based on rating life*, **page 88**.
- For applications where bearings are running under very low speeds or which are used under stationary conditions, very bad lubrication conditions or where occasional peak loads occur, use *Size selection based on static load*, **page 104**.

Note that there are applications where both selection criteria must be considered, for example where a duty cycle has occasional peak loads. Also, in applications where the bearing is lightly loaded, the minimum load requirement (*Requisite minimum load*, **page 106**) must also be considered.

After determining bearing size, and before going to the next step, check the items listed in *Checklist after the bearing size is determined*, **page 106**.

Other attributes of the bearing components, such as strength and suitability, are addressed elsewhere in the *Bearing selection process*, including *Lubrication*, **page 110**, and *Bearing execution*, **page 182**, as well as in the product sections. Consider these attributes, in addition to bearing size, to ensure you obtain best bearing performance.

Diagram 1

Performance and related bearing system components

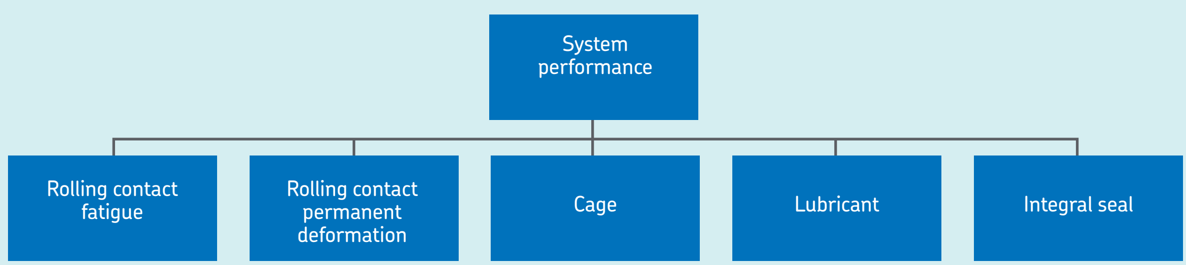
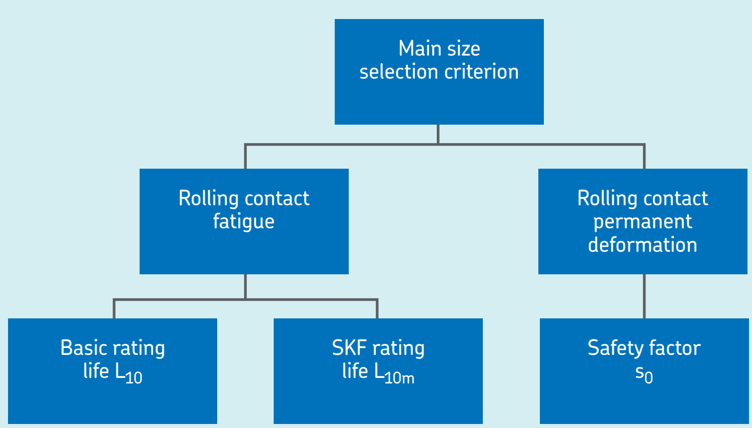


Diagram 2

Main selection criteria for bearing size and related bearing ratings and safety factor



Size selection based on rating life

For applications where bearings are running in typical operating conditions – i.e. normal speeds, good lubrication conditions and not highly or peak loaded – determine the appropriate bearing size based on the required bearing life, taking into account the possible effects of rolling contact fatigue (RCF).

This subsection describes the bearing rating life equations and the factors that must be determined to make the evaluation:

- *Bearing rating life* – the basis for bearing rating life, showing how to calculate basic rating life, L_{10}, and SKF rating life, L_{10m}
- *Basic dynamic load rating, C*, **page 91**
- *Equivalent dynamic bearing load, P*, **page 91**
- *Life modification factor,* a_{SKF}, **page 94**
- *Lubrication condition – the viscosity ratio,* κ, **page 102**
- *Fatigue load limit,* P_u, **page 104**
- *Contamination factor,* η_c, **page 104**

Bearing rating life

For estimating the expected bearing life, you can either use basic rating life, L_{10}, or SKF rating life, L_{10m}.

If you have experience with the operating conditions related to lubrication and contamination, and know that the conditions you are working with do not have a dramatic effect on the life of your bearings, use the basic rating life calculation, otherwise SKF recommends using the SKF rating life.

Bearing life definition

Bearing life is defined as the number of revolutions (or the number of operating hours) at a given speed that the bearing is capable of enduring before the first sign of metal fatigue (spalling) occurs on a rolling element or the raceway of the inner or outer ring.

Tests on seemingly identical bearings, under identical operating conditions, result in a large variation in the number of cycles, or time, needed to cause metal fatigue. Therefore, bearing life estimates based on rolling contact fatigue (RCF) are insufficiently accurate and so a statistical approach is needed to determine bearing size.

The basic rating life, L_{10}, is the fatigue life that 90% of a sufficiently large group of apparently identical bearings, operating under identical operating conditions, can be expected to attain or exceed.

To determine a relevant bearing size using the definition given here, compare the calculated rating life against the service life expectations of the bearing application, using experience from previous dimensioning where available. Otherwise, use the guidelines regarding specification life of various bearing applications provided in **table 1** and **table 2**.

Because of the statistical spread of bearing fatigue life, an observed time to failure for an individual bearing can be evaluated in relation to its rated life, only if the failure probability of that particular bearing is determined in relation to the general population of bearings running under similar conditions.

Numerous investigations on bearing failure, in a variety of applications, have confirmed that design guidelines based on 90% reliability, and use of dynamic safety factors, lead to robust bearing solutions in which typical fatigue failures are avoided.

Table 1

Guideline values of specification life for different machine types

Machine type	**Specification life** Operating hours
Household machines, agricultural machines, instruments, technical equipment for medical use	300 ... 3 000
Machines used for short periods or intermittently: electric hand tools, lifting tackle in workshops, construction equipment and machines	3000 ... 8 000
Machines used for short periods or intermittently where high operational reliability is required: lifts (elevators), cranes for packaged goods or slings of drums, etc.	8 000 ... 12 000
Machines for use 8 hours a day, but not always fully utilized: gear drives for general purposes, electric motors for industrial use, rotary crushers	10 000 ... 25 000
Machines for use 8 hours a day and fully utilized: machine tools, woodworking machines, machines for the engineering industry, cranes for bulk materials, ventilator fans, conveyor belts, printing equipment, separators and centrifuges	20 000 ... 30 000
Machines for continuous 24-hour use: rolling mill gear units, medium-sized electrical machinery, compressors, mine hoists, pumps, textile machinery	40 000 ... 50 000
Wind energy machinery, this includes main shaft, yaw, pitching gearbox, generator bearings	30 000 ... 100 000
Water works machinery, rotary furnaces, cable stranding machines, propulsion machinery for ocean-going vessels	60 000 ... 100 000
Large electric machines, power generation plant, mine pumps, mine ventilator fans, tunnel shaft bearings for ocean-going vessels	100 000 ... 200 000

Basic rating life

If you consider only the load and speed, you can use the basic rating life, L_{10}.

The basic rating life of a bearing in accordance with ISO 281 is

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

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

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

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

SKF rating life

For modern high-quality bearings, the calculated basic rating life can deviate significantly from the actual service life in a given application. Service life in a particular application depends not only on load and bearing size, but also on a variety of influencing factors including lubrication, degree of contamination, proper mounting and other environmental conditions.

ISO 281 uses a modified life factor to supplement the basic rating life. The life modification factor a_{SKF} applies the same concept of a fatigue load limit P_u (*Fatigue load limit, P_u*, **page 104**) as used in ISO 281. Values of P_u are listed in the product tables. Just as in ISO 281, to reflect three of the important operating conditions, the life modification factor a_{SKF} takes the lubrication conditions (*Lubrication condition – the viscosity ratio, κ*, **page 102**), the load level in relation to the bearing fatigue load limit, and a factor η_c for the contamination level (*Contamination factor, η_c*, **page 104**) into consideration using

$$L_{nm} = a_1\ a_{SKF}\ L_{10} = a_1\ a_{SKF} \left(\frac{C}{P}\right)^p$$

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

$$L_{nmh} = \left(\frac{10^6}{60\ n}\right) L_{nm}$$

where
L_{nm} = SKF rating life (at 100 – n[1])% reliability) [millions of revolutions]
L_{nmh} = SKF rating life (at 100 – n[1])% reliability) [operating hours]
L_{10} = basic rating life (at 90% reliability) [millions of revolutions]
a_1 = life adjustment factor for reliability (**table 3**, **page 90**, values in accordance with ISO 281)
a_{SKF} = SKF life modification factor
C = basic dynamic load rating [kN]
P = equivalent dynamic bearing load [kN]
n = rotational speed [r/min]
p = exponent of the life equation
= 3 for ball bearings
= 10/3 for roller bearings

For 90% reliability:
L_{nm} = SKF rating life (at 100 – n[1])% reliability) [million revolutions]

Becomes:
L_{10m} = SKF rating life [million revolutions]

Since the life adjustment factor a_1 is related to fatigue, it is less relevant for load levels P below the fatigue load limit P_u. Dimensioning with life adjustment factors reflecting very high reliability (such as 99%) will result in large bearings for given loads. In these cases, the bearing load must be checked against the minimum load requirement for the bearing. Calculating minimum load is described in *Requisite minimum load*, **page 106**.

Commonly used conversion factors for bearing life in units other than million revolutions are provided in **table 4**, **page 91**.

Table 2

Guideline values of specification life for axlebox bearings and units for railway vehicles

Type of vehicle	Specification life Million kilometres
Freight wagons to UIC specification based on continuously acting maximum axle load	0,8
Mass transit vehicles: suburban trains, underground carriages, light rail and tramway vehicles	1,5
Main line passenger coaches	3
Main line diesel and electric multiple units	3 … 4
Main line diesel and electric locomotives	3 … 5

1) The factor n represents the failure probability, which is the difference between the requisite reliability and 100%.

B.3 Bearing size

Calculating bearing life with variable operating conditions, fluctuating load

In some applications – for example, industrial gearboxes, vehicle transmissions or windmills – the operating conditions, such as the magnitude and direction of loads, speeds, temperatures and lubrication conditions, are continually changing. In these types of applications, bearing life cannot be calculated without first reducing the load spectrum or duty cycle of the application to a limited number of simplified load cases (**diagram 3**).

For continuously changing loads, each different load level can be accumulated and the load spectrum reduced to a histogram plotting constant-load blocks. Each block should characterize a given percentage or time-fraction during operation. 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 of relatively short duration.

Within each duty interval, the bearing load and operating conditions can be averaged to a representative, 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

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

where

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

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

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

The use of this calculation method is well suited for application conditions of varying load level and varying speed with known time fractions.

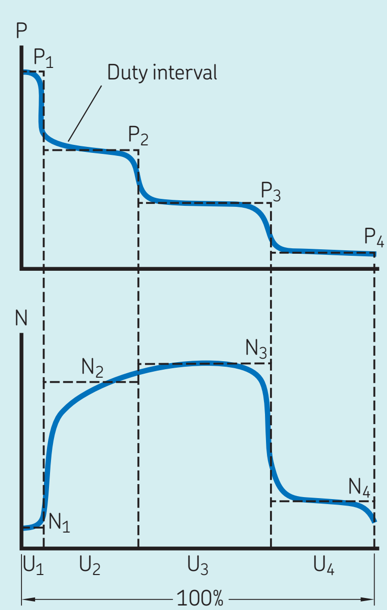
Table 3

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 3

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



Basic dynamic load rating, C

The basic dynamic load rating C is used for calculating basic rating life and SKF rating life for bearings that rotate under load. The C value is defined as: the bearing load that will result in an ISO 281 basic rating life of 1 000 000 revolutions. It is assumed that the load is constant in magnitude and direction and is radial for radial bearings and axial, centrically acting, for thrust bearings.

The basic dynamic load ratings for SKF bearings are determined in accordance with the procedures outlined in ISO 281, and apply to bearings made of chromium bearing steel, heat treated to a minimum hardness of 58 HRC, operating under normal conditions.

Dynamic load rating for SKF Explorer bearings

SKF Explorer bearings have undergone design, material and manufacturing improvements that require adjusted factors to calculate the dynamic load ratings in accordance with ISO 281. The SKF Explorer adjusted dynamic load ratings, which are higher than the ratings for SKF basic design bearings, are verified by extensive endurance testing.

To fully utilize the improved performance of SKF Explorer bearings, the SKF rating life calculation including the life modification factor a_{SKF} is recommended. In fact, it is the modified rating life of the bearing, L_{10m}, rather than the dynamic load rating, C, that provides the most valuable information regarding the endurance performance of a bearing. For detailed information, refer to *Life modification factor, a_{SKF}*, **page 94**.

Equivalent dynamic bearing load, P

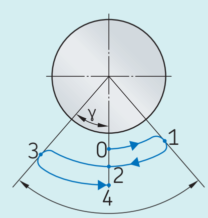
When calculating the bearing rating life, a value for equivalent dynamic bearing load is required for both basic bearing life and SKF bearing life equations.

The loads acting on a bearing are calculated according to the laws of mechanics using the external forces – such as forces from power transmission, work forces, gravitational or inertial forces – that are known or can be calculated.

In real-world circumstances, the loads acting on a bearing may not be constant, can act both radially and axially, and are subject to other factors that require the load calculations to be modified or, in some cases, simplified.

Table 4

Unit conversion factors for bearing life



The complete oscillation = 4 γ
(= from point 0 to point 4)

Basic units	**Conversion factor** Million revolutions	Operating hours	Million kilometres	Million oscillation cycles[1)]
1 million revolutions	1	$\frac{10^6}{60\ n}$	$\frac{\pi\ D}{10^3}$	$\frac{180}{2\ \gamma}$
1 operating hour	$\frac{60\ n}{10^6}$	1	$\frac{60\ n\ \pi\ D}{10^9}$	$\frac{180 \times 60\ n}{2\ \gamma\ 10^6}$
1 million kilometres	$\frac{10^3}{\pi\ D}$	$\frac{10^9}{60\ n\ \pi\ D}$	1	$\frac{180 \times 10^3}{2\ \gamma\ \pi\ D}$
1 million oscillation cycles[1)]	$\frac{2\ \gamma}{180}$	$\frac{2\ \gamma\ 10^6}{180 \times 60\ n}$	$\frac{2\ \gamma\ \pi\ D}{180 \times 10^3}$	1

D = vehicle wheel diameter [m]
n = rotational speed [r/min]
γ = oscillation amplitude (angle of max. deviation from centre position) [°]

1) Not valid for small amplitudes (γ < 10°).

Calculating equivalent dynamic bearing load

The load value, P, used in the bearing rating life equations is the equivalent dynamic bearing load. The equivalent dynamic bearing load is defined as: a hypothetical load, constant in magnitude and direction, that acts radially on radial bearings and axially and centrically on thrust bearings.

This hypothetical load, when applied, would have the same influence on bearing life as the actual loads to which the bearing is subjected (fig. 2).

If a bearing is loaded with simultaneously acting radial load F_r and axial load F_a that are constant in magnitude and direction, the equivalent dynamic bearing load P can be obtained from the general equation.

$$P = X F_r + Y F_a$$

where
- P = equivalent dynamic bearing load [kN]
- F_r = actual radial bearing load [kN]
- F_a = actual axial bearing load [kN]
- X = radial load factor for the bearing
- Y = axial load factor for the bearing

An axial load only influences the equivalent dynamic load P for a single row radial bearing if the ratio F_a/F_r exceeds a certain limiting factor e. With double row bearings, even light axial loads influence the equivalent load and have to be considered.

The same general equation also applies to spherical roller thrust bearings, which can accommodate both axial and radial loads.

Certain thrust bearings, such as thrust ball bearings and cylindrical and needle roller thrust bearings, can only accommodate pure axial loads. For these bearings, provided the load acts centrically, the equation is simplified to

$$P = F_a$$

Information and data required for calculating the equivalent dynamic bearing load for the different bearing types is provided in the relevant product sections.

Equivalent mean load

Other loads may vary with time. For these situations, an equivalent mean load must be calculated.

Mean load within a duty interval

Within each loading interval, the operating conditions can vary slightly from the nominal value. Assuming that the operating conditions, such as speed and load direction, are fairly constant and the magnitude of the load constantly varies between a minimum value F_{min} and a maximum value F_{max} (diagram 4), the mean load can be calculated using

$$F_m = \frac{F_{min} + 2F_{max}}{3}$$

Rotating load

If, as illustrated in diagram 5, the load on the bearing consists of a load F_1 which is constant in magnitude and direction, such as the weight of a rotor, and a rotating constant load F_2 such as an unbalanced load, the mean load can be calculated using

$$F_m = f_m (F_1 + F_2)$$

Values for the factor f_m are provided in diagram 6.

Peak load

High loads acting for short times (diagram 7) may not influence the mean load used in a fatigue life calculation. Evaluate such peak loads against the bearing static load rating C_0, using a suitable static safety factor s_0 (*Size selection based on static load*, page 104).

Fig. 2

Equivalent dynamic bearing load

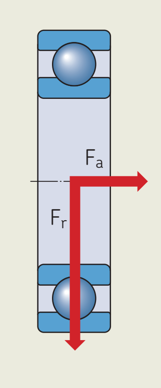


Diagram 4

Load averaging

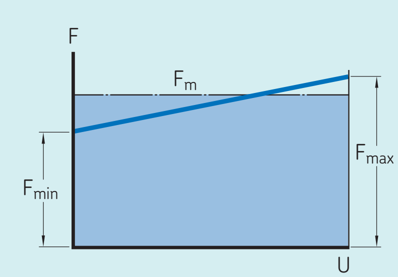
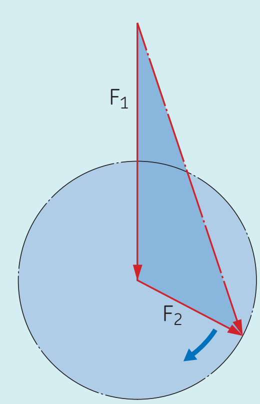


Diagram 5

Rotating load



SKF

Considerations when calculating equivalent dynamic bearing load

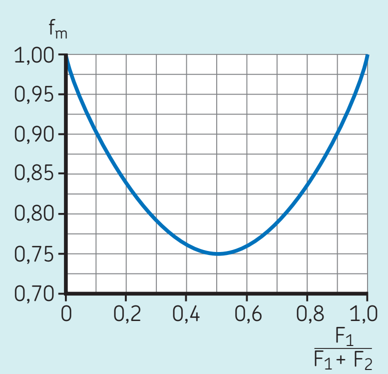
For the sake of simplification, when calculating the load components for bearings supporting a shaft, the shaft is considered as a statically determined beam resting on rigid, moment-free supports. Elastic deformations in the bearing, the housing or the machine frame are not considered, nor are the moments produced in the bearing as a result of shaft deflection. These simplifications are necessary if you are making bearing arrangement calculations without the aid of relevant computer software. The standardized methods for calculating basic load ratings and equivalent bearing loads are based on similar assumptions.

It is possible to calculate bearing loads based on the theory of elasticity, without making the above assumptions, but this requires the use of complex computer programs (SKF SimPro Quick and SKF SimPro Expert). In these programs, the bearings, shaft and housing are considered as resilient components of a system.

If external forces and loads – such as inertial forces or loads resulting from the weight of a shaft and its components – are not known, they can be calculated. However, when determining work forces and loads – such as rolling forces, moment loads, unbalanced loads and impact loads – it may be necessary to rely on estimates based on experience with similar machines or bearing arrangements.

Diagram 6

Rotating load



Geared transmissions

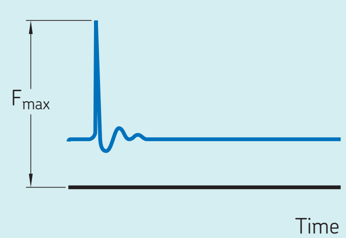
With geared transmissions, the theoretical tooth forces can be calculated from the power transmitted and the design characteristics of the gear teeth. However, there are additional dynamic forces, produced either by the gear, or by the input or output shaft. Additional dynamic forces from gears can be the result of pitch or form errors of the teeth and from unbalanced rotating components. Gears produced to a high level of accuracy have negligible additional forces. For lower precision gears, use the following gear load factors:

- pitch and form errors < 0,02 mm: 1,05 to 1,1
- pitch and form errors 0,02 to 0,1 mm: 1,1 to 1,3

Additional forces arising from the type and mode of operation of the machines that are coupled to the transmission can only be determined when the operating conditions, the inertia of the drive line and the behaviour of couplings or other connectors are known. Their influence on the rating lives of the bearings is included by using an "operation" factor that takes into account the dynamic effects of the system.

Diagram 7

Short time peak load



Belt drives

When calculating bearing loads for belt driven applications, "belt pull" must be taken into consideration. Belt pull, which is a circumferential load, depends on the amount of torque being transmitted. The belt pull must be multiplied by a factor whose value depends on the type of belt, belt tension and any additional dynamic forces. Belt manufacturers usually publish the values. However, should information not be available, the following can be used:

- toothed belts = 1,1 to 1,3
- V-belts = 1,2 to 2,5
- plain belts = 1,5 to 4,5

The larger values apply:

- where the distance between shafts is short
- for heavy or peak load type duty
- where belt tension is high

Life modification factor, a_{SKF}

The life modification factor a_{SKF} expands the scope of the basic rating life model, L_{10}, which depends purely on load and size, by taking the following important operational factors into account:

- the fatigue load limit in relation to the acting bearing equivalent load (P_u/P)
- the effect of the contamination level in the bearing (η_c)
- the lubrication condition (viscosity ratio κ)

This makes the resulting SKF rating life, L_{10m}, more encompassing than L_{10} when verifying bearing size selection:

$$L_{nm} = a_1\ a_{SKF}\ L_{10} = a_1\ a_{SKF} \left(\frac{C}{P}\right)^p$$

A graph for estimating a_{SKF} is shown in **diagram 8**. The horizontal axis represents the combined influence of load and contamination on fatigue. The viscosity ratio, κ, represents the lubrication conditions and their influence on fatigue.

Use **diagram 8** to see how operating conditions affect the basic rating life:

- **Area A** is dominated by very high load and/or severe indentations.
  The lubricating conditions in this domain can only marginally improve the expected fatigue life, so a potential life improvement depends on what dominates the relationship between the contamination level factor and the load level P_u/P. To achieve a greater SKF rating life, either the load must be reduced, or the cleanliness must be improved, or both.
- **Area B** offers high life modification factors, which is beneficial because a large life modification value will convert a low basic rating life sufficiently to produce a large SKF rating life.
  In this part of the graph, small deviations from estimated load level, cleanliness factor and lubrication conditions will greatly affect the life modification factor. Small changes to lubricating conditions, slightly higher loading and larger indentation severity (for example, from mounting or transport damage) may result in a change in a_{SKF} from 50 to 5. This would result in a 90% loss of SKF rating life. In cases where the SKF rating life consists of a large life modification factor a_{SKF} and a limited basic rating life L_{10}, the impact of variations in operating conditions should be evaluated in a sensitivity analysis.
- **Area C** is where the life modification factor is less sensitive to changes.
  Deviations from estimated load level, cleanliness factor and lubrication conditions (for example, from uncertainties in temperature) will not substantially affect the value of a_{SKF}, which means the resulting SKF rating life is more robust.
  In the load level domain, area C has the ranges:
  – $P_u \leq P \leq 0{,}5$ C for ball bearings
  – $P_u \leq P \leq 0{,}33$ C for roller bearings

Use the schematic a_{SKF} graph to evaluate how changes to operational conditions would affect the life modification factor. This can help you check whether a potential benefit is worth the effort. For example, you can see how:

- improved cleanliness (better sealing, filtration and assembly conditions) increases the contamination level factor η_c
- cooling or using a lubricant with higher viscosity increases the viscosity ratio κ
- choosing a larger bearing size increases the ratio P_u/P (and the basic rating life L_{10})
- using SKF Explorer bearings allows a more favourable scale on the horizontal axis for the combined effect of the η_c times P_u/P

The following graphs show plots of the life modification factor a_{SKF} for the four bearing types, as a function of $\eta_c(P_u/P)$, for SKF Explorer and SKF basic design bearings, and for different values of the viscosity ratio κ:

- **diagram 9**, **page 96**: radial ball bearings
- **diagram 10**, **page 97**: radial roller bearings
- **diagram 11**, **page 98**: thrust ball bearings
- **diagram 12**, **page 99**: thrust roller bearings

NOTE

The graphs in **diagram 9**, **10**, **11** and **12** are plotted for values and safety factors typically associated with fatigue load limits for other mechanical components. Considering the simplifications inherent in the SKF rating life equation, even if the operating conditions are accurately identified, it is not meaningful to use values of a_{SKF} in excess of 50.

Diagram 8

Factors influencing the life modification factor, a_{SKF}

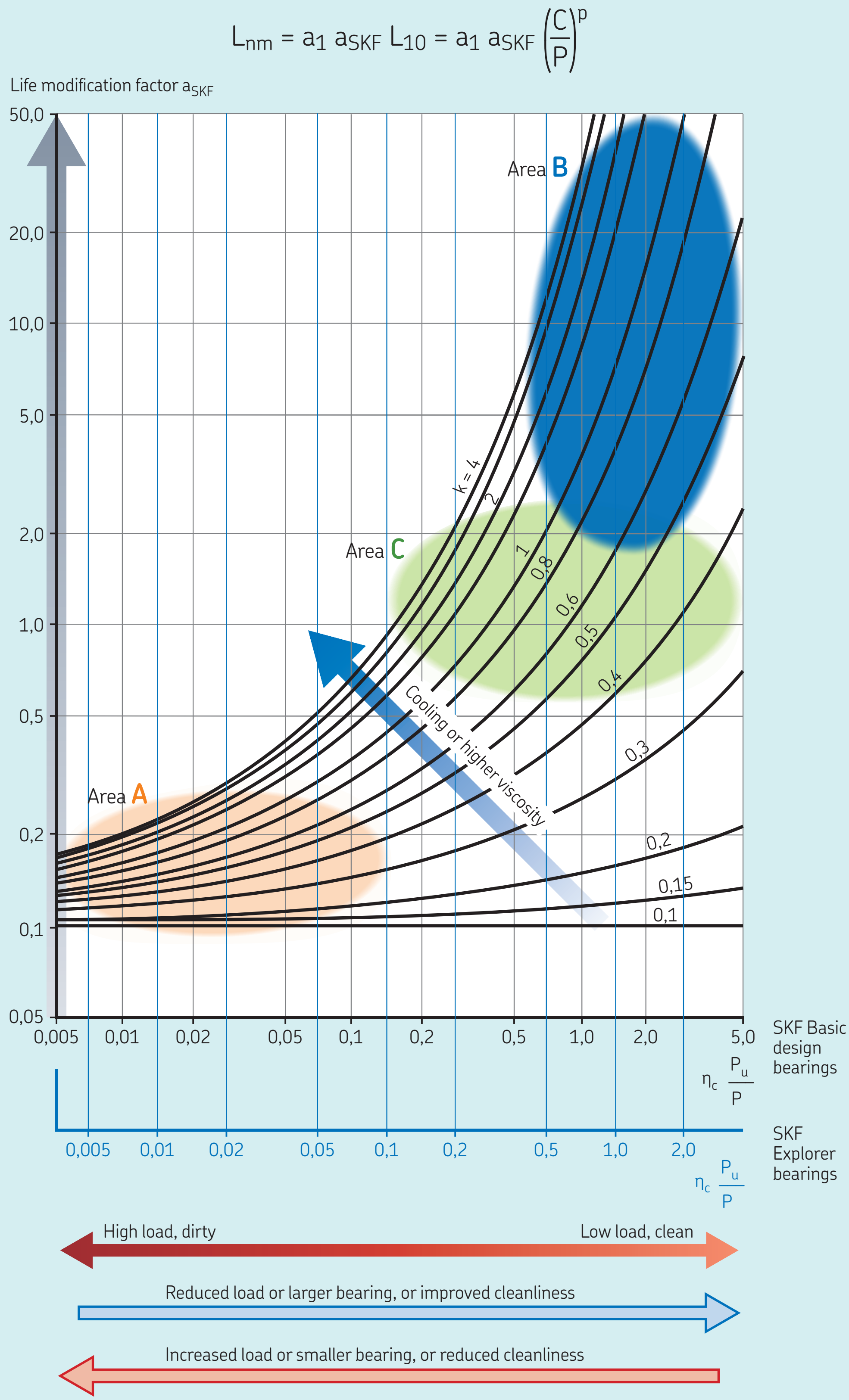


Diagram 9

Factor a_{SKF} for radial ball bearings

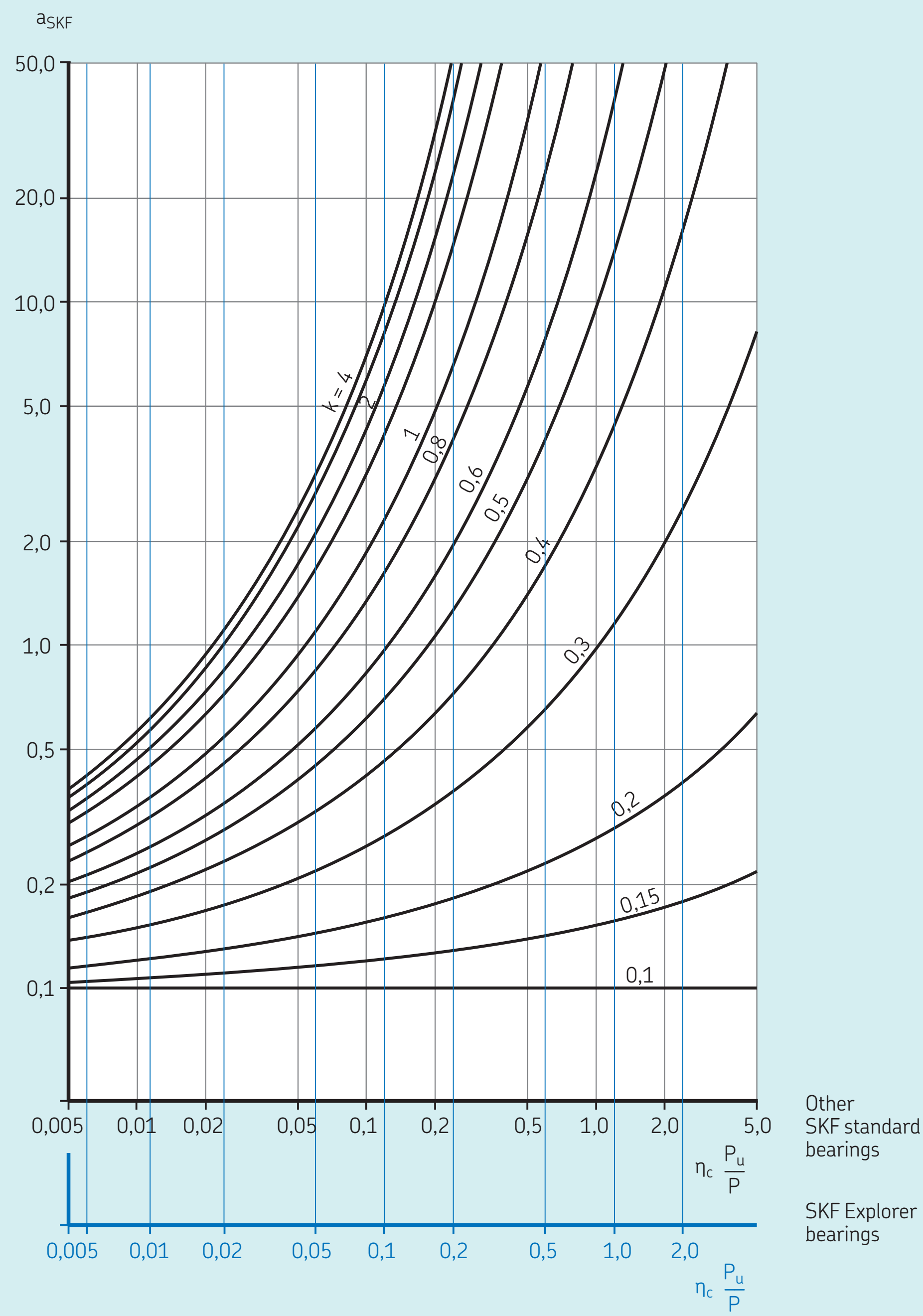


Diagram 10

Factor a_{SKF} for radial roller bearings

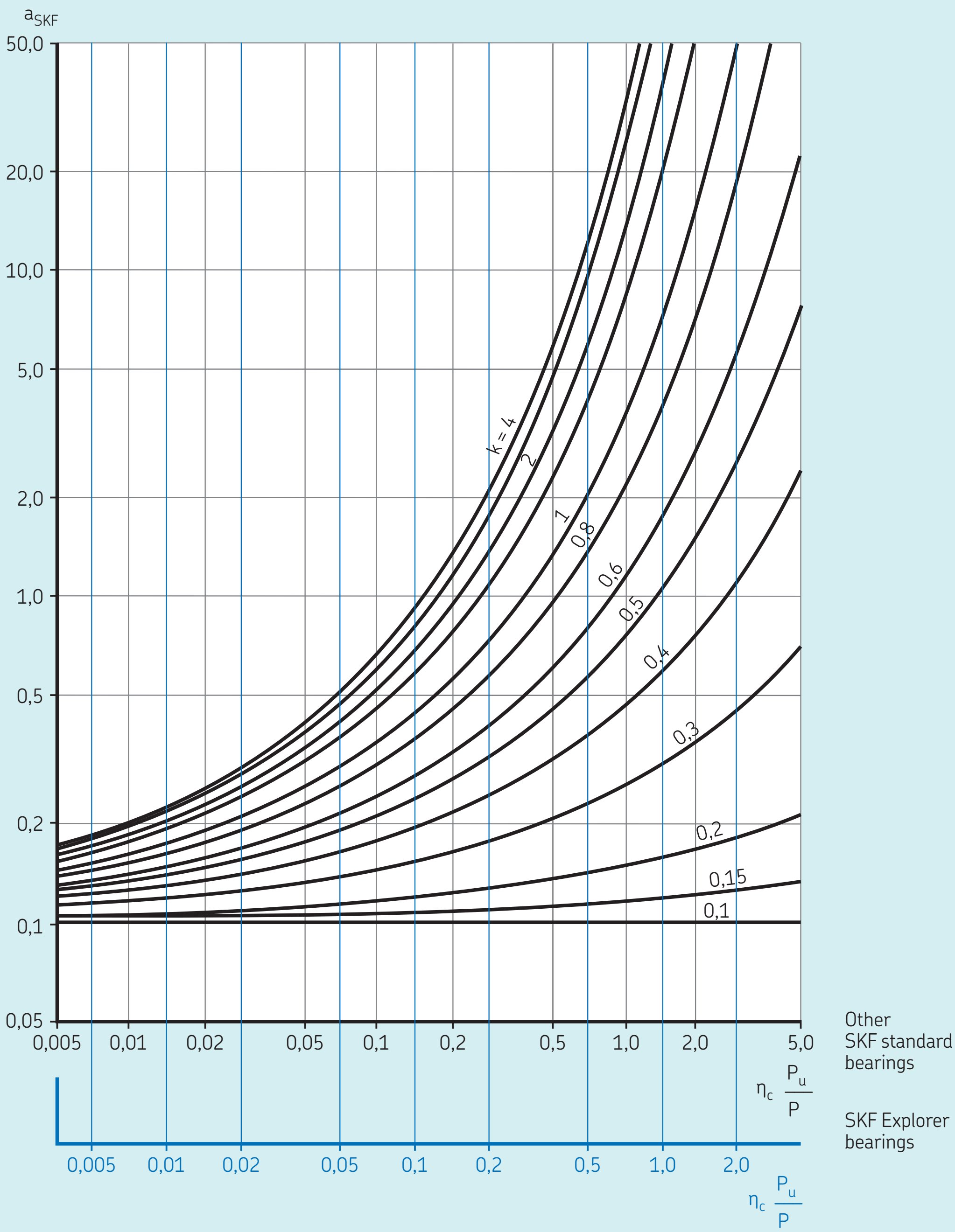


Diagram 11

Factor a_{SKF} for thrust ball bearings

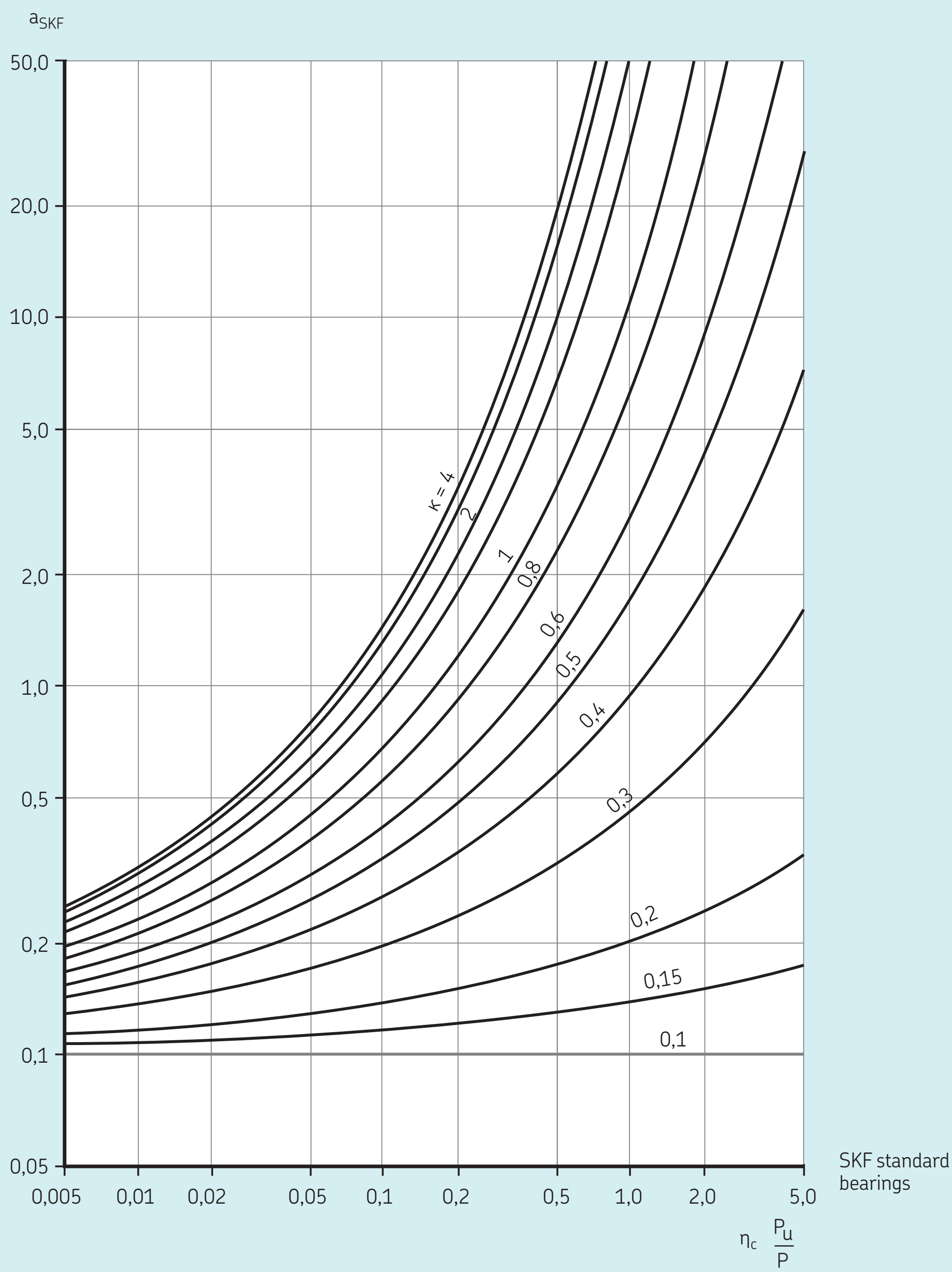


Diagram 12

Factor a_{SKF} for thrust roller bearings

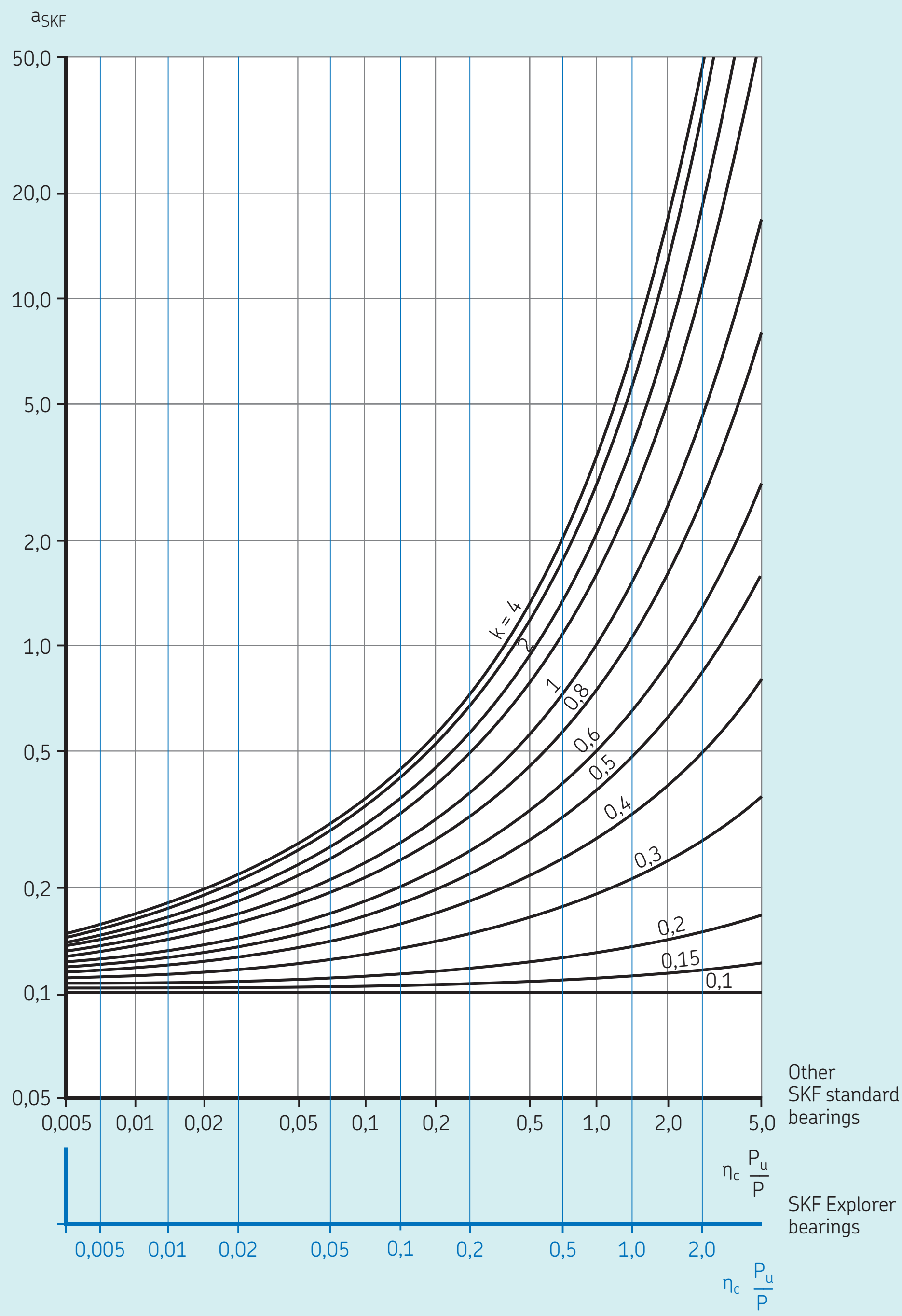


Diagram 13

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

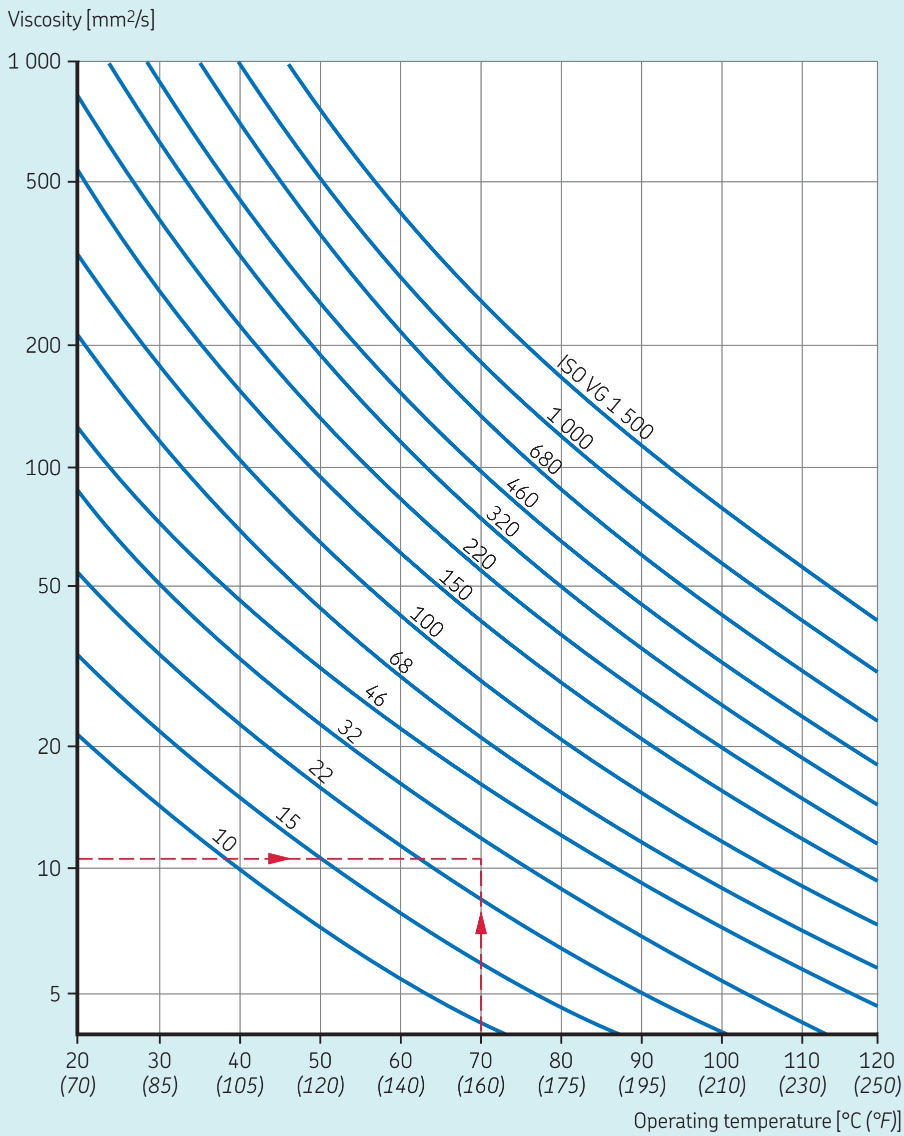
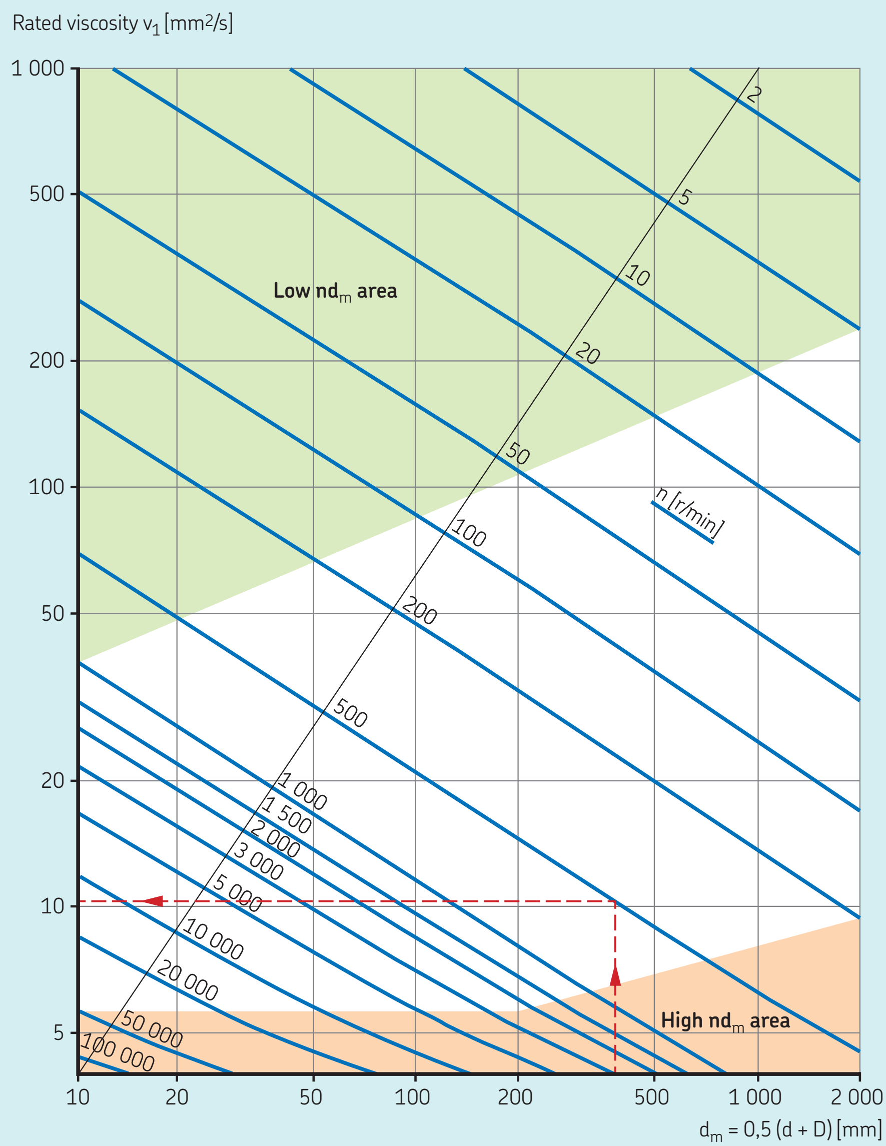


Diagram 14

Estimation of the rated viscosity ν_1



Low nd_m area, where $nd_m \leq 10\ 000$ mm/min.
At these lower nd_m values, AW or EP additives are needed to reduce wear.

High nd_m area, where $nd_m \geq 500\ 000$ mm/min for $d_m \leq 200$ mm, and $nd_m \geq 400\ 000$ mm/min for $d_m > 200$ mm.
At these higher nd_m values, operating temperature must be given more attention. Certain bearing types, such as spherical roller bearings, tapered roller bearings and spherical roller thrust bearings, normally have a higher operating temperature than others, such as deep groove ball bearings and cylindrical roller bearings, under comparable operating conditions.

Lubrication condition – the viscosity ratio, κ

When a bearing has reached its normal speed and operating temperature, the lubrication condition of the bearing is:

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

where

κ = lubrication condition of the bearing, i.e. viscosity ratio

ν = actual operating viscosity of the oil or the grease base oil [mm^2/s]

ν_1 = rated viscosity, function of the mean bearing diameter and rotational speed [mm^2/s]

The actual operating viscosity, ν, of the lubricant can be determined from the ISO viscosity grade of the oil, or the grease base oil, and the operating temperature of the bearing (**diagram 13**, **page 100**) .

You can determine the rated viscosity, ν_1, from **diagram 14**, **page 101**, using the bearing mean diameter d_m = 0,5 (d + D) [mm] and the rotational speed of the bearing, n [r/min]. Alternatively, you can use the *SKF Bearing Calculator* (skf.com/bearingcalculator).

Viscosity grades, in accordance with ISO 3448, are listed in **table 5**, along with the viscosity range for each grade at 40 °C *(105 °F)*.

The higher the κ value, the better the lubrication condition of the bearing and its expected rated life. This must be judged against the possible friction increase because of the higher oil viscosity. Therefore, most bearing applications are designed for a lubrication condition ranging from κ 1 to 4 (**diagram 15**). Alternatively, you can use the *SKF Bearing Calculator* (skf.com/bearingcalculator) to calculate the lubrication condition.

- κ = 4 indicates a regimen for which the rolling contact load is carried by the lubricant film – i.e. full film lubrication.
- κ > 4 (i.e. better than full film lubrication) will not further increase the rating of the bearing. However, κ > 4 may be useful in applications where the bearing temperature rise is small and additional lubrication condition reliability is desirable. This would apply, for example, to bearing applications with frequent start-stop running conditions or occasional temperature variations.
- κ < 0,1 indicates a regimen for which the rolling element load is carried by the contact of the asperities between rolling element and raceway – i.e. boundary lubrication. The use of fatigue life rating for lubrication conditions below 0,1 is not appropriate as it is beyond the applicability limits of the life rating model. Where κ < 0,1, select the bearing size on the basis of static loading criteria by means of the static safety factor, s_0 (*Size selection based on static load*, **page 104**).

κ value below 1

For lubrication conditions with 0,1 < κ < 1, take into account the following:

- If the κ value is low because of very low speed, base the bearing size selection on the static safety factor s_0 (*Size selection based on static load*, **page 104**).
- If the κ value is low because of low viscosity, counteract this by selecting a higher viscosity oil or by improving the cooling. Under these lubrication conditions, it is not appropriate to calculate the basic rating life L_{10} only, because it does not take into account the detrimental effects of inadequate lubrication of the bearing. Instead, to estimate the rolling contact fatigue life of the bearing, use the SKF rating life method.

Where κ < 1, EP/AW additives are recommended.

The speed factor nd_m is used to characterize the speed condition of the bearing.

- If the nd_m of the bearing is lower than 10 000, the application is operating under low-speed conditions (**diagram 14**, **page 101**). This regimen requires high oil viscosity to ensure that the rolling element load is carried by the lubricant film.
- Operating conditions leading to nd_m > 500 000 for d_m values up to 200 mm, and > 400 000 for larger d_m values, are typical of bearings operating at high speeds (**diagram 14**). At very high speeds, the rated viscosity drops to very low values. Lubrication conditions and κ values are generally high.

EP (extreme pressure) and AW (anti-wear) additives

EP/AW additives in the lubricant are used to improve the lubrication condition of the bearing in situations where small κ values are in use, e.g. when κ = 0,5. Furthermore, EP/AW additives are also used to prevent smearing between lightly loaded rollers and raceway, for example, when especially heavy rollers enter a loaded zone at a reduced speed.

For operating temperatures lower than 80 °C *(175 °F)*, EP/AW additives in the lubricant may extend bearing service life when κ is lower than 1 and the factor for the contamination level, η_c, is higher than 0,2 and the resulting a_{SKF} factor is lower than 3. Under those conditions, a value of κ_{EP} = 1 can be applied, in place of the actual κ value, in the calculation of a_{SKF} for a maximum advantage of up to a_{SKF} = 3.

Some modern EP/AW additives containing sulphur-phosphorus, which are most commonly used today, can reduce bearing life. Generally, SKF recommends testing chemical reactivity of EP/AW for operating temperatures lower than 80 °C *(175 °F)*.

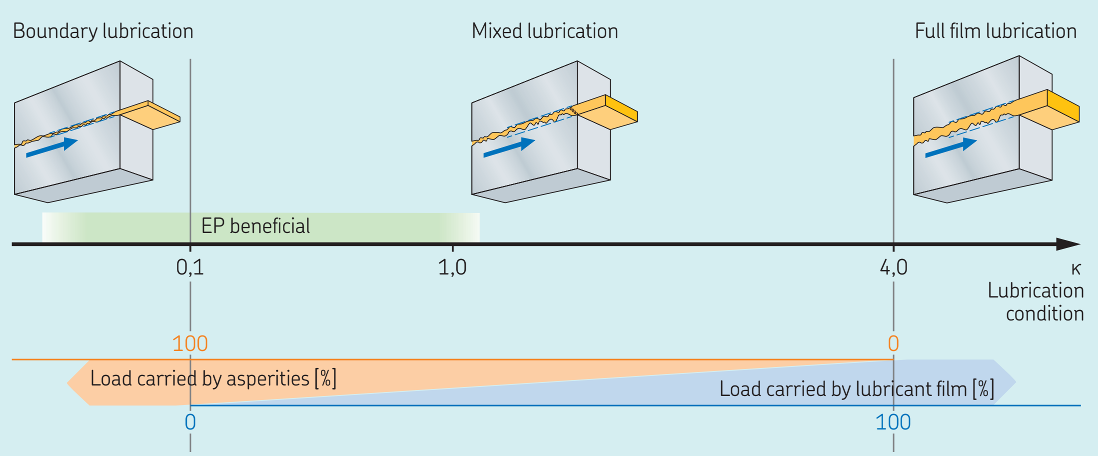
Table 5

Viscosity classification to ISO 3448

Viscosity grade	Kinematic viscosity limits at 40 °C (*105°F*) mean	min.	max.
–	mm²/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 15

Lubrication condition



Lubrication condition	κ	Size selection
Boundary lubrication Full asperity contact, wear without EP/AW additives, high friction	κ ≤ 0,1	static safety factor
Mixed lubrication Reducing asperity contact, wear and surface fatigue without EP/AW additives, friction reduced	0,1 < κ ≤ 4	SKF rating life and static safety factor[1]
Full film lubrication No asperity contact, increasing viscous frictional moment	κ > 4	SKF rating life (no life gain, possible higher temperatures) and static safety factor[1]

[1] This applies to peak load.

Fatigue load limit, P_u

The fatigue load limit P_u for a bearing is defined as the load level below which metal fatigue will not occur. For this to be valid, the lubricant film must fully separate the rolling elements from the raceways and no indentations, from contaminants or from damage related to handling, may exist on the rolling surfaces.

Contamination factor, η_c

The contamination factor, η_c, takes into account how the level of solid particle contamination of the lubricant influences the calculated bearing fatigue life. The particles cause indentations in the rolling surfaces of the bearing, and these indentations increase the local contact stress, which reduces the expected fatigue life (**fig. 3**).

- $\eta_c = 1$ means perfectly clean conditions without any indentations.
- $\eta_c \rightarrow 0$ means severely contaminated conditions resulting in pronounced indentations.

In the SKF rating life model, the contamination factor for a certain bearing acts as a stress raiser, by reducing the bearing fatigue load limit P_u (i.e. multiplying it by the contamination factor η_c).

Comparing the reduced fatigue load limit to the actual bearing load, the fatigue resistance value ($\eta_c P_u/P$) takes both the relative bearing load and the local stress field into account (**diagram 8**, **page 95**).

- Clean conditions (large contamination factor η_c) and a bearing load lower than the fatigue load limit results in a high resistance to fatigue.
- Contaminated conditions and a bearing load larger than the fatigue load limit results in a lower resistance to fatigue.

The stress-raising influence of contamination on bearing fatigue depends on a number of parameters, including: bearing size, relative lubricant condition, size and distribution of solid contaminant particles and types of contaminants (soft, hard, etc.). Therefore, it is not meaningful to specify precise values for the contamination factor η_c that would have general validity. However, guideline values in accordance with ISO 281 are listed in **table 6**.

To simplify calculation of the contamination factor η_c, use the *SKF Bearing Calculator* (skf.com/bearingcalculator).

A more detailed method for estimating the contamination factor η_c is described in a separate paper (*Method for estimating contamination factor, η_c, based on lubricant cleanliness*, skf.com/go/17000-B3).

Size selection based on static load

When any of the following conditions exist, bearing size should be selected or verified based on the static load that the bearing can accommodate, taking into account the possible effects of permanent deformation:

- The bearing is not rotating and is subjected to continuous high load or intermittent peak loads.
- The bearing makes slow oscillating movements under load.
- The bearing rotates and, in addition to the normal fatigue life dimensioning operating loads, has to sustain temporary high peak loads.
- The bearing rotates under load at low speed ($n < 10$ r/min) and is required to have only a limited life. In such a case, the rating life equations, for a given equivalent load P, would give such a low requisite basic dynamic load rating C, that a bearing selected on a fatigue life basis would be seriously overloaded in service.

In such conditions, the resulting deformation can include flattened areas on the rolling elements or indentations in the raceways. The indentations may be irregularly spaced around the raceway, or evenly spaced at positions corresponding to the spacing of the rolling elements. A stationary or slowly oscillating bearing supporting a load great enough to cause permanent deformation will generate high levels of vibration and friction when subjected to continuous rotation. It is also possible that the internal clearance will increase or the character of the housing and shaft fits may be affected.

Static load rating

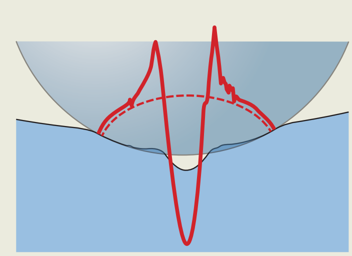
The basic static load rating C_0 is defined in ISO 76 as the load that results in a certain value of contact stress at the centre of contact of the most heavily loaded rolling element/raceway. The contact stress values are:

- 4 600 MPa for self-aligning ball bearings
- 4 200 MPa for all other ball bearings
- 4 000 MPa for all roller bearings

These stress values produce a total permanent deformation of the rolling element and raceway that is approximately 0,0001 of the rolling element diameter. The loads are purely radial for radial bearings and axial, centrically acting, for thrust bearings.

Fig. 3

Example of stress fields



Equivalent static bearing load

Loads comprising radial and axial components that are to be evaluated in relation to the static load rating C_0, must be converted into an equivalent static bearing load. This is defined as that hypothetical load (radial for a radial bearing and axial for a thrust bearing) which, when applied, would cause the same maximum rolling element load in the bearing as the actual loads to which the bearing is subjected. It is obtained from the general equation

$$P = X_0 F_r + Y_0 F_a$$

where
P_0 = equivalent static bearing load [kN]
F_r = actual radial bearing load [kN]
F_a = actual axial bearing load [kN]
X_0 = radial load factor for the bearing
Y_0 = axial load factor for the bearing

Information and data required for calculating the equivalent static bearing load P_0 is provided in the relevant product sections.

In the equation, use radial and axial component values (fig. 4) for the maximum load that can occur. If the load varies then consider the combination that induces the highest value of P_0.

Fig. 4

Equivalent static bearing load

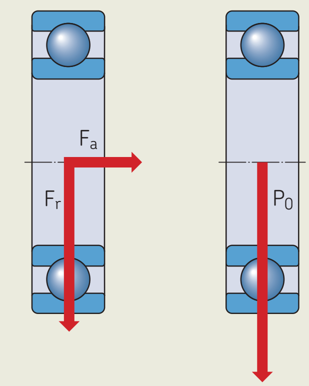


Table 6

Guideline values for factor η_c for different level of contamination

Conditions	Factor η_c[1)] for bearings with diameter d_m < 100	$d_m \geq$ 100 mm
Extreme cleanliness • Particle size of the order of the lubricant film thickness • Laboratory conditions	1	1
High cleanliness • Oil filtered through an extremely fine filter • Typical conditions: sealed bearings that are greased for life	0,8 ... 0,6	0,9 ... 0,8
Normal cleanliness • Oil filtered through a fine filter • Typical conditions: shielded bearings that are greased for life	0,6 ... 0,5	0,8 ... 0,6
Slight contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles and slight ingress of contaminants	0,5 ... 0,3	0,6 ... 0,4
Typical contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles, and ingress from surroundings	0,3 ... 0,1	0,4 ... 0,2
Severe contamination • Typical conditions: high levels of contamination due to excessive wear and/or ineffective seals • Bearing arrangement with ineffective or damaged seals	0,1 ... 0	0,1 ... 0
Very severe contamination • Typical conditions: contamination levels so severe that values of ηc are outside the scale, which significantly reduces the bearing life	0	0

1) The scale for η_c refers only to typical solid contaminants. Contamination by water or other fluids detrimental to bearing life is not included. Because of strong abrasive wear in highly contaminated environments (η_c = 0), the useful life of the bearing can be significantly shorter than the rating life.

B.3 Bearing size

Guideline values for static safety factor, s_0

The static safety factor s_0 is given by

$s_0 = C_0/P_0$

where
s_0 = static safety factor
C_0 = required basic static load rating [kN]
P_0 = equivalent static bearing load [kN]

Alternatively, you can calculate the required basic static load rating C_0.

Guideline values for the static safety factor s_0, based on experience, are listed for ball bearings in **table 7**, and roller bearings in **table 8**. The s_0 values given for continuous motion relate to the influence of permanent deformation on bearing performance – ranging from noticeable friction peaks, vibrations and reduced fatigue resistance (for the lowest s_0 values), to no influence on friction, vibration or fatigue life (for the highest s_0 values). The certainty of load level reflects how well the actual bearing load is known and/or can be predicted.

Requisite minimum load

In applications where the bearing size is determined by factors other than load – for example, shaft diameter constrained by critical speed – the bearing may be lightly loaded in relation to its size and carrying capacity. Where there are very light loads, failure mechanisms other than fatigue, such as skidding and smearing of raceways or cage damage, often prevail. To provide satisfactory operation, rolling bearings must always be subjected to a given minimum load. As a general rule, minimum loads of 0,01 C should be imposed on ball bearings and 0,02 C on roller bearings. More accurate minimum load requirements are given in the product sections.

The importance of applying a minimum load is greater in applications where there are rapid accelerations or rapid starts and stops, and where speeds exceed 50% of the limiting speeds listed in the product tables (*Speed limitations*, **page 135**). If minimum load requirements cannot be met, potential improvements are:

- Use a bearing with a smaller dimension series.
- Consider special lubrication or running-in procedures.
- Consider *NoWear coated bearings*, **page 1060**.
- Consider applying a preload (*Selecting preload*, **page 186**).

Checklist after the bearing size is determined

When you have worked through this section and determined bearing size, before continuing to the section on *Lubrication*, **page 110**, check the following by referring to the product sections:

- grease life for capped bearings
- allowed axial/radial loads and F_a/F_r ratios
- minimum load
- adjusted reference speed and limiting speed
- misalignment
- stabilization class

Table 7

Guideline values for the static safety factor s_0 – for continuous and/or occasional loads – ball bearings

Certainty of load level	**Continuous motion** Permanent deformation acceptance Yes	Some	No	**Infrequent motion** Permanent deformation acceptance Yes
High certainty For example, gravity loading and no vibration	0,5	1	2	0,4
Low certainty For example, peak loading	≥ 1,5	≥ 1,5	≥ 2	≥ 1

SKF life testing

SKF carries out life testing in the ISO 17025 accredited SKF Engineering and Research Centre in the Netherlands, together with the other SKF group research and testing facilities.

The purpose of this life testing is to improve the design, the materials and the manufacturing processes of bearing products, and the engineering analysis tools required for the design of bearing applications.

Typical life testing activities include tests on bearing population samples under different conditions, such as:

- full film lubrication conditions
- boundary and mixed lubrication conditions
- predefined contamination conditions of the lubricant

Apart from testing in different conditions, SKF life tests are performed to:

- verify the data published in product catalogues
- audit the quality of the manufacturing of SKF bearings
- research how lubricants and lubrication conditions influence bearing life
- support the development of rolling contact fatigue and friction models
- compare SKF products with competitors' products

Life tests are sophisticated and wide-ranging and are run under strictly controlled conditions. Post-test investigations with state-of-the-art equipment make it possible to investigate the factors that affect the life of the bearings in a systematic way.

As an example, the SKF Explorer bearing design is the result of optimizing influencing factors determined by analytical simulations and experimental verification.

Table 8

Guideline values for the static safety factor s_0 – for continuous and/or occasional loads – roller bearings[1)]

Certainty of load level	**Continuous motion** Permanent deformation acceptance Yes	Some	No	**Infrequent motion** Permanent deformation acceptance Yes
High certainty For example, gravity loading and no vibration	1	1,5	3	0,8
Low certainty For example, peak loading	≥ 2,5	≥ 3	≥ 4	≥ 2

1) For spherical roller thrust bearings, use $s_0 \geq 4$.

B.4 Lubrication

B.4 Lubrication

B.4 Lubrication

Rolling bearings must be adequately lubricated to operate reliably. The lubricant is required to reduce friction, inhibit wear, protect the bearing surfaces against corrosion and may also be needed to provide cooling. This section describes:

- how to select between grease or oil
- how to select a suitable grease
- how to select a suitable oil

For information on lubrication of sealed bearings, refer to the relevant product sections.

How lubrication relates to other selection criteria

Lubrication selection and lubricant properties greatly influence the operating temperature, which in turn influences:

- whether you should use grease or oil
- the relubrication interval required for grease
- whether oil lubrication is necessary, because circulating oil can be used to remove heat
- the lubrication condition – the viscosity ratio, κ, which influences the bearing size selection based on SKF rating life

Selecting grease or oil

The first step in the lubrication selection process is to decide whether to use grease or oil. In most cases, grease is the appropriate choice for open bearings.

Lubrication selection flow chart and criteria

A flow chart to help select the correct lubrication method is shown in **diagram 1**.

The main reasons to choose grease are:

- cost-effectiveness
- simplicity – grease is easily retained in the bearing and housing, thus requiring less complicated sealing arrangements compared with those for oil lubrication

The main exceptions to choosing grease are in applications where:

- operating conditions require a grease relubrication interval that is unacceptably short
- lubricating oil must be used for other purposes (such as in gearboxes)
- heat removal via circulating oil is required
- purging or removing used grease becomes cumbersome or expensive to handle

Estimating the relubrication interval for grease

Lubricating grease slowly degrades and therefore has a limited life. Grease life depends on the operating conditions of the bearing and the grease type. Rolling bearings therefore have to be relubricated if:

- the grease life is shorter than the specified bearing life
- the grease becomes contaminated

It is important to calculate the grease relubrication interval and if it is unacceptably short then, unless you use automatic (centralized) greasing (*Lubrication systems*, page 120), you should choose oil instead.

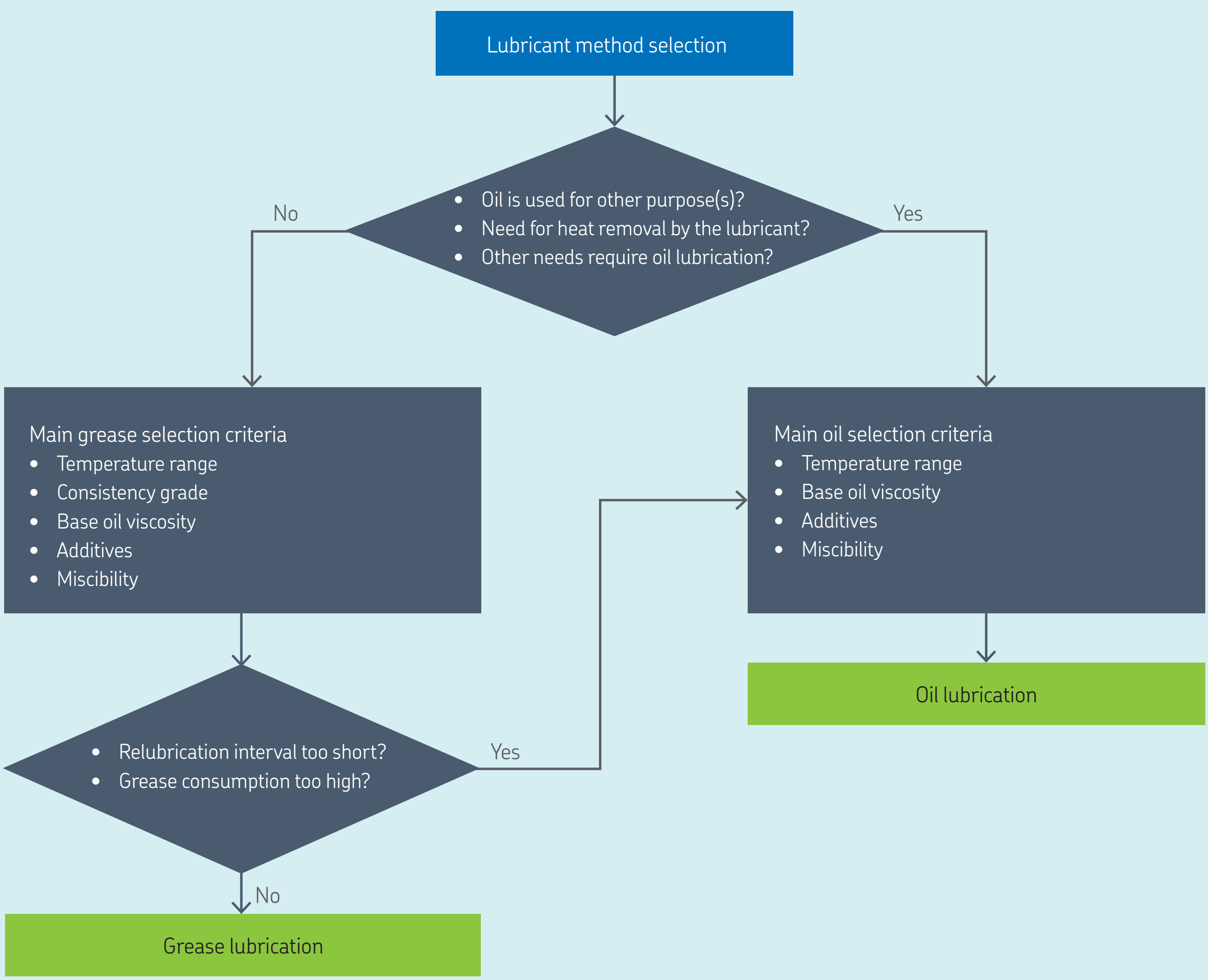
Relubrication should occur frequently enough to avoid grease deterioration having an adverse effect on the bearing life. Therefore, the SKF relubrication interval, t_f, is defined as the time period at the end of which there is only a 1% probability that the bearing will fail because of grease degradation. This represents the L_1 grease life. L_{10} grease life represents a 10% probability failure because of grease degradation. Grease life depends mainly on:

- bearing type and size
- speed
- load ratio C/P
- operating temperature
- grease type

As a rule, standard greases have a practical upper temperature limit of 100 °C *(210 °F)* on the ring with the highest temperature. Above this temperature, special greases or automatic (centralized) greasing systems should be used – otherwise, commonly the grease life would be too short.

Diagram 1

Process for selecting a suitable lubrication method for open bearings



Relubrication intervals

Use **diagram 2** to estimate the relubrication intervals t_f. The diagram is valid for bearings with a rotating inner ring on horizontal shafts under normal and clean operating conditions, using:

- the nd_m factor multiplied by the relevant bearing factor b_f where
  - n = rotational speed [r/min]
  - d_m = bearing mean diameter [mm] = 0,5 (d + D)
  - b_f = bearing factor dependent on bearing type and load conditions (**table 1**)
- the load ratio C/P

The relubrication interval t_f is the estimated number of operating hours that a good quality lithium soap grease with a mineral base oil can perform adequately when the operating temperature is 70 °C *(160 °F)*. High performance greases can extend relubrication intervals and grease life.

The relubrication intervals given in **diagram 2** must be adjusted according to **table 2**, **page 115**.

When the speed factor nd_m exceeds 70% of the recommended limits (**table 1**), check the influence of the selected lubricant on the operating temperature and speed.

In practice, relubrication intervals above 30 000 h are not reliable, because intervals of that length exceed the predictable performance life (because of lubricant ageing) of most greases.

Adjustments for relubrication intervals

Various adjustments for relubrication intervals are described in **table 2** under various operating conditions. You may also calculate lubrication intervals using the *SKF Bearing Calculator* (skf.com/bearingcalculator).

Determining grease quantity for initial fill and relubrication

Commonly, the free volume in bearings is completely filled during installation and the free volume in SKF plummer block housings is partly filled. SKF recommends that the free volume on each side of the bearing in a customer-designed housing is equal to the free volume of the bearing. For bearings with a metallic cage, the free volume in the bearing is approximately

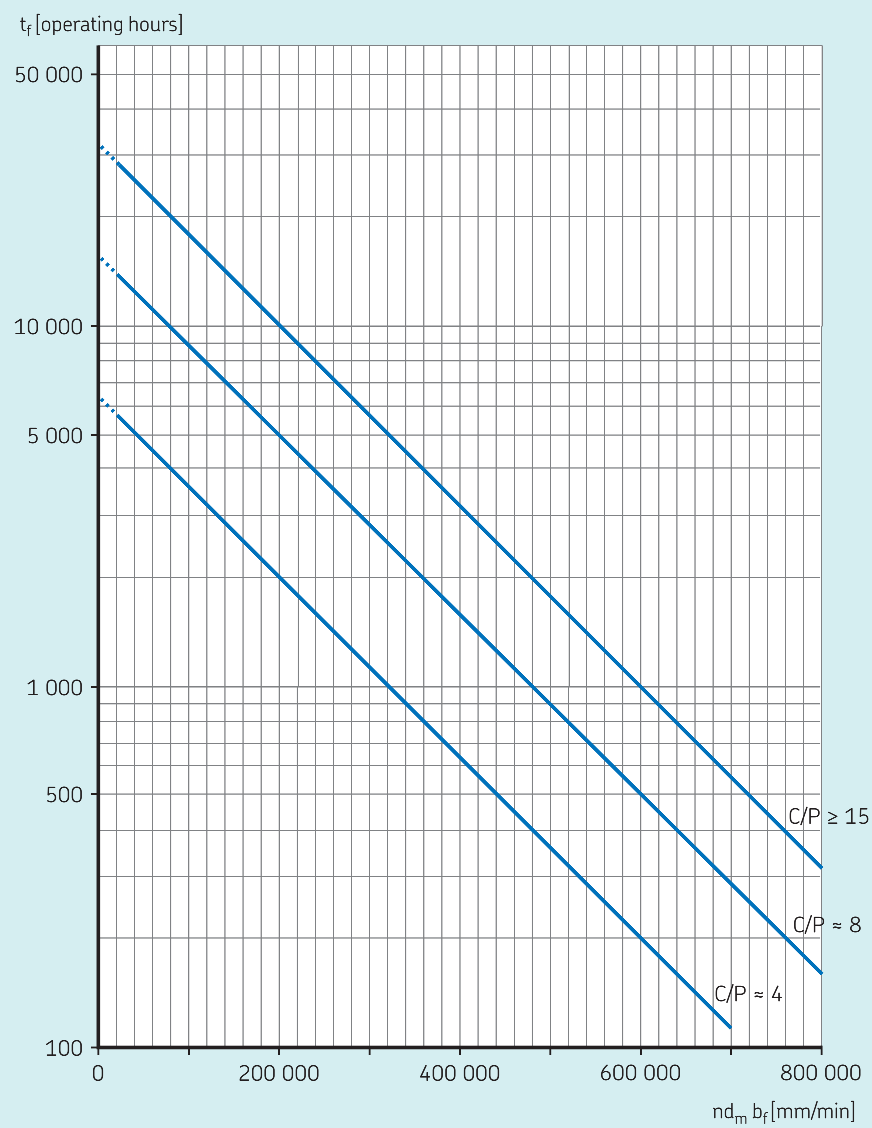
$$V = \frac{\pi}{4} B (D^2 - d^2) \times 10^{-3} - \frac{M}{7,8 \times 10^{-3}}$$

where

- V = free volume in the bearing [cm^3] (for standard grease, mass in grams multiplied by 0,9; for fluorinated grease, mass in grams multiplied by approximately 2)
- B = bearing width [mm]
- D = outside diameter [mm]
- d = bore diameter [mm]
- M = bearing mass [kg]

Diagram 2

Relubrication intervals at operating temperatures of 70 °C (*160 °F*)



For bearings with non-metallic cages, the formula gives a slight overestimation.

Depending on the intended method of relubrication, SKF recommends:

- relubrication from the side of the bearing (fig. 1, page 114)
  - initial fill: 40% of the free volume in the housing
  - replenishment quantity: $G_p = 0{,}005\ D\ B$
- relubrication through holes in the centre of the inner or outer ring (fig. 2, page 114)
  - initial fill: 20% of the free volume in the housing
  - replenishment quantity: $G_p = 0{,}002\ D\ B$

where

G_p = grease quantity to be added when replenishing [g]

D = bearing outside diameter [mm]

B = total bearing width [mm] (for tapered roller bearings use T, for thrust bearings use height H)

During a running-in period, excess grease in the bearing distributes or escapes. At the end of the running-in period, the operating temperature drops, indicating that the grease has been distributed.

In applications where bearings operate at very low speeds and good protection against contaminants and corrosion is required, SKF recommends filling 70% to 100% of the housing with grease.

Table 1

Bearing factors and recommended nd_m limits

Bearing type[1]	**Bearing factor** b_f	**Recommended nd_m limits for load ratio** C/P ≥ 15	C/P ≈ 8	C/P ≈ 4
–	–	mm/min		
Deep groove ball bearings	1	500 000	400 000	300 000
Angular contact ball bearings	1	500 000	400 000	300 000
Self-aligning ball bearings	1	500 000	400 000	300 000
Cylindrical roller bearings				
– non-locating bearing	1,5	450 000	300 000	150 000
– locating bearing, without external axial loads or with light but alternating axial loads	2	300 000	200 000	100 000
– locating bearing, with constantly acting light axial load	4	200 000	120 000	60 000
– without a cage, full complement[2]	4	NA[3]	NA[3]	20 000
Needle roller bearings				
– with a cage	3	350 000	200 000	100 000
Tapered roller bearings	2	350 000	300 000	200 000
Spherical roller bearings				
– when the load ratio $F_a/F_r \leq e$ and $d_m \leq 800$ mm				
series 213, 222, 238, 239	2	350 000	200 000	100 000
series 223, 230, 231, 232, 240, 248, 249	2	250 000	150 000	80 000
series 241	2	150 000	80 000	50 000
– when the load ratio $F_a/F_r \leq e$ and $d_m > 800$ mm				
series 238, 239	2	230 000	130 000	65 000
series 230, 231, 232, 240, 248, 249	2	170 000	100 000	50 000
series 241	2	100 000	50 000	30 000
– when the load ratio $F_a/F_r > e$ all series	6	150 000	50 000	30 000
CARB toroidal roller bearings				
– with a cage	2	350 000	200 000	100 000
– without a cage, full complement[2]	4	NA[3]	NA[3]	20 000
Thrust ball bearings	2	200 000	150 000	100 000
Cylindrical roller thrust bearings	10	100 000	60 000	30 000
Needle roller thrust bearings	10	100 000	60 000	30 000
Spherical roller thrust bearings				
– rotating shaft washer	4	200 000	120 000	60 000

[1] The bearing factors and recommended nd_m limits apply to bearings with standard internal geometry and standard cage execution. For alternative internal bearing design and special cage execution, contact the SKF application engineering service.
[2] The t_f value obtained from diagram 2 needs to be divided by a factor of 10.
[3] Not applicable, as a bearing with a cage is recommended for these C/P values.

B.4 Lubrication

Relubrication procedures

Select a relubrication procedure that suits the application and the relubrication interval t_f. SKF recommends one of the following procedures:

- **Manual relubrication by replenishment**
  is a convenient procedure. It enables uninterrupted operation and provides, when compared with continuous relubrication, a lower steady-state temperature.

- **Automatic (centralized) relubrication**
  avoids performance issues related to over- or under-greasing. This is also commonly used where there are multiple points to lubricate, or where access to positions is difficult, or where equipment is operated remotely with no local maintenance staff (**diagram 3**).

- **Continuous lubrication**
  is used when the estimated relubrication intervals are short because of the adverse effects of very severe contamination. Continuous lubrication of applications is recommended typically with nd_m values < 150 000 for ball bearings and < 75 000 for roller bearings. In these cases, the initial grease fill for the housing can be from 70% to 100 % (depending on the operation condition and housing seal), and the quantity for relubrication per unit of time is derived from the equations for G_p (*Determining grease quantity for initial fill and relubrication*, **page 112**) by spreading the required quantity over the relubrication interval.

There must be provision for the used grease to be purged from the housing. If an excess of used grease needs to be purged from the housing, contacting seals must allow for this (consider seal type and seal orientation). Otherwise, an escape hole should be provided in the housing – tubing is not allowed, because it can restrict grease escape. The escape hole should be plugged during high-pressure cleaning.

Where a variety of bearing types is used in a bearing arrangement, it is common practice to apply the shortest estimated relubrication interval from the bearings in the arrangement.

Fig. 1

Relubrication from the side

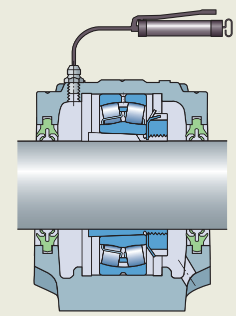


Fig. 2

Relubrication in the centre

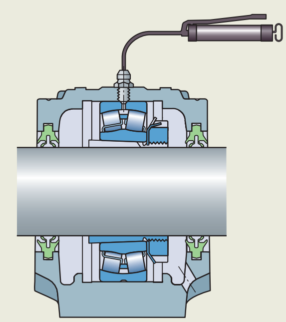


Diagram 3

Relubrication method and impact on performance

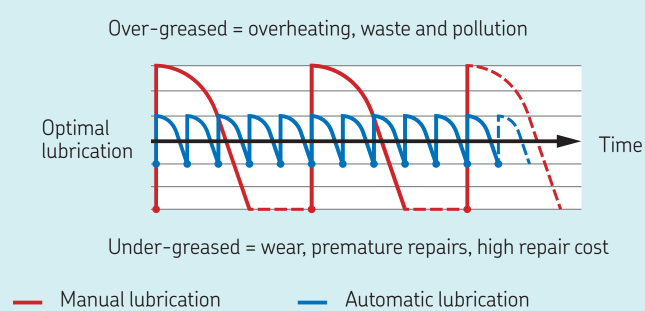


Table 2

Relubrication interval adjustments

Operating condition / bearing type	Description	Recommended adjustment of t_f	Reason for adjustment
Operating temperature	For every 15 °C *(27 °F)* above 70 °C *(160 °F)* up to the high temperature limit (HTL)	Halve the interval	To account for the accelerated ageing of grease at higher temperatures
	For 15 °C *(27 °F)* under 70 °C *(160 °F)*	Double the interval (maximum once)[1]	To account for the reduced risk of ageing of grease at lower temperatures
Shaft orientation	Bearings mounted on a vertical shaft	Halve the interval	The grease tends to leak out due to gravity
Vibration	High vibration or acceleration levels	Reduce the interval	Interval reduced depending on machine specific instructions (e.g. vibrating screen)
Outer ring rotation	Outer ring rotation or eccentric shaft weight	Calculate the speed as nD rather than nd_m	The grease has a shorter grease life under these conditions
Contamination	Contamination or presence of fluid contaminants	Adjust depending on the contamination level: **Low** Relubrication intervals are given by grease life. It is assumed that there will be no or slight ingress of contamination entering the bearing. **Medium** Some contaminants may enter the bearing. Some additional relubrication is required to remove contaminants. **High** There is a clear risk that contaminants will enter the bearing. Relubrication is required to remove aged grease and to remove contaminants. **Severe** Relubrication is primarily needed to flush the bearing and remove contaminants.	To reduce the damaging effects caused by contaminants
Bearing size	Bearings with a bore diameter d > 300 mm	Reduce the interval by a factor 0,5 initially. If grease samples taken before relubrication are found to be satisfactory, the relubrication interval can be increased gradually.	These are typically critical arrangements, which require strict, frequent relubrication programmes
Cylindrical roller bearings	Bearings fitted with J, JA, JB, MA, MB, ML, MP and PHA cages[2]	Halve the interval	These cage designs require higher oil bleeding from the grease

[1] For full complement and thrust bearings, do not extend the interval.
[2] For P, PH, M and MR cages, there is no need for adjustment.

Selecting a suitable grease

Selecting a suitable SKF grease

The assortment of SKF greases for rolling bearings provides adequate choice for most application requirements. These greases have been developed based on the latest knowledge of rolling bearing lubrication and their quality is continuously monitored.

Using LubeSelect and selection rules

SKF LubeSelect is an online tool that lists SKF greases that fulfil the demands of your specified operating conditions. The analysis performed by the tool is based on generalized selection rules that have been carefully developed by SKF lubrication experts.

The same selection rules are used in the *SKF bearing grease selection chart*, page 124, where the speed, temperature and load range are used as the primary operating parameters for selecting a suitable grease.

The most important technical specifications for SKF greases are provided in *Technical specifications for SKF greases*, page 126.

Temperature, speed, and load ranges for grease selection

The terms used to specify the ranges of temperature, speed and load, for grease lubricated bearings, are defined in **table 3** to **table 5**.

Consistency, NLGI

Consistency is a measure of the stiffness of the grease. Classification of greases by consistency is in accordance with the National Lubricating Grease Institute (NLGI), ISO 2137. Greases with a metallic soap thickener and a consistency grade of 1, 2 or 3 (soft to stiff) on the NLGI scale are typically used for rolling bearings. The most commonly used greases have a consistency of grade 2.

Table 3

Temperature ranges for greases

Range		**Temperature**	
–		°C	*°F*
L	Low	< 50	*< 120*
M	Medium	50 to 100	*120 to 210*
H	High	> 100	*> 210*
EH	Extremely high	> 150	*> 300*

Table 5

Load ranges for greases

Load range		**Load ratio** C/P
L	Low	≥ 15
M	Medium	≈ 8
H	High	≈ 4
VH	Very high	< 2

Table 4

Speed ranges for grease lubricated radial bearings

Speed range		**Speed factor** Ball bearings	Spherical roller, tapered roller, CARB toroidal roller bearings	Cylindrical roller bearings
		nd_m		
–		mm/min		
VL	Very low	–	< 30 000	< 30 000
L	Low	< 100 000	< 75 000	< 75 000
M	Medium	< 300 000	≤ 210 000	≤ 270 000
H	High	< 500 000	> 210 000	> 270 000
VH	Very high	≤ 700 000	–	–
EH	Extremely high	> 700 000	–	–

n = rotational speed [r/min]
d_m = bearing mean diameter [mm] = 0,5 (d + D)

Mechanical stability

During rotation of a bearing, the grease is mechanically worked and a change in consistency may result. This property is known as the mechanical stability of the grease and is measured in standardized tests, ASTM D217 and/or ASTM D1831. Greases that soften may leak from the bearing cavity. Those that stiffen may restrict bearing rotation or limit oil bleeding. The mechanical stability should not change drastically if operation is within the specified temperature range of the grease.

Corrosion protection

In applications where water or condensation is present, the corrosion inhibiting properties of the grease are very important. The corrosion inhibiting ability is determined by the properties of the rust inhibitor additive and/or the thickener type. The performance is measured using the EMCOR test, ISO 11007. For applications where water or condensation is present, the rating should be 0-0.

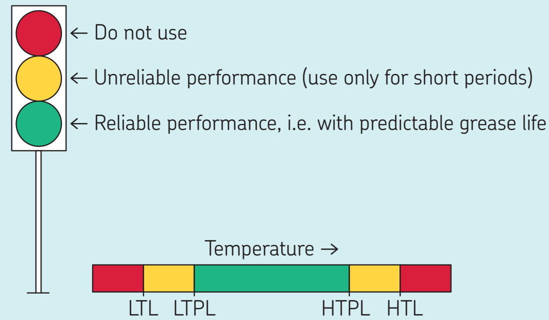
The SKF traffic light concept for grease temperature performance

The temperature range over which a grease can be used depends mainly on the type of base oil, thickener and additives. The relevant temperature limits are schematically illustrated in **diagram 4** in the form of a double traffic light, with additional details provided in **diagram 5**.

- The low temperature limit (LTL) is determined by the low temperature frictional torque test according to ASTM D1478 or IP 186. The LTL is determined by the temperature at which the starting torque is equal to 1 000 Nmm and the running torque is 100 Nmm.
- The high temperature limit (HTL) is the temperature at which a grease loses its consistency and becomes a fluid. It is determined using the dropping point (ISO 2176).

Diagram 4

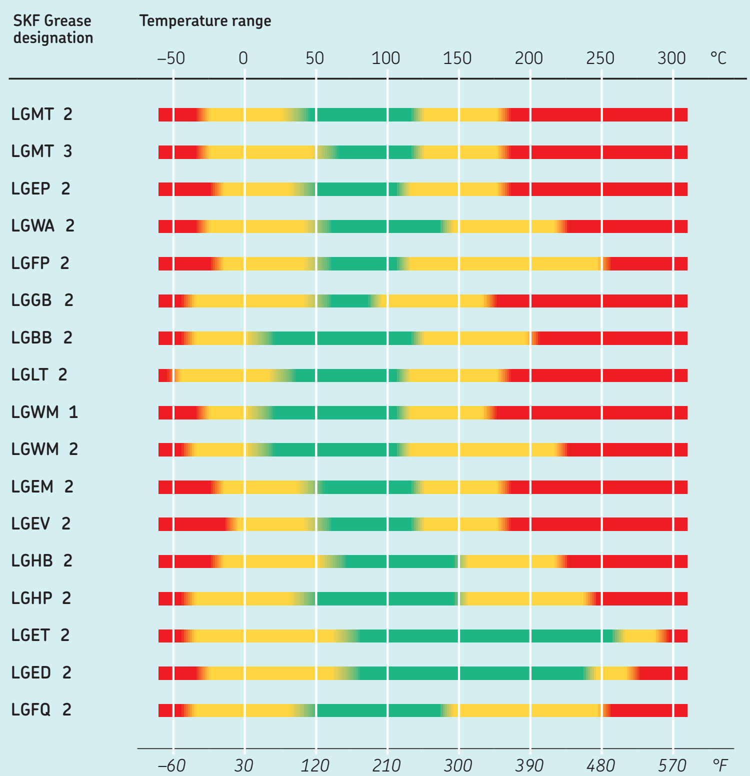
The SKF traffic light concept



LTL	Low temperature limit
LTPL	Low temperature performance limit
HTPL	High temperature performance limit
HTL	High temperature limit

Diagram 5

SKF traffic light concept – SKF greases[1)]



1) The low temperature performance limits (LTPL) are valid for roller bearings.
LTPL values for ball bearings are approx. 20 °C (*35 °F*) lower.

The low and high temperature limits for reliable operation, indicated by the green zone in **diagram 4**, **page 117**, are:

- low temperature performance limit (LTPL), defined as the temperature at which grease no longer shows sufficient oil bleed as measured in DIN 51817. The LTPL values for roller bearings are provided in **diagram 5**, **page 117**. The LTPL values for ball bearings are approximately 20 °C *(35 °F)* lower.
- high temperature performance limit (HTPL), determined by the SKF R0F grease life test

Within these two limits, the grease fulfils its function reliably and the relubrication interval or grease life is predictable. Because the definition of the temperature performance limits is not standardized internationally, care must be taken when interpreting data from grease suppliers other than SKF.

At temperatures above the high temperature performance limit (HTPL), grease degrades with increasing rapidity. Therefore, temperatures in the amber zone, between the high temperature performance limit (HTPL) and the high temperature limit (HTL), should only be allowed to occur for very short periods.

An amber zone also exists for low temperatures, between the low temperature limit (LTL) and the low temperature performance limit (LTPL). In this zone, the temperatures are too low to provide sufficient oil bleeding. The width of the amber zone depends on the grease type and bearing type. Serious damage can result when the bearings are operated continuously below the LTPL. Short periods in this zone, such as during a cold start, are generally not harmful because the heat caused by friction brings the bearing temperature into the green zone.

Additional factors and considerations when selecting a grease

Verify the lubrication condition, consider EP/AW additives

The lubrication condition, κ, is evaluated by using the base oil viscosity as described in *Lubrication condition – the viscosity ratio, κ*, **page 102**. In the lubrication condition domain defined by κ below 1, EP/AW additives are recommended.

EP/AW additives of the sulphur-phosphorus type, which are the most commonly used today, may also have a negative influence on the fatigue life of the bearings. This is because in the presence of humidity, which can never be completely avoided, sulphur and phosphorus acids are produced which induce a more aggressive chemical process at the rolling contact. This effect increases with temperature and, for temperatures above 80 °C *(175 °F)*, a lubricant with EP/AW additives should only be used after careful testing. SKF greases have been tested and can be used above 80 °C *(175 °F)* until the HTPL is reached.

Low speeds

Bearings that operate at very low to low speeds (**table 4**, **page 116**) under heavy loads should be lubricated with a grease that has a high viscosity base oil containing EP additives. The thickener should contribute to the surface separation. Sufficient oil bleeding should assure oil replenishment during operation.

Solid additives, such as graphite or molybdenum disulfide (MoS_2), should be considered for a speed factor $nd_m < 20\,000$ mm/min. SKF LGEV2 is successfully used up to $nd_m = 80\,000$.

Heavy and very heavy bearing loads

For bearings subjected to a load ratio C/P < 4, the calculated relubrication interval may be so short that it dictates the use of continuous relubrication or oil lubrication.

Miscibility with other greases

If it becomes necessary to change from one grease type to another, consider the miscibility of the greases and their ability to be mixed without adverse effects (**table 6** and **table 7**). If incompatible greases are mixed, the consistency of the grease mix can change dramatically such that bearing damage because of severe leakage could result. Note that PTFE-thickened greases are not compatible with other grease types.

Miscibility with preservation oils

The preservative oils with which SKF bearings are treated are compatible with the majority of lubricating greases, with the exception of synthetic fluorinated oil based greases using a PTFE thickener, for example, SKF LGET 2 grease. For PTFE-thickened greases, the bearing preservatives must be removed before applying the grease. White spirit is recommended as a solvent. Make sure all remnants of solvent have evaporated and then immediately apply the grease.

Assessing the suitability of non-SKF greases

Greases from suppliers other than SKF must be approved by the supplier. Use **diagram 6**, **page 120**, to evaluate the temperature performance and grease life prediction. Where relevant, take into account the considerations specified for SKF greases.

Table 6

Compatibility of base oil types

	Mineral oil	Ester oil	Polyglycol	Silicone-methyl	Silicone-phenyl	Polyphenyl-ether
Mineral oil	+	+	–	–	+	o
Ester oil	+	+	+	–	+	o
Polyglycol	–	+	+	–	–	–
Silicone-methyl	–	–	–	+	+	–
Silicone-phenyl	+	+	–	+	+	+
Polyphenylether	o	o	–	–	+	+

+ compatible
- incompatible
o individual testing required

Table 7

Compatibility of thickener types

	Lithium soap	Calcium soap	Sodium soap	Lithium complex soap	Calcium complex soap	Sodium complex soap	Barium complex soap	Aluminium complex soap	Clay	Polyurea
Lithium soap	+	o	–	+	–	o	o	–	o	o
Calcium soap	o	+	o	+	–	o	o	–	o	o
Sodium soap	–	o	+	o	o	+	+	–	o	o
Lithium complex soap	+	+	o	+	+	o	o	+	–	–
Calcium complex soap	–	–	o	+	+	o	–	o	o	+
Sodium complex soap	o	o	+	o	o	+	+	–	–	o
Barium complex soap	o	o	+	o	–	+	+	+	o	o
Aluminium complex soap	–	–	–	+	o	–	+	+	–	o
Clay	o	o	o	–	o	–	o	–	+	o
Polyurea	o	o	o	–	+	o	o	o	o	+

+ compatible
- incompatible
o individual testing required

Lubrication systems

Continuous lubrication can be achieved via singlepoint or multipoint automatic lubricators, e.g. SKF's SYSTEM 24 or SYSTEM MultiPoint.

Centralized lubrication systems, such as SKF MonoFlex, SKF ProFlex, SKF DuoFlex, SKF MultiFlex (**table 8**) and Lincoln Centro Matic, Quicklub and Dual Line can reliably deliver grease in a wide range of quantities.

For additional information about SKF lubrication systems, refer to skf.com/lubrication.

Selecting a suitable oil

Oil selection criteria

When you select a lubricating oil, the most important parameters are the viscosity and viscosity index, the temperature stability (which influences the choice of oil type) and the additive package (EP/AW and corrosion protection) that fits the operating conditions for the application.

Viscosity and viscosity index

The required viscosity is primarily given by the lubrication condition κ, at the expected operating temperature, evaluated as described in *Lubrication condition – the viscosity ratio, κ*, **page 102**. The viscosity index, VI, is the measure of how the oil viscosity changes with temperature. VI is a part of the selection process, in particular for applications that operate in a large temperature range. Oils with a VI of at least 95 are recommended.

Oil type

There are two broad categories of oil types – mineral and synthetic – with the following types of synthetic oils available:

- polyalphaolefins (PAO)
- esters
- polyglycols (PAG)

Choice of oil type is mainly determined by the temperature range in which the application is expected to operate.

- Mineral oils are generally favoured as the lubricant for rolling bearings.
- Synthetic oils should be considered for operational temperatures above 90 °C *(195 °F)* because of their improved thermal and oxidation resistance, or below –40 °C *(–40 °F)* because of their better properties at low temperatures.

The pour point of an oil is defined as the lowest temperature at which a lubricant will flow, but it must not be used as a functional limit when selecting oil type. If the temperature is above but near the pour point, the viscosity is still very high, which may impair pumping, filtering, and other characteristics.

The thickness of the hydrodynamic film is determined, in part, by the viscosity index (VI) and the pressure-viscosity coefficient. For most mineral oil based lubricants, the pressure-viscosity coefficient is similar, and you can use the generic values obtained from literature. However, for synthetic oils, the effect on viscosity to increasing pressure is determined by the chemical structure of its base stock. As a result, there is considerable variation in pressure-viscosity coefficients for different types of synthetic base stocks.

Because of the differences in the viscosity index and pressure-viscosity coefficient, the formation of a hydrodynamic lubricant film, when using a synthetic oil, may differ from that of a mineral oil with the same viscosity.

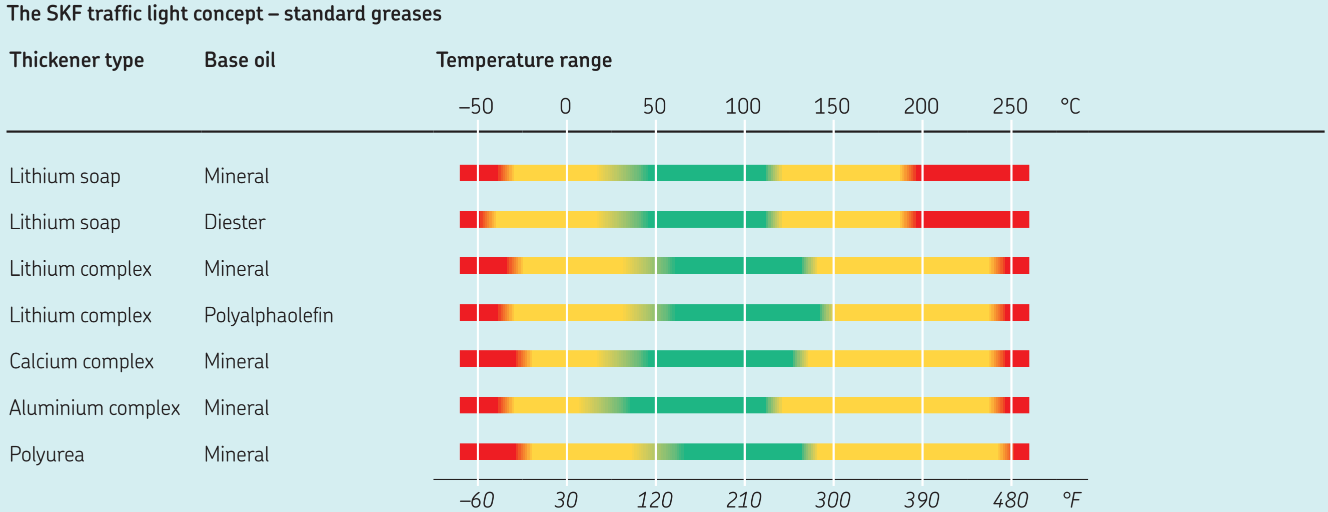
Regarding the lubrication condition for mineral and synthetic oils, the combined effect of the viscosity index and the pressure-viscosity coefficient normally cancel each other out.

The properties of the different oil types are summarized in **table 9**. For additional information about synthetic oils, contact the lubricant supplier.

Oils, and in particular synthetic oils, may interact with such things as seals, paint or water in a different way than mineral oils, so such effects, as well as miscibility, must be investigated.

Diagram 6

The SKF traffic light concept – standard greases



Additives

Lubricating oils usually contain additives of various kinds. The most important ones are antioxidants, corrosion protection agents, anti-foaming additives, and EP/AW additives. In the lubrication condition domain defined by $\kappa < 1$, EP/AW additives are recommended, but for temperatures above 80 °C *(175 °F)*, a lubricant with EP/AW additives should only be used after careful testing.

Oil change interval

The oil change interval depends on the operating conditions and the oil type. With oil-bath lubrication, it is generally sufficient to change the oil once a year, provided the operating temperature does not exceed 50 °C *(120 °F)*. Typically, at higher temperatures or with heavy contamination, the oil must be changed more often.

With oil circulation, the interval after which the oil needs to be changed is determined by an inspection of the oil quality, taking into account oxidation and the presence of water and abrasive particles. Oil life in circulation systems can be extended by removing particles and water from the oil.

A summary of oil change intervals for various systems and conditions is shown in **table 10**, **page 122**.

Table 8

SKF Centralized Lubrication Systems

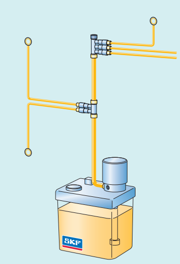
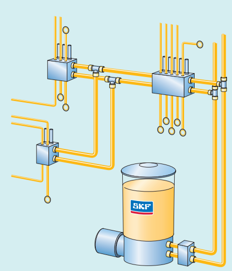
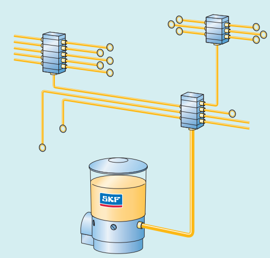
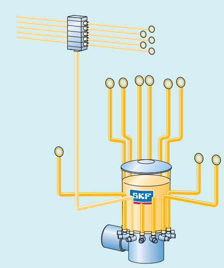
	SKF MonoFlex	**SKF DuoFlex**	**SKF ProFlex**	**SKF MultiFlex**
				
Type	Single-line	Dual-line	Progressive	Multi-line
Suitable lubricants	Oil Grease with NLGI grades from 000 to 2	Oil Grease with NLGI grades from 000 to 3	Oil Grease with NLGI grades from 000 to 2	Oil Grease with NLGI grades from 000 to 3
Application examples	Machine tools, printing, textile and off-highway applications	Metal working machines, pulp and paper industry, mining and cement plants, deck cranes, power plants	Printing and industrial presses machines, off-highway applications, wind turbines	Oil and gas industry, heavy industrial applications

Table 9

Properties of lubricating oil types

Properties		**Base oil type** Mineral	PAO	Ester	PAG
Pour point	[°C] *[°F]*	–30 .. 0 *–20 .. 30*	–50 .. –40 *–60 .. –40*	–60 .. –40 *–75 .. –40*	approx. –30 *approx. –20*
Viscosity index		low	moderate	high	high
Pressure-viscosity coefficient		high	moderate	low to moderate	moderate

Overview of main oil lubrication methods

The oil lubrication methods are:

- oil bath without circulating oil
- oil bath with self-circulating oil through bearing pumping action
- circulating oil with external pump
- oil jet method
- oil air method

The choice of the oil lubrication method depends mainly on:

- the bearing speed
- the need to remove heat
- the need to remove contaminants (solid particles or liquid)

SKF offers a wide range of products for oil lubrication that are not covered here. For additional information about SKF lubrication systems and related products, refer to skf.com/lubrication.

Oil bath without circulating oil

The simplest method of oil lubrication is the oil bath. The oil, which is picked up by the rotating components of the bearing, is distributed within the bearing and then flows back to the oil bath in the housing. Ideally, the oil level should reach the centre of the lowest rolling element (**fig. 3**) when the bearing is stationary. Oil levels higher than recommended will increase bearing temperature because of churning (*Bearing friction, power loss and starting torque*, **page 132**).

Oil bath with self-circulating oil

Oil from a bath is forced to circulate by different methods. Here are some examples:

- Oil is salvaged and directed to the bearings by means of drain and ducts (**fig. 4**).
- A dedicated component (ring, disc, etc.) picks up oil from an oil bath and transports it (**fig. 5**).
- The pumping effect of some bearing types can be used to circulate the oil. In **fig. 6**, the spherical thrust roller bearing pumps oil which returns to the thrust bearing by connecting ducts located under it.

All designs of such lubricating methods should be validated individually by tests.

Circulating oil without a bath

Circulating oil by means of an external oil pump, instead of an oil bath, is mainly used when it is needed to remove heat generated by the bearing and/or other sources. Oil circulation is also a good lubricating method for evacuating solid or liquid contaminants from the bearing to filters and/or oil/liquid separators. The design and layout of the oil drainage must ensure that there is no build-up of oil level (*Heat flow from adjacent parts or process*, **page 131**).

A basic circulating oil system (**fig. 7**) includes:

- oil pump
- filter
- oil reservoir
- oil cooling and/or heating system

Oil jet

The oil jet lubricating method (**fig. 8**) is an extension of circulating oil systems, and is used for bearings operating at very high speeds. The dimensioning of oil flow and corresponding jet size is selected so that the oil jet speed reaches at least 15 m/s.

Oil injectors must be positioned so that the oil jet penetrates the bearing between one of the rings and the cage. To prevent churning that can cause increased friction and temperature, the design and layout of the oil drainage must ensure that there is no oil level build-up.

Oil-air

The oil-air lubrication method (**fig. 9**), also called the oil-spot lubrication method, uses compressed air to transport small, accurately-metered quantities of oil as small droplets along the inside of the feed lines to an injector nozzle, where it is delivered to a bearing. This minimum-quantity lubrication method enables the bearings to operate at very high speeds at a relatively low operating temperature. The compressed air also cools the bearing and prevents dust or aggressive gases from entering. For additional information, refer to skf.com/super-precision.

Table 10

Summary of oil change intervals for different oil systems and operating conditions

Oil lubrication system	Typical operating conditions	Approximate oil change interval[1]
Oil bath or oil pick-up ring	Operating temperature < 50 °C *(120 °F)* Little risk of contamination	12 months
	Operating temperature 50 to 100 °C *(120 to 210 °F)* Some contamination	3 to 12 months
	Operating temperature > 100 °C *(210 °F)* Contaminated environment	3 months
Circulating oil or oil jet	All	Determined by test runs and regular inspection of the oil condition. Dependent on how frequently the total oil quantity is circulated and whether or not the oil is cooled.

1) More frequent oil changes are needed if the operating conditions are more demanding.

Fig. 3

Oil bath

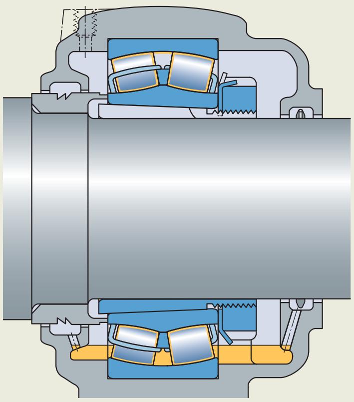


Fig. 4

Self-circulating oil by drain and ducts

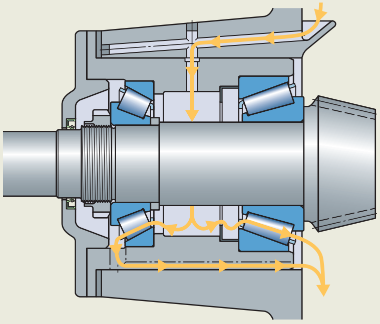


Fig. 5

Oil pick-up rings on SONL housing

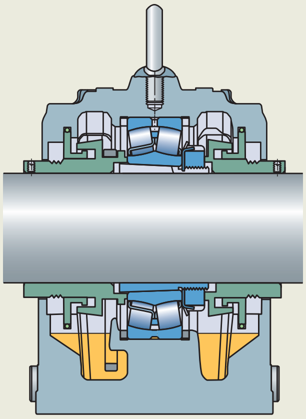


Fig. 6

Pumping effect in vertical shaft application

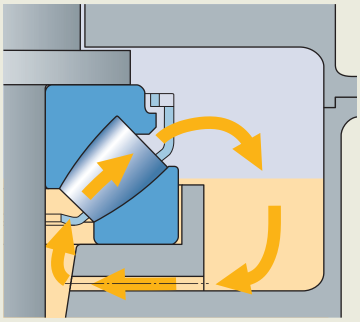


Fig. 8

Oil jet

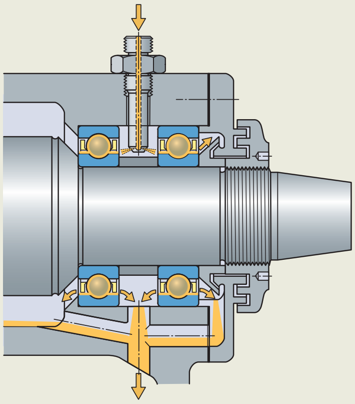


Fig. 7

Circulating oil

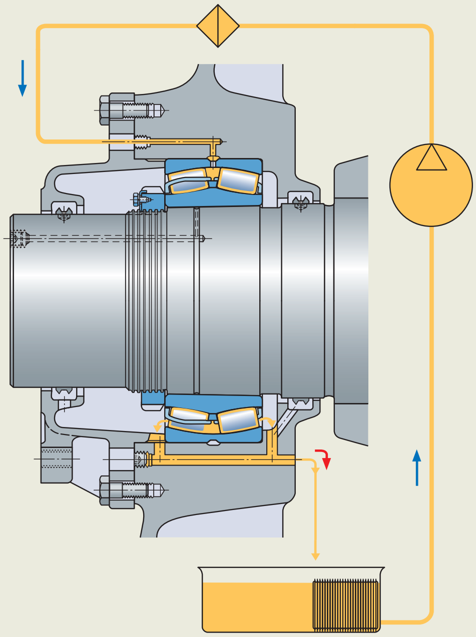
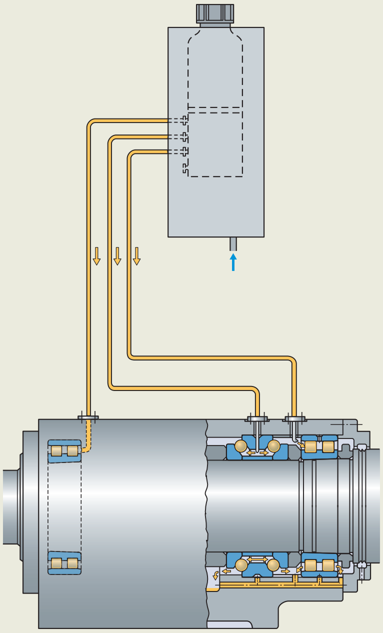


Fig. 9

Oil-air



B.4 Lubrication

SKF bearing grease selection chart

Grease	Description	Application example	Temperature range[1] LTL	HTPL	Temp.	Speed
LGMT 2	General purpose industrial and automotive	Automotive wheel bearings Conveyors and fans Small electric motors	–30 °C (–20 °F)	120 °C (250 °F)	M	M
LGMT 3	General purpose industrial and automotive	Bearings with d > 100 mm Vertical shaft or outer bearing ring rotation Car, truck and trailer wheel bearings	–30 °C (–20 °F)	120 °C (250 °F)	M	M
LGEP 2	Extreme pressure	Forming and press section of paper mills Work roll bearings in steel industry Heavy machinery, vibrating screens	–20 °C (–5 °F)	110 °C (230 °F)	M	L to M
LGWA 2	Wide temperature[3], extreme pressure	Wheel bearings in cars, trailers and trucks Washing machines Electric motors	–30 °C (–20 °F)	140 °C (285 °F)	M to H	L to M
LGGB 2	Biodegradable, low toxicity[4]	Agricultural and forestry equipment Construction and earthmoving equipment Water treatment and irrigation	–40 °C (–40 °F)	90 °C (195 °F)	L to M	L to M
LGFP 2	Food compatible	Food processing equipment Wrapping machines Bottling machines	–20 °C (–5 °F)	110 °C (230 °F)	M	M
LGFQ 2	Food compatible High load	Pellet presses Mills Mixers	–40 °C (–40 °F)	140 °C (285 °F)	L to H	VL to M
LGBB 2	Wind turbine blade and yaw bearing grease	Wind turbine blade and yaw slewing bearings	–40 °C (–40 °F)	120 °C (250 °F)	L to M	VL
LGLT 2	Low temperature, extremely high speed	Textile and machine tool spindles Small electric motors and robots Printing cylinders	–50 °C (–60 °F)	110 °C (230 °F)	L to M	M to EH
LGWM 1	Extreme pressure, low temperature	Main shaft of wind turbines Centralised lubrication systems Spherical roller thrust bearing applications	–30 °C (–20 °F)	110 °C (230 °F)	L to M	L to M
LGWM 2	High load, wide temperature	Main shaft of wind turbines Heavy duty off road or marine applications Snow exposed applications	–40 °C (–40 °F)	110 °C (230 °F)	L to M	L to M
LGEM 2	High viscosity plus solid lubricants	Jaw crushers Construction machinery Vibrating machinery	–20 °C (–5 °F)	120 °C (250 °F)	M	VL
LGEV 2	Extremely high viscosity with solid lubricants	Trunnion bearings Support and thrust rollers on rotary kilns and dryers Slewing ring bearings	–10 °C (–15 °F)	120 °C (250 °F)	M	VL
LGHB 2	EP high viscosity, high temperature[5]	Steel on steel plain bearings Dryer section of paper mills Work roll bearings and continuous casting in steel industry Sealed spherical roller bearings up to 150 °C (300 °F)	–20 °C (–5 °F)	150 °C (300 °F)	M to H	VL to M
LGHP 2	High performance polyurea grease	Electric motors Fans, even at high speed High speed ball bearings at medium and high temperatures	–40 °C (–40 °F)	150 °C (300 °F)	M to H	M to H
LGED 2	High temperature Harsh environment	Bakery/brick oven equipment Glass industry Vacuum pumps	–30 °C (–20 °F)	240 °C (464 °F)	VH	L to M
LGET 2	Extreme temperature	Bakery equipment (ovens) Wafer baking machines Textile dryers	–40 °C (–40 °F)	260 °C (500 °F)	VH	L to M

1) LTL = Low Temperature Limit. Defined by means of the IP 186 Low temperature torque test. HTPL = High Temperature Performance Limit
2) mm^2/s at 40 °C (105 °F) = cSt.
3) LGWA 2 can withstand peak temperatures of 220 °C (430 °F)
4) LGGB 2 can withstand peak temperatures of 120 °C (250 °F)
5) LGHB 2 can withstand peak temperatures of 200 °C (390 °F)

Load	Thickener / base oil	NLGI	Base oil viscosity[2)]	Vertical shaft	Fast outer ring rotation	Oscillating movements	Severe vibrations	Peak loads or frequent startup	Rust inhibiting properties
L to M	Lithium soap / mineral oil	2	110	●			+		+
L to M	Lithium soap / mineral oil	3	125	+	●		+		●
H	Lithium soap / mineral oil	2	200	●		●	+	+	+
L to H	Lithium complex soap / mineral oil	2	185	●	●	●	●	+	+
M to H	Lithium-calcium soap / synthetic ester oil	2	110	●		+	+	+	●
L to M	Aluminium complex / medical white oil	2	150	●					+
L to VH	Complex calcium sulphonate/PAO	1–2	320	●	●	+	+	+	+
M to H	Lithium complex soap / synthetic PAO oil	2	68			+	+	+	+
L	Lithium soap / synthetic PAO oil	2	18	●				●	●
H	Lithium soap / mineral oil	1	200			+		+	+
L to h	Complex calcium sulphonate / synthetic PAO oil / mineral oil	1–2	80	●	●	+	+	+	+
H to VH	Lithium soap / mineral oil	2	500	●		+	+	+	+
H to VH	Lithium-calcium soap / mineral oil	2	1 020	●		+	+	+	+
L to VH	Complex calcium sulphonate / mineral oil	2	425	●	+	+	+	+	+
L to M	Di-urea / mineral oil	2 to 3	96	+			●	●	+
H to VH	PTFE / synthetic fluorinated polyether oil	2	460	●	●	+	●	●	●
H to VH	PTFE / synthetic fluorinated polyether oil	2	400	●	+	+	●	●	●

Wide application greases

Special requirements

Low temperature

High loads

High temperature

● = Suitable ✚ = Recommended

B.4 Lubrication

Technical specifications for SKF greases

		LGMT 2	LGMT 3	LGEP 2	LGWA 2	LGGB 2	LGFP 2	LGFQ 2
DIN 51825 code		K2K-30	K3K-30	KP2G-20	KP2N-30	KPE 2K-40	K2G-20	KP1/2N-40
NLGI consistency grade		2	3	2	2	2	2	1–2
Thickener		Lithium	Lithium	Lithium	Lithium complex	Lithium/calcium	Aluminium complex	Complex calcium sulphonate
Colour		Red brown	Amber	Light brown	Amber	Off white	Transparent	Brown
Base oil type		Mineral	Mineral	Mineral	Mineral	Synthetic (Ester)	Medical white oil	Synthetic (PAO)
Operating temperature range	°C *°F*	–30 to +120 *(–20 to +250)*	–30 to +120 *(–20 to +250)*	–20 to +110 *(–5 to +230)*	–30 to +140 *(–20 to +285)*	–40 to +90 *(–40 to +195)*	–20 to +110 *(–5 to +230)*	–40 to +140 *(–40 to +284)*
Dropping point DIN ISO 2176	°C *°F*	>180 *(>355)*	>180 *(>355)*	>180 *(>355)*	>250 *(>480)*	>170 *(>340)*	>250 *(>480)*	>300 *(>570)*
Base oil viscosity 40 °C *(105 °F)* 100 °C *(210 °F)*	mm^2/s mm^2/s	110 11	125 12	200 16	185 15	110 13	150 15,3	320 30
Penetration DIN ISO 2137 60 strokes 100 000 strokes	10^{-1} mm 10^{-1} mm	265–295 +50 max. (325 max.)	220–250 280 max.	265–295 +50 max. (325 max.)	265–295 +50 max. (325 max.)	265–295 +50 max. (325 max.)	265–295 +30 max.	280–310 +30 max.
Mechanical stability Roll stability, 50 h at 80 °C *(175 °F)* V2F test	10^{-1} mm	+50 max. "M"	295 max. "M"	+50 max. "M"	+50 max. change "M"	+70 max. (350 max.)		–20 to +30 max.
Corrosion protection Emcor: – standard ISO 11007 – water washout test – salt water test (100% seawater)		0–0 0–0 0–1 [1]	0–0 0–0	0–0 0–0 1–1 [1]	0–0 0–0 [1]	0–0	0–0 [1]	0–0 0–0
Water resistance DIN 51 807/1, 3 h at 90 °C *(195 °F)*		1 max.	2 max.	1 max.	1 max.	0 max.	1 max.	1 max.
Oil separation DIN 51 817, 7 days at 40 °C *(105 °F)*, static	%	1–6	1–3	2–5	1–5	0,8–3	1–5	3 max.
Lubrication ability R2F, running test B at 120 °C *(250 °F)* R2F, cold chamber test, –30 °C *(–20 °F)*, +20 °C *(+70 °F)*		Pass	Pass	Pass	Pass 100 °C *(210 °F)*	Pass 100 °C *(210 °F)* [1]		Pass
Copper corrosion DIN 51 811		2 max. 110 °C *(230 °F)*	2 max. 130 °C *(265 °F)*	2 max. 110 °C *(230°F)*	2 max. 100 °C *(210°F)*		1 max. 120 °C *(250 °F)*	1b max. 100 °C *(210°F)*
Rolling bearing grease life R0F test L_{50} life at 10 000 r/min	h		1 000 min., 130 °C *(265 °F)*			>300, 120 °C *(250 °F)*	1 000, 110 °C *(230 °F)* [1]	
EP performance Wear scar DIN 51350/5, 1 400 N 4–ball test, welding load DIN 51350/4	mm N			1,4 max. 2 800 min.	1,6 max. 2 600 min.	1,8 max. 2 600 min.	1 100 min.	1 max. >4 000
Fretting corrosion ASTM D4170 FAFNIR test at +25 °C *(75 °F)*	mg			5,7 [1]				0,8 [1]
Low temperature torque IP186, starting torque IP186, running torque	Nmm [1] Nmm [1]	98, –30 °C *(–20 °F)* 58, –30 °C *(–20 °F)*	145, –30 °C *(–20 °F)* 95, –30 °C *(–20 °F)*	70, –20 °C *(–5 °F)* 45, –20 °C *(–5 °F)*	40, –30 °C *(–20 °F)* 30, –30 °C *(–20 °F)*		137, –30 °C *(–20 °F)* 51, –30 °C *(–20 °F)*	369, –40° C (–40°F) 223, –40 °C (–40°F)

Special requirements

Wide applications greases

1) Typical value

LGBB 2	LGLT 2	LGWM 1	LGWM 2	LGEM 2	LGEV 2	LGHB 2	LGHP 2	LGED 2	LGET 2
KP2G-40	K2G-50	KP1G-30	KP2G-40	KPF2K-20	KPF2K-10	KP2N-20	K2N-40	KFK2U-30	KFK2U-40
2	2	1	1–2	2	2	2	2–3	2	2
Lithium complex	Lithium	Lithium	Complex calcium sulphonate	Lithium	Lithium/calcium	Complex calcium sulphonate	Di-urea	PTFE	PTFE
Yellow	Beige	Brown	Yellow	Black	Black	Brown	Blue	Off white	Off white
Synthetic (PAO)	Synthetic (PAO)	Mineral	Synthetic (PAO)/ Mineral	Mineral	Mineral	Mineral	Mineral	Synthetic (fluorinated polyether)	Synthetic (fluorinated polyether)
–40 to +120 *(–40 to +250)*	–50 to +110 *(–60 to +230)*	–30 to +110 *(–20 to +230)*	–40 to +110 *(–40 to +230)*	–20 to +120 *(–5 to +250)*	–10 to +120 *(15 to 250)*	–20 to +150 *(–5 to +300)*	–40 to +150 *(–40 to +300)*	–30 to +240 *(–22 to +464)*	–40 to +260 *(–40 to +500)*
>200 *(390)*	>180 *(>355)*	>170 *(>340)*	>300 *(>570)*	>180 *(>355)*	>180 *(>355)*	>220 *(>430)*	>240 *(>465)*	>300 *(>570)*	>300 *(>570)*
68	18 4,5	200 16	80 8,6	500 32	1 020 58	425 26,5	96 10,5	460 42	400 38
265–295 +50 max.	265–295 +50 max.	310–340 +50 max.	280–310 +30 max	265–295 325 max.	265–295 325 max.	265–295 –20 to +50 (325 max.)	245–275 365 max.	265–295 271 1)	265–295 –
+50 max.			+50 max.	345 max. "M"	+50 max. "M"	–20 to +50 "M"	365 max.		±30 max. 130 °C *(265 °F)*
0–0 0–1 1)	0–1	0–0 0–0	0–0 0–0 0–0 1)	0–0 0–0	0–0 0–0 1) 0–0 1)	0–0 0–0 0–0 1)	0–0 0–0 0–0	0–0 1)	1–1 max.
1 max.	1 max.	1 max.	1 max.	1 max.	1 max.	1 max.	1 max.	1 max.	0 max.
4 max, 2,5 1)	<4	8–13	3 max.	1–5	1–5	1–3, 60 °C *(140 °F)*	1–5 1)		13 max. 30 h 200 °C *(390 °F)*
			Pass, 140 °C *(285 °F)* Pass, Pass	Pass, 100 °C *(210 °F)*		Pass, 140 °C *(285 °F)*	Pass		
1 max. 120 °C *(250 °F)*	1 max. 100 °C *(210 °F)*	2 max. 90 °C *(>195 °F)*	2 max. 100 °C *(210 °F)*	2 max. 100 °C *(210 °F)*	1 max. 100 °C *(210 °F)*	2 max. 150 °C *(300 °F)*	1 max. 150 °C *(300 °F)*	1 max. 100 °C *(210 °F)* 1)	1 max. 150 °C *(300 °F)*
	>1 000, 20 000 r/min 100 °C *(210 °F)*		1 824 1), 110 °C *(230 °F)*			>1 000, 130 °C *(265 °F)*	1 000 min. 150 °C *(300 °F)*	>700 at 220 °C *(430 °F)*	>1 000 1) at 220 °C *(428 °F)*
0,4 1) 5 500 1)	2 000 min.	1,8 max. 3 200 min. 1)	1,5 max. 1) 4 000 min. 1)	1,4 max. 3 000 min.	1,2 max. 3 000 min.	0,86 1) 4 000 min.		8 000 min.	8 000 min.
0–1 1)		5,5 1)	5,2 / 1,1 at –20 °C *(–5 °F)* 1)			0 1)	7 1)		
313, –40 °C *(–40 °F)* 75, –40 °C *(–40 °F)*	32, –50 °C *(–60 °F)* 21, –50 °C *(–60 °F)*	178, 0 °C *(32 °F)* 103, 0 °C *(32 °F)*	249, –40 °C *(–40 °F)* 184, –40 °C *(–40 °F)*	160, –20 °C *(–5 °F)* 98, –20 °C *(–5 °F)*	96, –10 °C *(14 °F)* 66, –10 °C *(14 °F)*	250, –20 °C *(–5 °F)* 133, –20 °C *(–5 °F)*	1 000, –40 °C *(–40 °F)* 280, –40 °C *(–40 °F)*		

High loads

Low temperatures

High temperatures

B.5 Operating temperature and speed

B.5 Operating temperature and speed

B.5 Operating temperature and speed

The relationships between the temperature and power loss of components within an application is complex and these factors, in turn, have interdependencies with many others such as bearing sizes, loads and lubrication conditions.

They influence many performance characteristics of an application and its parts, and do so in various ways depending on the operational state, such as at start-up or in normal operation, when steady-state conditions have been reached.

Estimating the operating temperature and verifying speed limitations is a critical aspect of the analysis of an application.

This section provides details of these primary relationships, and guidance on what to consider.

Bearing operating temperature and heat flow

Temperature has a major influence on many performance characteristics of an application. The heat flow to, from and within an application determines the temperature of its parts.

The operating temperature of a bearing is the steady-state temperature it attains when running and in thermal equilibrium with its surrounding elements. The operating temperature results from (**diagram 1**):

- the heat generated by the bearing, as a result of the combined bearing and seal frictional power loss
- the heat from the application transferred to the bearing via the shaft, housing, foundation and other elements in its surroundings
- the heat dissipated from the bearing via the shaft, housing, foundation, lubricant cooling system (if used) and other cooling devices

The bearing operating temperature depends as much on the application design as on the bearing generated friction. Therefore, the bearing, its adjacent parts and the application should all be thermally analysed.

Bearing size, operating temperature and lubrication conditions

For a given bearing type, the bearing size, operating temperature and lubrication conditions are interdependent as follows (**diagram 2**):

- Bearing size is selected based on bearing load, speed and lubrication conditions.
- Operating temperature is a function of the bearing load, size, speed and lubrication conditions.
- Lubrication conditions depend on the operating temperature, the viscosity of the lubricant and the speed.

These interdependencies are dealt with by taking an iterative approach to the analysis, in order to achieve an optimum design for a bearing arrangement and select the most appropriate components for it.

Thermal equilibrium

The operating temperature of a bearing reaches a steady state when there is thermal equilibrium – i.e. there is a balance between generated heat and dissipated heat.

Provided that the load ratio C/P > 10 and the speed is below 50% of the limiting speed n_{lim}, and there is no pronounced external heat input, then cooling via the surrounding air and foundation is usually sufficient to result in an operating temperature well below 100 °C (*210 °F*). Where these conditions are not met, perform a more detailed analysis, as additional heat dissipation may be required.

Generated heat

The heat generated is the sum of:

- heat generated by the bearing, as a result of the combined bearing and seal frictional power loss
- heat flow from adjacent parts or processes

Bearing frictional heat (power loss)

Bearing friction consists mainly of rolling friction, sliding friction, seal friction and oil drag losses (*Bearing friction, power loss and starting torque*, **page 132**).

Heat flow from adjacent parts or processes

In many applications, the bearings are in locations where they receive:

- heat from working parts of the machine, e.g. caused by friction in gears or shaft seals
- external heat, e.g. from hot steam going through a hollow shaft

The operating temperature of the bearings is influenced by this, in addition to their self-generated heat. Examples of such applications include:

- drying cylinders in paper machines
- calender rolls in plastic foil machines
- compressors
- hot gas fans

The heat input from adjacent parts within the application or from the process can be very pronounced and is typically very difficult to estimate. The rule is to insulate the bearing, as far as possible, from the additional heat flow.

Diagram 1

Bearing operating temperature as equilibrium between generated heat and dissipated heat

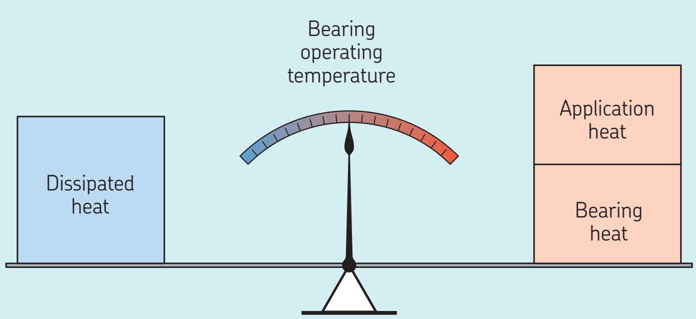
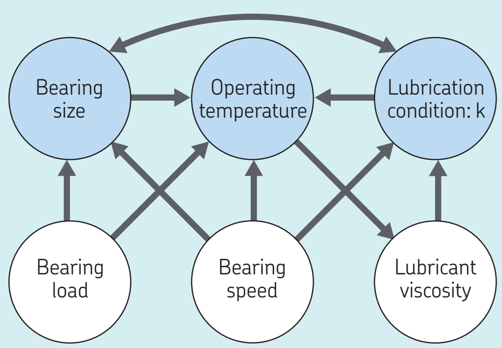


Diagram 2

Dependencies between bearing size, operating temperature and lubrication conditions



Dissipated heat

The heat dissipated is the sum of:

- heat dissipated by the shaft, housing and ambient airflow, e.g. cooling effects in arctic conditions
- heat dissipated via the lubricant or lubrication system

Bearing friction, power loss and starting torque

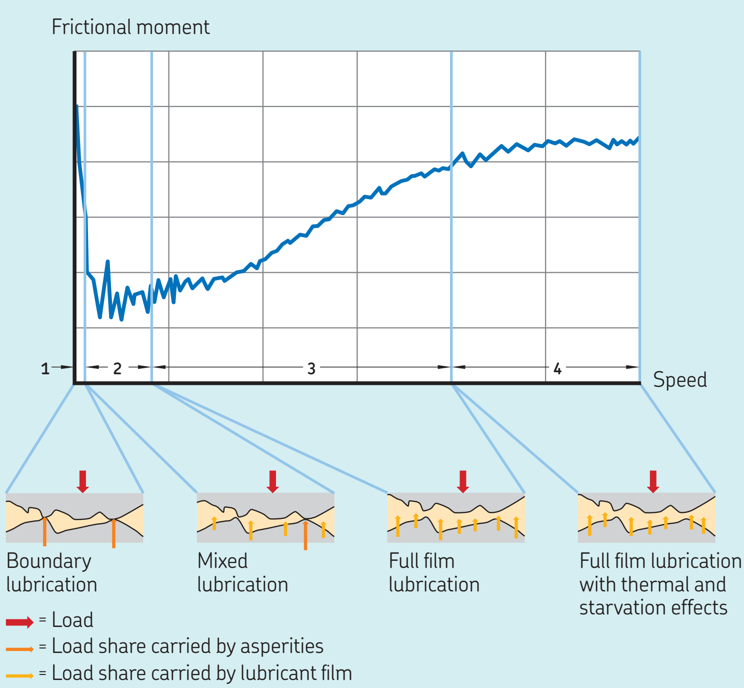
Bearing friction is not constant and depends on certain tribological phenomena that occur in the lubricant film between the rolling elements, raceways and cages.

Friction changes as a function of speed, in a bearing with a given lubricant, are shown in **diagram 3**. Four zones are distinguishable:

- **Zone 1 – Boundary lubrication condition**, in which only the asperities carry the load, and so friction between the moving surfaces is high.
- **Zone 2 – Mixed lubrication condition**, in which a separating oil film carries part of the load, with fewer asperities in contact, and so friction decreases.
- **Zone 3 – Full film lubrication condition**, in which the lubricant film carries the load, but with increased viscous losses, and so friction increases.
- **Zone 4 – Full film lubrication with thermal and starvation effects**, in which the inlet shear heating and kinematic replenishment reduction factors compensate partially for the viscous losses, and so friction evens off.

Diagram 3

Bearing frictional moment as a function of speed



SKF model of bearing friction

In the SKF model for calculating bearing friction, the total frictional moment, M, is derived from four sources:

$M = M_{rr} + M_{sl} + M_{seal} + M_{drag}$

where

M_{rr} = the rolling frictional moment, and includes effects of lubricant starvation and inlet shear heating [Nmm]

M_{sl} = the sliding frictional moment, and includes the effects of the quality of lubrication conditions [Nmm]

M_{seal} = the frictional moment from integral seals [Nmm]
Where bearings are fitted with contact seals, the frictional losses from the seals may exceed those generated in the bearing.

M_{seal} = the frictional moment from drag losses, churning, splashing, etc., in an oil bath [Nmm]

Calculating values for these four sources of friction is complex. Therefore, we recommend using the *SKF Bearing Calculator* (skf.com/bearingcalculator).

For detailed information on the calculations, refer to *The SKF model for calculating the frictional moment* (skf.com/go/17000-B5).

When the total frictional moment, M, of the bearing is known, you can calculate the bearing frictional power loss using

$P_{loss} = 1,05 \times 10^{-4}\, M\, n$

where

P_{loss} = bearing frictional power loss [W]
M = total frictional moment [Nmm]
n = rotational speed [r/min]

B.5 Operating temperature and speed

SKF

Starting torque

The starting torque of a rolling bearing is defined as the frictional moment that must be overcome by the bearing to start rotating, at an ambient temperature of 20 to 30 °C *(70 to 85 °F)*. Therefore, only the sliding frictional moment and the frictional moment of seals, if applied, are taken into consideration.

$M_{start} = M_{sl} + M_{seal}$

where
M_{start} = starting frictional moment [Nmm]
M_{sl} = sliding frictional moment [Nmm]
M_{seal} = frictional moment of the seals [Nmm]

We recommend using the *SKF Bearing Calculator* (skf.com/bearingcalculator) for calculating starting torque values.

Estimating heat dissipation from SKF plummer (pillow) blocks

For SKF plummer (pillow) block housings, you can use a model based on bearing size to estimate heat dissipation values.

Using **diagram 4**, you can estimate the heat dissipation per degree above ambient temperature, W_s, for a bearing with bearing mean diameter d_m in a plummer block housing, with the shaft exposed to the surrounding air.

The estimation is valid for SKF plummer block housings used with grease or oil bath lubrication and only where there is no significant heat input from external sources, such as steam heating of shafts or pronounced radiation from hot surfaces.

Estimating bearing operating temperature

If you are able to estimate a value for the heat dissipation from a bearing, W_s, you can estimate the operating temperature, T_{bear}, for a bearing in thermal equilibrium, under steady-state conditions, using

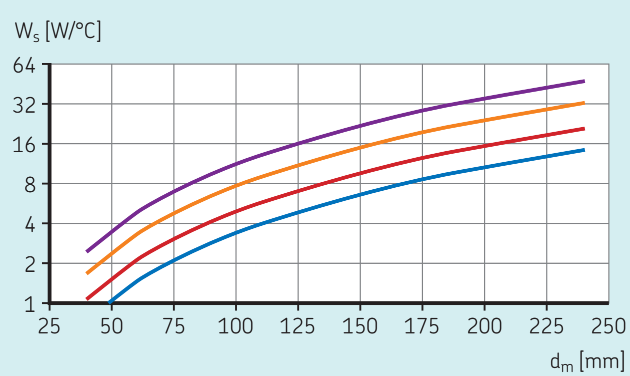
$T_{bear} = (P_{loss} / W_s) + T_{amb}$

where
T_{bear} = estimated average bearing operating temperature [°C]
P_{loss} = bearing frictional power loss [W]
W_s = total heat dissipation per degree above ambient temperature [W/°C]
T_{amb} = ambient temperature [°C]

Should the value of the estimated bearing operating temperature be too high for the application requirements – for example, resulting in a κ value that is too low, or a relubrication interval that is too short – a possible solution may be to reduce the operating temperature by means of a circulating oil lubrication system.

Diagram 4

Heat dissipation for SKF plummer block housings



Key	Foundation material	Velocity of surrounding air	Dissipation method
		m/s	
	concrete	0,5	by natural airflow
	steel	0,5	
	steel	2,5	by forced airflow
	steel	5	

Cooling via circulating oil

By circulating the oil, it is possible to cool it, and thereby remove heat from the bearing arrangement.

In **diagram 5**, the curved line shows the bearing frictional power loss, P_{loss}, and the angled line shows the heat dissipation, W_s.

Taking the heat dissipated via oil circulation into account, the bearing thermal equilibrium under steady-state conditions becomes:

$$P_{loss} = W_s (T_{bear} - T_{amb}) + P_{oil}$$

where
P_{loss} = bearing frictional power loss [W]
W_s = total heat dissipation per degree above ambient temperature [W/°C]
T_{bear} = estimated required bearing operating temperature [°C]
T_{amb} = the ambient temperature [°C]
P_{oil} = estimated power dissipated in the oil cooler [W]

Taking the heat dissipation via oil circulation into account, you can estimate the bearing operating temperature using

$$T_{bear} = ((P_{loss} - P_{oil}) / W_s) + T_{amb}$$

You can estimate the power that must be dissipated by oil cooling, for a given bearing temperature, using

$$P_{oil} = P_{loss} - W_s (T_{bear} - T_{amb})$$

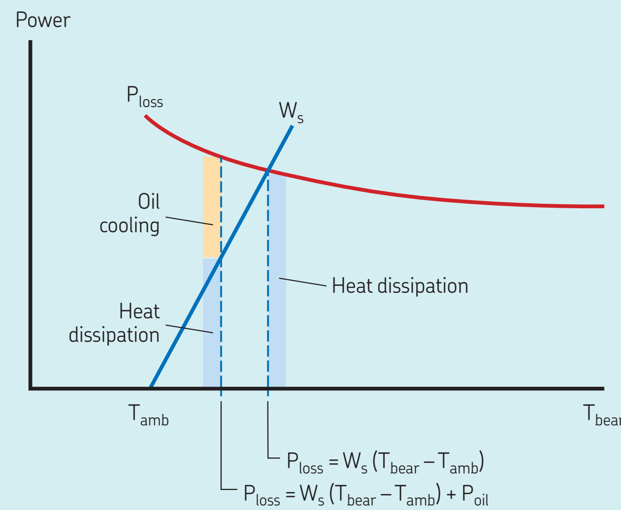
You can estimate the required oil flow, for a given quantity of power that must be dissipated by oil cooling (P_{oil}), using

$$Q = P_{oil} / (27 (T_{out} - T_{in}))$$

where
Q = required oil flow [l/min]
P_{oil} = power dissipated in the oil cooler [W]
T_{out} = oil temperature at the housing oil outlet [°C]
T_{in} = oil temperature at the housing oil inlet [°C]

Diagram 5

Relationship between power loss, heat dissipation and temperature



If you do not have values for T_{out} or T_{in}, you may assume a temperature difference of 5 to 10 °C *(10 to 20 °F)*.

The limit of cooling that is possible via circulating oil is determined by the degree of heat transfer that can be obtained from a given bearing. As a rule of thumb, you can determine the maximum oil flow, above which no significant temperature reduction is obtained, using

$$Q_{max} = (D\,B) / 12\,500$$

where
Q_{max} = maximum oil flow [l/min]
D = bearing outer diameter [mm]
B = bearing width [mm]

Further temperature-related checks

After you have estimated the operating temperature, check:

- that the temperature assumption for calculating bearing life (operating viscosity) was correct
- the lubricant selection and temperature limits
- the grease or oil change interval
- the cage and seal material limits

Speed limitations

The speed capability of a bearing is normally determined by the bearing operating temperature. However, for certain bearing types and arrangements, the mechanical limits of the bearing components may have a significant influence.

The product tables typically provide two speed ratings:

- the reference speed, which is based on thermal conditions
- the limiting speed, which is based on mechanical limits

Both speed ratings are cautionary limits, rather than strict prohibiting limits, but approaching either of them signals that deeper analysis of the operating conditions is required.

For bearings with contact seals, no reference speeds are listed in the product tables. Typically, the limiting speed determines the maximum speed for these bearings.

Approximate thermal speed limit based on ISO standard conditions

The reference speed listed in the product tables is based on the SKF friction model and derived from thermal equilibrium under the ISO 15312 standardized operating and cooling conditions. Its main purpose is to provide a quick assessment of the speed capabilities of a bearing. You can also use it to estimate a thermal speed limit.

The ISO reference speed is valid for open bearings only, operating under the following conditions:

- predefined reference heat dissipation
- light loads
  - radial load P = 0,05 C_0 for radial bearings
  - axial load P = 0,02 C_0 for thrust bearings
- nominal temperature increase of 50 °C *(90 °F)* above an ambient reference temperature of 20 °C *(70 °F)*
- oil lubrication with mineral oil without EP additives
  - ISO VG32 for radial bearings
  - ISO VG68 for thrust bearings
- clean conditions
- sufficient operating clearance (*Selecting initial internal clearance*, **page 183**)
- horizontal shaft, rotating inner ring and stationary outer ring

The ISO standard does not provide reference conditions for sealed bearings.

The ISO standard, established for oil lubrication, is also valid for grease lubrication, provided a lithium based grease with mineral base oil having a viscosity between 100 and 200 mm²/s is used. Grease lubricated bearings may, however, undergo a temperature peak during initial start-up, requiring a running-in period before they reach their steady-state operating temperature.

Adjusted reference speed

The ISO reference speed is valid for a standardized set of operating conditions including standardized heat dissipation. Therefore, SKF recommends calculating the adjusted reference speed considering the actual load and lubricant viscosity in your application. Do this using the *SKF Bearing Calculator* (skf.com/bearingcalculator). However, this reference speed adjustment does not include the data regarding the actual heat dissipation for your application, so a conservative approach to the result is recommended. To include effects from heat dissipation, a detailed thermal analysis is required.

Mechanical speed limit

The limiting speed indicated in the product tables is a maximum speed valid for the standard bearing execution that should not be exceeded unless the bearing design and the application is adapted to a higher speed.

The limiting speed is determined by:

- the form stability or strength of the cage
- lubrication of the cage guiding surfaces
- centrifugal and gyratory forces acting on the rolling elements
- other speed-limiting factors, such as seals and the lubricant for sealed bearings

NOTE

Some open ball bearings have very low friction, and the reference speeds listed for them might be higher than their limiting speeds. Do not use only the mechanical speed limit. Also calculate the adjusted reference speed. The lower of the two sets the speed limit.

Speeds above the reference or limiting speed

It is possible to operate a bearing at speeds above its reference speed, its adjusted reference speed, or even the limiting speed. Before doing so, first make a detailed thermal analysis, and take whatever further measures may be required, such as use of special cage executions, or consider using high precision bearings. Regarding management of the effects of increased speed, consider the following options:

- Control the resulting increase in bearing temperature by additional cooling.
- Compensate for any reduction in bearing clearance resulting from increased bearing temperature.
- Revise the housing fitting tolerance choice to ensure that the influence of increased bearing temperature does not impair the axial displaceability of non-locating bearing outer rings.
- Revise the bearing tolerance class, together with the geometrical precision of the shaft and housing seats, to ensure these are sufficient to avoid excessive vibration.
- Consider using an alternative cage execution that is suitable for higher speed operation, in particular when approaching or exceeding the limiting speed.
- Ensure that the lubricant and lubrication method used are compatible with the higher operating temperature and the cage execution.
- Check that the relubrication interval is still acceptable, particularly for grease lubricated bearings. Oil lubrication may be required.

B.6 Bearing interfaces

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B.6 Bearing interfaces

Bearing seats on shafts and in housings, and components which locate a bearing axially, have a significant impact on bearing performance. To fully exploit the load carrying ability of a bearing, its rings or washers should be fully supported around their complete circumference and across the entire width of the raceway. Bearing seats should be manufactured to adequate geometrical and dimensional tolerances and be uninterrupted by grooves, holes or other features.

In this section you can find recommendations and requirements for designing bearing interfaces, including:

- criteria when selecting bearing fits
- recommended fits for standard conditions
- tables to help determine minimum, maximum and probable values of clearance or interference between the bearing and its seat
- recommendations for specifying geometrical tolerances of bearing seats
- recommendations for the axial support of bearing rings
- further design considerations for bearing interfaces

The ISO tolerance system

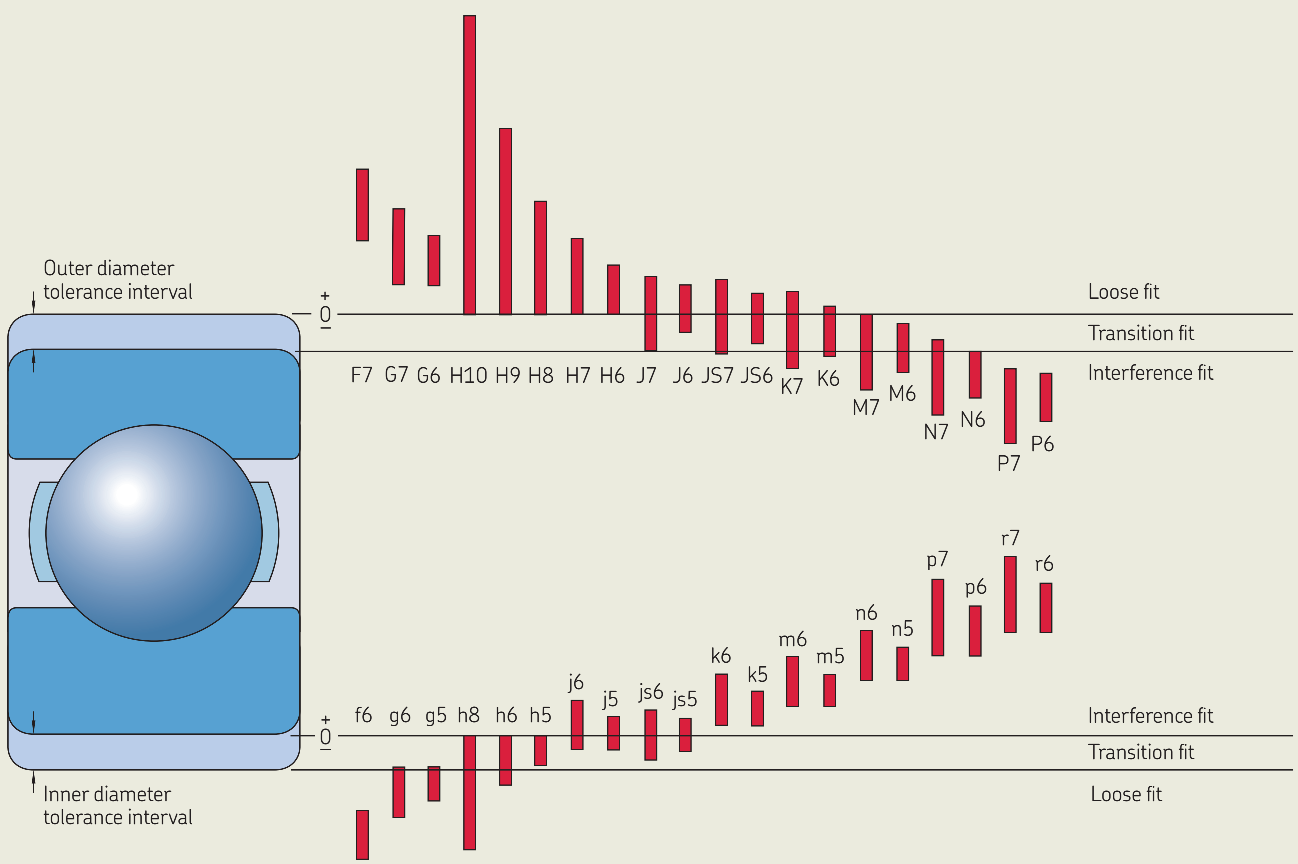
Fits for rolling bearings are typically specified with standard tolerance classes for holes and shafts as described in ISO 286-2. As bearings are typically manufactured to ISO tolerances (*Tolerances*, **page 36**), the selection of the tolerance class for the bearing seat determines the fit. The position and width of the tolerance intervals of commonly used tolerance classes relative to the bearing bore and outside diameter tolerances are illustrated in **fig. 1**, which is valid for bearings with Normal tolerances and of medium size. It is important to note that the ISO tolerance classes for rolling bearings and for holes and shafts are different. The tolerances for each size vary over the full range of actual sizes. You should therefore select the respective tolerance classes for bearing seats based on the actual bearing size for your application.

Selecting fits

Fits can be selected by following the recommendations for bearing seat diameter tolerances (*Seat tolerances for standard conditions*, **page 148**). These recommendations will provide adequate solutions for the majority of applications. However, they do not cover all details of a specific application and so you may find that adjustments may be necessary. When selecting fits, you should consider the following topics.

Fig. 1

Position and width of shaft and housing tolerance classes



B.6 Bearing interfaces

Conditions of rotation

Conditions of rotation refer to the relative motion between a bearing ring and the load acting upon it (table 1). Essentially, there are three different conditions:

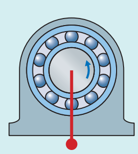
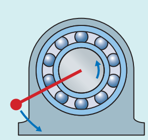
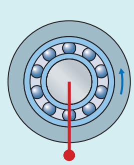
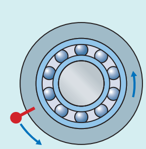
- **Rotating loads**
  These loads occur where either the bearing ring or the applied load is stationary while the other rotates. A bearing ring mounted with a loose fit will creep on its seat when subjected to a rotating load, and this can lead to fretting corrosion and eventually wear. To prevent this from happening, an adequate interference fit, between the ring subjected to rotating load and its seat, is required. For the purpose of selecting fits, loads that oscillate (such as loads acting on connecting rod bearings) are considered to be rotating loads.

- **Stationary loads**
  These loads occur where both the bearing ring and the applied load are stationary or both are rotating at the same speed. Under these conditions, a bearing ring normally does not creep and there is no risk of fretting corrosion or wear. In this case, the ring does not need to have an interference fit.

- **Direction of load indeterminate**
  This refers to variable or alternating external loads, sudden load peaks, vibration or unbalanced loads in high-speed applications. These give rise to changes in the direction of load, which cannot be accurately described. Where the direction of load is indeterminate and particularly where heavy loads are involved, there is a risk of fretting corrosion or wear. You should use an interference fit for both rings. The same fit as for a rotating load is normally suitable. Where the outer ring should be able to move axially in its housing, a loose fit must be used. However, a loose fit can result in housing wear. Where this cannot be tolerated, either protect the bearing seat surface or select a bearing that accommodates the axial displacement within itself (cylindrical roller, needle roller or CARB bearing). These bearings can be mounted with an interference fit for both rings.

Table 1

Conditions of rotation

Operating conditions	Schematic illustration	Load condition	Recommended fits
Rotating inner ring Stationary outer ring Constant load direction		Rotating inner ring load Stationary outer ring load	Interference fit for the inner ring Loose fit for the outer ring possible
Rotating inner ring Stationary outer ring Load rotates with the inner ring		Stationary inner ring load Rotating outer ring load	Loose fit for the inner ring possible Interference fit for the outer ring
Stationary inner ring Rotating outer ring Constant load direction		Stationary inner ring load Rotating outer ring load	Loose fit for the inner ring possible Interference fit for the outer ring
Stationary inner ring Rotating outer ring Load rotates with outer ring		Rotating inner ring load Stationary outer ring load	Interference fit for the inner ring Loose fit for the outer ring possible

Magnitude of load

The ring of a bearing deforms proportionately to the load. For rotating inner ring loads, this deformation can loosen the interference fit between the inner ring and shaft, causing the ring to creep on its shaft seat. The heavier the load, the tighter the interference fit required. The required interference can be estimated using:

$$\Delta = 2{,}5\sqrt{F_r \frac{d}{B}}$$

where
Δ = required interference [µm]
d = bearing bore diameter [mm]
B = bearing width [mm]
F_r = radial load [kN]

Where sudden load peaks or vibration occurs, a tighter fit can be required.

Temperature differences

In operation, bearing rings normally reach a temperature that is higher than that of the components to which they are fitted. This can loosen the fit on the shaft seat, while outer ring expansion can prevent the desired axial displacement in the housing.

Rapid start-up can loosen the inner ring fit when the frictional heat generated by the bearing is not dissipated quickly enough. In some cases, friction from seals can generate enough heat to loosen the inner ring fit.

External heat and the direction of heat flow can have an effect on fits. Steady-state and transient conditions must be considered. For additional information about temperature differences, refer to *Selecting internal clearance or preload*, **page 182**.

Precision requirements

To minimize deflections and vibration in precision or high-speed applications, interference or transition fits are recommended.

Design and material of the shaft and housing

Distortion of the bearing rings caused by shaft or housing design, for example by discontinuities of the seat or uneven wall thickness, should be avoided.

For split housings, SKF generally recommends loose fits. The tighter (less loose) the fit is in a split housing, the higher are the requirements for the geometrical tolerances of the seat. Split housings machined to tight tolerances, such as SKF plummer block housings, can be used for transition fits up to K7.

Bearings mounted in thin-walled housings or on hollow shafts require tighter interference fits than those recommended for robust cast iron housings or solid shafts (*Tolerances for seats on hollow shafts*, **page 146**).

Shafts or housings made of materials other than steel or cast iron may require different fits depending on material strength and thermal properties.

Ease of mounting and dismounting

Loose fits are beneficial for easy mounting and dismounting. In applications where interference fits are required for both the shaft and housing seat, separable bearings or bearings with a tapered bore should be considered. Bearings with a tapered bore can be mounted on tapered sleeves (**fig. 2**) or on a tapered shaft seat (**fig. 3**).

Axial displacement of the bearing in the non-locating position

When a non-locating bearing needs to be able to move axially on its seat, the ring subjected to the stationary load should have a loose fit. For additional information about bearings in the non-locating position, refer to *Arrangements and their bearing types*, **page 70**.

Fig. 2

Bearing with a tapered bore mounted on a plain shaft with an adapter sleeve

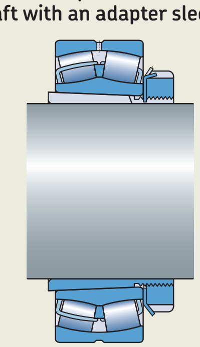
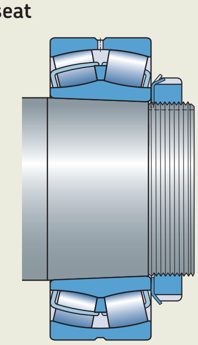


Fig. 3

Bearing with a tapered bore mounted on a tapered seat



Tolerances for bearing seats and abutments

Dimensional tolerances for bearing seats are dictated by the required fit. Precision requirements of the application will direct you to which bearing tolerance class to use (*Bearing execution*, **page 182**) and, consequently, what run-out tolerance of the seat is needed. The run-out of the seat is specified by the total radial run-out of the seat surface and the total axial run-out of the abutment (ISO 1101, 18.16).

For bearings with Normal tolerances in general industrial applications, seats are typically machined to the following tolerances:

- shaft seats to grade IT6 dimensional tolerances and grade IT5 total run-out tolerances
- housing seats to grade IT7 dimensional tolerances and grade IT6 total run-out tolerances

Suitable combinations of tolerance grades are listed in **table 2**. The tolerance zone for the total radial run-out is limited to half of the ISO tolerance grade, because the run-out tolerance is specified as a difference in radii of two coaxial cylinders, and the ISO tolerance grade refers to the diameter.

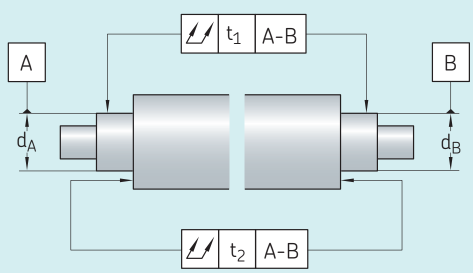
For seats of bearings mounted on withdrawal or adapter sleeves, wider diameter tolerances are permissible. The total run-out tolerances should be the same as for bearings on cylindrical seats.

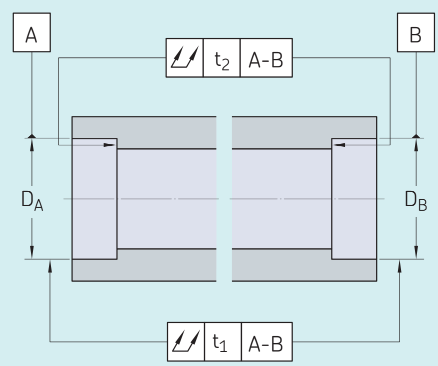
Tolerance values for ISO tolerance grades are listed in **table 3**.

Table 2

Tolerance grades for bearing seats[1)]

Application requirements	**Shaft seat**			**Housing seat**		
	Dimensional tolerance grade	**Geometrical tolerance grades**		**Dimensional tolerance grade**	**Geometrical tolerance grades**	
		Radial run-out t_1	Axial run-out t_2		Radial run-out t_1	Axial run-out t_2
Bearing to Normal tolerances (moderate speed and running accuracy)	IT6	IT5/2	IT5	IT7	IT6/2	IT6
Bearing to P6 tolerances (higher speeds or running accuracy)	IT5	IT4/2	IT4	IT6	IT5/2	IT5
Bearing to P5 tolerances (high speeds and running accuracy)	IT4	IT3/2	IT3	IT5	IT4/2	IT4





1) For very high-speed and high-precision applications, use SKF super-precision bearings and reduced IT tolerances (skf.com/super-precision).

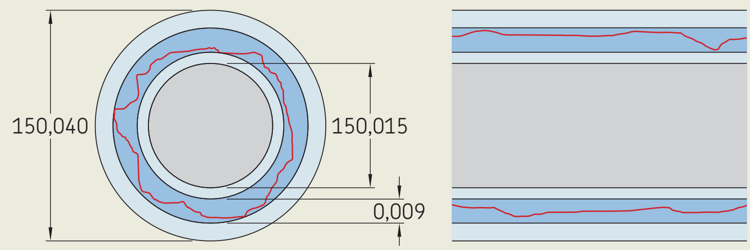
Example

A deep groove ball bearing 6030 is to be used in an electric motor. The bearing accommodates normal to heavy loads (0,05 C < P ≤ 0,1 C), and requirements for speed and precision are moderate. An interference fit on the shaft is required. For this fit, the shaft diameter should be 150 m6Ⓔ. The total radial run-out should be within IT5/2 (from table 3: 18/2 = 9 µm), and the total axial run-out of the abutment should be within IT5 (from table 3: 18 µm).

The dimensional tolerance zone in grey and the tolerance zone for the total radial run-out in blue are shown in fig. 4. The blue zone can be located at any place within the grey zone, but must not be wider than 9 µm.

Fig. 4

Dimensional and total run-out tolerances zones



Dimensional tolerance: 150 m6Ⓔ
Total radial run-out 9 µm
Actual dimensions of the shaft seat

Table 3

Values of ISO tolerance grades

Nominal dimension >	≤	Tolerance grades IT3 max.	IT4	IT5	IT6	IT7	IT8	IT9
mm		µm						
1	**3**	2	3	4	6	10	14	25
3	**6**	3	4	5	8	12	18	30
6	**10**	3	4	6	9	15	22	36
10	**18**	3	5	8	11	18	27	43
18	**30**	4	6	9	13	21	33	52
30	**50**	4	7	11	16	25	39	62
50	**80**	5	8	13	19	30	46	74
80	**120**	6	10	15	22	35	54	87
120	**180**	8	12	18	25	40	63	100
180	**250**	10	14	20	29	46	72	115
250	**315**	12	16	23	32	52	81	130
315	**400**	13	18	25	36	57	89	140
400	**500**	15	20	27	40	63	97	155
500	**630**	-	-	32	44	70	110	175
630	**800**	-	-	36	50	80	125	200
800	**1 000**	-	-	40	56	90	140	230
1 000	**1 250**	-	-	47	66	105	165	260
1 250	**1 600**	-	-	55	78	125	195	310
1 600	**2 000**	-	-	65	92	150	230	370
2 000	**2 500**	-	-	78	110	175	280	440

Tolerances for seats on hollow shafts

When a bearing is mounted on a hollow shaft using an interference fit, the shaft experiences more elastic deformation than a solid shaft. As a result, the effectiveness of the fit is less than for the same size solid shaft. The effectiveness of an interference fit on a hollow shaft depends on certain diameter ratios (**fig. 5**):

- the diameter ratio of the hollow shaft $c_i = d_i / d$
  For diameter ratios $c_i \leq 0,5$ the reduction of effectiveness is negligible.
- the diameter ratio of the bearing inner ring $c_e = d / d_e$
  When the average outside diameter of the inner ring d_e is not known, the diameter ratio can be estimated from

$$c_e = \frac{d}{k\,(D - d) + d}$$

where
c_e = diameter ratio of the bearing inner ring
d = bearing bore diameter [mm]
D = bearing outside diameter [mm]
k = adjustment factor
= 0,25 for self-aligning ball bearings in the 22 and 23 series
= 0,25 for cylindrical roller bearings
= 0,3 for other bearings

For shaft diameter ratios $c_i > 0,5$ the diameter tolerance determined for a seat on a solid shaft should be adjusted to achieve the same effectiveness of the fit on the hollow shaft. This can be done with the following procedure.

1. Determine the mean probable interference for the tolerance selected for a seat on a solid shaft, Δ_S (*Tolerances and resultant fits*, **page 153**).
2. Determine the required increase of interference for the seat on the hollow shaft from **diagram 1,** based on the diameter ratios c_i and c_e.
3. Calculate the required mean probable interference for the seat on the hollow shaft and select the tolerance class accordingly.

Example

A 6208 deep groove ball bearing with d = 40 mm and D = 80 mm is to be mounted on a hollow shaft with a diameter ratio $c_i = 0,8$. What is the appropriate tolerance class for the shaft seat?

The bearing is subjected to normal loads, and a tolerance class k5 is appropriate for a seat on a solid shaft.

- The diameter ratio of the bearing inner ring is

$$c_e = \frac{40}{0,3\,(80 - 40) + 40} = 0,77$$

- The mean probable interference on a solid shaft is
  $\Delta_S = (22 + 5) / 2 = 13,5$ µm (**table 14**, **page 160**, k5 for a 40 mm shaft diameter)
- The increase in interference for the seat on the hollow shaft is
  $\Delta_H / \Delta_S = 1,7$ (**diagram 1**, $c_i = 0,8$ and $c_e = 0,77$)
- The requisite interference for the seat on the hollow shaft is
  $\Delta_H = 1,7 \times 13,5 = 23$ µm
- The appropriate tolerance class for the seat on the hollow shaft is m6 (**table 14**, mean probable interference, (33 + 13) / 2 = 23 µm)

Fig. 5

Seat on a hollow shaft

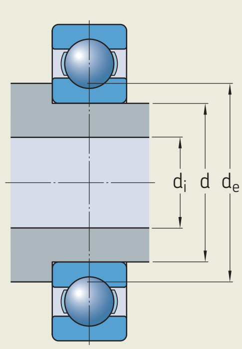
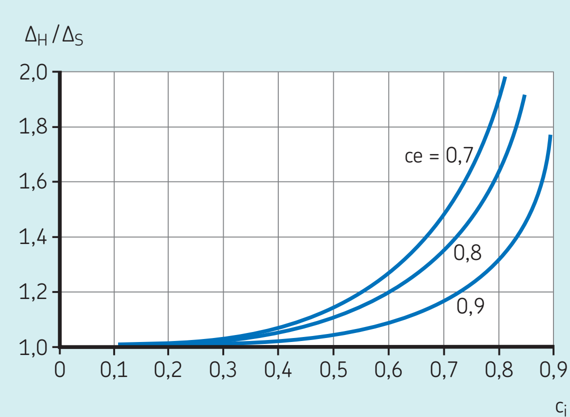


Diagram 1

Relationship of interference Δ_H, needed for a hollow steel shaft, to the known interference Δ_S for a solid steel shaft



Tolerances for tapered seats

For tapered shaft seats, SKF recommends the following tolerances (**fig. 6**):

- The permissible deviation for the slope of the taper is a ± tolerance in accordance with IT7/2. The bearing width B is the nominal size, which determines the standard tolerance values. The permissible deviation for the slope of the taper can be determined using

$$\Delta_k = \frac{IT7/2}{B}$$

The permissible range of dispersion of the slope of the taper can be determined using

$$V_k = 1/k \pm \frac{IT7/2}{B}$$

where
Δ_k = the permissible deviation of the slope of the taper
V_k = the permissible range of dispersion of the slope of the taper
B = bearing width [mm]
IT7 = the value of the tolerance grade, based on the bearing width [mm]
k = factor for the taper
= 12 for taper 1:12
= 30 for taper 1:30

- The roundness tolerance is defined as "distance t between two concentric circles in each radial plane along the tapered surface of the shaft". t is the value of tolerance grade IT5/2, based on the diameter d. Where a high degree of precision is required, IT4/2 should be used instead.
- The straightness is defined as "In each axial plane through the tapered shaft, the tolerance zone is limited by two parallel lines a distance t apart". t is the value of tolerance grade IT5/2, based on the diameter d.

Taper position

Only dimensional and geometrical tolerances of the taper are indicated in **fig. 6**. The axial position of the taper requires additional specifications. When specifying the axial position, you should also take into account the axial drive-up distance of the bearing, which is required to achieve a suitable interference fit.

Checking tolerances

To check whether a tapered shaft seat is within its tolerances, SKF recommends measuring it with a special taper gauge, based on saddles and gauging pins. More practical, but less accurate measurement methods include ring gauges, taper gauges and sine bars. For information about SKF measuring devices, refer to skf.com (*GRA 30 ring gauges* and *DMB taper gauges*).

Surface texture of bearing seats

The surface texture of a bearing seat has less of an impact on bearing performance compared to the dimensional and geometrical tolerances of the seat. However, the texture of the mating surfaces affects smoothing, which can reduce the interference in a fit. The surface texture should be limited to ensure the required fit is obtained.

Guideline values for the roughness profile parameter Ra are listed in **table 4**. These recommendations apply to ground seats, which are normally assumed for shaft seats. For housing seats, which are normally fine-turned, the Ra values may be one class higher. For applications where some loss of interference is not critical, rougher surfaces than recommended in **table 4** can be used.

Table 4

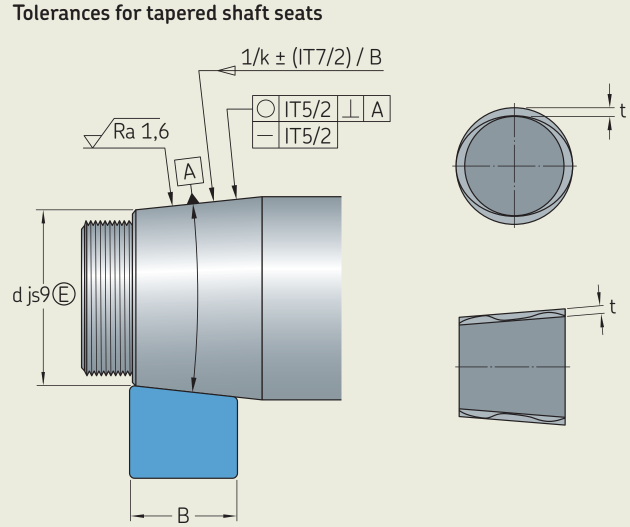
Surface roughness of bearing seats

Seat diameter		**Ra** (guideline values for ground seats)		
d, D		Diameter tolerance grade		
>	≤	IT7	IT6	IT5
mm		µm		
–	**80**	1,6	0,8	0,4
80	**500**	1,6	1,6	0,8
500	**1 250**	3,2[1]	1,6	1,6

[1] When using the oil injection method for mounting, Ra should not exceed 1,6 µm.

Fig. 6

Tolerances for tapered shaft seats



Seat tolerances for standard conditions

The following tables provide recommendations for tolerances of shaft and housing seats. They are valid for standard applications but do not cover all details of a specific application. The information under *Selecting fits*, page 140, and *Tolerances for bearing seats and abutments*, page 144, should be additionally considered.

These recommendations are valid for bearings with Normal dimensional tolerances. They can also be used for bearings to P6 dimensional tolerances. The tighter P6 tolerance zone changes the resulting fit only slightly.

Recommended seat tolerances for metric bearings:

- For solid steel shafts:
  - Radial ball bearings (table 5, except insert bearings)
  - Radial roller bearings (table 6, except needle roller bearings)
  - Thrust ball bearings and spherical roller thrust bearings (table 7, page 150)
- For cast iron and steel housings:
  - Radial bearings (table 8, page 151)
  - Thrust bearings (table 9, page 152)

For the following bearing types, recommendations are listed in the product sections:

- Insert bearings, *Design considerations*, page 356
- Needle roller bearings, relevant sections under *Needle roller bearings*, page 903
- Cylindrical roller thrust bearings, *Design considerations*, page 885
- Needle roller thrust bearings, *Design considerations*, page 903
- Inch tapered roller bearings, *Design considerations*, page 687

All ISO tolerance classes used in the tables are valid with the envelope requirement (such as H7Ⓔ), in accordance with ISO 14405-1. For practical reasons, symbol Ⓔ is not indicated in the tables.

Table 5

Tolerances for solid steel shafts – seats for radial ball bearings[1)]

Conditions	Shaft diameter	Dimensional tolerance[2)]	Total radial run-out tolerance[3)]	Total axial run-out tolerance[3)]	Ra
	mm	–	–	–	µm
Rotating inner ring load or direction of load indeterminate					
Light loads (P ≤ 0,05 C)	≤ 17	js5	IT4/2	IT4	0,4
	> 17 to 100	j6	IT5/2	IT5	0,8
	> 100 to 140	k6	IT5/2	IT5	1,6
Normal to heavy loads (0,05 C < P ≤ 0,1 C)	≤ 10	js5	IT4/2	IT4	0,4
	> 10 to 17	j5	IT4/2	IT4	0,4
	> 17 to 100	k5	IT4/2	IT4	0,8
	> 100 to 140	m5	IT4/2	IT4	0,8
	> 140 to 200	m6	IT5/2	IT5	1,6
	> 200 to 500	n6	IT5/2	IT5	1,6
	> 500	p7	IT6/2	IT6	3,2
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable		g6[4)]	IT5/2	IT5	1,6
Easy axial displacement of inner ring on shaft unnecessary		h6	IT5/2	IT5	1,6
Axial loads only		j6	IT5/2	IT5	1,6

1) For insert bearings, refer to *Design considerations*, page 356.
2) The envelope requirement (symbol Ⓔ from ISO 14405-1) is not shown but applies to all tolerance classes.
3) Values listed are for bearings to Normal tolerances. For bearings with tighter tolerance classes, use the recommendations in table 2, page 144.
4) Depending on bearing size, a shifted g6Ⓔ tolerance may be needed to obtain a loose fit.

Bearings with a tapered bore

Bearings with a tapered bore are always mounted with an interference fit for the inner ring. The fit is determined by the distance through which the inner ring is driven up on a tapered seat or sleeve. For detailed information, refer to the information in the product sections:

- *Self-aligning ball bearings*, **page 438**
- *Spherical roller bearings*, **page 774**
- *CARB toroidal roller bearings*, **page 842**

For seats of bearings mounted on tapered sleeves, wider diameter tolerances are permissible. The total run-out tolerances should be the same as for bearings on cylindrical seats (*Tolerances for bearing seats and abutments*, **page 144**).

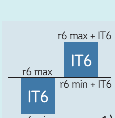
Suitable tolerances are listed in **table 10**, **page 152**. They are valid for moderate speeds and moderate precision requirements.

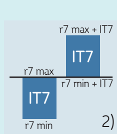
Table 6

Tolerances for solid steel shafts – seats for radial roller bearings[1)]

Conditions	Shaft diameter	Dimensional tolerance[2)]	Total radial run-out tolerance[3)]	Total axial run-out tolerance[3)]	Ra
	mm	–	–	–	µm
Rotating inner ring load or direction of load indeterminate					
Light loads (P ≤ 0,05 C)	≤ 25	j6	IT5/2	IT5	0,8
	> 25 to 60	k6	IT5/2	IT5	0,8
	> 60 to 140	m6	IT5/2	IT5	0,8
Normal to heavy loads (0,05 C < P ≤ 0,1 C)	≤ 30	k6	IT5/2	IT5	0,8
	> 30 to 50	m5	IT5/2	IT5	0,8
	> 50 to 65	n5	IT5/2	IT5	0,8
	> 65 to 100	n6	IT5/2	IT5	0,8
	> 100 to 280	p6	IT5/2	IT5	1,6
	> 280 to 500	r6	IT5/2	IT5	1,6
	> 500	r7	IT6/2	IT6	3,2
Heavy to very heavy loads and high peak loads under difficult operating conditions (P > 0,1 C)	> 50 to 65	n5	IT5/2	IT5	0,8
	> 65 to 85	n6	IT5/2	IT5	0,8
	> 85 to 140	p6	IT5/2	IT5	0,8
	> 140 to 300	r6	IT5/2	IT5	1,6
	> 300 to 500	r6 + IT6[4)]	IT5/2	IT5	1,6
	> 500	r7 + IT7[4)]	IT6/2	IT6	3,2
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable		g6[5)]	IT5/2	IT5	1,6
Easy axial displacement of inner ring on shaft unnecessary		h6	IT5/2	IT5	1,6
Axial loads only		j6	IT5/2	IT5	1,6

1) For needle roller bearings, refer to the relevant sections under *Needle roller bearings*, **page 581**.
2) The envelope requirement (symbol Ⓔ from ISO 14405-1) is not shown but applies to all tolerance classes.
3) Values listed are for bearings to Normal tolerances. For bearings with tighter tolerance classes, use the recommendations in **table 2**, **page 144**.
4) Shifted tolerance field.





5) Depending on bearing size, a shifted g6Ⓔ tolerance may be needed to obtain a loose fit.

Table 7

Tolerances for solid steel shafts – seats for thrust bearings[1]

Conditions	Shaft diameter	Dimensional tolerance[2]	Total radial run-out tolerance	Total axial run-out tolerance	Ra
	mm	–	–	–	µm
Axial loads only on thrust ball bearings					
		h6	IT5/2	IT5	1,6[3]
Combined radial and axial loads on spherical roller thrust bearings					
Stationary load on shaft washer	all	j6	IT5/2	IT5	1,6[3]
Rotating load on shaft washer, or direction of oad indeterminate	≤ 200	k6	IT5/2	IT5	1,6[3]
	> 200 to 400	m6	IT5/2	IT5	1,6
	> 400	n6	IT5/2	IT5	1,6

1) For cylindrical roller thrust bearings, refer to *Design considerations*, page 885. For needle roller thrust bearings, refer to *Design considerations*, page 903.
2) The envelope requirement (symbol Ⓔ from ISO 14405-1) is not shown but applies to all tolerance classes.
3) For d ≤ 80 mm, use Ra = 0,8 µm.

Table 8

Tolerances for cast iron and steel housings – seats for radial bearings[1)]

	Conditions	Dimensional tolerance[2)3)]	Total radial run-out tolerance	Total axial run-out tolerance	Ra[6)]	Displacement of outer ring
		–	–	–	µm	–
For non-split housings only	**Rotating outer ring load**					
	Heavy loads on bearings in thin-walled housings, heavy peak loads (P > 0,1 C)	P7	IT6/2	IT6	3,2	Cannot be displaced
	Normal to heavy loads (P > 0,05 C)	N7	IT6/2	IT6	3,2	Cannot be displaced
	Light and variable loads (P ≤ 0,05 C)	M7	IT6/2	IT6	3,2	Cannot be displaced
	Direction of load indeterminate					
	Heavy peak loads	M7	IT6/2	IT6	3,2	Cannot be displaced
	Normal to heavy loads (P > 0,05 C), axial displacement of outer ring unnecessary	K7[5)]	IT6/2	IT6	3,2	In most cases, cannot be displaced
For non-split housings and split housings	**Direction of load indeterminate**					
	Light to normal loads (P ≤ 0,1 C), axial displacement of outer ring desirable	J7	IT6/2	IT6	3,2	In most cases, can be displaced
	Stationary outer ring load					
	Loads of all kinds	H7[3)]	IT6/2	IT6	3,2	Can be displaced
	Light to normal loads (P ≤ 0,1 C) with simple working conditions	H8[3)]	IT6/2	IT6	3,2	Can be displaced
	Thermal expansion of the shaft	G7[4)]	IT6/2	IT6	3,2	Can be displaced

1) For drawn cup, alignment and combined needle roller bearings, refer to *Shaft and housing tolerances*, page 610.
2) The envelope requirement (symbol Ⓔ from ISO 14405-1) is not shown but applies to all tolerance classes.
3) For large bearings (D > 250 mm), or temperature differences between the outer ring and housing > 10 °C *(18 °F)*, tolerance class G7Ⓔ should be used instead of tolerance class H7Ⓔ.
4) For large bearings (D > 500 mm), or temperature differences between the outer ring and housing > 10 °C *(18 °F)*, tolerance class F7Ⓔ should be used instead of tolerance class G7Ⓔ.
5) A split housing is allowed provided housing halves are well aligned during machining of the housing, with relief chamfers at the split.
6) For D > 500 mm, use Ra = 6,3 µm.

Table 9

Tolerances for cast iron and steel housings – seats for thrust bearings[1)]

Conditions	Dimensional tolerance[2)]	Total axial run-out tolerance	Ra	Remarks
	–	–	µm	–
Axial loads only				
Thrust ball bearings	H8	IT7	6,3	For less precise bearing arrangements, there can be a radial clearance of up to 0,001 D.
Spherical roller thrust bearings where separate bearings provide radial location	–	IT6		Housing washer must be fitted with an adequate radial gap so that no radial load can act on the thrust bearings.
Combined radial and axial loads on spherical roller thrust bearings				
Stationary load on housing washer arrangements	H7	IT6	3,2[3)]	For additional information, refer to *Design considerations*, page 918.
Rotating load on housing washer	M7	IT6	3,2[3)]	

1) For cylindrical roller thrust bearings, refer to *Design considerations*, page 885. For needle roller thrust bearings, refer to *Design considerations*, page 903.
2) The envelope requirement (symbol Ⓔ from ISO 14405-1) is not shown but applies to all tolerance classes.
3) For D < 80 mm, use Ra = 1,6 µm.

Table 10

Tolerances for seats of bearings mounted on tapered sleeves

Shaft diameter d Nominal		Diameter tolerance h9Ⓔ		Total radial run-out IT5/2
>	≤	U	L	max.
mm		µm		mm
10	**18**	0	-43	4
18	**30**	0	-52	5
30	**50**	0	-62	6
50	**80**	0	-74	7
80	**120**	0	-87	8
120	**180**	0	-100	9
180	**250**	0	-115	10
250	**315**	0	-130	12
315	**400**	0	-140	13
400	**500**	0	-155	14
500	**630**	0	-175	16
630	**800**	0	-200	18
800	**1 000**	0	-230	20
1 000	**1 250**	0	-260	24

Tolerances and resultant fits

The tables in this section provide information about bearing tolerances, seat tolerances and resultant fits (**fig. 7**). These should enable you to determine easily the maximum and minimum values of fits when using ISO tolerance classes for bearing seats and bearings with Normal tolerances for the bore and outside diameter. The *SKF Bearing Calculator* (skf.com/bearingcalculator) provides a similar function for every individual bearing.

The tables cannot be used for tapered roller bearings when d ≤ 30 mm or D ≤ 150 mm or for thrust bearings when D ≤ 150 mm. The diameter tolerances for these bearings deviate from the Normal tolerances for other rolling bearings.

The tables list:

- the upper and lower limits of bore or outside diameter deviations for bearings with Normal tolerances
- the upper and lower limits of shaft or housing bore diameter deviations for relevant tolerance classes in accordance with ISO 2862
- the smallest and largest values of the theoretical interference (–) or clearance (+)
- the smallest and largest values of the ±3σ probable interference (–) or clearance (+)

The appropriate values for shaft seats are listed for the following tolerance classes:

- f5, f6, g5, g6, h5 (**table 11**, **page 154**)
- h6, h8, h9, j5, j6 (**table 12**, **page 156**)
- js4, js5, js6, js7, k4 (**table 13**, **page 158**)
- k5, k6, m5, m6, n5 (**table 14**, **page 160**)
- n6, p6, p7, r6, r7 (**table 15**, **page 162**)
- r6+IT6, r7+IT7 (**table 16**, **page 164**)

The appropriate values for housing seats are listed for the following tolerance classes:

- F7, G6, G7, H5, H6 (**table 17**, **page 166**)
- H7, H8, H9, H10, J6 (**table 18**, **page 168**)
- J7, JS5, JS6, JS7, K5 (**table 19**, **page 170**)
- K6, K7, M5, M6, M7 (**table 20**, **page 172**)
- N6, N7, P6, P7 (**table 21**, **page 174**)

Fig. 7

Shaft and housing tolerances and resultant fits

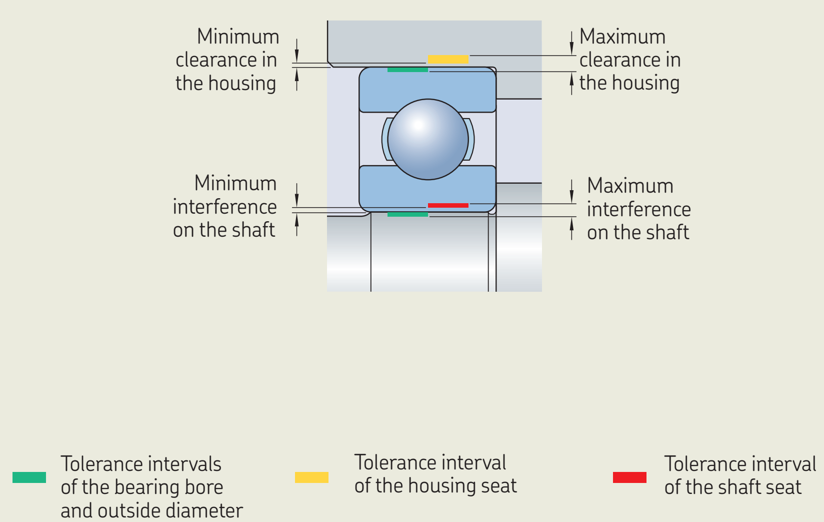
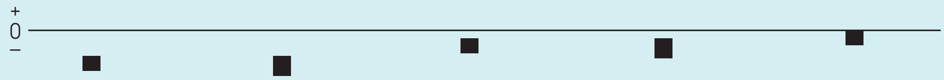


Table 11

Shaft tolerances and resultant fits

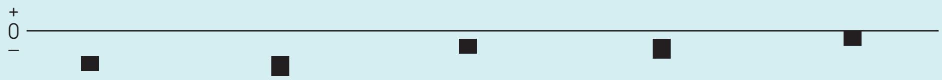


Shaft Normal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1)] Tolerance classes									
				f5Ⓔ		f6Ⓔ		g5Ⓔ		g6Ⓔ		h5Ⓔ	
				Deviations (shaft diameter) Theoretical interference (–) Probable interference (–)									
over	incl.	low	high										
mm		µm		µm									
–	**3**	–8	0	–6	–10	–6	–12	–2	–6	–2	–8	0	–4
				–2	+10	–2	+12	–6	+6	–6	+8	–8	+4
				–1	+9	0	+10	–5	+5	–4	+6	–7	+3
3	**6**	–8	0	–10	–15	–10	–18	–4	–9	–4	–12	0	–5
				+2	+15	+2	+18	–4	+9	–4	+12	–8	+5
				+3	+14	+4	+16	–3	+8	–2	+10	–7	+4
6	**10**	–8	0	–13	–19	–13	–22	–5	–11	–5	–14	0	–6
				+5	+19	+5	+22	–3	+11	–3	+14	–8	+6
				+7	+17	+7	+20	–1	+9	–1	+12	–6	+4
10	**18**	–8	0	–16	–24	–16	–27	–6	–14	–6	–17	0	–8
				+8	+24	+8	+27	–2	+14	–2	+17	–8	+8
				+10	+22	+10	+25	0	+12	0	+15	–6	+6
18	**30**	–10	0	–20	–29	–20	–33	–7	–16	–7	–20	0	–9
				+10	+29	+10	+33	–3	+16	–3	+20	–10	+9
				+12	+27	+13	+30	–1	+14	0	+17	–8	+7
30	**50**	–12	0	–25	–36	–25	–41	–9	–20	–9	–25	0	–11
				+13	+36	+13	+41	–3	+20	–3	+25	–12	+11
				+16	+33	+17	+37	0	+17	+1	+21	–9	+8
50	**80**	–15	0	–30	–43	–30	–49	–10	–23	–10	–29	0	–13
				+15	+43	+15	+49	–5	+23	–5	+29	–15	+13
				+19	+39	+19	+45	–1	+19	–1	+25	–11	+9
80	**120**	–20	0	–36	–51	–36	–58	–12	–27	–12	–34	0	–15
				+16	+51	+16	+58	–8	+27	–8	+34	–20	+15
				+21	+46	+22	+52	–3	+22	–2	+28	–15	+10
120	**180**	–25	0	–43	–61	–43	–68	–14	–32	–14	–39	0	–18
				+18	+61	+18	+68	–11	+32	–11	+39	–25	+18
				+24	+55	+25	+61	–5	+26	–4	+32	–19	+12
180	**250**	–30	0	–50	–70	–50	–79	–15	–35	–15	–44	0	–20
				+20	+70	+20	+79	–15	+35	–15	+44	–30	+20
				+26	+64	+28	+71	–9	+29	–7	+36	–24	+14
250	**315**	–35	0	–56	–79	–56	–88	–17	–40	–17	–49	0	–23
				+21	+79	+21	+88	–18	+40	–18	+49	–35	+23
				+29	+71	+30	+79	–10	+32	–9	+40	–27	+15
315	**400**	–45	0	–62	–87	–62	–98	–18	–43	–18	–54	0	–25
				+22	+87	+22	+98	–22	+43	–22	+54	–40	+25
				+30	+79	+33	+87	–14	+35	–11	+43	–32	+17
400	**500**	–45	0	–68	–95	–68	–108	–20	–47	–20	–60	0	–27
				+23	+95	+23	+108	–25	+47	–25	+60	–45	+27
				+32	+86	+35	+96	–16	+38	–13	+48	–36	+18
500	**630**	–50	0	–76	–104	–76	–120	–22	–50	–22	–66	0	–28
				+26	+104	+26	+120	–28	+50	–28	+66	–50	+28
				+36	+94	+39	+107	–18	+40	–15	+53	–40	+18

B.6 Bearing interfaces

Table 11

Shaft tolerances and resultant fits



Shaft Nominal diameter d		**Bearing** Bore diameter tolerance $t_{\Delta dmp}$		**Shaft diameter deviations, resultant fits**[1)] Tolerance classes									
				f5Ⓔ		f6Ⓔ		g5Ⓔ		g6Ⓔ		h5Ⓔ	
				Deviations (shaft diameter) / Theoretical interference (–) / Probable interference (–)									
over	incl.	low	high										
mm		µm		µm									
630	**800**	–75	0	–80	–112	–80	–130	–24	–56	–24	–74	0	–32
				+5	+112	+5	+130	–51	+56	–51	+74	–75	+32
				+17	+100	+22	+113	–39	+44	–34	+57	–63	+20
800	**1 000**	–100	0	–86	–122	–86	–142	–26	–62	–26	–82	0	–36
				–14	+122	–14	+142	–74	+62	–74	+82	–100	+36
				0	+108	+6	+122	–60	+48	–54	+62	–86	+22
1 000	**1 250**	–125	0	–98	–140	–98	–164	–28	–70	–28	–94	0	–42
				–27	+140	–27	+164	–97	+70	–97	+94	–125	+42
				–10	+123	–3	+140	–80	+53	–73	+70	–108	+25
1 250	**1 600**	–160	0	–110	–160	–110	–188	–30	–80	–30	–108	0	–50
				–50	+160	–50	+188	–130	+80	–130	+108	–160	+50
				–29	+139	–20	+158	–109	+59	–100	+78	–139	+29
1 600	**2 000**	–200	0	–120	–180	–120	–212	–32	–92	–32	–124	0	–60
				–80	+180	–80	+212	–168	+92	–168	+124	–200	+60
				–55	+155	–45	+177	–143	+67	–133	+89	–175	+35

1) Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Table 12

Shaft tolerances and resultant fits

Shaft Normal diameter d		**Bearing** Bore diameter tolerance $t_{\Delta dmp}$		**Shaft diameter deviations, resultant fits**[1)] Tolerance classes									
				h6Ⓔ		h8Ⓔ		h9Ⓔ		j5Ⓔ		j6Ⓔ	
>	≤	L	U	Deviations (shaft diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
mm		µm		µm									
–	3	–8	0	0	–6	0	–14	0	–25	+2	–2	+4	–2
				–8	+6	–8	+14	–8	+25	–10	+2	–12	+2
				–6	+4	–6	+12	–5	+22	–9	+1	–10	0
3	6	–8	0	0	–8	0	–18	0	–30	+3	–2	+6	–2
				–8	+8	–8	+18	–8	+30	–11	+2	–14	+2
				–6	+6	–5	+15	–5	+27	–10	+1	–12	0
6	10	–8	0	0	–9	0	–22	0	–36	+4	–2	+7	–2
				–8	+9	–8	+22	–8	+36	–12	+2	–15	+2
				–6	+7	–5	+19	–5	+33	–10	0	–13	0
10	18	–8	0	0	–11	0	–27	0	–43	+5	–3	+8	–3
				–8	+11	–8	+27	–8	+43	–13	+3	–16	+3
				–6	+9	–5	+24	–5	+40	–11	+1	–14	+1
18	30	–10	0	0	–13	0	–33	0	–52	+5	–4	+9	–4
				–10	+13	–10	+33	–10	+52	–15	+4	–19	+4
				–7	+10	–6	+29	–6	+48	–13	+2	–16	+1
30	50	–12	0	0	–16	0	–39	0	–62	+6	–5	+11	–5
				–12	+16	–12	+39	–12	+62	–18	+5	–23	+5
				–8	+12	–7	+34	–7	+57	–15	+2	–19	+1
50	80	–15	0	0	–19	0	–46	0	–74	+6	–7	+12	–7
				–15	+19	–15	+46	–15	+74	–21	+7	–27	+7
				–11	+15	–9	+40	–9	+68	–17	+3	–23	+3
80	120	–20	0	0	–22	0	–54	0	–87	+6	–9	+13	–9
				–20	+22	–20	+54	–20	+87	–26	+9	–33	+9
				–14	+16	–12	+46	–12	+79	–21	+4	–27	+3
120	180	–25	0	0	–25	0	–63	0	–100	+7	–11	+14	–11
				–25	+25	–25	+63	–25	+100	–32	+11	–39	+11
				–18	+18	–15	+53	–15	+90	–26	+5	–32	+4
180	250	–30	0	0	–29	0	–72	0	–115	+7	–13	+16	–13
				–30	+29	–30	+72	–30	+115	–37	+13	–46	+13
				–22	+21	–18	+60	–17	+102	–31	+7	–38	+5
250	315	–35	0	0	–32	0	–81	0	–130	+7	–16	+16	–16
				–35	+32	–35	+81	–35	+130	–42	+16	–51	+16
				–26	+23	–22	+68	–20	+115	–34	+8	–42	+7
315	400	–40	0	0	–36	0	–89	0	–140	+7	–18	+18	–18
				–40	+36	–40	+89	–40	+140	–47	+18	–58	+18
				–29	+25	–25	+74	–23	+123	–39	+10	–47	+7
400	500	–45	0	0	–40	0	–97	0	–155	+7	–20	+20	–20
				–45	+40	–45	+97	–45	+155	–52	+20	–65	+20
				–33	+28	–28	+80	–26	+136	–43	+11	–53	+8

Table 12

Shaft tolerances and resultant fits

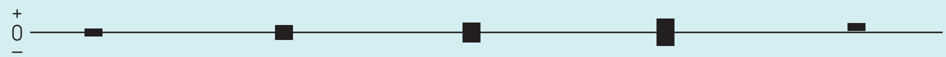
Shaft Normal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1)] Tolerance classes h6Ⓔ		h8Ⓔ		h9Ⓔ		j5Ⓔ		j6Ⓔ	
				Deviations (shaft diameter) Theoretical interference (–)/clearance (+) Probable interference (–)/clearance (+)									
>	≤	L	U										
mm		µm		µm									
500	**630**	–50	0	0	–44	0	–110	0	–175	–	–	–22	–22
				–50	+44	–50	+110	–50	+175	–	–	–72	+22
				–37	+31	–31	+91	–29	+154	–	–	–59	+9
630	**800**	–75	0	0	–50	0	–125	0	–200	–	–	+25	–25
				–75	+50	–75	+125	–75	+200	–	–	–100	+25
				–58	+33	–48	+98	–45	+170	–	–	–83	+8
800	**1 000**	–100	0	0	–56	0	–140	0	–230	–	–	+28	–28
				–100	+56	–100	+140	–100	+230	–	–	–128	+28
				–80	+36	–67	+107	–61	+191	–	–	–108	+8
1 000	**1 250**	–125	0	0	–66	0	–165	0	–260	–	–	+33	–33
				–125	+66	–125	+165	–125	+260	–	–	–158	+33
				–101	+42	–84	+124	–77	+212	–	–	–134	+9
1 250	**1 600**	–160	0	0	–78	0	–195	0	–310	–	–	+39	–39
				–160	+78	–160	+195	–160	+310	–	–	–199	+39
				–130	+48	–109	+144	–100	+250	–	–	–169	+9
1 600	**2 000**	–200	0	0	–92	0	–230	0	–370	–	–	+46	–46
				–200	+92	–200	+230	–200	+370	–	–	–246	+46
				–165	+57	–138	+168	–126	+296	–	–	–211	+11

1) Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, **page 153**.

B.6 Bearing interfaces

Table 13

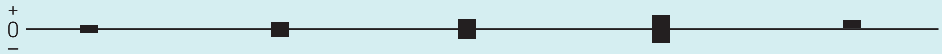
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1)] Tolerance classes									
				js4Ⓔ		js5Ⓔ		js6Ⓔ		js7Ⓔ		k4Ⓔ	
over	incl.	low	high	Deviations (shaft diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
mm		µm		µm									
–	3	–8	0	+1,5	–1,5	+2	–2	+3	–3	+5	–5	+3	0
				–9,5	+1,5	–10	+2	–11	+3	–13	+5	–11	0
				–8,5	+0,5	–9	+1	–9	+1	–11	+3	–10	–1
3	6	–8	0	+2	–2	+2,5	–2,5	+4	–4	+6	–6	+5	+1
				–10	+2	–10,5	+2,5	–12	+4	–14	+6	–13	–1
				–9	+1	–9	+1	–10	+2	–12	+4	–12	–2
6	10	–8	0	+2	–2	+3	–3	+4,5	–4,5	+7,5	–7,5	+5	+1
				–10	+2	–11	+3	–12,5	+4,5	–15,5	+7,5	–13	–1
				–9	+1	–9	+1	–11	+3	–13	+5	–12	–2
10	18	–8	0	+2,5	–2,5	+4	–4	+5,5	–5,5	+9	–9	+6	+1
				–10,5	+2,5	–12	+4	–13,5	+5,5	–17	+9	–14	–1
				–9,5	+1,5	–10	+2	–11	+3	–14	+6	–13	–2
18	30	–10	0	+3	–3	+4,5	–4,5	+6,5	–6,5	+10,5	–10,5	+8	+2
				–13	+3	–14,5	+4,5	–16,5	+6,5	–20,5	+10,5	–18	–2
				–10,5	+1,5	–12	+2	–14	+4	–17	+7	–16	–4
30	50	–12	0	+3,5	–3,5	+5,5	–5,5	+8	–8	+12,5	–12,5	+9	+2
				–15,5	+3,5	–17,5	+5,5	–20	+8	–24,5	+12,5	–21	–2
				–13,5	+1,5	–15	+3	–16	+4	–20	+8	–19	–4
50	80	–15	0	+4	–4	+6,5	–6,5	+9,5	–9,5	+15	–15	+10	+2
				–19	+4	–21,5	+6,5	–24,5	+9,5	–30	+15	–25	–2
				–15,5	+1,5	–18	+3	–20	+5	–25	+10	–22	–5
80	120	–20	0	+5	–5	+7,5	–7,5	+11	–11	+17,5	–17,5	+13	+3
				–25	+5	–27,5	+7,5	–31	+11	–37,5	+17,5	–33	–3
				–22	+2	–23	+3	–25	+5	–31	+11	–30	–6
120	180	–25	0	+6	–6	+9	–9	+12,5	–12,5	+20	–20	+15	+3
				–31	+6	–34	+9	–37,5	+12,5	–45	+20	–40	–3
				–27	+2	–28	+3	–31	+6	–37	+12	–36	–7
180	250	–30	0	+7	–7	+10	–10	+14,5	–14,5	+23	–23	+18	+4
				–37	+7	–40	+10	–44,5	+14,5	–53	+23	–48	–4
				–32	+2	–34	+4	–36	+6	–43	+13	–43	–9
250	315	–35	0	+8	–8	+11,5	–11,5	+16	–16	+26	–26	+20	+4
				–4	+8	–46,5	+11,5	–51	+16	–61	+26	–55	–4
				–37	+2	–39	+4	–42	+7	–49	+14	–49	–10
315	400	–40	0	+9	–9	+12,5	–12,5	+18	–18	+28,5	–28,5	+22	+4
				–49	+9	–52,5	+12,5	–58	+18	–68,5	+28,5	–62	–4
				–42	+2	–44	+4	–47	+7	–55	+15	–55	–11
400	500	–45	0	+10	–10	+13,5	–13,5	+20	–20	+31,5	–31,5	+25	+5
				–55	+10	–58,5	+13,5	–65	+20	–76,5	+31,5	–70	–5
				–48	+3	–49	+4	–53	+8	–62	+17	–63	–12
500	630	–50	0	–	–	+14	–14	+22	–22	+35	–35	–	–
				–	–	–64	+14	–72	+22	–85	+35	–	–
				–	–	–54	+4	–59	+9	–69	+19	–	–

Table 13

Shaft tolerances and resultant fits

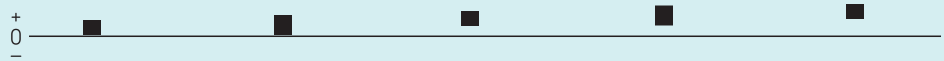


Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1) Tolerance classes									
				js4Ⓔ		js5Ⓔ		js6Ⓔ		js7Ⓔ		k4Ⓔ	
over	incl.	low	high	Deviations (shaft diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
mm		µm		µm									
630	**800**	–75	0	–	–	+16	–16	+25	–25	+40	–40	–	–
				–	–	–91	+16	–100	+25	–115	+40	–	–
				–	–	–79	+4	–83	+8	–93	+18	–	–
800	**1 000**	–100	0	–	–	+18	–18	+28	–28	+45	–45	–	–
				–	–	–118	+18	–128	+28	–145	+45	–	–
				–	–	–104	+4	–108	+8	–118	+18	–	–
1 000	**1 250**	–125	0	–	–	+21	–21	+33	–33	+52	–52	–	–
				–	–	–146	+21	–158	+33	–177	+52	–	–
				–	–	–129	+4	–134	+9	–145	+20	–	–
1 250	**1 600**	–160	0	–	–	+25	–25	+39	–39	+62	–62	–	–
				–	–	–185	+25	–199	+39	–222	+62	–	–
				–	–	–164	+4	–169	+9	–182	+22	–	–
1 600	**2 000**	–200	0	–	–	+30	–30	+46	–46	+75	–75	–	–
				–	–	–230	+30	–246	+46	–275	+75	–	–
				–	–	–205	+5	–211	+11	–225	+25	–	–

1) Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

Table 14

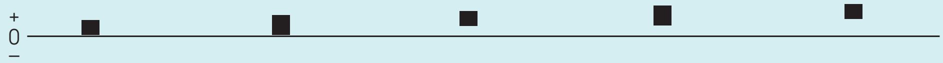
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1)] Tolerance classes									
				k5Ⓔ		k6Ⓔ		m5Ⓔ		m6Ⓔ		n5Ⓔ	
				Deviations (shaft diameter) / Theoretical interference (–) / Probable interference (–)									
>	≤	L	U										
mm		µm		µm									
–	**3**	–8	0	+4	0	+6	0	+6	+2	+8	+2	+8	+4
				–12	0	–14	0	–14	–2	–16	–2	–16	–4
				–11	–1	–12	–2	–13	–3	–14	–4	–15	–5
3	**6**	–8	0	+6	+1	+9	+1	+9	+4	+12	+4	+13	+8
				–14	–1	–17	–1	–17	–4	–20	–4	–21	–8
				–13	–2	–15	–3	–16	–5	–18	–6	–20	–9
6	**10**	–8	0	+7	+1	+10	+1	+12	+6	+15	+6	+16	+10
				–15	–1	–18	–1	–20	–6	–23	–6	–24	–10
				–13	–3	–16	–3	–18	–8	–21	–8	–22	–12
10	**18**	–8	0	+9	+1	+12	+1	15	+7	+18	+7	+20	+12
				–17	–1	–20	–1	–23	–7	–26	–7	–28	–12
				–15	–3	–18	–3	–21	–9	–24	–9	–26	–14
18	**30**	–10	0	+11	+2	+15	+2	+17	+8	+21	+8	+24	+15
				–21	–2	–25	–2	–27	–8	–31	–8	–34	–15
				–19	–4	–22	–5	–25	–10	–28	–11	–32	–17
30	**50**	–12	0	+13	+2	+18	+2	+20	+9	+25	+9	+28	+17
				–25	–2	–30	–2	–32	–9	–37	–9	–40	–17
				–22	–5	–26	–6	–29	–12	–33	–13	–37	–20
50	**80**	–15	0	+15	+2	+21	+2	+24	+11	+30	+11	+33	+20
				–30	–2	–36	–2	–39	–11	–45	–11	–48	–20
				–26	–6	–32	–6	–35	–15	–41	–15	–44	–24
80	**120**	–20	0	+18	+3	+25	+3	+28	+13	+35	+13	+38	+23
				–38	–3	–45	–3	–48	–13	–55	–13	–58	–23
				–33	–8	–39	–9	–43	–18	–49	–19	–53	–28
120	**180**	–25	0	+21	+3	+28	+3	+33	+15	+40	+15	+45	+27
				–46	–3	–53	–3	–58	–15	–65	–15	–70	–27
				–40	–9	–46	–10	–52	–21	–58	–22	–64	–33
180	**250**	–30	0	+24	+4	+33	+4	+37	+17	+46	+17	+51	+31
				–54	–4	–63	–4	–67	–17	–76	–17	–81	–31
				–48	–10	–55	–12	–61	–23	–68	–25	–75	–37
250	**315**	–35	0	+27	+4	+36	+4	+43	+20	+52	+20	+57	+34
				–62	–4	–71	–4	–78	–20	–87	–20	–92	–34
				–54	–12	–62	–13	–70	–28	–78	–29	–84	–42
315	**400**	–40	0	+29	+4	+40	+4	+46	+21	+57	+21	+62	+37
				–69	–4	–80	–4	–86	–21	–97	–21	–102	–37
				–61	–12	–69	–15	–78	–29	–86	–32	–94	–45
400	**500**	–45	0	+32	+5	+45	+5	+50	+23	+63	+23	+67	+40
				–77	–5	–90	–5	–95	–23	–108	–23	–112	–40
				–68	–14	–78	–17	–86	–32	–96	–35	–103	–49

Table 14

Shaft tolerances and resultant fits

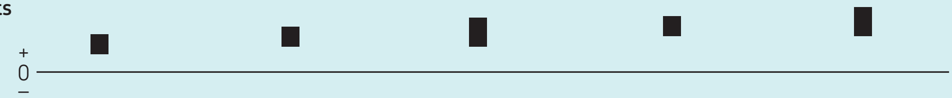


Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1] Tolerance classes									
				k5Ⓔ		k6Ⓔ		m5Ⓔ		m6Ⓔ		n5Ⓔ	
				Deviations (shaft diameter) / Theoretical interference (–) / Probable interference (–)									
>	≤	L	U										
mm		µm		µm									
500	**630**	–50	0	+29	0	+44	0	+55	+26	+70	+26	+73	+44
				–78	0	–94	0	–105	–26	–120	–26	–122	–44
				–68	–10	–81	–13	–94	–36	–107	–39	–112	–54
630	**800**	–75	0	+32	0	+50	0	+62	+30	+80	+30	+82	+50
				–107	0	–125	0	–137	–30	–155	–30	–157	–50
				–95	–12	–108	–17	–125	–42	–138	–47	–145	–62
800	**1 000**	–100	0	+36	0	+56	0	+70	+34	+90	+34	+92	+56
				–136	0	–156	0	–170	–34	–190	–34	–192	–56
				–122	–14	–136	–20	–156	–48	–170	–54	–178	–70
1 000	**1 250**	–125	0	+42	0	+66	0	+82	+40	+106	+40	+108	+66
				–167	0	–191	0	–207	–40	–231	–40	–233	–66
				–150	–17	–167	–24	–190	–57	–207	–64	–216	–83
1 250	**1 600**	–160	0	+50	0	+78	0	+98	+48	+126	+48	+128	+78
				–210	0	–238	0	–258	–48	–286	–48	–288	–78
				–189	–21	–208	–30	–237	–69	–256	–78	–267	–99
1 600	**2 000**	–200	0	+60	0	+92	0	+118	+58	+150	+58	+152	+92
				–260	0	–292	0	–318	–58	–350	–58	–352	–92
				–235	–25	–257	–35	–293	–83	–315	–93	–327	–117

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

Table 15

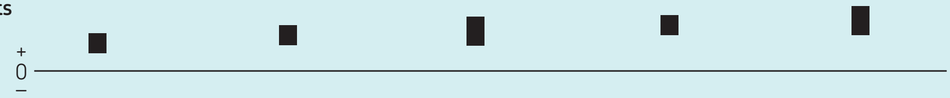
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1)] Tolerance classes									
				n6Ⓔ		p6Ⓔ		p7Ⓔ		r6Ⓔ		r7Ⓔ	
				Deviations (shaft diameter) / Theoretical interference (–) / Probable interference (–)									
over	incl.	low	high										
mm		µm		µm									
50	**80**	–15	0	+39	+20	+51	+32	+62	+32	–	–	–	–
				–54	–20	–66	–32	–77	–32	–	–	–	–
				–50	–24	–62	–36	–72	–38	–	–	–	–
80	**100**	–20	0	+45	+23	+59	+37	+72	+37	+73	+51	+86	+51
				–65	–23	–79	–37	–92	–37	–93	–51	–106	–51
				–59	–29	–73	–43	–85	–44	–87	–57	–99	–58
100	**120**	–20	0	+45	+23	+59	+37	+72	+37	+76	+54	+89	+54
				–65	–23	–79	–37	–92	–37	–96	–54	–109	–54
				–59	–29	–73	–43	–85	–44	–90	–60	–102	–61
120	**140**	–25	0	+52	+27	+68	+43	+83	+43	+88	+63	+103	+63
				–77	–27	–93	–43	–108	–43	–113	–63	–128	–63
				–70	–34	–86	–50	–100	–51	–106	–70	–120	–71
140	**160**	–25	0	+52	+27	+68	+43	+83	+43	+90	+65	+105	+65
				–77	–27	–93	–43	–108	–43	–115	–65	–130	–65
				–70	–34	–86	–50	–100	–51	–108	–72	–122	–73
160	**180**	–25	0	+52	+27	+68	+43	+83	+43	+93	+68	+108	+68
				–77	–27	–93	–43	–108	–43	–118	–68	–133	–68
				–70	–34	–86	–50	–100	–51	–111	–75	–125	–76
180	**200**	–30	0	+60	+31	+79	+50	+96	+50	+106	+77	+123	+77
				–90	–31	–109	–50	–126	–50	–136	–77	–153	–77
				–82	–39	–101	–58	–116	–60	–128	–85	–143	–87
200	**225**	–30	0	+60	+31	+79	+50	+96	+50	+109	+80	+126	+80
				–90	–31	–109	–50	–126	–50	–139	–80	–156	–80
				–82	–39	–101	–58	–116	–60	–131	–88	–146	–90
225	**250**	–30	0	+60	+31	+79	+50	+96	+50	+113	+84	+130	+84
				–90	–31	–109	–50	–126	–50	–143	–84	–160	–84
				–82	–39	–101	–58	–116	–60	–135	–92	–150	–94
250	**280**	–35	0	+66	+34	+88	+56	+108	+56	+126	+94	+146	+94
				–101	–34	–123	–56	–143	–56	–161	–94	–181	–94
				–92	–43	–114	–65	–131	–68	–152	–103	–169	–106
280	**315**	–35	0	+66	+34	+88	+56	+108	+56	+130	+98	+150	+98
				–101	–34	–123	–56	–143	–56	–165	–98	–185	–98
				–92	–43	–114	–65	–131	–68	–156	–107	–173	–110
315	**355**	–40	0	+73	+37	+98	+62	+119	+62	+144	+108	+165	+108
				–113	–37	–138	–62	–159	–62	–184	–108	–205	–108
				–102	–48	–127	–73	–146	–75	–173	–119	–192	–121
355	**400**	–40	0	+73	+37	+98	+62	+119	+62	+150	+114	+171	+114
				–113	–37	–138	–62	–159	–62	–190	–114	–211	–114
				–102	–48	–127	–73	–146	–75	–179	–125	–198	–127
400	**450**	–45	0	+80	+40	+108	+68	+131	+68	+166	+126	+189	+126
				–125	–40	–153	–68	–176	–68	–211	–126	–234	–126
				–113	–52	–141	–80	–161	–83	–199	–138	–219	–141

Table 15

Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance $t_{\Delta dmp}$		Shaft diameter deviations, resultant fits[1] Tolerance classes									
				n6Ⓔ		p6Ⓔ		p7Ⓔ		r6Ⓔ		r7Ⓔ	
				Deviations (shaft diameter) / Theoretical interference (–) / Probable interference (–)									
over	incl.	low	high										
mm		µm		µm									
450	**500**	–45	0	+80	+40	+108	+68	+131	+68	+172	+132	+195	+132
				–125	–40	–153	–68	–176	–68	–217	–132	–240	–132
				–113	–52	–141	–80	–161	–83	–205	–144	–225	–147
500	**560**	–50	0	+88	+44	+122	+78	+148	+78	+194	+150	+220	+150
				–138	–44	–172	–78	–198	–78	–244	–150	–270	–150
				–125	–57	–159	–91	–182	–94	–231	–163	–254	–166
560	**630**	–50	0	+88	+44	+122	+78	+148	+78	+199	+155	+225	+155
				–138	–44	–172	–78	–198	–78	–249	–155	–275	–155
				–125	–57	–159	–91	–182	–94	–236	–168	–259	–171
630	**710**	–75	0	+100	+50	+138	+88	+168	+88	+225	+175	+255	+175
				–175	–50	–213	–88	–243	–88	–300	–175	–330	–175
				–158	–67	–196	–105	–221	–110	–283	–192	–308	–197
710	**800**	–75	0	+100	+50	+138	+88	+168	+88	+235	+185	+265	+185
				–175	–50	–213	–88	–243	–88	–310	–185	–340	–185
				–158	–67	–196	–105	–221	–110	–293	–202	–318	–207
800	**900**	–100	0	+112	+56	+156	+100	+190	+100	+266	+210	+300	+210
				–212	–56	–256	–100	–290	–100	–366	–210	–400	–210
				–192	–76	–236	–120	–263	–127	–346	–230	–373	–237
900	**1 000**	–100	0	+112	+56	+156	+100	+190	+100	+276	+220	+310	+220
				–212	–56	–256	–100	–290	–100	–376	–220	–410	–220
				–192	–76	–236	–120	–263	–127	–356	–240	–383	–247
1 000	**1 120**	–125	0	+132	+66	+186	+120	+225	+120	+316	+250	+355	+250
				–257	–66	–311	–120	–350	–120	–441	–250	–480	–250
				–233	–90	–287	–144	–317	–153	–417	–274	–447	–283
1 120	**1 250**	–125	0	+132	+66	+186	+120	+225	+120	+326	+260	+365	+260
				–257	–66	–311	–120	–350	–120	–451	–260	–490	–260
				–233	–90	–287	–144	–317	–153	–427	–284	–457	–293
1 250	**1 400**	–160	0	+156	+78	+218	+140	+265	+140	+378	+300	+425	+300
				–316	–78	–378	–140	–425	–140	–538	–300	–585	–300
				–286	–108	–348	–170	–385	–180	–508	–330	–545	–340
1 400	**1 600**	–160	0	+156	+78	+218	+140	+265	+140	+408	+330	+455	+330
				–316	–78	–378	–140	–425	–140	–568	–330	–615	–330
				–286	–108	–348	–170	–385	–180	–538	–360	–575	–370
1 600	**1 800**	–200	0	+184	+92	+262	+170	+320	+170	+462	+370	+520	+370
				–384	–92	–462	–170	–520	–170	–662	–370	–720	–370
				–349	–127	–427	–205	–470	–220	–627	–405	–670	–420
1 800	**2 000**	–200	0	+184	+92	+262	+170	+320	+170	+492	+400	+550	+400
				–384	–92	–462	–170	–520	–170	–692	–400	–750	–400
				–349	–127	–427	–205	–470	–220	–657	–435	–700	–450

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Table 16

Shaft tolerances and resultant fits

Shaft Nominal diameter d over	incl.	Bearing Bore diameter tolerance $t_{\Delta dmp}$ low	high	Shaft diameter deviations, resultant fits[1] Tolerance classes r6+IT6 Deviations (shaft diameter) Theoretical interference (–) Probable interference (–)		r7+IT7	
mm		µm		µm			
315	**355**	–40	0	+180 –220 –209	+144 –144 –155	+222 –262 –248	+165 –165 –179
355	**400**	–40	0	+186 –226 –215	+150 –150 –161	+228 –268 –254	+171 –171 –185
400	**450**	–45	0	+206 –251 –239	+166 –166 –178	+252 –297 –282	+189 –189 –204
450	**500**	–45	0	+212 –257 –245	+172 –172 –184	+258 –303 –288	+195 –195 –210
500	**560**	–50	0	+238 –288 –274	+194 –194 –208	+290 –340 –323	+220 –220 –237
560	**630**	–50	0	+243 –293 –279	+199 –199 –213	+295 –345 –328	+225 –225 –242
630	**710**	–75	0	+275 –350 –333	+225 –225 –242	+335 –410 –387	+255 –255 –278
710	**800**	–75	0	+285 –360 –343	+235 –235 –252	+345 –420 –397	+265 –265 –288
800	**900**	–100	0	+322 –422 –401	+266 –266 –287	+390 –490 –462	+300 –300 –328
900	**1 000**	–100	0	+332 –432 –411	+276 –276 –297	+400 –500 –472	+310 –310 –338
1 000	**1 120**	–125	0	+382 –507 –482	+316 –316 –341	+460 –585 –552	+355 –355 –388
1 120	**1 250**	–125	0	+392 –517 –492	+326 –326 –351	+470 –595 –562	+365 –365 –398

Table 16

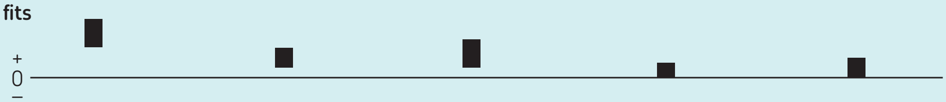
Shaft tolerances and resultant fits

Shaft Nominal diameter d		**Bearing** Bore diameter tolerance $t_{\Delta dmp}$		**Shaft diameter deviations, resultant fits**[1] Tolerance classes			
				r6+IT6		r7+IT7	
				Deviations (shaft diameter)			
				Theoretical interference (–)			
over	incl.	low	high	Probable interference (–)			
mm		µm		µm			
1 250	**1 400**	–160	0	+456	+378	+550	+425
				–616	–378	–710	–425
				–586	–408	–669	–466
1 400	**1 600**	–160	0	+486	+408	+580	+455
				–646	–408	–740	–455
				–616	–438	–699	–496
1 600	**1 800**	–200	0	+554	+462	+670	+520
				–754	–462	–870	–520
				–718	–498	–820	–570
1 800	**2 000**	–200	0	+584	+492	+700	+550
				–784	–492	–900	–550
				–748	–528	–850	–600

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

Table 17

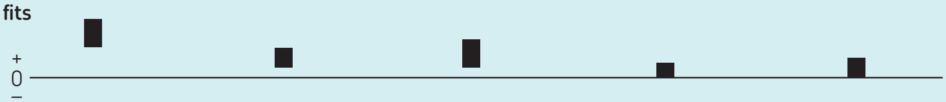
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$			Housing bore diameter deviations, resultant fits[1)] Tolerance classes									
					F7Ⓔ		G6Ⓔ		G7Ⓔ		H5Ⓔ		H6Ⓔ	
over	incl.	low	high	Deviations (housing bore diameter) / Theoretical clearance (+) / Probable clearance (+)										
mm		µm		µm										
6	**10**	0	–8		+13	+28	+5	+14	+5	+20	0	+6	0	+9
					+13	+36	+5	+22	+5	+28	0	+14	0	+17
					+16	+33	+7	+20	+8	+25	+2	+12	+2	+15
10	**18**	0	–8		+16	+34	+6	+17	+6	+24	0	+8	0	+11
					+16	+42	+6	+25	+6	+32	0	+16	0	+19
					+19	+39	+8	+23	+9	+29	+2	+14	+2	+17
18	**30**	0	–9		+20	+41	+7	+20	+7	+28	0	+9	+0	+13
					+20	+50	+7	+29	+7	+37	0	+18	0	+22
					+23	+47	+10	+26	+10	+34	+2	+16	+3	+19
30	**50**	0	–11		+25	+50	+9	+25	+9	+34	0	+11	0	+16
					+25	+61	+9	+36	+9	+45	0	+22	0	+27
					+29	+57	+12	+33	+13	+41	+3	+19	+3	+24
50	**80**	0	–13		+30	+60	+10	+29	+10	+40	0	+13	0	+19
					+30	+73	+10	+42	+10	+53	0	+26	0	+32
					+35	+68	+14	+38	+15	+48	+3	+23	+4	+28
80	**120**	0	–15		+36	+71	+12	+34	+12	+47	0	+15	0	+22
					+36	+86	+12	+49	+12	+62	0	+30	0	+37
					+41	+81	+17	+44	+17	+57	+4	+26	+5	+32
120	**150**	0	–18		+43	+83	+14	+39	+14	+54	0	+18	0	+25
					+43	+101	+14	+57	+14	+72	0	+36	0	+43
					+50	+94	+20	+51	+21	+65	+5	+31	+6	+37
150	**180**	0	–25		+43	+83	+14	+39	+14	+54	0	+18	0	+25
					+43	+108	+14	+64	+14	+79	0	+43	0	+50
					+51	+100	+21	+57	+22	+71	+6	+37	+7	+43
180	**250**	0	–30		+50	+96	+15	+44	+15	+61	0	+20	0	+29
					+50	+126	+15	+74	+15	+91	0	+50	0	+59
					+60	+116	+23	+66	+25	+81	+6	+44	+8	+51
250	**315**	0	–35		+56	+108	+17	+49	+17	+69	0	+23	0	+32
					+56	+143	+17	+84	+17	+104	0	+58	0	+67
					+68	+131	+26	+75	+29	+92	+8	+50	+9	+58
315	**400**	0	–40		+62	+119	+18	+54	+18	+75	0	+25	0	+36
					+62	+159	+18	+94	+18	+115	0	+65	0	+76
					+75	+146	+29	+83	+31	+102	+8	+57	+11	+65
400	**500**	0	–45		+68	+131	+20	+60	+20	+83	0	+27	0	+40
					+68	+176	+20	+105	+20	+128	0	+72	0	+85
					+83	+161	+32	+93	+35	+113	+9	+63	+12	+73
500	**630**	0	–50		+76	+146	+22	+66	+22	+92	0	+28	0	+44
					+76	+196	+22	+116	+22	+142	0	+78	0	+94
					+92	+180	+35	+103	+38	+126	+10	+68	+13	+81
630	**800**	0	–75		+80	+160	+24	+74	+24	+104	0	+32	0	+50
					+80	+235	+24	+149	+24	+179	0	+107	0	+125
					+102	+213	+41	+132	+46	+157	+12	+95	+17	+108

Table 17

Housing tolerances and resultant fits



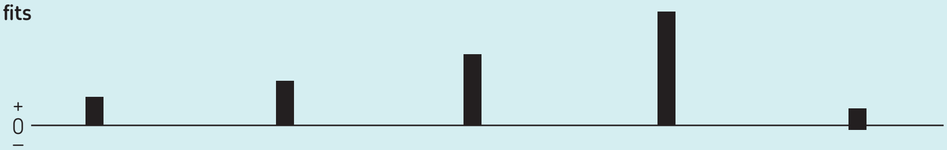
Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance $t_{\Delta Dmp}$		**Housing bore diameter deviations, resultant fits**[1] Tolerance classes									
				F7Ⓔ		G6Ⓔ		G7Ⓔ		H5Ⓔ		H6Ⓔ	
over	incl.	low	high	Deviations (housing bore diameter) / Theoretical clearance (+) / Probable clearance (+)									
mm		µm		µm									
800	**1 000**	0	–100	+86	+176	+26	+82	+26	+116	0	+36	0	+56
				+86	+276	+26	+182	+26	+216	0	+136	0	+156
				+113	+249	+46	+162	+53	+189	+14	+122	+20	+136
1 000	**1 250**	0	–125	+98	+203	+28	+94	+28	+133	0	+42	0	+66
				+98	+328	+28	+219	+28	+258	0	+167	0	+191
				+131	+295	+52	+195	+61	+225	+17	+150	+24	+167
1 250	**1 600**	0	–160	+110	+235	+30	+108	+30	+155	0	+50	0	+78
				+110	+395	+30	+268	+30	+315	0	+210	0	+238
				+150	+355	+60	+238	+70	+275	+21	+189	+30	+208
1 600	**2 000**	0	–200	+120	+270	+32	+124	+32	+182	0	+60	0	+92
				+120	+470	+32	+324	+32	+382	0	+260	0	+292
				+170	+420	+67	+289	+82	+332	+25	+235	+35	+257
2 000	**2 500**	0	–250	+130	+305	+34	+144	+34	+209	0	+70	0	+110
				+130	+555	+34	+394	+34	+459	0	+320	0	+360
				+189	+496	+77	+351	+93	+400	+30	+290	+43	+317

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Table 18

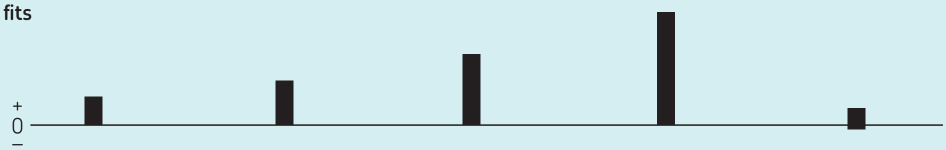
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		Housing bore diameter deviations, resultant fits[1) Tolerance classes									
				H7Ⓔ		H8Ⓔ		H9Ⓔ		H10Ⓔ		J6Ⓔ	
over	incl.	low	high	Deviations (housing bore diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
mm		µm		µm									
6	**10**	0	–8	0	+15	0	+22	0	+36	0	+58	–4	+5
				0	+23	0	+30	0	+44	0	+66	–4	+13
				+3	+20	+3	+27	+3	+41	+3	+63	–2	+11
10	**18**	0	–8	0	+18	0	+27	0	+43	0	+70	–5	+6
				0	+26	0	+35	0	+51	0	+78	–5	+14
				+3	+23	+3	+32	+3	+48	+3	+75	–3	+12
18	**30**	0	–9	0	+21	0	+33	0	+52	0	+84	–5	+8
				0	+30	0	+42	0	+61	0	+93	–5	+17
				+3	+27	+3	+39	+4	+57	+4	+89	–2	+14
30	**50**	0	–11	0	+25	0	+39	0	+62	0	+100	–6	+10
				0	+36	0	+50	0	+73	0	+111	–6	+21
				+4	+32	+4	+46	+5	+68	+5	+106	–3	+18
50	**80**	0	–13	0	+30	0	+46	0	+74	0	+120	–6	+13
				0	+43	0	+59	0	+87	0	+133	–6	+26
				+5	+38	+5	+54	+5	+82	+6	+127	–2	+22
80	**120**	0	–15	0	+35	0	+54	0	+87	0	+140	–6	+16
				0	+50	0	+69	0	+102	0	+155	–6	+31
				+5	+45	+6	+63	+6	+96	+7	+148	–1	+26
120	**150**	0	–18	0	+40	0	+63	0	+100	0	+160	–7	+18
				0	+58	0	+81	0	+118	0	+178	–7	+36
				+7	+51	+7	+74	+8	+110	+8	+170	–1	+30
150	**180**	0	–25	0	+40	0	+63	0	+100	0	+160	–7	+18
				0	+65	0	+88	0	+125	0	+185	–7	+43
				+8	+57	+10	+78	+10	+115	+11	+174	0	+36
180	**250**	0	–30	0	+46	0	+72	0	+115	0	+185	–7	+22
				0	+76	0	+102	0	+145	0	+215	–7	+52
				+10	+66	+12	+90	+13	+132	+13	+202	+1	+44
250	**315**	0	–35	0	+52	0	+81	0	+130	0	+210	–7	+25
				0	+87	0	+116	0	+165	0	+245	–7	+60
				+12	+75	+13	+103	+15	+150	+16	+229	+2	+51
315	**400**	0	–40	0	+57	0	+89	0	+140	0	+230	–7	+29
				0	+97	0	+129	0	+180	0	+270	–7	+69
				+13	+84	+15	+114	+17	+163	+18	+252	+4	+58
400	**500**	0	–45	0	+63	0	+97	0	+155	0	+250	–7	+33
				0	+108	0	+142	0	+200	0	+295	–7	+78
				+15	+93	+17	+125	+19	+181	+20	+275	+5	+66
500	**630**	0	–50	0	+70	0	+110	0	+175	0	+280	–	–
				0	+120	0	+160	0	+225	0	+330	–	–
				+16	+104	+19	+141	+21	+204	+22	+308	–	–

Table 18

Housing tolerances and resultant fits

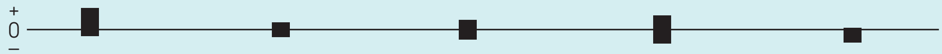


Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		**Housing bore diameter deviations, resultant fits**[1)] Tolerance classes									
				H7Ⓔ		H8Ⓔ		H9Ⓔ		H10Ⓔ		J6Ⓔ	
over	incl.	low	high	Deviations (housing bore diameter) Theoretical interference (–)/clearance (+) Probable interference (–)/clearance (+)									
mm		µm		µm									
630	**800**	0	–75	0	+80	0	+125	0	+200	0	+320	–	–
				0	+155	0	+200	0	+275	0	+395	–	–
				+22	+133	+27	+173	+30	+245	+33	+362	–	–
800	**1 000**	0	–100	0	+90	0	+140	0	+230	0	+360	–	–
				0	+190	0	+240	0	+330	0	+460	–	–
				+27	+163	+33	+207	+39	+291	+43	+417	–	–
1 000	**1 250**	0	–125	0	+105	0	+165	0	+260	0	+420	–	–
				0	+230	0	+290	0	+385	0	+545	–	–
				+33	+197	+41	+249	+48	+337	+53	+492	–	–
1 250	**1 600**	0	–160	0	+125	0	+195	0	+310	0	+500	–	–
				0	+285	0	+355	0	+470	0	+660	–	–
				+40	+245	+51	+304	+60	+410	+67	+593	–	–
1 600	**2 000**	0	–200	0	+150	0	+230	0	+370	0	+600	–	–
				0	+350	0	+430	0	+570	0	+800	–	–
				+50	+300	+62	+368	+74	+496	+83	+717	–	–
2 000	**2 500**	0	–250	0	+175	0	+280	0	+440	0	+700	–	–
				0	+425	0	+530	0	+690	0	+950	–	–
				+59	+366	+77	+453	+91	+599	+103	+847	–	–

1) Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

Table 19

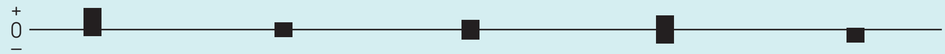
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		Housing bore diameter deviations, resultant fits[1)] Tolerance classes									
				J7Ⓔ		JS5Ⓔ		JS6Ⓔ		JS7Ⓔ		K5Ⓔ	
				Deviations (housing bore diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
over	incl.	low	high										
mm		µm		µm									
6	**10**	0	–8	–7	+8	–3	+3	–4,5	+4,5	–7,5	+7,5	–5	+1
				–7	+16	–3	+11	–4,5	+12,5	–7,5	+15,5	–5	+9
				–4	+13	–1	+9	–3	+11	–5	+13	–3	+7
10	**18**	0	–8	–8	+10	–4	+4	–5,5	+5,5	–9	+9	–6	+2
				–8	+18	–4	+12	–5,5	+13,5	–9	+17	–6	+10
				–5	+15	–2	+10	–3	+11	–6	+14	–4	+8
18	**30**	0	–9	–9	+12	–4,5	+4,5	–6,5	+6,5	–10,5	+10,5	–8	+1
				–9	+21	–4,5	+13,5	–6,5	+15,5	–10,5	+19,5	–8	+10
				–6	+18	–2	+11	–4	+13	–7	+16	–6	+8
30	**50**	0	–11	–11	+14	–5,5	+5,5	–8	+8	–12,5	+12,5	–9	+2
				–11	+25	–5,5	+16,5	–8	+19	–12,5	+23,5	–9	+13
				–7	+21	–3	+14	–5	+16	–9	+20	–6	+10
50	**80**	0	–13	–12	+18	–6,5	+6,5	–9,5	+9,5	–15	+15	–10	+3
				–12	+31	–6,5	+19,5	–9,5	+22,5	–15	+28	–10	+16
				–7	+26	–3	+16	–6	+19	–10	+23	–7	+13
80	**120**	0	–15	–13	+22	–7,5	+7,5	–11	+11	–17,5	+17,5	–13	+2
				–13	+37	–7,5	+22,5	–11	+26	–17,5	+32,5	–13	+17
				–8	+32	–4	+19	–6	+21	–12	+27	–9	+13
120	**150**	0	–18	–14	+26	–9	+9	–12,5	+12,5	–20	+20	–15	+3
				–14	+44	–9	+27	–12,5	+30,5	–20	+38	–15	+21
				–7	+37	–4	+22	–7	+25	–13	+31	–10	+16
150	**180**	0	–25	–14	+26	–9	+9	–12,5	+12,5	–20	+20	–15	+3
				–14	+51	–9	+34	–12,5	+37,5	–20	+45	–15	+28
				–6	+43	–3	+28	–6	+31	–12	+37	–9	+22
180	**250**	0	–30	–16	+30	–10	+10	–14,5	+14,5	–23	+23	–18	+2
				–16	+60	–10	+40	–14,5	+44,5	–23	+53	–18	+32
				–6	+50	–4	+34	–6	+36	–13	+43	–12	+26
250	**315**	0	–35	–16	+36	–11,5	+11,5	–16	+16	–26	+26	–20	+3
				–16	+71	–11,5	+46,5	–16	–51	–26	+61	–20	+38
				–4	+59	–4	+39	–7	+42	–14	+49	–12	+30
315	**400**	0	–40	–18	+39	–12,5	+12,5	–18	+18	–28,5	+28,5	–22	+3
				–18	+79	–12,5	+52,5	–18	+58	–28,5	+68,5	–22	+43
				–5	+66	–4	+44	–7	+47	–15	+55	–14	+35
400	**500**	0	–45	–20	+43	–13,5	+13,5	–20	+20	–31,5	+31,5	–25	+2
				–20	+88	–13,5	+58,5	–20	+65	–31,5	+76,5	–25	+47
				–5	+73	–4	+49	–8	+53	–17	+62	–16	+38
500	**630**	0	–50	–	–	–14	+14	–22	+22	–35	+35	–	–
				–	–	–14	+64	–22	+72	–35	+85	–	–
				–	–	–4	+54	–9	+59	–19	+69	–	–

Table 19

Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		Housing bore diameter deviations, resultant fits[1] Tolerance classes									
				J7Ⓔ		JS5Ⓔ		JS6Ⓔ		JS7Ⓔ		K5Ⓔ	
				Deviations (housing bore diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
over	incl.	low	high										
mm		µm		µm									
630	**800**	0	–75	–	–	–16	+16	–25	+25	–40	+40	–	–
				–	–	–16	+91	–25	+100	–40	+115	–	–
				–	–	–4	+79	–8	+83	–18	+93	–	–
800	**1 000**	0	–100	–	–	–18	+18	–28	+28	–45	+45	–	–
				–	–	–18	+118	–28	+128	–45	+145	–	–
				–	–	–4	+104	–8	+108	–18	+118	–	–
1 000	**1 250**	0	–125	–	–	–21	+21	–33	+33	–52	+52	–	–
				–	–	–21	+146	–33	+158	–52	+177	–	–
				–	–	–4	+129	–9	+134	–20	+145	–	–
1 250	**1 600**	0	–160	–	–	–25	+25	–39	+39	–62	+62	–	–
				–	–	–25	+185	–39	+199	–62	+222	–	–
				–	–	–4	+164	–9	+169	–22	+182	–	–
1 600	**2 000**	0	–200	–	–	–30	+30	–46	+46	–75	+75	–	–
				–	–	–30	+230	–46	+246	–75	+275	–	–
				–	–	–5	+205	–11	+211	–25	+225	–	–
2 000	**2 500**	0	–250	–	–	–35	+35	–55	+55	–87	+87	–	–
				–	–	–35	+285	–55	+305	–87	+337	–	–
				–	–	–5	+255	–12	+262	–28	+278	–	–

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Table 20

Housing tolerances and resultant fits

Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		Housing bore diameter deviations, resultant fits[1)] Tolerance classes									
				K6Ⓔ		K7Ⓔ		M5Ⓔ		M6Ⓔ		M7Ⓔ	
over	incl.	low	high	Deviations (housing bore diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
mm		µm		µm									
6	**10**	0	–8	–7	+2	–10	+5	–10	–4	–12	–3	–15	0
				–7	+10	–10	+13	–10	+4	–12	+5	–15	+8
				–5	+8	–7	+10	–8	+2	–10	+3	–12	+5
10	**18**	0	–8	–9	+2	–12	+6	–12	–4	–15	–4	–18	0
				–9	+10	–12	+14	–12	+4	–15	+4	–18	+8
				–7	+8	–9	+11	–10	+2	–13	+2	–15	+5
18	**30**	0	–9	–11	+2	–15	+6	–14	–4	–17	–4	–21	0
				–11	+11	–15	+15	–14	+4	–17	+5	–21	+9
				–8	+8	–12	+12	–12	+2	–14	+2	–18	+6
30	**50**	0	–11	–13	+3	–18	+7	–16	–5	–20	–4	–25	0
				–13	+14	–18	+18	–16	+6	–20	+7	–25	+11
				–10	+11	–14	+14	–13	+3	–17	+4	–21	+7
50	**80**	0	–13	–15	+4	–21	+9	–19	–6	–24	–5	–30	0
				–15	+17	–21	+22	–19	+7	–24	+8	–30	+13
				–11	+13	–16	+17	–16	+4	–20	+4	–25	+8
80	**120**	0	–15	–18	+4	–25	+10	–23	–8	–28	–6	–35	0
				–18	+19	–25	+25	–23	+7	–28	+9	–35	+15
				–13	+14	–20	+20	–19	+3	–23	+4	–30	+10
120	**150**	0	–18	–21	+4	–28	+12	–27	–9	–33	–8	–40	0
				–21	+22	–28	+30	–27	+9	–33	+10	–40	+18
				–15	+16	–21	+23	–22	+4	–27	+4	–33	+11
150	**180**	0	–25	–21	+4	–28	+12	–27	–9	–33	–8	–40	0
				–21	+29	–28	+37	–27	+16	–33	+17	–40	+25
				–14	+22	–20	+29	–21	+10	–26	+10	–32	+17
180	**250**	0	–30	–24	+5	–33	+13	–31	–11	–37	–8	–46	0
				–24	+35	–33	+43	–31	+19	–37	+22	–46	+30
				–16	+27	–23	+33	–25	+13	–29	+14	–36	+20
250	**315**	0	–35	–27	+5	–36	+16	–36	–13	–41	–9	–52	0
				–27	+40	–36	+51	–36	+22	–41	+26	–52	+35
				–18	+31	–24	+39	–28	+14	–32	+17	–40	+23
315	**400**	0	–40	–29	+7	–40	+17	–39	–14	–46	–10	–57	0
				–29	+47	–40	+57	–39	+26	–46	+30	–57	+40
				–18	+36	–27	+44	–31	+18	–35	+19	–44	+27
400	**500**	0	–45	–32	+8	–45	+18	–43	–16	–50	–10	–63	0
				–32	+53	–45	+63	–43	+29	–50	+35	–63	+45
				–20	+41	–30	+48	–34	+20	–38	+23	–48	+30
500	**630**	0	–50	–44	0	–70	0	–	–	–70	–26	–96	–26
				–44	+50	–70	+50	–	–	–70	+24	–96	+24
				–31	+37	–54	+34	–	–	–57	+11	–80	+8

Table 20

Housing tolerances and resultant fits

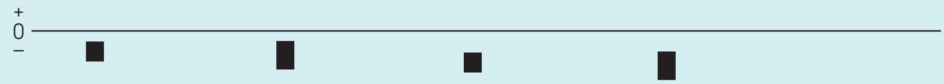
Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance $t_{\Delta Dmp}$		**Housing bore diameter deviations, resultant fits**[1] Tolerance classes									
				K6Ⓔ		K7Ⓔ		M5Ⓔ		M6Ⓔ		M7Ⓔ	
				Deviations (housing bore diameter) / Theoretical interference (–)/clearance (+) / Probable interference (–)/clearance (+)									
over	incl.	low	high										
mm		µm		µm									
630	**800**	0	–75	–50	0	–80	0	–	–	–80	–30	–110	–30
				–50	+75	–80	+75	–	–	–80	+45	–110	+45
				–33	+58	–58	+53	–	–	–63	+28	–88	+23
800	**1 000**	0	–100	–56	0	–90	0	–	–	–90	–34	–124	–34
				–56	+100	–90	+100	–	–	–90	+66	–124	+66
				–36	+80	–63	+73	–	–	–70	+46	–97	+39
1 000	**1 250**	0	–125	–66	0	–105	0	–	–	–106	–40	–145	–40
				–66	+125	–105	+125	–	–	–106	+85	–145	+85
				–42	+101	–72	+92	–	–	–82	+61	–112	+52
1 250	**1 600**	0	–160	–78	0	–125	0	–	–	–126	–48	–173	–48
				–78	+160	–125	+160	–	–	–126	+112	–173	+112
				–48	+130	–85	+120	–	–	–96	+82	–133	+72
1 600	**2 000**	0	–200	–92	0	–150	0	–	–	–158	–58	–208	–58
				–92	+200	–150	+200	–	–	–150	+142	–208	+142
				–57	+165	–100	+150	–	–	–115	+107	–158	+92
2 000	**2 500**	0	–250	–110	0	–175	0	–	–	–178	–68	–243	–68
				–110	+250	–175	+250	–	–	–178	+182	–243	+182
				–67	+207	–116	+191	–	–	–135	+139	–184	+123

[1] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Table 21

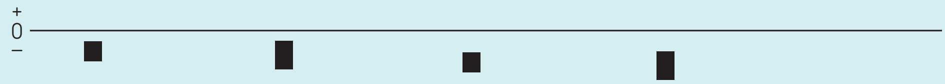
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance $t_{\Delta Dmp}$		Housing bore diameter deviations, resultant fits[1) Tolerance classes							
				N6Ⓔ		N7Ⓔ		P6Ⓔ		P7Ⓔ	
ove	incl.	low	high	Deviations (housing bore diameter) Theoretical interference (–)/clearance (+) Probable interference (–)/clearance (+)							
mm		µm		µm							
6	**10**	0	–8	–16	–7	–19	–4	–21	–12	–24	–9
				–16	+1	–19	+4	–21	–4	–24	–1
				–14	–1	–16	+1	–19	–6	–21	–4
10	**18**	0	–8	–20	–9	–23	–5	–26	–15	–29	–11
				–20	–1	–23	+3	–26	–7	–29	–3
				–18	–3	–20	0	–24	–9	–26	–6
18	**30**	0	–9	–24	–11	–28	–7	–31	–18	–35	–14
				–24	–2	–28	+2	–31	–9	–35	–5
				–21	–5	–25	–1	–28	–12	–32	–8
30	**50**	0	–11	–28	–12	–33	–8	–37	–21	–42	–17
				–28	–1	–33	+3	–37	–10	–42	–6
				–25	–4	–29	–1	–34	–13	–38	–10
50	**80**	0	–13	–33	–14	–39	–9	–45	–26	–51	–21
				–33	–1	–39	+4	–45	–13	–51	–8
				–29	–5	–34	–1	–41	–17	–46	–13
80	**120**	0	–15	–38	–16	–45	–10	–52	–30	–59	–24
				–38	–1	–45	+5	–52	–15	–59	–9
				–33	–6	–40	0	–47	–20	–54	–14
120	**150**	0	–18	–45	–20	–52	–12	–61	–36	–68	–28
				–45	–2	–52	+6	–61	–18	–68	–10
				–39	–8	–45	–1	–55	–24	–61	–17
150	**180**	0	–25	–45	–20	–52	–12	–61	–36	–68	–28
				–45	+5	–52	+13	–61	–11	–68	–3
				–38	–2	–44	+5	–54	–18	–60	–11
180	**250**	0	–30	–51	–22	–60	–14	–70	–41	–79	–33
				–51	+8	–60	+16	–70	–11	–79	–3
				–43	0	–50	+6	–62	–19	–69	–13
250	**315**	0	–35	–57	–25	–66	–14	–79	–47	–88	–36
				–57	+10	–66	+21	–79	–12	–88	–1
				–48	+1	–54	+9	–70	–21	–76	–13
315	**400**	0	–40	–62	–26	–73	–16	–87	–51	–98	–41
				–62	+14	–73	+24	–87	–11	–98	–1
				–51	+3	–60	+11	–76	–22	–85	–14
400	**500**	0	–45	–67	–27	–80	–17	–95	–55	–108	–45
				–67	+18	–80	+28	–95	–10	–108	0
				–55	+6	–65	+13	–83	–22	–93	–15
500	**630**	0	–50	–88	–44	–114	–44	–122	–78	–148	–78
				–88	+6	–114	+6	–122	–28	–148	–28
				–75	–7	–98	–10	–109	–41	–132	–44

Table 21

Housing tolerances and resultant fits



Housing bore diameter deviations, resultant fits[1)]
Tolerance classes

Deviations (housing bore diameter)
Theoretical interference (–)/clearance (+)
Probable interference (–)/clearance (+)

Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance $t_{\Delta Dmp}$		N6Ⓔ		N7Ⓔ		P6Ⓔ		P7Ⓔ	
over	incl.	low	high								
mm		µm		µm							
630	800	0	–75	–100	–50	–130	–50	–138	–88	–168	–88
				–100	+25	–130	+25	–138	–13	–168	–13
				–83	+8	–108	+3	–121	–30	–146	–35
800	1 000	0	–100	–112	–56	–146	–56	–156	–100	–190	–100
				–112	+44	–146	+44	–156	0	–190	0
				–92	+24	–119	+17	–136	–20	–163	–27
1 000	1 250	0	–125	–132	–66	–171	–66	–186	–120	–225	–120
				–132	+59	–171	+59	–186	+5	–225	+5
				–108	+35	–138	+26	–162	–19	–192	–28
1 250	1 600	0	–160	–156	–78	–203	–78	–218	–140	–265	–140
				–156	+82	–203	+82	–218	+20	–265	+20
				–126	+52	–163	+42	–188	–10	–225	–20
1 600	2 000	0	–200	–184	–92	–242	–92	–262	–170	–320	–170
				–184	+108	–242	+108	–262	+30	–320	+30
				–149	+73	–192	+58	–227	–5	–270	–20
2 000	2 500	0	–250	–220	–110	–285	–110	–305	–195	–370	–195
				–220	+140	–285	+140	–305	+55	–370	+55
				–177	+97	–226	+81	–262	+12	–311	–4

[1)] Values are valid for most bearings with Normal tolerances. For exceptions, refer to *Tolerances and resultant fits*, page 153.

B.6 Bearing interfaces

Provisions for mounting and dismounting

Particularly when large bearings are involved, SKF recommends that during the design stage you make provisions to facilitate mounting and dismounting, including:

- slots or recesses machined in the shaft or housing shoulders so that withdrawal tools can be used (fig. 8)
- threaded holes in the housing shoulders so that bolts can be used for dismounting (fig. 9)
- oil supply ducts and distribution grooves in the shaft to enable the oil injection method to be used (fig. 10)

  Recommended dimensions for oil supply ducts and distribution grooves are listed in table 22, and for threaded holes in table 23. When using the oil injection method, Ra should not exceed 1,6 µm.

Fig. 8

Slots or recesses in the shaft to apply withdrawal tools

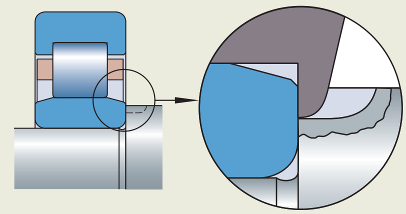


Fig. 9

Threaded holes in the housing to push bearing with bolts from its seat

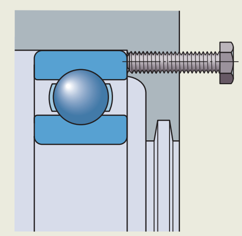


Fig. 10

Duct and groove for oil injection

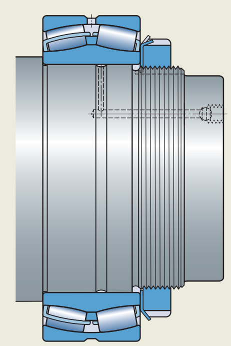
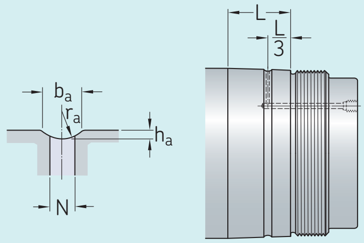


Table 22

Recommended dimensions for oil supply ducts and distribution grooves

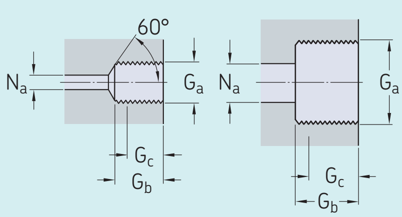


Seat diameter >	≤	**Dimensions** b_a	h_a	r_a	N
mm		mm			
–	**100**	3	0,5	2,5	2,5
100	**150**	4	0,8	3	3
150	**200**	4	0,8	3	3
200	**250**	5	1	4	4
250	**300**	5	1	4	4
300	**400**	6	1,25	4,5	5
400	**500**	7	1,5	5	5
500	**650**	8	1,5	6	6
650	**800**	10	2	7	7
800	**1 000**	12	2,5	8	8

L = width of bearing seat

Table 23

Design and recommended dimensions for threaded holes for connecting oil supply



Design A Design B

Thread G_a	**Design**	**Dimensions** G_b	G_c 1)	N_a max.
–	–	mm		
M6	**A**	10	8	3
G 1/8	**A**	12	10	3
G 1/4	**A**	15	12	5
G 3/8	**B**	15	12	8
G 1/2	**B**	18	14	8
G 3/4	**B**	20	16	8

1) Effective threaded length

Axial location of bearing rings

Typically, it is not sufficient to use an interference fit alone to axially locate a bearing ring on a cylindrical seat. Common ways of locating bearing rings axially include:

- shaft or housing shoulders
- lock nuts or threaded rings (fig. 11 and fig. 12)
- end plates or housing covers (fig. 13 and fig. 14)
- distance rings, which support against adjacent parts (fig. 15)
- snap rings (fig. 16)

Any axial location should be able to accommodate the axial loads that may be applied to the bearing.

Bearings with a tapered bore

Depending on conditions and requirements, common ways of axially locating the inner ring of a bearing with a tapered bore are:

- a lock nut for bearings mounted on a tapered seat (fig. 17)
- an adapter sleeve only (fig. 18), if no precise axial positioning is required and the axial loads do not exceed the friction between sleeve and shaft
- an adapter sleeve and a distance ring (fig. 19), if precise axial positioning is required or elevated axial loads occur
- a withdrawal sleeve with a distance ring (or shaft shoulder) and lock nut (fig. 20)

Abutments and fillets

When designing abutments, allow enough space to avoid contact between rotating and stationary parts.

Shaft and housing fillet dimensions should always be smaller than the bearing chamfer radii. Heavily loaded shafts can require large fillets and a spacing collar may be necessary (fig. 21).

Appropriate abutment and fillet dimensions are listed in the product tables.

Fig. 11

Inner ring supported by a lock nut and shaft shoulder

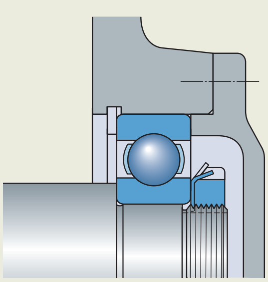


Fig. 13

Inner ring supported by an end plate and shaft shoulder

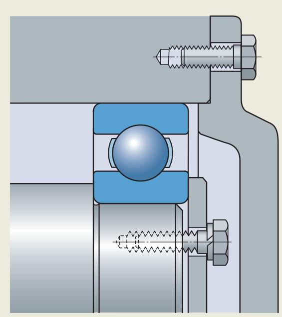


Fig. 15

Inner ring supported by a distance ring and lock nut

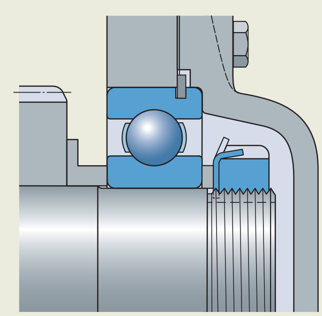


Fig. 12

Outer ring supported by a threaded ring

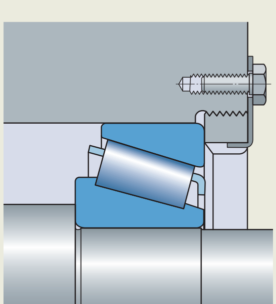


Fig. 14

Outer ring supported by housing cover and housing shoulder

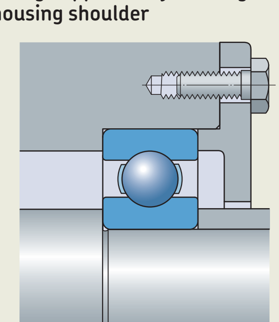
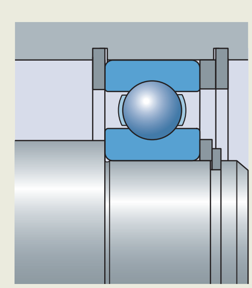


Fig. 16

Bearing supported axially by snap rings and shaft shoulder



Radially free mounted bearings for axial load

You may want to use individual bearings in a bearing arrangement to separately accommodate the radial and axial component of the load. A typical arrangement is to use a cylindrical roller bearing and a four-point contact ball bearing (fig. 22).

When using an individual bearing to accommodate the axial load, you should ensure that this bearing is not subjected to unintended radial loads by:

- designing the bore diameter of its housing to be approximately 1 mm larger than the bearing outer diameter
- not clamping its outer ring in the axial direction to permit its free radial positioning

Also consider the use of an anti-rotation pin. The designation suffix N2 indicates that the bearing has two locating slots in the outer ring.

Raceways on shafts and in housings

In order to save space, the rolling elements of cylindrical, needle or tapered roller bearings can run directly on raceways on the shaft and/or in the housing. To fully exploit the load carrying capacity, the raceways should comply with certain requirements, including:

- suitable material properties such as cleanliness, hardness and heat treatment
- suitable roughness and surface texture
- adequate tolerances for profile, roundness and total run-out

For additional information, contact the SKF application engineering service.

Fig. 17

Bearing on a tapered seat, supported by a lock nut

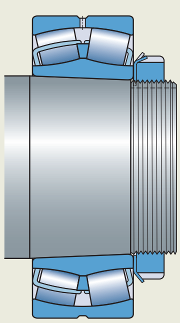


Fig. 19

Bearing on an adapter sleeve, positioned by a distance ring

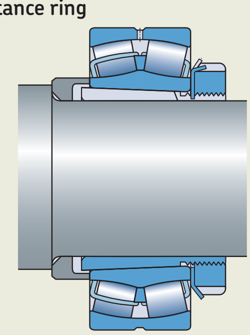


Fig. 21

Spacing collar designed not to contact the shaft fillet

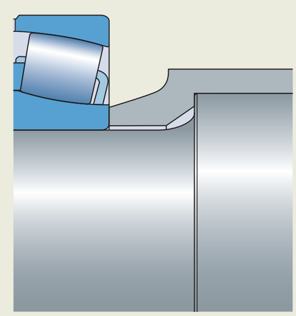


Fig. 18

Bearing on an adapter sleeve

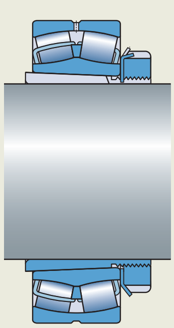


Fig. 20

Bearing on a withdrawal sleeve

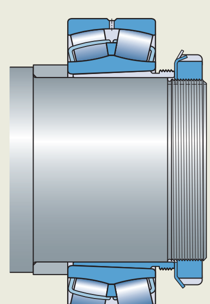
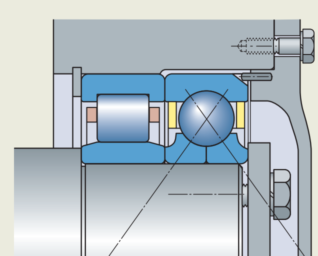


Fig. 22

Cylindrical roller bearing for radial load and four-point contact ball bearing for axial load



B.7 Bearing execution

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B.7 Bearing execution

As part of the bearing selection process, when the bearing type, size, and fit have been determined, additional factors must be considered to enable you to further define the final variant of the bearing.

In this section you can find recommendations and requirements for selecting:

- the bearing internal clearance or preload
- the bearing tolerances
- the appropriate cage (where applicable)
- integral seals (where applicable)
- additional options, such as coatings and other features to meet any special needs/ requirements

Selecting internal clearance or preload

Bearing internal clearance (**fig. 1**) is defined as the total distance through which one bearing ring can be moved relative to the other in the radial direction (radial internal clearance) or in the axial direction (axial internal clearance).

Initial internal clearance is the internal clearance in the bearing prior to mounting.

Mounted clearance is the internal clearance in the bearing after mounting but prior to operation.

Operating clearance is the internal clearance in the bearing when it is in operation and has reached a stable temperature.

In most applications, the initial internal clearance in a bearing is greater than its operating clearance. This is because of the effects of (**fig. 2**):

- interference fits with the shaft and/or housing
- thermal expansion of the bearing rings and associated components

Bearings must have the appropriate operating clearance to operate satisfactorily (*Importance of selecting correct clearance/ preload*).

In most cases, bearings require a certain degree of clearance (*Selecting initial internal clearance*). However, in some cases, they may require preload (i.e. negative clearance, refer to *Selecting preload*, **page 186**).

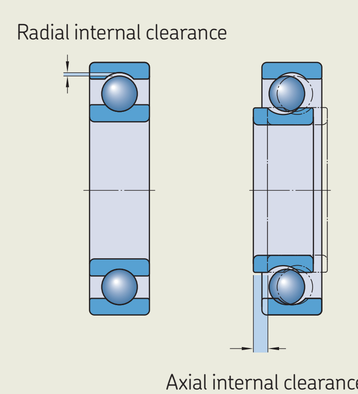
As a general rule:

- Ball bearings should have an operating clearance that is virtually zero.
- Cylindrical, needle, spherical and CARB toroidal roller bearings typically require at least a small operational clearance.
- Tapered roller and angular contact ball bearings should have a small operational clearance, except in applications where a high degree of stiffness or positional control is required, in which case they can be mounted with a degree of preload.

Sections *Selecting initial internal clearance* and *Selecting preload*, describe the influencing factors that you must consider and provide the methods by which you can calculate the initial internal clearance needed to achieve the degree of operational clearance/ preload required by your application.

Fig. 1

Bearing internal clearance



Importance of selecting correct clearance/preload

The operating clearance or preload in a bearing influences, among other things, the friction, load zone size and fatigue life of a bearing. The relationship between these parameters is shown in **diagram 1**. The diagram is generalized and based on rolling bearings under radial load.

For general applications, the operating clearance range should be within the recommended zone shown in **diagram 1**.

Selecting initial internal clearance

The operating clearance required for a bearing to perform satisfactorily is application dependent (*Importance of selecting correct clearance/preload*).

You must ensure that the bearing has a minimum initial internal clearance of a size that, when it is reduced by the effects of mounting and other influences, is equal to or greater than the required minimum operating clearance.

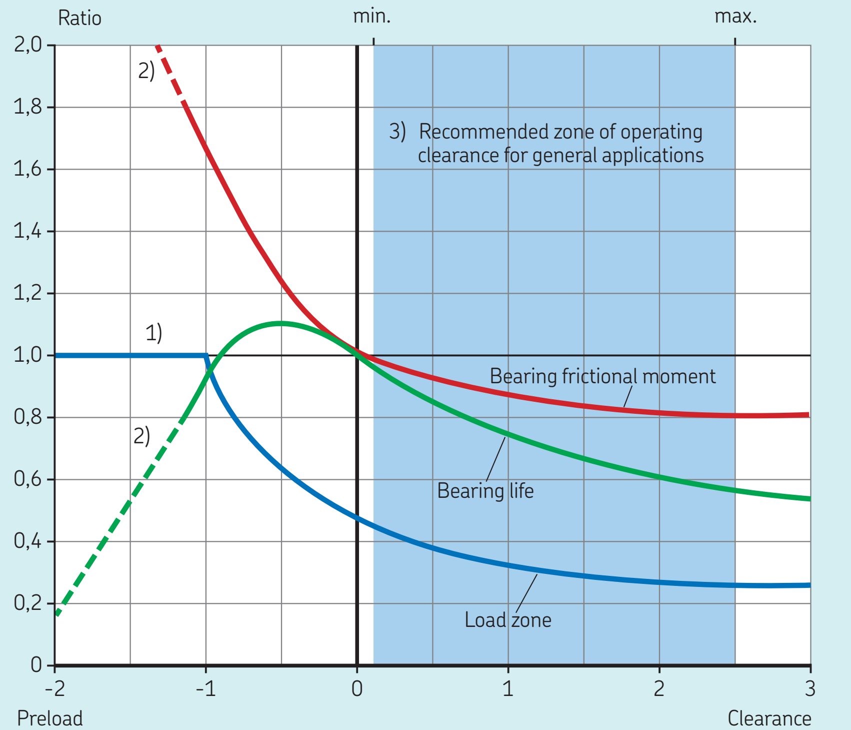
To achieve this, follow this procedure:

- consider the reduction of clearance caused by interference fits (**page 184**)
- consider the reduction of clearance caused by temperature difference between the shaft, bearing rings and housing (**page 184**)
- consider the reduction of clearance caused by other influences (**page 185**)
- consider the required minimum initial internal clearance (**page 185**)
- select the required minimum initial internal clearance (**page 185**)

In case of doubt, contact the SKF application engineering service for support.

Diagram 1

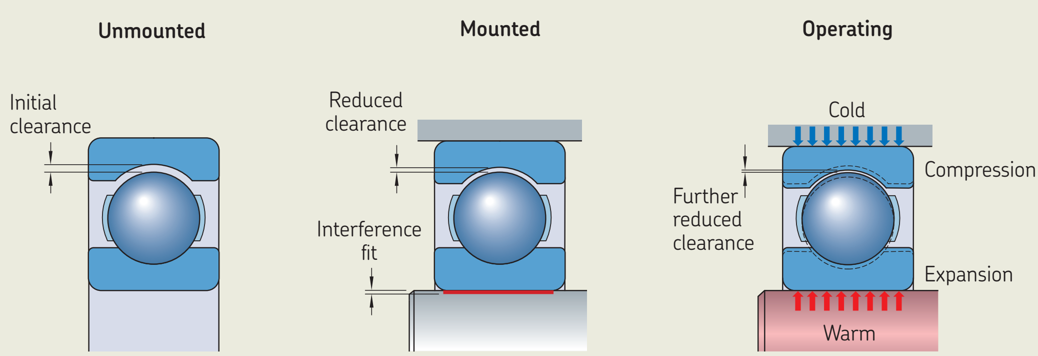
Operational clearance or preload versus main operational criteria



1) A ratio of 1 means that all rolling elements are loaded.
2) The dotted line indicates potentially unstable working conditions.
3) Limits of range are determined by initial clearance, fit and temperature effects.

Fig. 2

Initial internal clearance and operating clearance



Range of initial internal clearance

Bearing types for adjusted bearing arrangements – such as angular contact ball bearings, tapered roller bearings and spherical roller thrust bearings – have their internal clearance set during mounting. The internal clearance of such an arrangement, even though set by adjustment during mounting, will nevertheless have a range.

For other bearing types, the initial internal clearance is determined during their manufacture. ISO has defined five clearance classes for specifying the degree of initial internal clearance in a bearing (*Internal clearance*, **page 26**). Each clearance class represents a range of values. The size of the ranges varies depending on bearing type and size. Clearance class details are listed in relevant product sections.

Initial clearances greater than Normal, such as C3 or even C4 clearance classes, are very common today. This is because modern bearings take higher loads and require tighter interference fits, and typical operating conditions are different, compared to when the clearance classes were defined.

For universally matchable single row angular contact ball bearings and matched tapered roller bearings, double row angular contact ball bearings and four-point contact ball bearings, values for the axial internal clearance are given instead of radial internal clearance, because the axial clearance is of greater practical importance for these bearing types. Radial internal clearance is related to axial internal clearance and that relationship is determined by the bearing type and its internal geometry. For detailed information, refer to the product sections.

Clearance reduction caused by interference fits

An interference fit causes clearance reduction because inner rings are expanded and outer rings are compressed. The reduction equals the effective interference fit multiplied by a reduction factor using

$$\Delta r_{fit} = \Delta_1 f_1 + \Delta_2 f_2$$

where

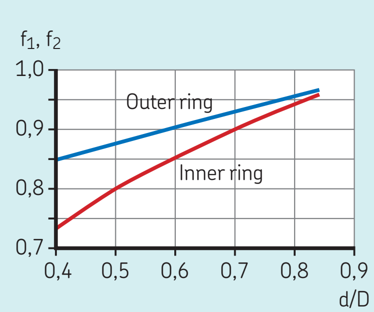
Δr_{fit} = clearance reduction caused by the fit [µm]
f_1 = reduction factor for the inner ring
f_2 = reduction factor for the outer ring
Δ_1 = effective interference between the inner ring and shaft [µm]
Δ_2 = effective interference between the outer ring and housing [µm]

Reduction factors valid for a solid steel shaft and a thick-walled cast iron or steel housing can be obtained from **diagram 2** as a function of the ratio of the bearing bore diameter d to the outside diameter D. For the effective interference value, use the maximum probable interference value listed in the appropriate tables in *Tolerances and resultant fits*, **page 153**.

For a more detailed analysis, consider using SKF calculation tools, such as *SKF Bearing Calculator* (skf.com/bearingcalculator), SKF SimPro Quick or SKF SimPro Expert, or contact the SKF application engineering service.

Diagram 2

Factors for clearance reduction caused by interference fits



Clearance reduction caused by temperature difference between shaft, bearing rings and housing

The temperature behaviour of an application can create a difference in temperature between a bearing inner ring and outer ring, which changes the mounted bearing clearance/preload. For a steel shaft and steel or cast iron housing, the change can be estimated using

$$\Delta r_{temp} = 0{,}012\ \Delta T\ d_m$$

where

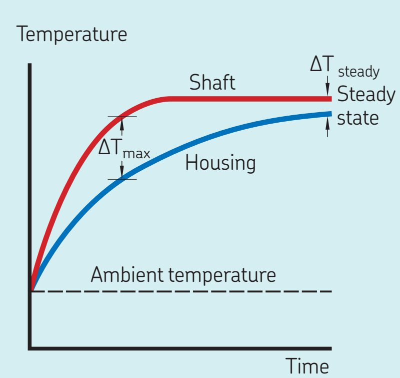
Δr_{temp} = clearance reduction caused by temperature difference [mm]
ΔT = temperature difference between inner and outer ring [°C]
d_m = the bearing mean diameter [mm] = (d + D)/2

Steady state

The operating temperature of a bearing reaches a steady state when there is thermal equilibrium (**page 131**) – i.e. there is a balance between generated heat and dissipated heat. In the common case where the ambient temperature of the surroundings of the housing of a bearing arrangement is cooler than its shaft, a steady-state temperature gradient is developed that results in the inner ring of the bearing being hotter than the outer ring (ΔT_{steady} in **diagram 3**).

Diagram 3

Temperature differences during start-up going into steady state



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Start-up

During start-up, the temperature gradient over the bearing is largely determined by the transient heat flow. Among the various components in contact with the bearing, the one that has the smallest thermal capacity will rise in temperature faster than the one that has the largest thermal capacity. Therefore, the start-up sequence can result in a larger temperature differential between bearing inner and outer ring than in the steady-state condition. It results in a temperature peak during start-up (ΔT_{max} in **diagram 3**). This is especially pronounced in machines that either are working outdoors in a cold climate or have a heated shaft.

Higher speeds

Whether during start-up or at steady state, higher speeds generate larger frictional losses. This typically results in a larger temperature differential between the bearing inner and outer ring and therefore a need for larger initial clearance.

Other influences on clearance/preload

Axial clamping of a ring results in a small increase of its diameter. Normally, this has a negligible influence. For machines where there is a large axial load on any of the rings, or where two bearings (e.g. angular contact ball bearings or tapered roller bearings, with or without distance rings) are clamped axially, the influence on clearance or preload from the axial compression and the radial expansion must be considered.

Misalignment beyond the limits specified in the product sections will reduce the clearance which, because of unfavourable load distribution, will result in reduced service life and increased friction.

Where light alloy materials are used, the temperature differences between rings and shaft or housing may have a more pronounced influence on the clearance of the bearing.

Required minimum initial internal clearance

The required minimum initial internal clearance can be estimated using

$$r = r_{op} + \Delta r_{fit} + \Delta r_{temp} + \Delta r_{other}$$

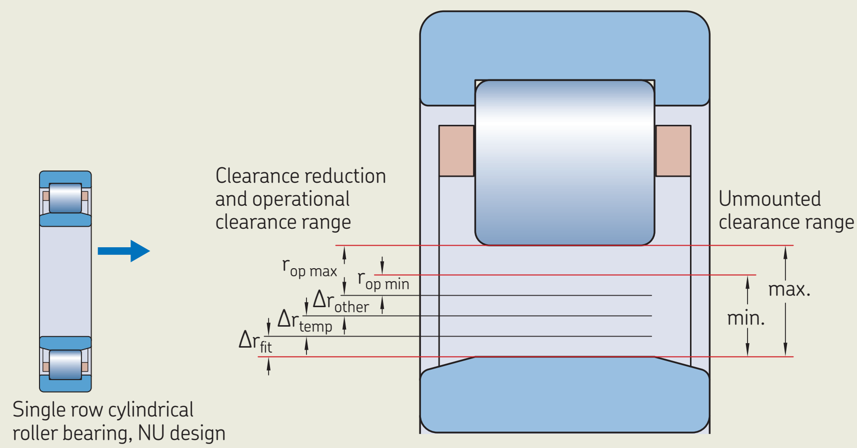
where

r = required minimum initial internal clearance [µm]
r_{op} = required operating clearance [µm]
Δr_{fit} = clearance change caused by the maximum expected fits [µm]
Δr_{temp} = maximum clearance change expected from the temperature difference during start-up or in steady state [µm]
Δr_{other} = maximum clearance change expected from other effects such as axial clamping [µm]

- Bearing types for adjusted bearing arrangements – such as angular contact ball bearings, tapered roller bearings or spherical roller thrust bearings – have their internal clearance set during mounting (*Mounting adjusted bearing arrangements*, **page 203**).
- For other bearing types, select a bearing clearance class (*Internal clearance*, **page 26**: Normal, C3, C4, etc.) whose minimum clearance is equal to or greater than the estimated minimum initial internal clearance (**fig.3**). Then verify whether the resulting maximum clearance of the selected clearance class is acceptable for the application. If the maximum clearance, for whatever reason, is too large then consider choosing a reduced clearance group – e.g. C3L, which includes only the lower half of the C3 clearance group range.

Fig.3

Relationship between minimum initial internal clearance and bearing clearance class



The minimum clearance of the bearing clearance class must be equal to or greater than the estimated minimum initial internal clearance.

Selecting preload

Depending on the application, there may be a need to preload a bearing arrangement. For example, if a high degree of stiffness or positional control is required then preload may be suitable. Similarly, where there is a very light or no external load on the bearing in operation then preload may be required to ensure a minimum load.

Applying the preload is typically done by measuring a force, sometimes a displacement over a distance or path, or by measuring the frictional torque during mounting.

Empirical preload values can be obtained from proven designs and can be applied to similar designs. For new designs, SKF recommends calculating the appropriate preload range by using SKF SimPro Quick or SKF SimPro Expert and then checking it by testing in the application. The agreement between the calculation and the actual application depends on how closely the estimated operating temperature and elastic behaviour of the associated components – most importantly the housing – coincide with the actual conditions in operation. In this context, the effects of start-up at low ambient temperature must be included in the testing.

If the shaft temperature in operation is higher than the housing temperature, the preload, which was adjusted at ambient temperature during mounting, will change. Since thermal growth of a shaft makes it larger both in the axial and in the radial direction, the back-to-back arrangements are less sensitive to thermal effects than the face-to-face arrangements.

When adjusting preload in a bearing system, it is important that the established preload value is attained with the least possible variation. To reduce variation when mounting tapered roller bearings, the shaft should be turned several times to ensure that the rollers are in correct contact with the guide flange of the inner ring.

Considerations for preload

Depending on the bearing type, preload may be either radial or axial. Super-precision cylindrical roller bearings, for example, can only be preloaded radially because of their design, while angular contact ball bearings or tapered roller bearings can only be preloaded axially.

Single tapered roller bearings or angular contact ball bearings are generally mounted together with a second bearing of the same type and size in a back-to-back (load lines diverge, **fig. 4**) or face-to-face (load lines converge, **fig. 5**) arrangement. The same is true for single row angular contact ball bearings.

The distance L between the pressure centres is longer when the bearings are arranged back-to-back compared to bearings that are arranged face-to-face. The back-to-back arrangement can accommodate larger tilting moments.

Preloading with springs

By preloading bearings it is possible to reduce the noise in, for example, small electric motors or similar applications. In this example, the bearing arrangement comprises a preloaded single row deep groove ball bearing at each end of the shaft (**fig. 6**). The simplest method of applying preload is to use a wave spring. The spring acts on the outer ring of one of the two bearings. This outer ring must be able to be axially displaced.

The preload force remains practically constant, even when there is axial displacement of the bearing as a result of thermal elongation.

The requisite preload force can be estimated using

$$F = k\,d$$

where

F = preload force [kN]
k = a factor, described in the following text
d = bearing bore diameter [mm]

Fig. 4

Back-to-back arrangement, tapered roller bearings

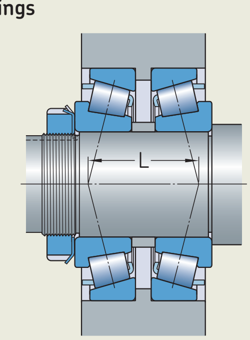


Fig. 5

Face-to-face arrangement, tapered roller bearings

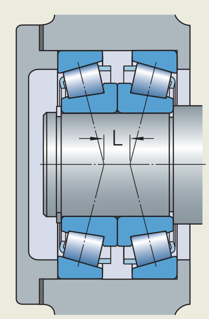
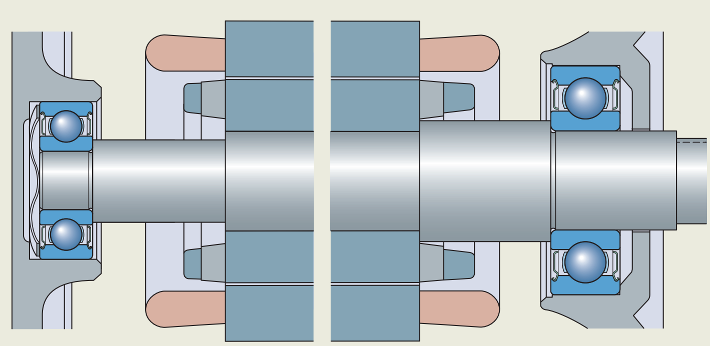


Fig. 6

Preloading with springs



For small electric motors, values of between 0,005 and 0,01 are used for the factor k. If preload is used primarily to protect the bearing from the damage caused by external vibrations when stationary, then greater preload is required and k = 0,02 should be used.

Spring loading is also a common method of applying preload to angular contact ball bearings in high-speed grinding spindles. The method is not suitable for bearing applications where a high degree of stiffness is required, where the direction of axial load changes, or where undefined peak loads can occur.

For additional information, refer to *Bearing preload*, (skf.com/go/17000-B7).

Bearing tolerance class

The dimensional and geometrical tolerances of bearings are described by their tolerance classes (*Tolerances*, **page 36**). In addition to the Normal, P6 and P5 tolerance classes, SKF also manufactures bearings with even narrower tolerances. These include P4, UP and other tolerance classes. For information about SKF bearings that have a tolerance class better than P5, refer to skf.com/super-precision.

Select the tolerance class for a bearing based on the application requirements for precision of rotation and operational speed (**diagram 4**).

If the application requirements for precision of rotation are moderate (*Selecting fits*, **page 140**) and operational speed are moderate (*Speed limitations*, **page 135**), then choose a Normal tolerance class. If the requirements for precision of rotation and/or operational speed are greater than moderate, then choose an appropriately more accurate tolerance class (**diagram 4**).

For detailed information about standard tolerances, please refer to the product sections.

Cages

The main cage types are described in *Components and materials*, **page 24**. Additionally, information about standard cages, and possible cage options, for a particular bearing type is given in the relevant product section. If a bearing with a non-standard cage is required, check availability before ordering.

There are fundamental design differences between bearings which, together with the influence of bearing size, make certain cage designs necessary. For example:

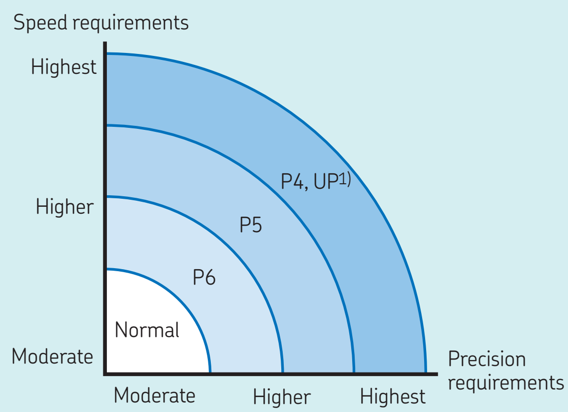
- some bearing types need either split or snap-type cages, because they are assembled after the rings and rolling elements have been sub-assembled
- other bearing types need roller-guided cages, to be self-containing
- bearings of a certain combination of size and series need ring-guided cages, to limit contact stress between rolling elements and cage

Given the specific functional demands and quantity of bearings being manufactured, the material and manufacturing methods are chosen to provide the most reliable and cost-effective cage.

Cages are mechanically stressed during bearing operation by frictional, impact, centrifugal and inertial forces. They can also be chemically influenced by certain organic solvents or coolants, lubricants, and lubricant additives. Therefore, the material type used for a cage has a significant influence on the suitability of a rolling bearing for a particular application.

Diagram 4

Bearing tolerance class related to precision of rotation and operational speed



1) For information about SKF bearings that have a tolerance class better than P5, refer to skf.com/super-precision.

Steel cages

Steel cages can be used at operating temperatures up to 300 °C *(570 °F)*.

Sheet steel cages

Stamped sheet steel cages are made of low carbon steel. These lightweight cages have relatively high strength and, for some bearing types, can be surface treated to further reduce friction and wear in critical conditions.

Machined steel cages

Machined steel cages are normally made of non-alloyed structural steel. To reduce friction and wear, some machined steel cages are surface treated.

Machined steel cages are not affected by the mineral or synthetic oil-based lubricants normally used for rolling bearings, or by the organic solvents used to clean bearings.

Brass cages

Brass cages can be used at operating temperatures up to 250 °C *(480 °F)*.

Sheet brass cages

Stamped sheet brass cages are used for some small and medium-size bearings. In applications such as refrigeration compressors that use ammonia, machined brass or steel cages should be used.

Machined brass cages

Most brass cages are machined from cast or wrought brass. They are unaffected by most common bearing lubricants, including synthetic oils and greases, and can be cleaned using organic solvents.

Polymer cages

Polyamide 66

Polyamide 66 (PA66) is the most commonly used material for injection moulded cages. This material, with or without glass fibres, is characterized by a favourable combination of strength and elasticity. The mechanical properties, such as strength and elasticity, of polymer materials are temperature dependent and subject to ageing. The factors that most influence the ageing process are temperature, time and the medium (lubricant) to which the polymer is exposed. The relationship between these factors for glass fibre reinforced PA66 is shown in **diagram 5**. Cage life decreases with increasing temperature and the aggressiveness of the lubricant.

Therefore, whether polyamide cages are suitable for a specific application depends on the operating conditions and life requirements. The classification of lubricants into “aggressive” and “mild” is reflected by the “permissible operating temperature” for cages made of glass fibre reinforced PA66 with various lubricants (**table 1**). The permissible operating temperature in **table 1** is defined as the temperature that provides a cage ageing life of at least 10 000 operating hours.

Some media are even more “aggressive” than those specified in **table 1**. A typical example is ammonia, used as a refrigerant in compressors. In those cases, cages made of glass fibre reinforced PA66 should not be used at operating temperatures above 70 °C *(160 °F)*.

Polyamide loses its elasticity at low temperatures. Therefore, cages made of glass fibre reinforced PA66 should not be used in applications where the continuous operating temperature is below –40 °C *(–40 °F)*.

Polyamide 46

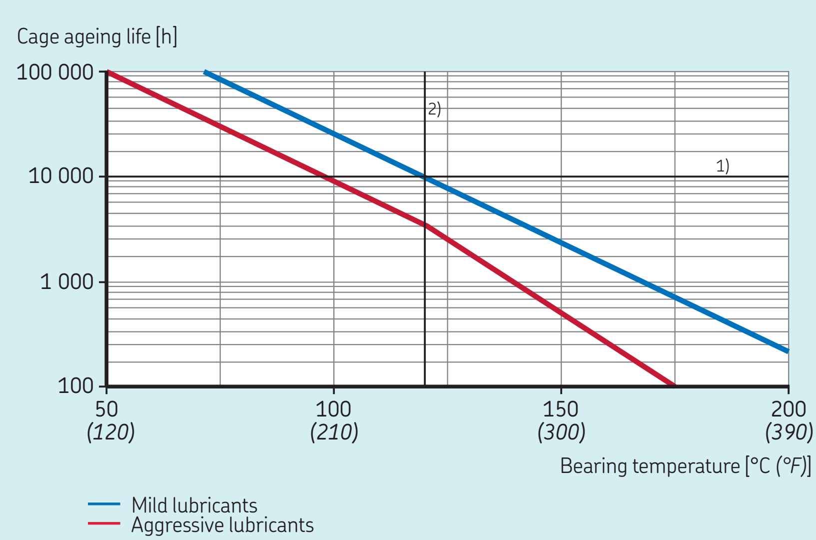
Glass fibre reinforced polyamide 46 (PA46) is the standard cage material for some small and medium-size CARB toroidal roller bearings. The permissible operating temperature is 15 °C *(25 °F)* higher than for glass fibre reinforced PA66.

Polyetheretherketone

Glass fibre reinforced polyetheretherketone (PEEK) is more suitable for demanding conditions regarding high speeds, chemical resistance or high temperatures than PA66 and PA46. The exceptional properties of PEEK provide a superior combination of strength and flexibility, high operating temperature range, and high chemical and wear resistance. Because of these outstanding features, PEEK cages are commonly available for hybrid and/or super-precision ball and cylindrical roller bearings. The material does not show signs of ageing by temperature or oil additives up to 200 °C *(390 °F)*. However, the maximum temperature for high-speed use is limited to 150 °C *(300 °F)* as this is the softening temperature of the polymer.

Diagram 5

Cage ageing life for glass fibre reinforced PA66



1) The permissible operating temperature is defined as the temperature that provides a cage ageing life of at least 10 000 operating hours.
2) Generally, “aggressive” lubricants have a permissible temperature that is less than 120 °C *(250 °F)*.

Cages made of other materials

In addition to the materials described previously, SKF bearings for special applications may be fitted with cages made of other engineered polymers, light alloys or special cast iron. For additional information about alternative cage materials, contact SKF.

Integral sealing

Integral sealing can significantly prolong bearing service life by keeping lubricant in the bearing and contaminants out.

The various types of capping devices that are available for SKF bearings are described in *Components and materials*, **page 24**.

Information about which integral seal options are available for a particular bearing type is given in the relevant product section.

Additional options

Coatings

Coating is a well-established method to upgrade materials and to provide bearings with additional benefits for specific application conditions. Various coating methods developed by SKF are available and have been proven successful in many applications.

Black oxide

Black oxide coating of rings and rollers improves reliability and performance in highly demanding applications, especially under low load conditions and high vibration. In addition, it improves corrosion protection and lubricant adhesion on the bearing surfaces.

SKF also supplies customized black oxide coating layers that are optimized for best tribological results and highest bearing performance, produced using well-defined processes and fine tuned to the individual steel grade, bearing type and size. SKF's evaluation and quality control technology for the black oxide application process includes a scanning electron microscope and a patented examination method.

NoWear

NoWear is a wear-resistant surface coating that applies a low-friction carbon coating on the bearing inner ring raceway(s) and/or the rolling elements. It can withstand long periods of operation under marginal lubrication conditions. For additional information, refer to *NoWear coated bearings*, **page 1060**

INSOCOAT

INSOCOAT bearings are standard bearings that have the external surfaces of their inner or outer ring plasma-sprayed with an aluminium oxide, impregnated with a resin sealant, to form a coating. It offers resistance to the damage that can be caused by the passage of stray electric current through the bearing. For additional information, refer to *INSOCOAT bearings*, **page 1030**.

Other coatings are available that provide an alternative to stainless steel bearings (especially for ready-to-mount bearing units) that are used in a corrosive environment.

Table 1

Permissible operating temperatures for PA66 cages with various bearing lubricants

Lubricant	Permissible operating temperature[1]	
–	°C	°F
Mineral oils		
Oils without EP additives, e.g. machine or hydraulic oils	120	*250*
Oils with EP additives, e.g. industrial and automotive gearbox oils	110	*230*
Oils with EP additives, e.g. automotive rear axle and differential gear oils (automotive), hypoid gear oils	100	*210*
Synthetic oils		
Polyglycols, poly-alpha-olefins	120	*250*
Diesters, silicones	110	*230*
Phosphate esters	80	*175*
Greases		
Lithium greases	120	*250*
Polyurea, bentonite, calcium complex greases	120	*250*

For sodium and calcium greases and other greases with a maximum operating temperature ≤ 120 °C *(250 °F)*, the maximum temperature for a polyamide cage is the same as the maximum operating temperature for the grease.

[1] Measured on the outside surface of the outer ring; defined as the temperature that provides a cage ageing life of at least 10 000 operating hours.

Features for special requirements

SKF supplies many more bearing variants, in addition to those presented in the product sections, for accomplishing various tasks and satisfying special application needs. Among the more common special variants manufactured by SKF are:

- special chamfers – e.g. with a larger radius or with a modified shape (fig. 7)
- additional anti-rotation slots in the outer ring (standard for some bearing types, such as four-point contact ball bearings) (table 2, fig. 8)
- threaded holes in the rings to accommodate eye bolts to ease lifting (fig. 9)
- special greases
- sensors – e.g. to aid mounting (fig. 10) or for monitoring speed and direction of rotation (fig. 11)
- measuring reports, material certificate, additional inspections
- tailor-made bearings and units (fig.12 and fig.13)

Fig. 7

Bearing with special-shaped chamfer

Fig. 8

Bearing with additional anti-rotation slots in the outer ring

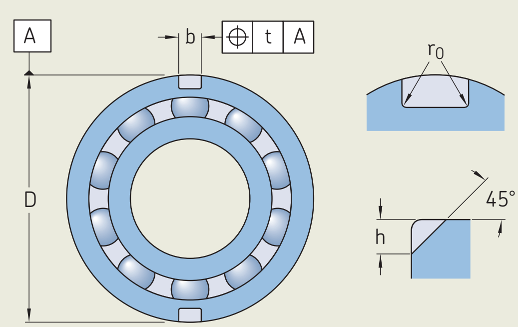


Fig. 9

Eye bolts in bearing ring to ease lifting

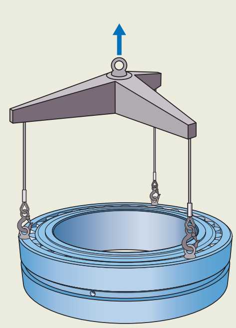


Fig. 10

Mounting with SKF SensorMount

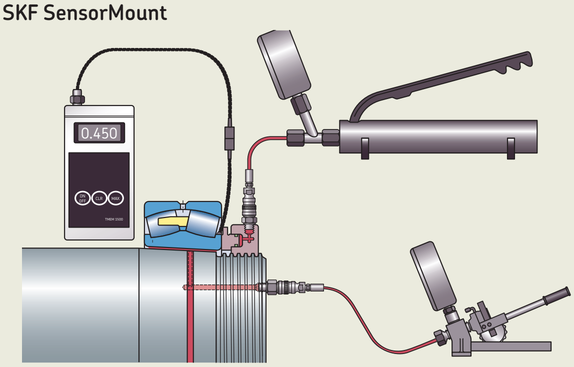
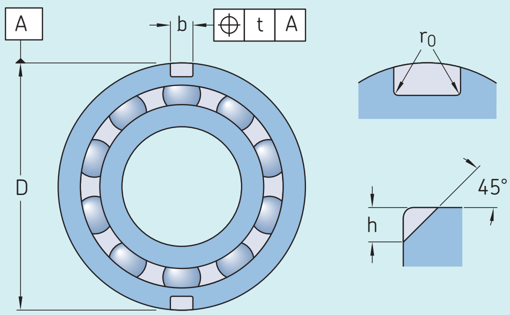


Table 2

Locating slots in the outer ring of four-point contact ball bearings



Outside diameter		**Dimensions**						**Tolerance**[1)]
		Diameter series 2			Diameter series 3			
D		h	b	r_0	h	b	r_0	t
>	≤							U
mm		mm						mm
35	**45**	2,5	3,5	0,5	–	–	–	0,2
45	**60**	3	4,5	0,5	3,5	4,5	0,5	0,2
60	**72**	3,5	4,5	0,5	3,5	4,5	0,5	0,2
72	**95**	4	5,5	0,5	4	5,5	0,5	0,2
95	**115**	5	6,5	0,5	5	6,5	0,5	0,2
115	**130**	6,5	6,5	0,5	8,1	6,5	1	0,2
130	**145**	8,1	6,5	1	8,1	6,5	1	0,2
145	**170**	8,1	6,5	1	10,1	8,5	2	0,2
170	**190**	10,1	8,5	2	11,7	10,5	2	0,2
190	**210**	10,1	8,5	2	11,7	10,5	2	0,2
210	**240**	11,7	10,5	2	11,7	10,5	2	0,2
240	**270**	11,7	10,5	2	11,7	10,5	2	0,2
270	**400**	12,7	10,5	2	12,7	10,5	2	0,4

1) Other tolerances are in accordance with ISO 20515.

Fig. 11

Motor encoder unit

Fig.12

Special bearing used in pulp and paper manufacturing

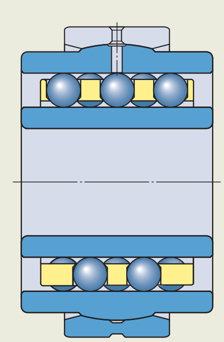


Fig.13

SKF Agri Hub for seeding disc

B.8

Sealing, mounting and dismounting

B.8 Sealing, mounting and dismounting

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This section is the last step in the *Bearing selection process* and it covers:

- **External sealing**
  How to select appropriate seals for rolling bearing applications and the different types of seal available.
- **Mounting and dismounting**
  The preparation and guidelines for mounting and dismounting bearings.
- **Inspection and monitoring**
  Various aspects of inspecting and monitoring bearings in operation for the purpose of preventing problems, and an introduction to troubleshooting.

External sealing

Bearing arrangements generally include a shaft, bearings, housing(s), lubricant, associated components, and seals. Seals are vital to the cleanliness of the lubricant and the service life of the bearings.

The section on *Integral sealing*, **page 189**, gives a general description of the integral seals used in capped bearings. For detailed information, refer to the relevant product sections.

This section describes seals outside the bearing, and how they affect bearing performance. Because of their importance for bearing applications, this section deals exclusively with non-contact and contact shaft seals, their various designs and executions.

Seal selection criteria

Seals for bearing applications should provide maximum protection with a minimum amount of friction and wear, under the prevailing operating conditions. Because bearing performance and service life are so closely tied to the effectiveness and cleanliness of the lubricant, the seal is a key component. For additional information on the influence of solid contaminants on bearing performance, refer to *Contamination factor*, η_c, **page 104**

Many factors must be considered when selecting the most suitable seal for a particular bearing-shaft-housing system. These include:

- the lubricant type: oil or grease
- the contaminant type: particles or fluid or both
- the circumferential speed at the seal lip
- the shaft arrangement: horizontal or vertical
- possible shaft misalignment or deflection
- run-out and concentricity
- available space
- seal friction and the resulting temperature increase
- environmental influences
- cost
- required operating time
- maintenance requirements

For additional information, refer to *Power transmission seals*, (skf.com/seals).

Seal types

The purpose of a seal is to retain lubricant and prevent any contaminants from entering into a controlled environment.

There are several basic seal types:

- non-contact seals
- contact seals
- static seals

Non-contact radial shaft seals form a narrow gap between the stationary and the rotating component. The gap can be arranged axially, radially or in combination. Non-contact seals, which range from simple gap-type seals to multi-stage labyrinths (**fig. 1**), do not wear.

Seals in contact with sliding surfaces are called contact seals and are used to seal passages between machine components that move relative to each other, either linearly or circumferentially.

The most common contact seal is the radial shaft seal (**fig. 2**), which is installed between a stationary and a rotating component.

Seals between stationary surfaces are called static seals. Their effectiveness depends on the radial or axial deformation of their cross section when installed. Gaskets (**fig. 3**) and O-rings (**fig. 4**) are typical examples of static seals.

Fig. 1

Multi-stage labyrinth seal

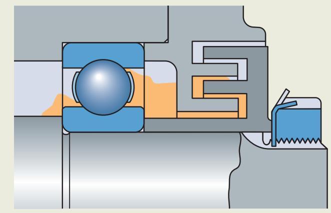


Fig. 2

Radial shaft seal

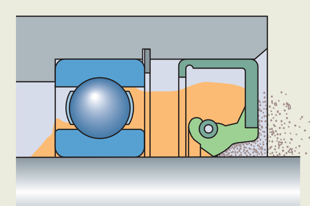


Fig. 3

Gasket

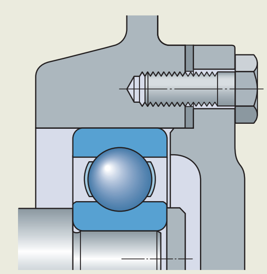
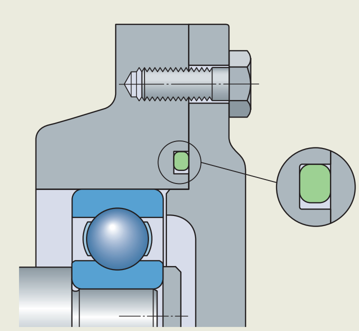


Fig. 4

O-ring



Non-contact seals

The simplest seal used outside a bearing is the gap-type seal, which creates a small gap between the shaft and housing cover (fig. 5). This type of seal is mainly used for grease lubricated applications that operate in dry, dust-free environments. To enhance the effectiveness of this seal, one or more concentric grooves can be machined in the housing cover bore at the shaft end (fig. 6). The grease emerging through the gap fills the grooves and helps to prevent entry of contaminants.

With oil lubrication and horizontal shafts, helical grooves can be machined into the shaft or housing bore, either right-handed or left-handed, depending on the direction of shaft rotation (fig. 7). These grooves are designed to return emerging oil to the bearing; therefore, it is essential that the shaft rotates in one direction only.

Other shapes can be machined into the shaft. Non-helical grooves may be used on the shaft and in the housing; these function as disruptors/flingers. Additional shaft collars can prevent oil leakage, whatever the direction of rotation.

Single or multi-stage labyrinth seals, typically used with grease lubrication, are considerably more effective than simple gap-type seals, but are also more expensive. Their effectiveness can be further improved by periodically applying grease, via a duct, to the labyrinth passages. The passages of the labyrinth seal can be arranged axially (fig. 8) or radially (fig. 9), depending on the housing type (split or non-split), mounting procedures, available space, etc. The radial gaps of the labyrinth (fig. 8) remain unchanged when axial displacement of the shaft occurs in operation; therefore, the gaps can be very narrow. Where angular misalignment of the shaft relative to the housing can occur, labyrinths with inclined passages can be used (fig. 10).

Effective and inexpensive labyrinth seals can be made using SKF sealing washers (fig. 11). Sealing effectiveness increases with the number of washer sets and can be further improved by incorporating flocked washers. For additional information on these sealing washers, refer to *Power transmission seals*, (skf.com/seals).

Rotating discs (fig. 12) are often fitted to the shaft to act as a shield. Flingers, grooves or discs are also used with oil lubrication. The oil from the flinger is collected in a channel in the housing and returned to the housing sump through suitable ducts (fig. 13).

Fig. 5

External gap-type seal

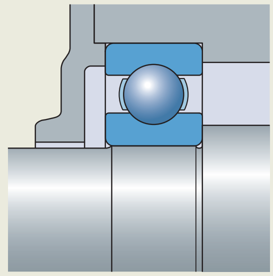


Fig. 6

External gap-type seal with concentric grooves

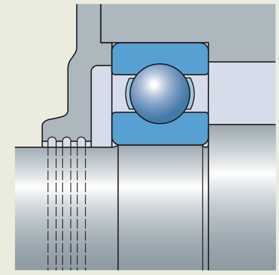


Fig. 7

External gap-type seal with helical grooves

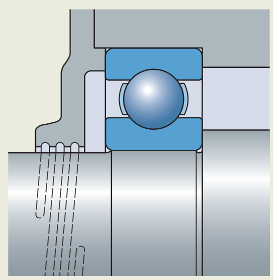


Fig. 8

External labyrinth seal, passages arranged axially

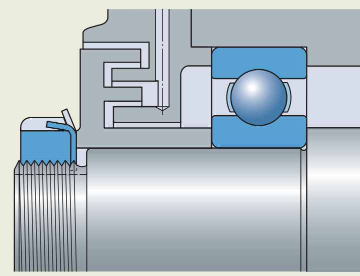


Fig. 9

External labyrinth seal, passages arranged radially

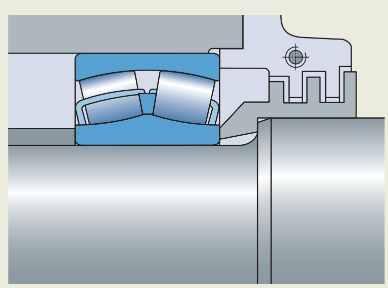


Fig. 10

External labyrinth seal, with inclined passages

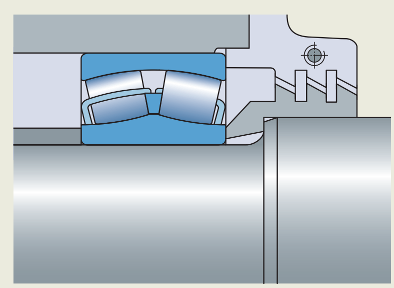


Fig. 11

Labyrinth seal consisting of multiple SKF sealing washers

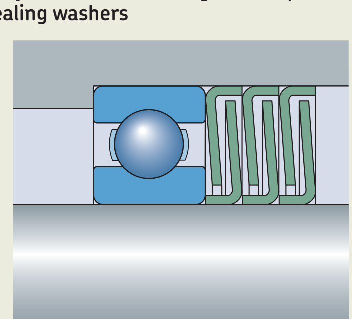
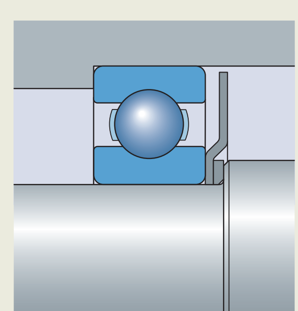


Fig. 12

Rotating disc acting as a shield



Contact seals

There are four common types of contact seals:

- radial shaft seals
- V-ring seals
- axial clamp seals
- mechanical seals

The seal type selected for a particular application typically depends on:

- the primary purpose of the seal (to retain lubricant or exclude contaminants)
- the type of lubricant (oil, grease or other)
- the operating conditions (speed, temperature, pressure and environment)

Radial shaft seals

Radial shaft seals (**fig. 14** and **fig. 15**) are contact seals that are used for oil and grease lubricated applications. For detailed information, refer to the SKF catalogue *Industrial shaft seals*. These ready-to-mount components typically consist of a metal reinforcement or casing, a synthetic rubber body, a seal lip and a garter spring. The seal lip is pressed against the shaft by the garter spring. Depending on the seal material and medium to be retained and/or excluded, commonly used materials for radial shaft seals can be used at temperatures between –55 °C *(–65 °F)* and +200 °C *(390 °F)*.

The seal counterface, that part of the shaft where the seal lip makes contact, is of vital importance to sealing effectiveness. The surface hardness of the counterface should be at least 45 HRC at a depth of at least 0,3 mm. The surface texture should be in accordance with ISO 4288 and within the guidelines of Ra = 0,2 to 0,5 µm. In applications where speeds are low, lubrication is good, and contamination levels are minimal, a lower hardness can be acceptable. For oil lubrication, to avoid the pumping effect induced by helical grinding marks, SKF recommends plunge grinding the counterface.

If the primary purpose of the radial shaft seal is lubricant retention, the seal should be mounted with the lip facing inward (**fig. 14**). If the primary purpose is to exclude contaminants, the lip should face outward, away from the bearing (**fig. 15**).

SKF can also supply machined polyurethane radial shaft seals.

Fig. 14

Radial shaft seal, mounted for lubricant retention

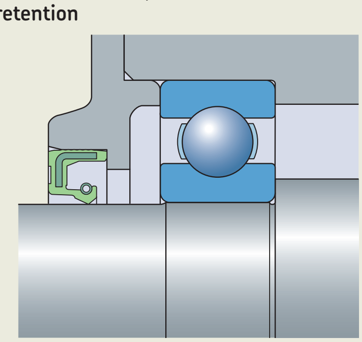


Fig. 13

Oil caught by rotating flinger at the seal

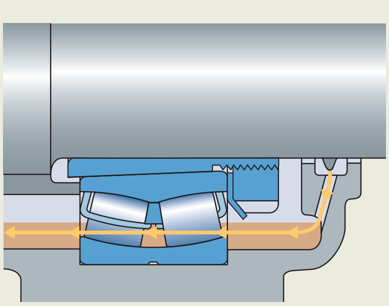
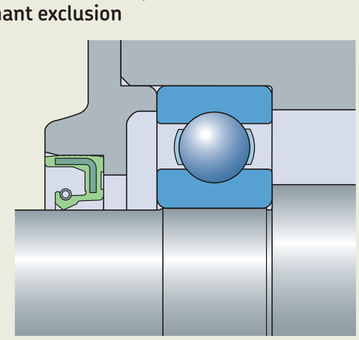


Fig. 15

Radial shaft seal, mounted for contaminant exclusion



⚠ WARNING

Safety precautions for fluoro rubber and Polytetrafluoroethylene

Fluoro rubber (FKM) and Polytetrafluoroethylene (PTFE) are very stable and harmless up to normal operating temperatures of 200 °C *(390 °F)*. However, if exposed to temperatures above 300 °C *(570 °F)*, such as fire or the open flame of a cutting torch, FKM and PTFE give off hazardous fumes. These fumes can be harmful if inhaled, as well as if they contact the eyes. In addition, once the seals have been heated to such temperatures, they are dangerous to handle even after they have cooled. Therefore, they should never come in contact with the skin.

If it is necessary to handle bearings with seals that have been subjected to high temperatures, such as when dismounting the bearing, the following safety precautions should be observed:

- Always wear protective goggles, gloves and appropriate breathing apparatus.
- Place all of the remains of the seals in an airtight plastic container marked with a symbol for “material will etch”.
- Follow the safety precautions in the appropriate safety data sheet (SDS).

If there is contact with the seals, wash hands with soap and plenty of water and, if contact has been made with the eyes, flush eyes with plenty of water and consult a doctor immediately. If the fumes have been inhaled, consult a doctor immediately.

The user is responsible for the correct use of the product during its service life and its proper disposal. SKF takes no responsibility for the improper handling of FKM or PTFE, or for any injury resulting from their use.

V-ring seals

V-ring seals (**fig. 16**) can be used with either oil or grease lubrication. The elastic rubber body of the seal grips the shaft and rotates with it, while the seal lip exerts a light axial pressure on a stationary component, such as a housing. Depending on the material, V-rings can be used at operating temperatures between –40 °C *(–40 °F)* and +200 °C *(390 °F)*. They are simple to install and permit relatively large angular misalignments of the shaft at low speeds.

The recommended counterface surface finish (surface texture) depends on the circumferential speed (**table 1**). At circumferential speeds above 8 m/s, the body of the seal must be located axially on the shaft. At speeds above 12 m/s, the body must be prevented from lifting from the shaft. A sheet metal support ring can be used to do this. When circumferential speeds exceed 15 m/s, the seal lip lifts away from the counterface and the V-ring becomes a gap-type seal.

V-ring seals have good sealing abilities, which can be attributed to the body of the seal, which acts as a flinger, repelling dirt and fluids. As a result, these seals are generally arranged outside the housing in grease lubricated applications and inside the housing, with the lip pointing away from the bearing, in oil lubricated applications. Used as a secondary seal, V-rings protect the primary seal from excessive contaminants and moisture.

For added protection in extremely contaminated applications, SKF also supplies MVR seals (**fig. 17** and SKF catalogue *Industrial shaft seals*).

Axial clamp seals

Axial clamp seals (**fig. 18**) are used as secondary seals for large-diameter shafts in applications where protection is required for the primary seal. They are clamped in position on a non-rotating component and seal axially against a rotating counterface. For this type of seal, it is sufficient if the counterface is fine-turned and has a surface texture of Ra = 2,5 µm.

Mechanical seals

Mechanical seals (**fig. 19**) are used to seal grease or oil lubricated applications, where speeds are relatively low and operating conditions arduous. Mechanical seals consist of two sliding steel rings with finely finished sealing surfaces and two Belleville rubber compound washers, which position the sliding rings in the housing bore and provide the necessary preload force to the sealing surfaces. There are no special requirements for the mating surfaces in the housing bore.

Other seals

Felt seals (**fig. 20**) are generally used with grease lubrication. They are simple, cost-effective and can be used at circumferential speeds of up to 4 m/s and at operating temperatures up to 100 °C *(210 °F)*. The counterface should be ground to a surface texture of Ra ≤ 3,2 µm. The effectiveness of a felt seal can be improved substantially by mounting a simple labyrinth seal as a secondary seal. Before being inserted in the housing groove, felt seals should be soaked in oil at about 80 °C *(175 °F)* prior to mounting.

Metal seals (**fig. 21**) are simple, cost-effective and space-saving seals for grease lubricated bearings. The seals are clamped against either the inner or outer ring and exert a resilient axial pressure against the other ring. After a certain running-in period, a narrow gap forms and these become non-contact seals.

Table 1

Recommended counterface surface finish

Circumferential speed		Surface finish Ra	
m/s	ft/min.	µm	µin.
>10	>1 969	0,4–0,8	16–32
5–10	984–1 969	0,8–1,6	32–64
1–5	199–984	1,6–2,0	64–80
<1	<199	2,0–2,5	80–100

The surface finish must not be lower than Ra = 0,05 µm (2 µin.).

Fig. 16

V-ring seal

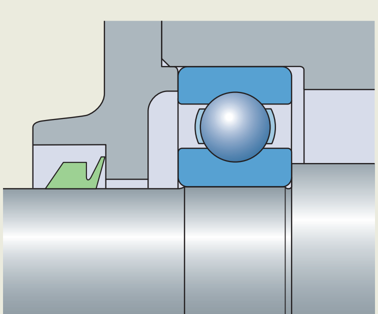


Fig. 17

MVR seal designs

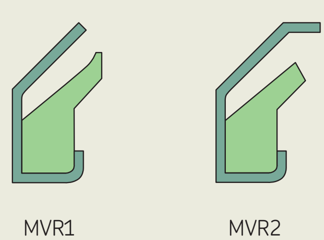
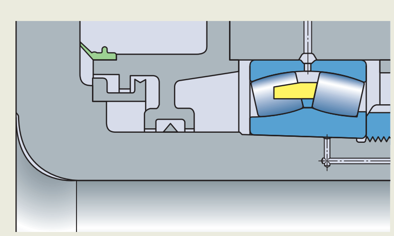


Fig. 18

Axial clamp seal



Mounting and dismounting

Rolling bearings are reliable machine elements that can provide long service life, if they are mounted properly. Proper mounting requires experience, accuracy, a clean work environment, correct working methods and the appropriate tools. SKF offers a comprehensive assortment of high-quality tools for this purpose. For detailed information, refer to *Maintenance products*, (skf.com/mapro).

Mounting bearings correctly is often more difficult than it appears, especially where large bearings are concerned. As part of the SKF Services and Solutions program, SKF offers seminars and hands-on training courses. Mounting and maintenance assistance may also be available from your local SKF company or SKF Authorized Distributor.

The information provided in this section is quite general and is intended primarily to indicate what must be considered by machine and equipment designers to facilitate bearing mounting and dismounting. It includes:

- Mounting
- Test running
- Machines on standby
- Dismounting

Further reading on bearing mounting and dismounting

- *SKF bearing maintenance handbook* (ISBN 978-91-978966-4-1)
- Mounting instructions for individual bearings (skf.com/mount)

Fig. 19

Mechanical seal

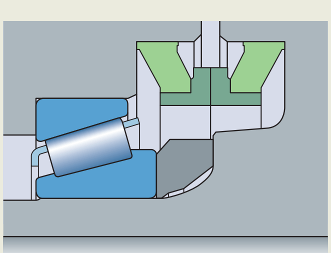


Fig. 20

Felt seal

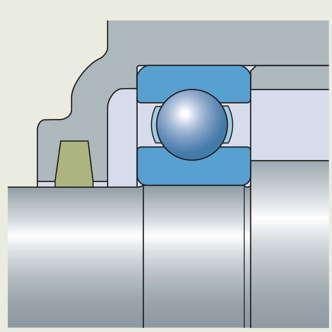
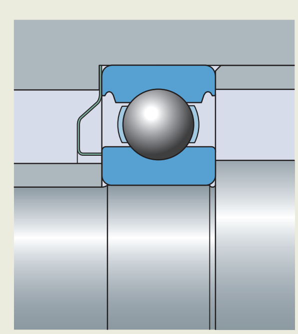


Fig. 21

Metal seal



Mounting

Before mounting, be sure you have all the necessary parts, tools, equipment and data available and ready to use. Review any drawings or instructions to determine the correct sequence and orientation in which components are to be assembled. Leave the bearings in their original packages until immediately before mounting so that they are not exposed to any contaminants. If there is a risk that the bearings have become contaminated because of improper handling or damaged packaging, they should be washed, dried and inspected before mounting.

Assembly area

Bearings should be mounted in a dry, dust-free area, away from machines producing swarf and dust. When bearings have to be mounted in an unprotected area, which is often the case with large bearings, steps should be taken to protect the bearing and mounting position from contaminants such as dust, dirt and moisture. This can be done by covering or wrapping bearings and machine components with plastic or foil.

Checking associated components

Housings, shafts, seals and other components of the bearing-shaft-housing system should be checked to make sure they are clean. This is particularly important for lubrication holes and threaded holes, lead-ins or grooves where remnants of previous machining operations might have collected. Also, make sure that all unpainted surfaces of cast housings are free of core sand and that any burrs are removed.

When all components have been cleaned and dried, check the dimensional and geometrical tolerances of each piece. The bearings only perform satisfactorily if the associated components comply with the prescribed tolerances. The diameters of cylindrical shaft and housing seats are usually checked with a micrometer, or internal gauge, at two cross sections and in four directions (**fig. 22**). Tapered shaft seats can be checked using a *GRA 30 series ring gauge* or a *DMB* or *9205 series taper gauge* refer to skf.com, or a sine bar.

Removing the preservative

Normally, the preservative applied to new bearings does not need to be removed. It is only necessary to wipe off the outside and bore surfaces. However, if the lubricant to be used is not compatible with the preservative, the bearing should be washed and dried carefully. Bearings capped with seals or shields are filled with grease and should not be washed prior to mounting.

Bearing handling and safety

SKF recommends use of personal protection clothing and equipment, such as gloves, safety shoes and goggles, as well as carrying and lifting tools (**fig. 23**) that have been specially designed for handling bearings. Using the proper tools enhances safety, while saving time and effort.

When handling hot or oily bearings, SKF recommends wearing the appropriate heat or oil resistant gloves (**fig. 24**).

For large, heavy bearings, use lifting tackle that supports the bearing from the bottom (**fig. 25**). A spring between the hook and tackle can facilitate positioning of the bearing onto the shaft.

To ease lifting, large bearings can be provided, on request, with threaded holes in the ring side faces to accommodate eye bolts. These holes are designed to bear only the weight of the bearing, because the size and depth of the hole is limited by the ring thickness. Make sure that the eye bolts are only subjected to load in the direction of the shank axis (**fig. 26**).

Fig. 22

Measuring of cylindrical shaft and housing seats

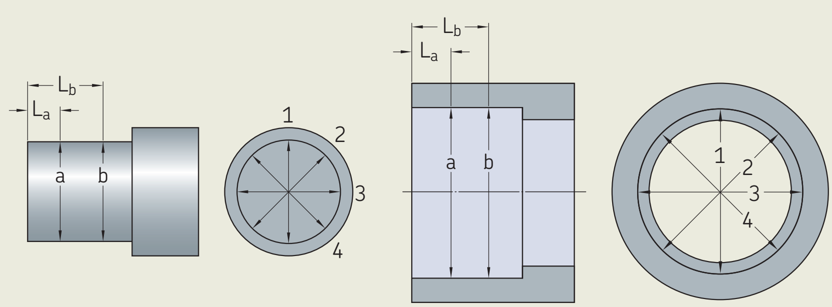


Fig. 23

Carrying tool

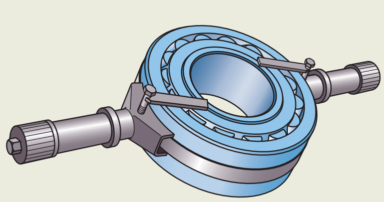


Fig. 24

Heat resistant glove

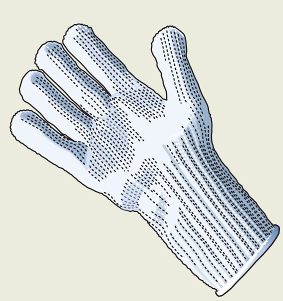
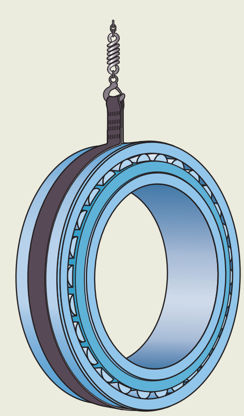


Fig. 25

Lifting of heavy bearings



Methods and tools

Depending on the bearing type and size, mechanical, thermal or hydraulic methods are used for mounting (table 2, page 202). Bearing sizes are categorized as follows:

- small → d ≤ 80 mm
- medium-size → 80 mm < d < 200 mm
- large → d ≥ 200 mm

In all cases, it is important that the bearing rings, cages and rolling elements or seals are never struck directly with any hard object and that the mounting force is never applied through the rolling elements.

For an interference fit, the mating surfaces should be coated with a thin layer of light oil. For a loose fit, the mating surfaces should be coated with SKF anti-fretting agent.

Mounting bearings with a cylindrical bore

Non-separable bearings

With non-separable bearings, the ring that requires the tighter fit is usually mounted first.

Separable bearings

With separable bearings, the inner ring can be mounted independently of the outer ring, which simplifies mounting, particularly where both rings have an interference fit. When mounting the shaft and inner ring assembly into the housing containing the outer ring, careful alignment is required to avoid scoring the raceways and rolling elements. When mounting cylindrical or needle roller bearings with an inner ring without flanges or with a flange on one side only, a guiding sleeve should be used (fig. 27). The outside diameter of the sleeve should be the same as the raceway diameter of the inner ring and should be machined to tolerance class d10Ⓔ for cylindrical roller bearings, and to tolerance 0/–0,025 mm for needle roller bearings.

Cold mounting

If the fit is not too tight, small bearings can be driven into position by applying light hammer blows to a bearing fitting tool (fig. 28). The tool enables the mounting force to be applied centrally.

If a bearing has to be pressed onto the shaft and into the housing bore at the same time, the mounting force must be applied equally to both rings and the abutment surfaces of the mounting tool must lie in the same plane. Whenever possible, mounting should be done with an SKF bearing fitting tool (fig. 28).

With self-aligning bearings, the use of an intermediate mounting ring prevents the outer ring from tilting and swivelling when the bearing and shaft assembly is introduced into the housing bore (fig. 29). The balls of larger self-aligning ball bearings in the 12 and 13 series protrude from the sides of the bearing, therefore the mounting ring must have a recess.

Fig. 26

Loading of eye bolts only in direction of the shank axis

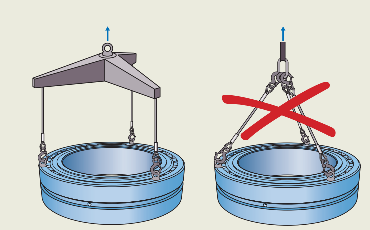


Fig. 27

Use of a guiding sleeve for mounting of a cylindrical roller bearing

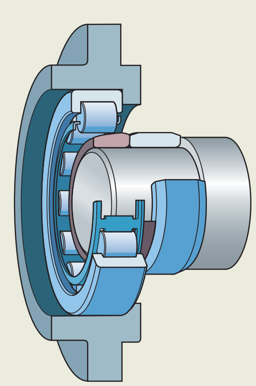


Fig. 28

Bearing fitting tool

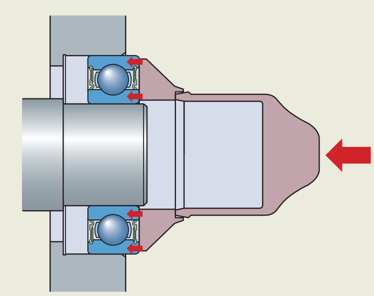


Fig. 29

Mounting a self-aligning ball bearing by use of an intermediate mounting ring

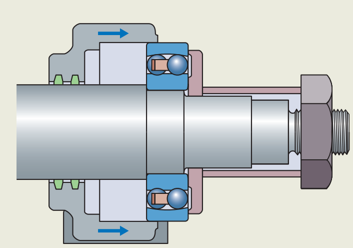
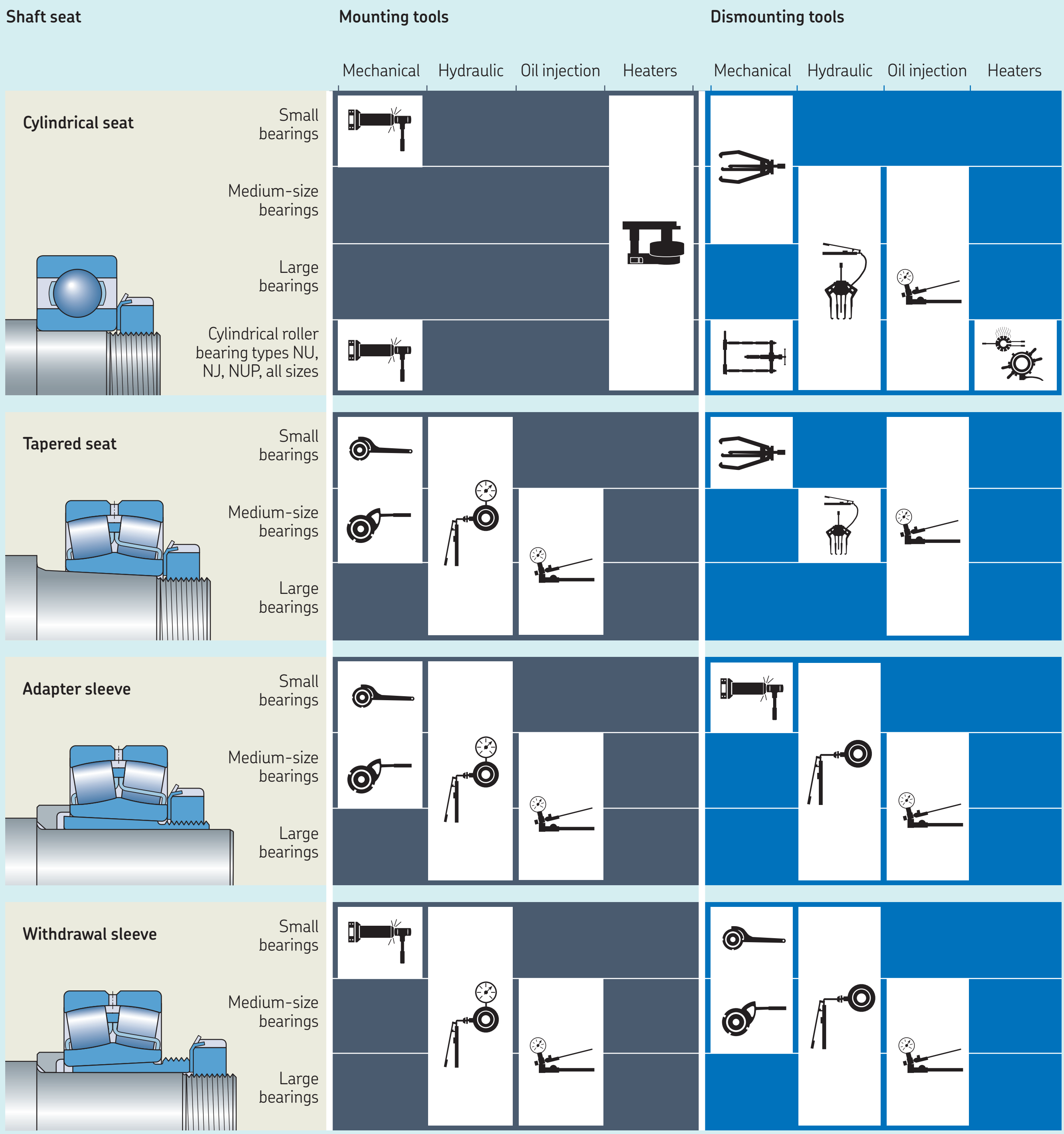
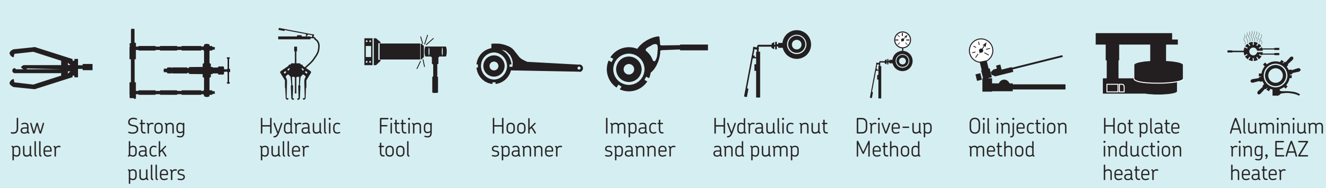


Table 2

SKF methods and tools





Hot mounting

It is generally not possible to mount larger bearings without heating either the bearing or the housing, as the force required to mount a bearing increases considerably with increasing bearing size.

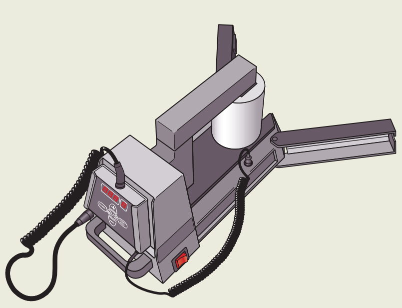
The requisite difference in temperature between the bearing ring and shaft or housing depends on the degree of interference and the diameter of the bearing seat. Generally, open bearings must not be heated to more than 120 °C *(250 °F)*. SKF does not recommend heating bearings capped with seals or shields above 80 °C *(175 °F)*. However, if higher temperatures are necessary, make sure that the temperature does not exceed the permissible temperature of either the seal or grease, whichever is lowest.

When heating bearings, local overheating must be avoided. To heat bearings evenly and reliably, SKF recommends using SKF electric induction heaters (**fig. 30**). If hot-plates are used, the bearing must be turned over a number of times. The seals on sealed bearings should never contact the heating plate directly. Place a ring between the plate and bearing. Read and follow the safety precautions on **page 197**.

For additional information about these mounting methods, refer to the *SKF bearing maintenance handbook*.

Fig. 30

SKF electric induction heater



Mounting adjusted bearing arrangements

The following recommendations refer only to the adjustment of the mounted clearance for bearing arrangements with single row angular contact ball bearings or tapered roller bearings.

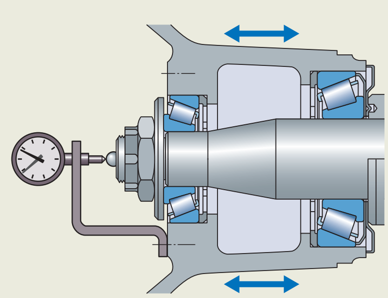
The mounted clearance of single row angular contact ball bearings and single row tapered roller bearings is only established when the bearing is adjusted against a second bearing. Usually, they are arranged back-to-back or face-to-face, and one bearing ring is axially displaced until a given clearance or preload is obtained. For information about bearing preload, refer to *Selecting preload*, **page 186**.

The appropriate value for the clearance to be obtained when mounting depends on the bearing size and arrangement, and operating conditions such as load and temperature. Since there is a definite relationship between the radial and axial clearance of angular contact ball bearings and tapered roller bearings, it is sufficient to specify one value, generally the axial clearance for the arrangement. This specified value is then obtained by measuring the clearance during adjustment and by loosening or tightening a nut on the shaft or a threaded ring in the housing bore or by inserting calibrated washers or shims between one of the bearing rings and its abutment. The actual method used to adjust and measure the clearance depends on whether this is an occasional or repetitive process.

One way to check the axial clearance in a bearing arrangement is to use a dial indicator attached to the hub (**fig. 31**). When adjusting tapered roller bearings and measuring clearance, the shaft or housing should be turned through several revolutions in both directions to be sure that there is proper contact of the roller ends with the guide flange on the inner ring. Without proper contact, the measured result will not be correct.

Fig. 31

Checking of axial clearance using a dial indicator



Mounting bearings with a tapered bore

For bearings with a tapered bore, inner rings are always mounted with an interference fit. The degree of interference is determined by how far the bearing is driven up onto a tapered shaft seat or an adapter or withdrawal sleeve. As the bearing is driven up the tapered seat, its radial internal clearance is reduced. This reduction in clearance, or the axial drive-up distance, can be measured to determine the degree of interference and the proper fit. Recommended values of clearance reduction and axial drive-up are listed in the relevant product section.

The SKF Drive-up Method is a reliable and well-proven method for mounting SKF bearings on tapered seats. For additional information, refer to the *SKF Drive-up Method Program* (skf.com/drive-up).

Small and medium-size bearings

Bearings with bore diameters up to 80 mm ($d \leq 80$ mm), can be driven up onto a tapered seat using either a bearing fitting tool or, preferably, a lock nut. For adapter sleeves, use the sleeve nut that can be tightened with a hook or impact spanner. Withdrawal sleeves can be driven into the bearing bore using a bearing fitting tool or a nut. Starting from a 50 mm thread, SKF hydraulic nuts can also be used.

Medium-size and large bearings

Larger bearings, with bore diameters greater than 80 mm ($d > 80$ mm), require considerably more force to mount. Therefore, SKF hydraulic nuts should be used. Where applicable, SKF also recommends using shafts and sleeves with grooves and ducts for the oil injection method. When combining the two methods, bearing mounting and dismounting becomes much faster, easier and safer. For additional information about the oil injection equipment required for both the hydraulic nut and the oil injection method, refer to skf.com/mapro and skf.com/mount.

Mounting with SKF hydraulic nuts

Bearings with a tapered bore can be mounted with the aid of an SKF hydraulic nut:

- on a tapered shaft seat (fig. 32)
- on an adapter sleeve (fig. 33)
- on a withdrawal sleeve (fig. 34)

The hydraulic nut can be positioned onto a threaded section of the shaft (fig. 32), or onto the thread of a sleeve (fig. 33 and fig. 34). The annular piston abuts the inner ring of the bearing (fig. 32 and fig. 33) or a stop on the shaft, which can be either a nut on a shaft thread (fig. 34) or a plate attached to the end of the shaft. Pumping oil into the hydraulic nut displaces the piston axially with the force needed to drive the inner ring up the taper for accurate and safe mounting.

Oil injection method

With the oil injection method, oil under high pressure is injected via ducts and distribution grooves between the bearing and bearing seat to form an oil film. This oil film separates the mating surfaces and considerably reduces the friction between them. This method is typically used when mounting bearings directly on tapered shaft seats (fig. 35). The necessary ducts and grooves should be an integral part of the shaft design. This method can also be used to mount bearings on adapter or withdrawal sleeves if they are equipped with the relevant features, ducts and grooves.

A spherical roller bearing mounted on a withdrawal sleeve with oil ducts is shown in fig. 36. Oil is injected between all mating surfaces so that the withdrawal sleeve can be pressed into the bearing bore as the bolts are tightened.

Verifying the interference fit

During mounting, the degree of interference is normally determined by one of the following methods:

- measuring the clearance reduction
- measuring the lock nut tightening angle
- measuring the axial drive-up
- measuring the inner ring expansion

For self-aligning ball bearings, feeling the clearance reduction by swivelling the outer ring is an additional method (*Mounting*, page 447).

Fig. 32

Mounting on tapered seat with aid of a hydraulic nut

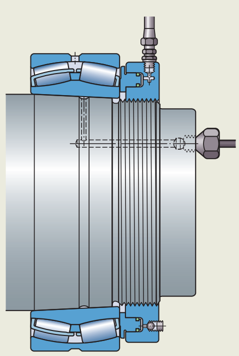


Fig. 33

Mounting on adapter sleeve with aid of a hydraulic nut

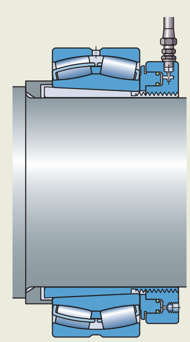


Fig. 34

Mounting on withdrawal sleeve with aid of a hydraulic nut

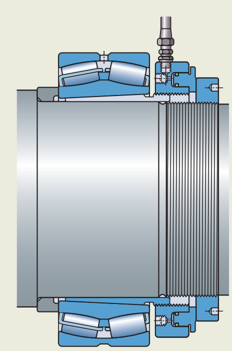


Fig. 35

Mounting on tapered seat with aid of oil injection

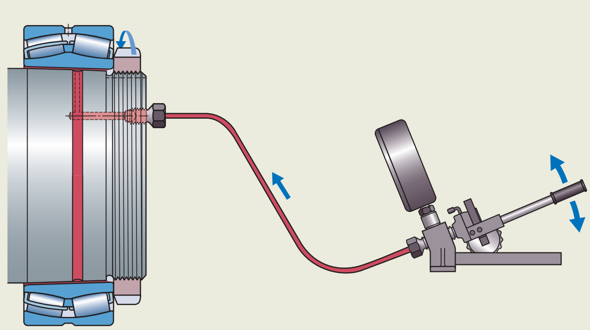
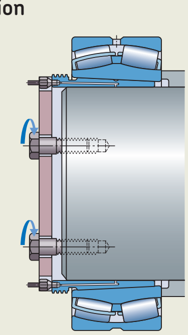


Fig. 36

Mounting on withdrawal sleeve with aid of oil injection



Measuring the clearance reduction

A feeler gauge is most often used to measure the radial internal clearance in medium-size and large spherical and CARB toroidal roller bearings. Recommended values for the reduction of radial internal clearance to obtain the correct interference fit are listed in the relevant product section.

Before mounting, measure the clearance between the outer ring and upper-most roller (**fig. 37**). During mounting, measure the clearance between the inner or outer ring and lowest roller, depending on the bearing internal design (**fig. 38**).

Before measuring, rotate the inner or outer ring a few times. Both bearing rings and the roller complement must be centrically arranged relative to each other.

For larger bearings, especially those with a thin-walled outer ring, the measurements are affected by the elastic deformation of the rings, caused by the weight of the bearing or the force to draw the feeler gauge blade through the gap between the raceway and an unloaded roller. To establish the "true" clearance before and after mounting, use the following procedure (**fig. 39**):

1. Measure the clearance "c" at the 12 o'clock position for a standing bearing or at the 6 o'clock position for an unmounted bearing hanging from the shaft.
2. Measure the clearance "a" at the 9 o'clock position and "b" at the 3 o'clock position without moving the bearing.
3. Obtain the "true" radial internal clearance with relatively good accuracy from 0,5 (a + b + c).

Measuring the lock nut tightening angle

This method can be used when mounting bearings with a bore diameter d ≤ 120 mm. Recommended values for the tightening angle α are listed in the relevant product section.

Before starting the final tightening procedure, push the bearing up onto the tapered seat until it is firmly in position. By tightening the nut through the recommended angle α (**fig. 40**), the bearing is driven up over the proper distance on the tapered seat. The bearing inner ring then has the requisite interference fit. The residual clearance should be checked if possible.

Fig. 39

Procedure to establish the "true" clearance before and after mounting

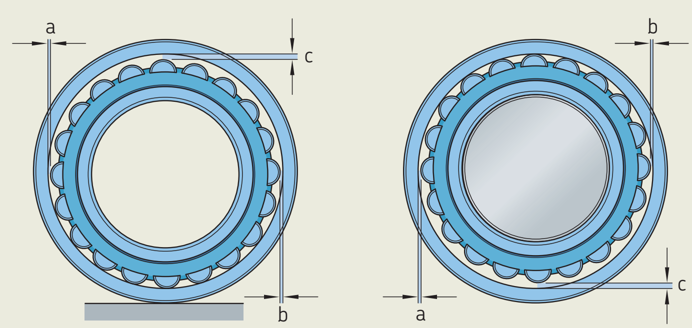


Fig. 37

Measuring of internal clearance before mounting

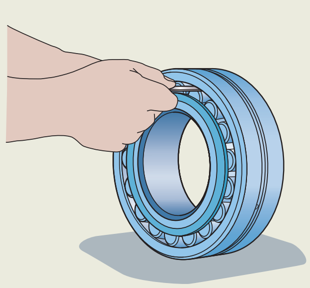


Fig. 38

Measuring of internal clearance during mounting

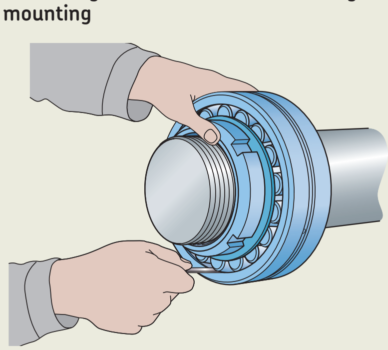
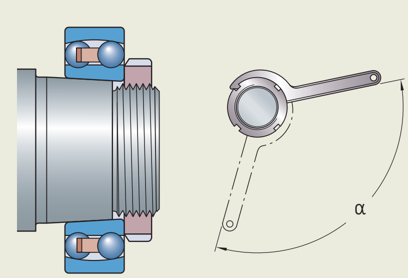


Fig. 40

Tightening angle α



Measuring the axial drive-up

Bearings with a tapered bore can be mounted by measuring the axial drive-up of the inner ring on its seat. Recommended values for the required axial drive-up are listed in the relevant product section.

However, the SKF Drive-up Method is recommended for medium-size and large bearings. This method provides a reliable and easy way to determine the degree of interference. The correct fit is achieved by controlling the axial displacement of the bearing from a pre-determined position. The equipment for the SKF Drive-up Method is shown in **fig. 41**. It includes an SKF hydraulic nut (**1**) fitted with a dial indicator (**2**), and a hydraulic pump (**3**) fitted with a pressure gauge (**4**).

The SKF Drive-up Method is based on a two-step mounting procedure (**fig. 42**):

- Step 1
  Push the bearing to its starting position by applying the prescribed pressure to the hydraulic nut.
- Step 2
  Increase the pressure on the hydraulic nut so the bearing inner ring is pushed further on its tapered seat to its final position. The prescribed displacement is measured by the dial indicator.

Recommended values for the requisite oil pressure to reach the start position and the axial displacement to reach the final position for individual bearings are available from the *SKF Drive-up Method Program* (skf.com/drive-up).

Measuring the inner ring expansion

Measuring the inner ring expansion is a quick and accurate method for determining the correct position of large spherical and CARB toroidal roller bearings on their seats (d ≥ 340 mm, depending on the series). To apply this method, use common hydraulic mounting tools and a SensorMount, which consists of a bearing with a sensor embedded in the inner ring and a dedicated hand-held indicator (**fig. 43**). Aspects such as bearing size, shaft material and design (solid or hollow), and surface finish do not need any special consideration.

Test running

Once assembly is complete, an application should undergo a test run to determine that all components are operating properly. During a test run, the application should run under partial load and, where there is a wide speed range, at low or moderate speeds.

IMPORTANT: A rolling bearing should never be started up unloaded and then rapidly accelerated to high speed, as there is a significant risk that the rolling elements will slide and damage the raceways. A minimum bearing load needs to be applied (refer to *Minimum load* in the relevant product section).

Any noise or vibration can be checked using SKF condition monitoring equipment. Normally, bearings produce an even "purring" noise. Whistling or screeching indicates inadequate lubrication. An uneven rumbling or hammering is, in most cases, caused by the presence of contaminants in the bearing or to bearing damage caused during mounting.

An increase in bearing temperature immediately after start-up is normal. In the case of grease lubrication, the temperature does not drop until the grease has been evenly distributed in the bearing arrangement, after which an equilibrium temperature is reached. Unusually high or constantly peaking temperatures indicate too much lubricant in the arrangement, too heavy preload or that the bearing is radially or axially distorted. Other causes could be that associated components have not been made or mounted correctly, or that the seals are generating too much heat.

Fig. 41

Equipment for the SKF Drive-up Method

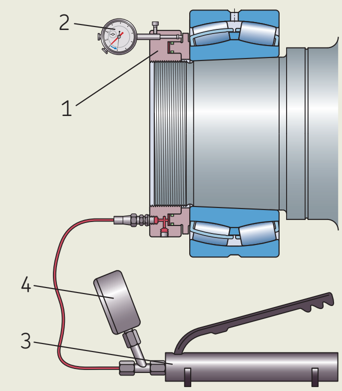


Fig. 42

Two-step mounting procedure for the SKF Drive-up Method

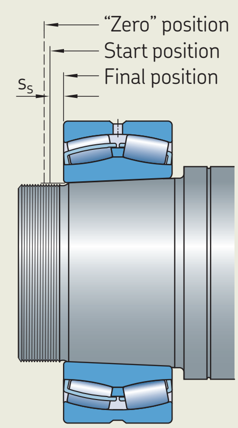
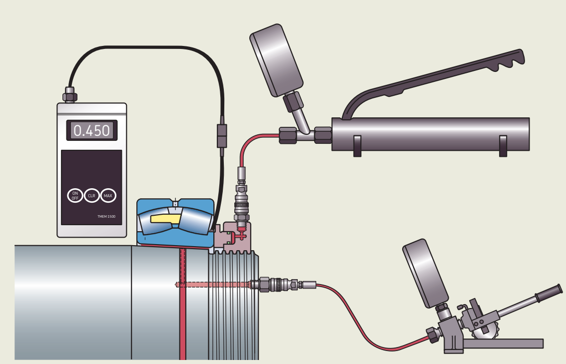


Fig. 43

Mounting with SKF SensorMount



During the test run, or immediately afterwards, check the seals, any lubrication systems and all fluid levels. If noise and vibration levels are severe, it is advisable to check the lubricant for signs of contamination.

Machines on standby

Machines on standby should be rotated or run as frequently as possible to redistribute the lubricant within the bearings and change the position relative to the raceways to reduce the risk of false brinelling and standstill corrosion.

Dismounting

There are several reasons why bearings may need to be dismounted. For example, the bearings may need to be replaced or they may have to be removed to access other components. If bearings are to be used again after dismounting, the force used to dismount them must never be applied through the rolling elements.

With separable bearings, the ring with the rolling element and cage assembly can be removed independently of the other ring. With non-separable bearings, the ring having the looser fit should be withdrawn from its seat first. To dismount a bearing with an interference fit, the tools described in the following section can be used. The choice of tools depends on the bearing type, size and fit (**table 2**, **page 202**). Bearing sizes are categorized as follows:

- small → d ≤ 80 mm
- medium-size → 80 mm < d < 200 mm
- large → d ≥ 200 mm

Dismounting bearings fitted on a cylindrical shaft seat

Cold dismounting

Small bearings can be dismounted from a shaft by applying light hammer blows via a suitable drift to the ring side face, or preferably by using a mechanical puller. The claws must be applied to the inner ring or an adjacent component (**fig. 44**). Dismounting is made easier if slots for the claws of a puller are provided in the shaft and/or housing shoulders. Alternatively, tapped holes in the housing shoulder can be provided to accommodate push-out bolts (**fig. 45**).

Medium-size and large bearings generally require greater force than a mechanical tool can provide. Therefore, SKF recommends using either hydraulically assisted tools or the oil injection method, or both. Using the oil injection method assumes that the necessary oil supply ducts and distribution grooves have been designed into the shaft (**fig. 46**).

Fig. 44

Dismounting with aid of a mechanical puller

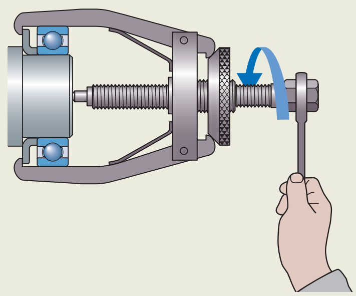


Fig. 45

Dismounting with aid of push-out bolts

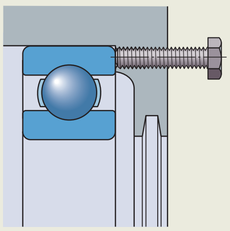
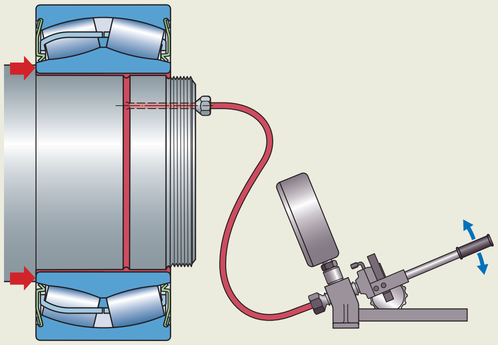


Fig. 46

Dismounting with aid of oil injection



Hot dismounting

Dismounting by heating is a suitable method when removing the inner rings of needle roller bearings or cylindrical roller bearings of the type NU, NJ and NUP. Two different tools for this purpose are common: heating rings and adjustable induction heaters.

Heating rings are typically used to mount and dismount the inner ring of small to medium-size bearings that are all the same size. Heating rings are made of light alloy. They are radially slotted and equipped with insulated handles (**fig. 47**).

If inner rings with different diameters are dismounted frequently, SKF recommends using an adjustable induction heater. These heaters (**fig. 48**) heat the inner ring rapidly without heating the shaft.

Special, fixed induction heaters have been developed to dismount the inner rings of large cylindrical roller bearings (**fig. 49**).

Induction heaters and heating rings are available from SKF. For additional information, refer to the *SKF bearing maintenance handbook* or skf.com/mapro.

⚠ WARNING
Fire hazard. Never use an open flame for hot dismounting.

Dismounting bearings fitted on a tapered shaft seat

Small bearings can be dismounted using a mechanical or hydraulic puller that engages the inner ring. Self-centring pullers equipped with spring-operated arms should be used to simplify the procedure and avoid damage to the bearing seat. If it is not possible to apply the claws of the puller to the inner ring, withdraw the bearing via the outer ring or use a puller in combination with a pulling plate (**fig. 50**).

It is much easier and safer to dismount medium-size and large bearings when the oil injection method is used. This method injects oil, under high pressure, between the two tapered mating surfaces, via a supply duct and a distribution groove. This significantly reduces the friction between the two surfaces and separates the bearing from its seat (**fig. 51**).

⚠ WARNING
To avoid the risk of serious injury, attach a provision such as a lock nut or end plate to the shaft end to limit the bearing travel when it suddenly comes loose.

Fig. 47

Heating ring

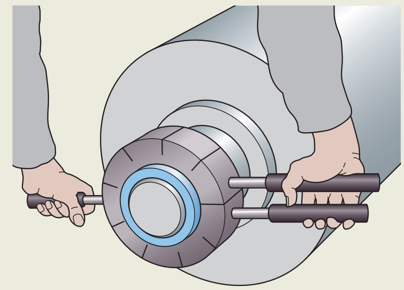


Fig. 48

Adjustable induction heater

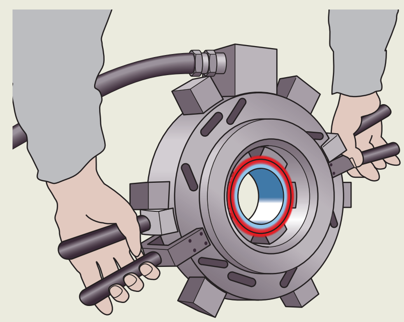


Fig. 49

Special fixed induction heater

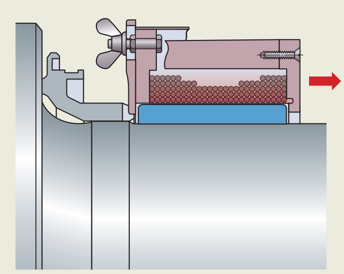


Fig. 50

Dismounting with aid of a puller

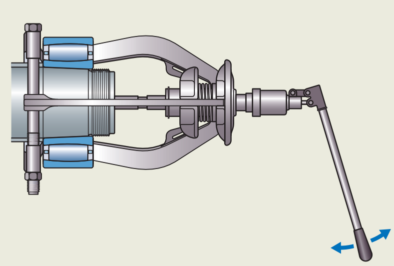
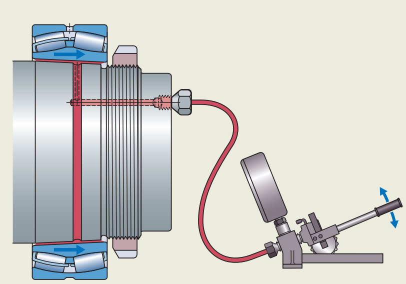


Fig. 51

Dismounting with aid of oil injection – a stop is provided



Dismounting bearings fitted on an adapter sleeve

To dismount small bearings fitted on an adapter sleeve and a plain shaft, loosen the sleeve lock nut a few turns, then use a hammer of suitable size to tap a small steel block evenly around the bearing inner ring side face (fig. 52).

For small bearings fitted on an adapter sleeve and a stepped shaft with a spacing collar between the shoulder and the bearing side face, loosen the adapter sleeve lock nut a few turns and apply a couple of sharp hammer blows to a bearing fitting tool abutting the sleeve lock nut (fig. 53).

Using a hydraulic nut for dismounting bearings fitted on an adapter sleeve and a stepped shaft with a spacing collar makes bearing dismounting easy. However, to use this method, you should mount a suitable stop that abuts the piston of the hydraulic nut (fig. 54). If the sleeves are provided with oil supply ducts and distribution grooves, dismounting becomes easier because the oil injection method can be used.

Fig. 52

Dismounting by tapping a small steel block with an appropriate hammer

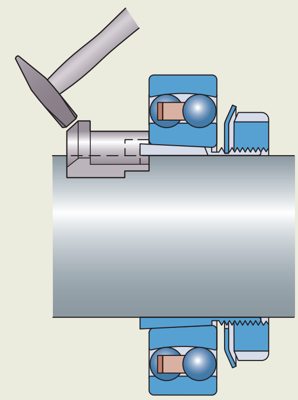


Fig. 53

Dismounting by a couple of sharp hammer blows applied to a bearing fitting tool abutting the sleeve lock nut

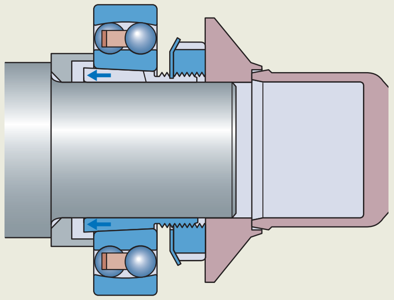
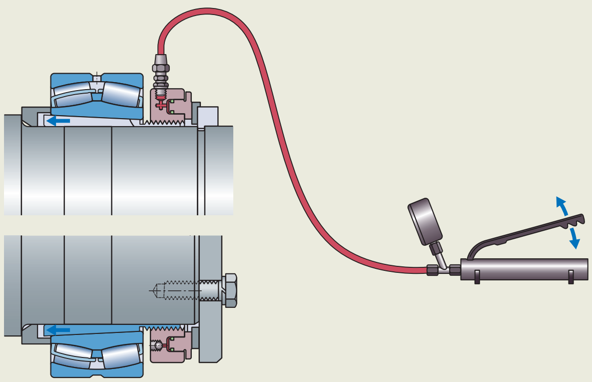


Fig. 54

Dismounting from adapter sleeve with aid of a hydraulic nut



Dismounting bearings fitted on a withdrawal sleeve

When dismounting a bearing fitted on a withdrawal sleeve, the locking device (for example a lock nut or end plate) must be removed.

Small and medium-size bearings can be dismounted with a lock nut and a hook or impact spanner (fig. 55).

Medium-size and large bearings fitted on a withdrawal sleeve can be easily dismounted using a hydraulic nut.

Withdrawal sleeves with a bore diameter ≥ 200 mm are provided, as standard, with two oil supply ducts and distribution grooves in both the bore and outside surface. When using the oil injection method, two hydraulic pumps or oil injectors and appropriate extension pipes are needed (fig. 56).

⚠ WARNING

To avoid the risk of serious injury, attach a stop behind the hydraulic nut at the shaft end (fig. 57). The stop prevents the withdrawal sleeve and hydraulic nut from shooting off the shaft if the sleeve separates suddenly from its seat.

Fig. 55

Dismounting with a lock nut and a hook or impact spanner

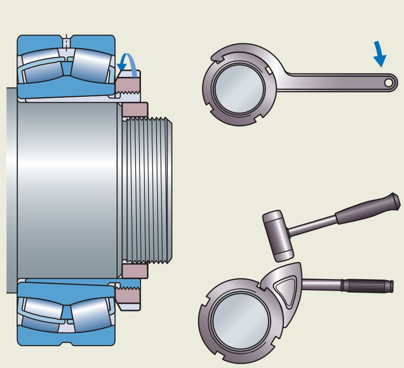


Fig. 56

Dismounting from withdrawal sleeve with oil injection

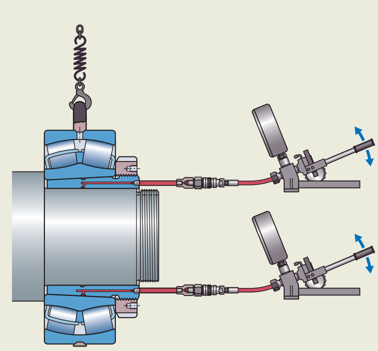
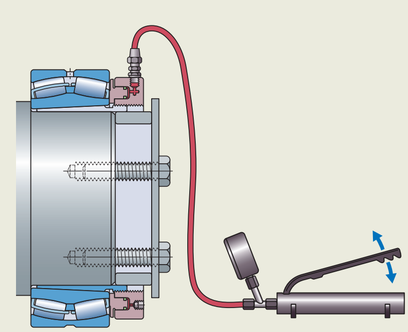


Fig. 57

Dismounting from withdrawal sleeve with aid of a hydraulic nut



Inspection and monitoring

This section describes various aspects of inspecting and monitoring bearings in operation for the purpose of preventing problems. It also gives an introduction to troubleshooting and links to more detailed troubleshooting procedures.

Inspection during operation

Spotting early indications of bearing damage makes it possible to replace bearings during regularly scheduled maintenance. This avoids otherwise costly unscheduled machine downtime if a bearing fails. Important parameters for monitoring machine condition include noise, temperature and vibration.

Bearings that are worn or damaged usually exhibit identifiable symptoms (*Troubleshooting*, **page 213**). There can be a number of possible causes and this section helps identify some of these.

For practical reasons, not all machines or machine functions can be monitored using advanced systems. In these cases, trouble can be detected by looking at or listening to the machine. However, if deterioration can be detected by human senses, damage may already be extensive. Using objective technologies, such as advanced vibration analysis, means damage can be detected before it becomes problematic (**diagram 1**). By using condition-monitoring instruments and the SKF enveloped acceleration technology, the pre-warning time can be maximized.

An example of how damage can progress is shown in **fig. 58** and shown conceptually in **diagram 1**. A damage scenario may follow this sequence:

1 Bearing starts to show abrasive wear.
2 First spall, detected by SKF enveloped acceleration technology.
3 Spalling has developed to an extent that the damage can be detected by standard vibration monitoring.
4 Advanced spalling causes high vibration and noise levels and an increase in operating temperature.
5 Severe damage occurs: fatigue fracture of the bearing inner ring.
6 Catastrophic failure occurs with secondary damage to other components.

Monitoring noise and vibration

A common method used to try to identify deterioration or damage in a bearing is to listen. Bearings in good condition produce a soft purring noise. Grinding, squeaking and other irregular sounds usually indicate that the bearings are in poor condition, or that something is wrong. However, sound monitoring is of limited use. SKF recommends vibration monitoring. It is more thorough and allows better monitoring of bearings and rotating equipment.

Vibration monitoring is based on three fundamental facts:

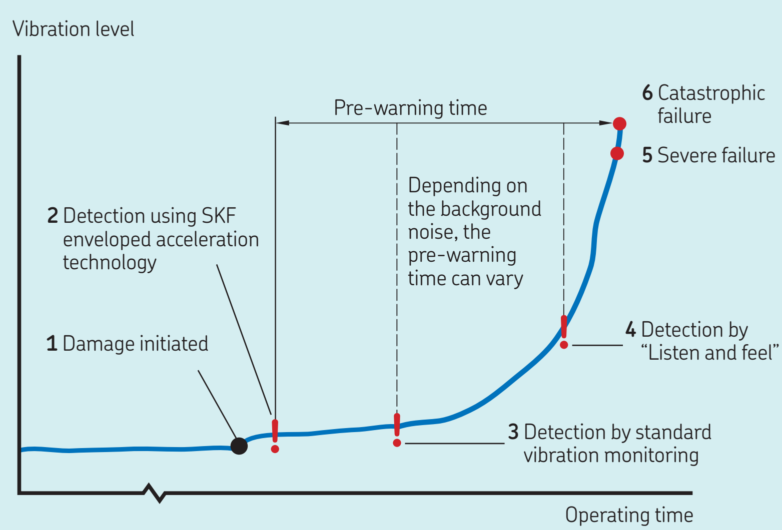
- All machines vibrate.
- The onset of a mechanical problem is generally accompanied by an increase in vibration levels.
- The nature of the fault can be determined by analyzing the vibration characteristics.

Fig. 58

Progress of damage

Diagram 1

The advantage of advanced condition monitoring



Monitoring temperature

It is important to monitor the operating temperature at bearing positions. If the operating conditions have not been altered, a sudden increase in temperature is often an indication of developed bearing damage and possible imminent failure of the bearing. However, keep in mind that a natural temperature rise can last up to one or two days immediately after first machine start-up and after each relubrication when using grease.

Monitoring lubrication conditions

Bearings can only achieve maximum performance levels with adequate lubrication. The lubrication conditions of a bearing should therefore be monitored closely. The condition of the lubricant itself should also be assessed periodically, preferably by taking samples and having them analysed.

SKF recommends the following general guidelines for lubrication-related inspection activities:

- Check for lubricant leaks in the areas surrounding the bearing positions.
- Keep protective collars and labyrinth seals filled with grease for maximum protection.
- Check that automatic lubricating systems are functioning properly and providing the appropriate amount of lubricant to the bearings.
- Check the lubricant level in sumps and reservoirs, and replenish as necessary.
- Where manual grease lubrication is employed, relubricate according to schedule.
- Where oil lubrication is used, change oil according to schedule.
- Always make sure that the specified lubricant is used.

Inspection during a machine shutdown

When a machine is not in operation, it is an opportunity to assess the condition of bearings, seals, seal counterfaces, housings, and lubricant. A general inspection can often be done by removing a housing cover or cap. If a bearing appears to be damaged, it should be dismounted and thoroughly inspected.

Shaft and belt alignment, and a thorough inspection of the machine foundation and exterior, can also be done during a machine shutdown.

Any condition, whether it is a missing shim or a deteriorating foundation, can negatively affect machine performance. The sooner any problem is identified, the sooner corrective action can begin. It is far less costly to replace bearings and associated components during a regularly scheduled shutdown than during unscheduled downtime that unexpectedly takes the machine out of service.

Inspecting bearings

Bearings are not always easily accessible. However, when bearings are partially or fully exposed, visual checks can be made. The most practical time to inspect bearings is during routine maintenance.

When inspecting a mounted bearing, SKF recommends following these general guidelines:

- **Preparation**
  - Clean the external surface of the machine.
  - Remove the housing cover, or the housing cap, to expose the bearing.
  - Take lubricant samples for analysis. For oil lubrication, take samples from the sump/reservoir. For grease lubricated open bearings, take samples from various positions within the bearing and surroundings. Inspect the condition of the lubricant. Impurities can often be detected by spreading a thin layer of the lubricant on a sheet of paper and examining it under a light.
  - Clean the exposed external surfaces of the bearing with a lint-free cloth.
- **Inspection**
  - Inspect the exposed external surfaces of the bearing for corrosion. Inspect the bearing rings for any abnormal signs.
  - For sealed bearings, inspect the seals for wear or damage.
  - Where possible, rotate the shaft very slowly and feel for uneven resistance in the bearing; an undamaged bearing turns smoothly.
- **Detailed inspection of grease lubricated bearings**
  Grease lubricated open bearings in split plummer blocks can be subjected to a more detailed in-situ inspection as follows:

  - Remove all grease around the bearing.
  - Remove as much grease from the bearing as possible using a non-metallic scraper.
  - Clean the bearing with a petroleum-based solvent by spraying the solvent into the bearing. Rotate the shaft very slowly while cleaning it, and continue to spray until the solvent ceases to collect dirt and grease. For large bearings that contain a build-up of severely oxidized lubricant, clean them with a strong alkaline solution containing up to 10% caustic soda and 1% wetting agent.
  - Dry the bearing, and surrounding parts, with a lint-free cloth or clean, moisture-free compressed air (but do not rotate or spin the bearing).
  - Inspect the bearing raceways, cage(s) and rolling elements for spalls, marks, scratches, streaks, discolouration and mirror-like areas. Where applicable, measure the radial internal clearance of the bearing (to determine if wear has taken place) and confirm that it is within the expected range.
  - If the condition of the bearing is satisfactory, apply the appropriate grease to the bearing and the housing and immediately close the housing. If bearing damage is evident, dismount the bearing and protect it from corrosion. Then conduct a full analysis.

- **General recommendations**
  - Take photographs throughout the inspection process to help document the condition of the bearing, lubricant and machine in general.
  - Check the condition of the grease at different places and compare with fresh grease (**fig. 59**). Keep a representative sample of the grease for further analysis.
  - Certain large and medium-size bearings are suitable for reconditioning. For additional information, refer to the *SKF bearing maintenance handbook* and publication *SKF Remanufacturing Services*.

Inspecting seal counterfaces

To be effective, a seal lip must run on a smooth counterface. If the counterface is worn or damaged, the seal lip will cease to function properly.

When inspecting the seal counterface, also check for corrosion, shaft wear, scratches, dents, lip wear, lip tears and so on. If corrosion is evident but not severe, use a fine wet/dry abrasive paper to remove it, and then make sure all remnants are also removed. Worn counterface parts of the shaft can be repaired using SKF Speedi-Sleeve.

⚠ WARNING
Avoid inhaling, ingesting or contacting solvents and alkaline solutions. These can cause skin and eye burns or damage to respiratory or digestive tract. If necessary, seek medical assistance.

Fig. 59

Condition of the grease

Fresh grease: brown colour

Used grease: colour turned greyish

Troubleshooting

Bearings that are not operating properly usually exhibit identifiable symptoms. The best way to identify these symptoms, and take corrective action at an early stage, is to establish a plant-wide condition monitoring programme.

In cases where condition monitoring equipment is not available or practical, the section *Troubleshooting* of the *SKF bearing maintenance handbook* presents some useful hints to help identify the most common symptoms, their causes and, whenever possible, some practical solutions. Depending on the degree of bearing damage, some symptoms may be misleading and, in many cases, are the result of secondary damage. To effectively troubleshoot bearing problems, it is necessary to analyse the symptoms according to those first observed in the application. This is dealt with in more detail in the publication *Bearing damage and failure analysis*.

C

Bearing selection examples

Bearing selection examples

This section contains several worked examples that show the *Bearing selection process*, **page 60**, applied to various machines and application cases.

Each example is presented as a number of steps that generally follows the sequence in the bearing selection process. However, interdependencies in any particular application case may require looping back and forth between the process steps and where this occurs it is fully described in the example.

C.1 Vibrating screen

This example shows the bearing selection process applied to an application case in which a vibrating screen manufacturer is selecting the bearings for a new machine.

The steps in the example follow the sequence in the bearing selection process. Refer to sections B.1 – B.8 for a full description of each process step.

Performance and operating conditions

The new machine is a free circular motion vibrating screen. The vibrator unit is composed of a shaft with two bearings and counterweights. This means the main radial load rotates with the shaft while the outer ring is stationary. The application drawing is shown in fig. 1.

The relevant performance requirements, operating conditions and input parameters for the bearing selection are:

- mass of screen box without charge: G = 6 100 kg
- shaft diameter: 140 mm
- rotational speed: n = 756 r/min
- angular velocity ($n \times 2\pi/60$): ω = 79,2 rad/s
- radius of vibration: r = 8,1 mm
- distance between the centres of gravity of the counterweights and shaft axis: R = 80 mm
- distance between the bearings: 3 m
- lubrication method: grease
- operating temperature of the bearings: T = 75 °C *(165 °F)*
- environment: the screen may be located outdoors, in harsh, dusty and humid conditions
- required SKF rating life: 20 000 h

Bearing type and arrangement

A locating/non-locating bearing arrangement is used. The bearing on the drive side is the locating bearing. This limits axial displacement of the transmission pulley, which saves energy and increases belt life. The opposite bearing is non-locating to accommodate axial displacement caused by thermal expansion of the shaft.

The distance between the bearings is 3 metres, and the vibrating screen structure is made of welded and bolted steel parts. Shaft deflection and misalignment of supports under load require bearings that can compensate for misalignment.

Spherical roller bearings are selected for this new vibrating screen (fig. 2), which is the typical solution. They can carry high loads and accommodate misalignment between the inner and outer ring without any reduction of their service life.

Fig. 1

Free circular motion vibrating screen

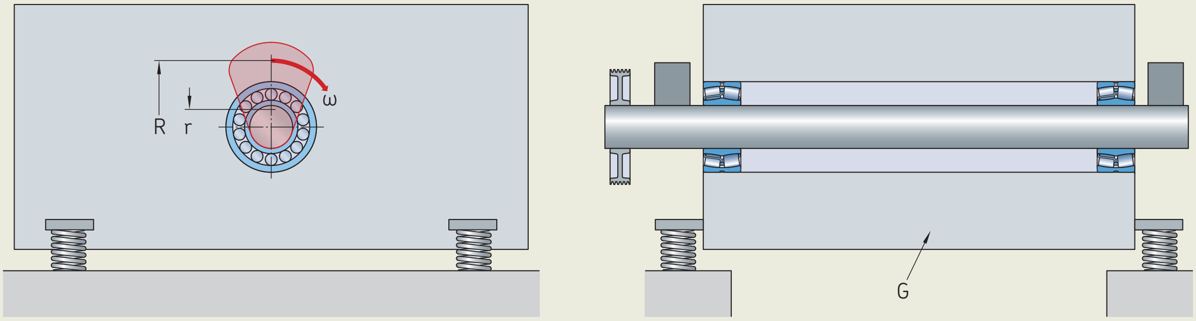
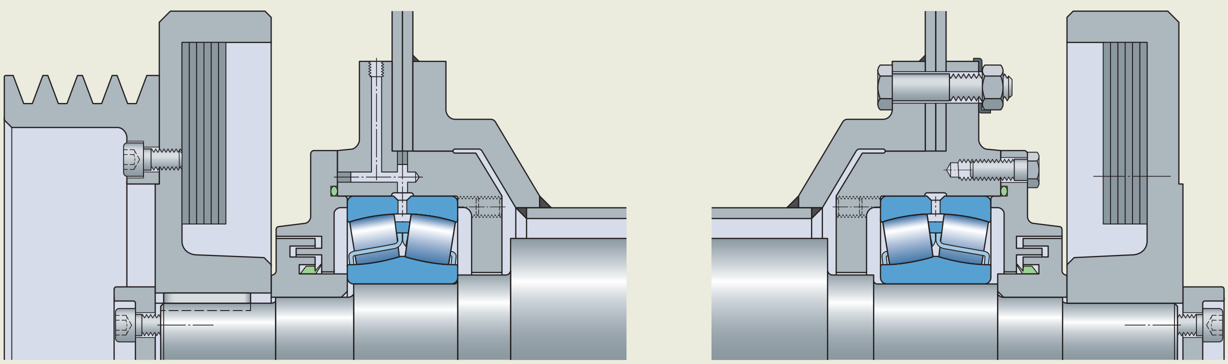


Fig. 2

Bearing arrangement



Bearing size

A shaft diameter of 140 mm is needed to transmit the required shaft drive torque and limit shaft deflection.

SKF supplies bearings in the 223 series for vibratory applications. Based on the required shaft diameter, the 22328 CCJA/W33VA405 is selected. We will check its size using the SKF rating life.

Product data for 22328 CCJA/W33VA405 is on page 800.

For vibrating screens, the equivalent dynamic bearing load, P, can be estimated using:

$$P = \left(\frac{1{,}2 \times G \times r \times \omega^2}{2}\right) = \left(\frac{1{,}2 \times 6\,100 \times 0{,}0081 \times 79{,}2^2}{2}\right)$$

$$= 186 \text{ kN}$$

The load ratio C/P = 1 357/186 = 7,3

SKF rating life

$L_{10mh} = a_{SKF}\, L_{10h}$

1. Lubrication condition – the viscosity ratio, κ

$\kappa = \nu/\nu_1$

The rated viscosity ν_1 = 10 mm²/s (diagram 14, page 101).

A viscosity ratio, κ, of about 4 is targeted to operate in full film lubrication conditions, therefore ν should be about 40 mm²/s.

You need to verify the viscosity ratio after you have selected your lubricant.

2. Contamination factor, η_c

Given:

- contamination conditions are typical (i.e. open bearings, no filtering, wear particles and ingress from surrounding and harsh environment)
- d_m = 220 mm

then, using table 6, page 105, η_c = 0,2

3. Life modification factor, a_{SKF}

Given:

- $\kappa = 4$
- $\eta_c\, P_u/P = 0,2 \times 132/186 = 0,14$
- 22328 CCJA/W33VA405 is an SKF Explorer bearing

then, using **diagram 10**, **page 97**, for radial roller bearings, $a_{SKF} = 1,3$

$$L_{10mh} = a_{SKF} \left(\frac{10^6}{60\, n}\right) \left(\frac{C}{P}\right)^{10/3}$$

$$= 1,3 \times (10^6/(60 \times 756))\,(7,3)^{10/3} = 21\,500 \text{ h} > 20\,000 \text{ h}$$

Conclusion

SKF bearing 22328 CCJA/W33VA405 is a suitable size to meet the rating life requirements.

Lubrication

Selecting grease or oil

On **page 113**, **table 1** provides limits for the nd_m value, up to which grease lubrication is normally a suitable solution in terms of relubrication intervals at normal temperatures.

Input values:

- spherical roller bearing in the 223 series
- $C/P = 7,3$
- $n\, d_m = 756 \times (140 + 300)/2 = 166\,320$

From **table 1**, **page 113**, the recommended nd_m limit for $C/P \approx 8$ is 150 000, which is somewhat below the actual nd_m value. The operating conditions are at the limits where grease lubrication is suitable, and you can expect short relubrication intervals. But this is not an issue for vibrating screens, and you can select grease lubrication.

Grease selection

You can find a suitable SKF grease using the *SKF bearing grease selection chart*, **page 124**. Grease selection criteria are:

- temperature: 75 °C *(165 °F)* → M
- speed: $n\, d_m \approx 166\,000$ → M to H
- load: $C/P \approx 8$ → M
- severe vibrations
- humid outdoor conditions → good rust inhibiting properties

SKF LGEP2 is a suitable choice provided a viscosity ratio, κ, of 4 is confirmed.

LGEP2 has the following properties:

- $\nu = 200$ mm²/s at 40 °C *(105 °F)*
- $\nu = 16$ mm²/s at 100 °C *(210 °F)*
- operating viscosity at 75 °C *(165 °F)* is around 40 mm²/s, based on **diagram 13**, **page 100**.
- $\kappa = \nu/\nu_1 = 40/10 = 4$ is confirmed

Relubrication interval and quantity

Experience suggests relubricating the bearings in the vibrating screen every 75 h with 30 g of grease. The short intervals are needed to push out contamination, while the reduced quantity limits heat generation caused by high grease volumes.

Using the standard relubrication interval from **diagram 2**, **page 112**, and input values gives:

- $n\, d_m\, b_f = 166\,320 \times 2 \approx 330\,000$
- $C/P \approx 8$

The relubrication interval is 1 700 h. This needs to be reduced, with contamination and vibration considered (**table 2**, **page 115**), confirming approximately the experienced values used for vibrating screen bearings.

Relubrication quantity is:

$$G_p = 0,002\, D\, B = 0,002 \times 300 \times 102 = 61 \text{ g}$$

Standard relubrication of the bearings every 75 h with 30 g of grease will maintain adequate lubrication condition.

Initial grease fill

The free volume in the bearing, which should be filled with grease, is approximately:

$$V = \frac{\pi}{4}\, B\, (D^2 - d^2) \times 10^{-3} - \frac{M}{7,8 \times 10^{-3}}$$

$$V = 3,14/4 \times 102 \times (300^2 - 140^2) \times 10^{-3} - 36,5/0,0078 = 957 \text{ cm}^3$$

For a filling degree of 50%, you need about 430 g of grease per bearing.

Operating temperature and speed

Experience from similar applications is broad and a bearing operating temperature between 70 to 80 °C (*160 to 175 °F*) can be assumed.

The screen charge is at ambient temperature and there are no other external sources generating heat. The speed is < 50% of the limiting speed. Although the load ratio C/P < 10, no detailed thermal analysis is required.

The actual operating temperature should be checked on the real machine.

The bearing frictional losses are 1 900 W per bearing, calculated with the *SKF Bearing Calculator* (skf.com/bearingcalculator).

Bearing interfaces

The radial load turns in phase with the rotating inner ring, while the outer ring stands still. Therefore, the inner ring has a stationary load condition and the outer ring a rotating load condition. An interference fit is needed between the outer ring and the housing. A loose fit can be used between the inner ring and the shaft.

The standard fit recommendations are listed in **table 1**.

There are reasons for choosing dimensional tolerances other than the standard fits:

- Choose f6Ⓔ for easy axial displacement of the inner ring. To reduce the risk of fretting corrosion, consider hardening the shaft seat.
- Select P6Ⓔ (tighter tolerances) to improve outer ring support and bearing service life.

Additional recommendations

The following additional factors are recommended:

- The bearing centre should be aligned with the frame centre of the vibrating screen (**fig. 3**).
- The housing wall thickness should be greater than 40% of the bearing width.
- Design the housing to be as symmetrical as possible, so it has the same thickness on both sides of the vibrating screen frame, in order to avoid housing deformation (**fig. 4**).
- Machine threads in the housing to make it easier when dismounting the housing from the screen body and the bearing of the housing by the use of bolts (**fig. 5** and **fig. 6**, **page 220**).

Table 1

Seat tolerances for standard conditions

	Dimensional tolerance	Total radial run-out tolerance	Total axial run-out tolerance	Ra
Shaft	g6Ⓔ	IT5/2	IT5	1,6 µm
Housing	P7Ⓔ	IT6/2	IT6	3,2 µm

Fig. 3

Aligning the bearing centrally with the vibrating screen frame

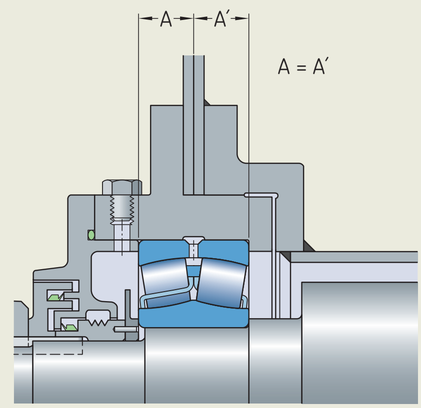
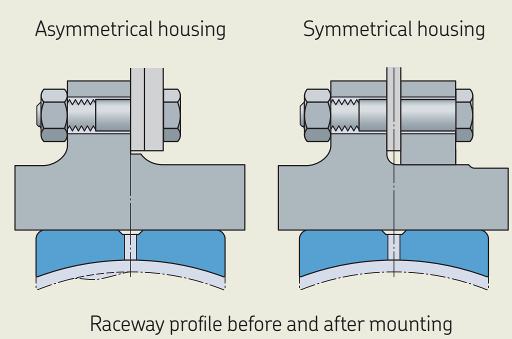


Fig. 4

Symmetrical housing prevents raceway deformation



Bearing execution

The bearing selected for this application is a spherical roller bearing for vibratory applications (*Designs and variants* **page 775**).

These bearings are identified by the designation suffixes VA405 and VA406. They have a C4 internal clearance, which is required because of the interference fit of the outer ring in combination with the temperature difference between inner and outer rings, particularly during start-up situations. Their hardened window-type cages reduce friction and wear in the bearing when operating under rotating outer ring load and high acceleration conditions, resulting in a lower operating temperature and longer lubricant life.

The VA406 execution is intended for the non-locating support and has a PTFE coated bore. This helps to prevent fretting corrosion, which can occur because of the loose fit and vibration.

Sealing, mounting and dismounting

Vibrating screen designs generally use labyrinth seals to protect the rolling bearings. With this type of seal, it is important to maintain a sufficient quantity of grease in the labyrinth gaps so that dirt and moisture are kept away from the bearings. Quantities and relubrication intervals should be adjusted according to the operator's observations.

Check the total radial run-out of the housing seat when the housing is mounted to the screen frame. Inadmissible deformation might occur and can require corrective action.

Overall conclusions

- The 22328 CCJA/W33VA405 bearing meets the rating life requirement.
- SKF grease LGEP2 is appropriate for the given operating conditions.
- Maintenance and condition monitoring aspects have not been included in this example. For additional information about SKF offers for vibrating screens, refer to the information on the SKF website under *Industry Solutions*.

Fig. 5

Bolts used to dismount the housing from the screen body

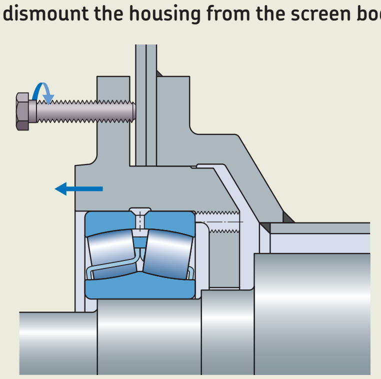
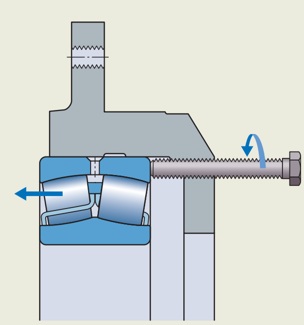


Fig. 6

Bolts used to dismount the bearing from the housing



C.2 Rope sheave

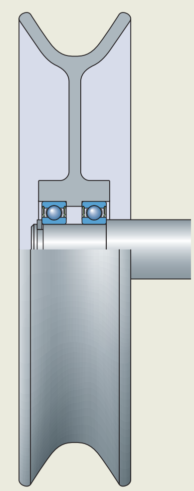
This example shows the bearing selection process applied to an application case in which bearings are to be selected for the rope sheaves on a new paper machine.

A paper machine manufacturer wants to build a new machine using rope sheaves of their standard design. The end customer requires the sheaves to be maintenance free for five years.

The steps in the example follow the sequence in the bearing selection process. Some steps, such as *Bearing size*, require more than one iteration if the calculation is dependent on a subsequent stage in the process. This is indicated in the heading (for example, *Bearing size (step 2)*, **page 224**). Refer to sections **B.1 – B.8** for a full description of each process step.

Fig. 1

Traditional rope sheave used in paper machines



Performance and operating conditions

The rope sheaves (**fig. 1**) are positioned between rolls/cylinders of the paper machine and rotate all the time that the machine is in operation. In this application the outer ring of each rope sheave rotates continuously. The operating conditions are:

- rotational speed: 2 450 r/min
- radial load: 1,1 kN created by the weight of the sheave and by the rope tension, shared between the bearings
- axial load: zero – because of the orientation of the sheaves, the rope creates no axial load
- environment: hot and humid, with 80 °C *(175 °F)* ambient temperature

Bearing type and arrangement

Because loads are low and speeds moderate, rope sheaves use two deep groove ball bearings. For a long, maintenance-free period sealed bearings are required. SKF deep groove ball bearings are available with various seal executions.

A floating bearing arrangement is used, where each bearing locates the sheave axially in one direction and the whole arrangement is able to move axially over a small distance between the two end positions.

Bearing size

The manufacturer's existing rope sheave design uses two 6207-2RS1 bearings. SKF has replaced the RS1 seal with the RSH seal. In this example we check the suitability of 6207-2RSH bearings (**page 274**).

The next step in the selection process is to determine the method, on which to base the size selection. The bearings are running in typical operating conditions and, therefore, rolling contact fatigue is the probable failure mode. We base the size selection on rating life.

Basic rating life

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

As there is no axial load, the equivalent dynamic bearing load, P, on each bearing is equal to the radial load divided by two.

- P = 0,55 kN
- load ratio C/P = 49

The basic rating life L_{10h} = 804 800 h. This is much longer than the required maintenance-free period of 5 years (43 800 h).

Conclusion

- With such a high basic rating life at 2 450 r/min, it is recommended to check that the bearing is sufficiently loaded to maintain ball rolling and avoid ball sliding. This will be done after the lubrication is checked, because lubricant viscosity influences the requisite minimum load.
- Grease life should be checked to see if the bearing fulfills the end-customer's requirement.

The SKF rating life, L_{10mh}, will be calculated after the lubrication is checked and the operating temperature and speed are checked, because lubricant viscosity influences the result. This will be done in *Bearing size (step 2)*, **page 224**.

Lubrication

Bearing 6207-2RSH is filled with MT33 grease (**table 2**, **page 245**). The operating temperature should be defined before continuing.

Operating temperature and speed

When the load ratio C/P > 10, the operating temperature is below 100 °C *(210 °F)*, the operating speed is below 50% of the limiting speed, and there is no pronounced external heat input, a detailed thermal analysis is not required. In this example:

- load ratio: C/P = 49 > 10
- operating speed: 2 450 r/min < 0,5 × 6 300 (limiting speed)
- From experience of rope sheaves operating in similar conditions, the bearing operating temperature is about 90 °C *(195 °F)*.

So a detailed thermal analysis is not required.

Lubrication (step 2)

1. Grease life MT33

Grease life can be estimated using **diagram 1**, **page 246**. Because the bearing outer ring rotates, for grease life estimations, nD is used instead of nd_m (**table 2**, **page 115**).

Then, using the input values:

- nD = 2 450 × 72 = 176 400
- MT33 grease with a grease performance factor, GPF = 1
- operating temperature of about 90 °C *(195 °F)*

The grease life, L_{10h}, is about 12 500 hours, which is less than the required 5-year maintenance-free period.

2. Grease life WT

The SKF bearing 6207-2RSH is available with the grease WT, which has a GPF = 4. It is a polyurea-type grease with an ester base oil, **table 3**, **page 245**.

From **diagram 1**, **page 246** the grease life, $L_{10h,}$ is 50 000 hours, which is greater than 5 years.

Conclusion

The SKF bearing 6207-2RSH with the grease WT fulfills the requirement in terms of grease life.

Bearing size (step 2)

From the conclusions in *Bearing size*, **page 223**, the minimum load needs to be checked and, now the lubrication has been selected, the SKF rating life can be verified.

Minimum load

Using the minimum load equation from *Loads*, **page 254**, the minimum load, F_{rm}, is given by:

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

where:
k_r = 0,025
ν = 210 mm²/s
When determining the minimum load, to cover all critical operating conditions, use the highest oil viscosity that might occur. This will be at the lowest temperature, which is 20 °C *(70 °F)*. Base oil viscosity of WT grease at 40 °C *(105 °F)* is 70 mm²/s ≈ ISO VG 68. Estimated from **diagram 13**, **page 100**, or calculated with the *SKF Bearing Calculator* (skf.com/bearingcalculator), for WT grease ν = 210 mm²/s at 20 °C *(70 °F)*.
d_m = (d+D)/2 = (35+72)/2 = 53,5 mm

Therefore:
F_{rm} = 0,44 kN < 0,55 kN, so the bearing 6207-2RSH/WT is adequate.

SKF rating life

$L_{10mh} = a_{SKF}\, L_{10h}$

Because P < P_u, fatigue is not a factor (*Fatigue load limit, P_u*, **page 104**). However, it is useful to verify the lubrication condition (viscosity ratio) and life modification factor.

1. Lubrication condition – the viscosity ratio, κ

$\kappa = \nu/\nu_1$

The following are used:

- ν_1 is determined from **diagram 14**, **page 101**
- with d_m = 53,5 and n = 2 450 r/min, ν_1 is close to 12 mm²/s

For WT, the base oil viscosity at 90 °C *(195 °F)* can be estimated from **diagram 13**, **page 100**, or calculated with the *SKF Bearing Calculator* (skf.com/bearingcalculator) and is 12 mm²/s.

Viscosity ratio, κ = 12/12 = 1

2. Life modification factor, a_{SKF}

To determine the life modification factor for radial ball bearings, **diagram 9**, **page 96** is used, with:

- P = 0,55 kN
- κ = 1
- P_u = 0,655 kN
- η_c = 0,6
  The contamination factor is chosen based on **table 6**, **page 105**.
- SKF 6207-2RSH/WT is an SKF Explorer bearing.

With $\eta_c P_u/P$ = 0,7 and using **diagram 9**, **page 96**, the a_{SKF} of about 50 is much greater than 1, so the SKF rating life is far above the required life.

Conclusion

The bearing SKF 6207-2RSH/WT is adequate in terms of fatigue life.

Bearing interfaces

The bearing inner rings have a stationary load condition and no spacer between the inner rings in the cross-located arrangement. They are mounted with a loose fit for easy mounting. The recommended fit for standard conditions is g6Ⓔ (**table 5**, **page 148**).

The outer rings have a rotating load condition, so they are mounted with an interference. The recommended fit for standard conditions is M7Ⓔ (**table 8**, **page 151**), which has a probable interference range of –25 to +8 (**table 20**, **page 172**).

Bearings in rope sheaves of paper machines should always have an interference for the outer ring (application handbook *Rolling bearings in paper machines*). To achieve this select N6Ⓔ, which has a probable interference range of –29 to –5 (**table 21**, **page 174**). For geometrical tolerances and surface roughness, standard recommendations can be applied.

The tolerances for the bearing seats are:

	Dimensional tolerance	Total radial run-out tolerance	Total axial run-out tolerance	Ra
Inner ring	g6Ⓔ	IT5/2	IT5	1,6 µm
Outer ring	N6Ⓔ	IT6/2	IT6	3,2 µm

Bearing execution

Intial internal clearance

The current design uses bearings with Normal initial clearance. The interference fit on the outer ring reduces the internal clearance. We determine the operational clearance for both Normal and C3 initial clearance, to select the most appropriate bearing execution.

1. Initial internal clearance

Refer to *Bearing data*, **page 250**. Values obtained from **table 6**, **page 252**.

	Normal	C3
min./avg./max.	6 / 13 / 20 µm	15 / 24 / 33 µm

2. Clearance reduction caused by interference fits

There is no interference on the inner ring, therefore use:
$\Delta r_{fit} = \Delta_2 f_2$ (*Clearance reduction caused by interference fits*, **page 184**)

Obtain values for:

- factor, f_2 (**diagram 2**, **page 184**)
- probable fits for housings, Δ_2 (**table 21**, **page 174**)

Results:

d/D		0,49
f_2		0,87
Δ_2	min./avg./max.	–29 / –17 / –5 µm
Δr_{fit}	min./avg./max.	**–25** / **–15** / **–4** µm

3. Internal clearance after mounting

	Normal	C3
min./avg./max.	**–19** / **–2** / **6** µm	**–10** / **9** / **29** µm

At least C3 clearance is required. Analysis with SKF proprietary software, considering the effects from smoothing of the mating surfaces and the probability that maximum fit reduction coincides with minimum bearing clearance, provides the following values for a bearing with C3 internal clearance:

min./avg./max.	**–2** / **16** / **32** µm

A small negative clearance is not critical for ball bearings. C3 clearance is adequate for this application.

Seals

It is not recommended to use shields (suffix 2Z) instead of contact seals (suffix 2RSH) in this application because there is a risk of grease leakage with outer ring rotation. The 2RSH seal design has the advantage of being more resistant to washout (high-pressure cleaning) that happens in paper machines, and so this will increase service life.

Consider hybrid bearings

Depending on the paper machine and position of the rope sheave, the sheave may face higher operating temperatures, which will reduce the grease life. To increase grease life, the use of hybrid bearings (ceramic balls instead of steel ones) of the same size can increase the grease life by at least a factor of two.

Consider design change

By changing the rope sheave hub design so that the bearing's inner ring rotates instead of the outer ring, grease life is increased. The speed factor will be $n\ d_m = 131\ 000$ instead of $nD = 176\ 400$.

The grease life, L_{10h}, of the 6207-2RSH/C3WT bearing will increase from 50 000 h to 61 000 h.

SKF has developed a rope sheave hub to take the above consideration into account. The bearings have ceramic balls, and WT grease, and their inner rings rotate (fig. 2). An enhanced design has been created using special bearings. For additional information, see the handbook *Rolling bearings in paper machines*.

Sealing, mounting and dismounting

Sometimes, simple labyrinth seals are added to protect the bearing integral seals further.

The normal mounting and dismounting methods are applicable.

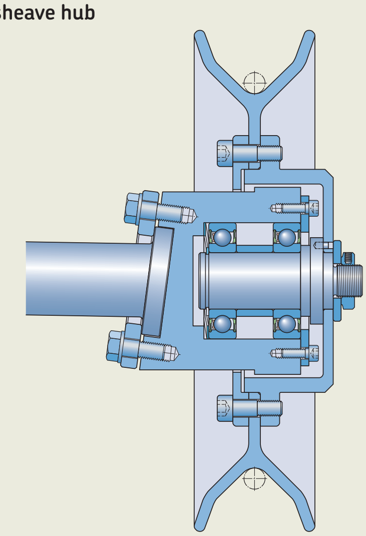
Overall conclusions

The bearing that fulfills the requirements is a sealed and greased SKF Explorer bearing 6207-2RSH/C3WT.

For more demanding operating conditions, or to achieve an even longer maintenance-free period, SKF can provide other solutions.

Fig. 2

SKF rope sheave hub



C.3 Centrifugal pump

This example shows the bearing selection process applied to an application case in which modification is required to a centrifugal pump.

The pump manufacturer wants to improve the efficiency of an existing centrifugal process pump by modifying the impeller. As a result, the bearing loads will be greater, and so the current bearing selection needs to be checked to verify that it can cope with the change. The application drawing is shown in **fig. 1**.

The steps in the example follow the sequence in the bearing selection process. Refer to sections **B.1 – B.8** for a full description of each process step.

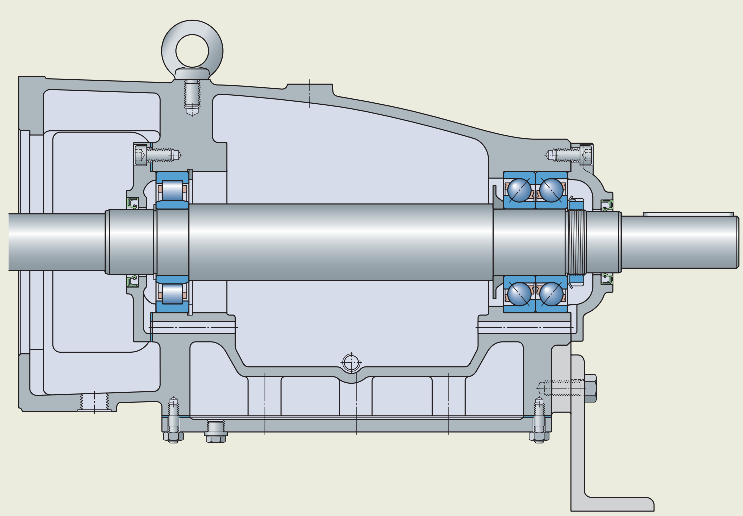
Performance and operating conditions

The operating conditions are:

- rotational speed: n = 3 000 r/min
- lubrication:
  - method: oil bath
  - oil viscosity grade: ISO VG 68
- for the non-locating support – a cylindrical roller bearing, NU 311 ECP:
  - max. radial load: F_r = 3,29 kN
  - estimated operating temperature: T = 70 °C *(160 °F)*

Fig. 1

The centrifugal pump and its bearing arrangement



- for the locating support – a pair of universally matchable single row angular contact ball bearings, 7312 BECBP, arranged back-to-back:
  - max. radial load: F_r = 1,45kN
  - max. axial load: F_a = 11,5 kN
  - estimated operating temperature: T = 85 °C *(185 °F)*

Following pump industry standards, the basic rating life L_{10h} should be at least 16 000 h at maximum load conditions.

Bearing type and arrangement

A cylindrical roller bearing is used as the non-locating support and a pair of universally matchable single row angular contact ball bearings are used as the locating support.

The cylindrical roller bearing, of type NU, is used for the following reasons:

- It can accommodate, within itself, thermal expansion of the shaft.
- The inner ring is separable from the outer ring, with rollers and cage – this simplifies assembly of the pump and the use of interference fits on both inner and outer ring.

For the pair of universally matchable single row angular contact ball bearings:

- Ball bearings with a 40° angle are well suited to accommodate high axial loads and medium to high speeds.
- The bearings are arranged back-to-back, with the inner rings clamped and mounted with an interference fit to the shaft. Because the clearance of the pair is controlled by clamping the inner rings, the outer rings can be positioned in the housing between a shoulder and a cover, without the need for precise clamping.

Both bearing housing seats are machined in one clamping position, which guarantees good alignment. Misalignment is less than 2 minutes of arc, which is within the acceptable misalignment limits for the angular contact ball bearing pair and cylindrical roller bearing.

Conclusion

The current selection of bearing type and arrangement is adequate for this application.

Bearing size, non-locating support

The given operating conditions, and the effects of rolling contact fatigue, indicate that bearing size should be determined using the basic rating life and SKF rating life.

Product data for NU 311 ECP is on page 522.

Basic rating life

$$L_{10h} = \left(\frac{10^6}{60\,n}\right)\left(\frac{C}{P}\right)^p$$

From *Loads*, page 509, P = F_r. Therefore, the load ratio C/P = 156/3,29 = 47

$$L_{10h} = \left(\frac{10^6}{60 \times 3\,000}\right)\left(\frac{156}{3{,}29}\right)^{3{,}33} > 1\,000\,000 \text{ h}$$

The bearing is oversized.

SKF rating life

$L_{10mh} = a_{SKF}\, L_{10h}$

1. Lubrication condition – the viscosity ratio, κ

$\kappa = \nu/\nu_1$

Given:
oil viscosity grade = ISO VG 68
operating temperature = 70 °C *(160 °F)*

then, using diagram 13, page 100, ν = 20 mm²/s

Given:
n = 3 000 r/min
d_m = 0,5 (55 + 120) = 87,5 mm

then, using diagram 14, page 101, ν_1 = 7 mm²/s

Therefore, κ = 20/7 = 2,8

2. Contamination factor, η_c

Given:

- contamination conditions are typical (i.e. open bearings, no filtering, wear particles and ingress from surrounding environment)
- d_m = 87,5 mm

then, using table 6, page 105, η_c = 0,2

C.3 Centrifugal pump

Given:
$P_u = 18{,}6$ kN
$P = F_r = 3{,}29$ kN (*Loads*, **page 509**)

then $\eta_c P_u/P = 0{,}2 \times 18{,}6/3{,}29 = 1{,}13$

3. Life modification factor, a_{SKF}

Given:
$\kappa = 2{,}8$
$\eta_c\, P_u/P = 1{,}13$
NU 311 ECP is an SKF Explorer bearing

then, using **diagram 10**, **page 97**, $a_{SKF} = 50$

Given:
$L_{10h} > 1\,000\,000$ h

then $L_{10mh} > 50 \times 1\,000\,000$ h
then $L_{10mh} >> 1\,000\,000$ h indicating that the bearing is oversized for the operating conditions.

Minimum load

The fact that the basic rating life and SKF rating life are both very high and above the required bearing life indicates that the bearing may be too lightly loaded.

Using the minimum load equation from *Loads*, **page 509**, the minimum radial load, F_{rm}, required to avoid skidding and roller slip for cylindrical roller bearings is given by:

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

Given:
$d_m = 87{,}5$ mm
$k_r = 0{,}15$
$n = 3\,000$ r/min
$n_r = 6\,000$ r/min

then $F_{rm} = 0{,}94$ kN $< F_r = 3{,}29$ kN

Conclusion

The bearing is oversized / lightly loaded. Options are:

- Continue to use the current bearing. There is no risk that the bearing will be damaged due to being too lightly loaded.
- Downsize the bearing, and in so doing reduce cost. Consider one of the following:
  - Keep the shaft diameter the same, but use the smaller NU 2 series bearing NU 211 ECP (refer to the product section).
  - Reduce the shaft diameter one step, provided the shaft design permits (strength and stiffness), and use the smaller NU 2 series bearing NU 210 ECP (refer to the product section).

However, both of these downsizing actions require design modifications to the adjacent components.

Bearing size, locating support

The given operating conditions, and the effects of rolling contact fatigue, indicate that bearing size should be determined using the basic rating life and SKF rating life.

Product data for 7312 BECBP is on **page 414**

Basic rating life

$$L_{10h} = \left(\frac{10^6}{60\,n}\right) \left(\frac{C}{P}\right)^p$$

From *Loads*, **page 398**:

$C = 1{,}62\, C_{\text{single bearing}} = 1{,}62 \times 104 = 168{,}5$ kN

From *Loads*, **page 398**, for bearing pairs arranged back-to-back:

$F_a/F_r = 11{,}5/1{,}45 > 1{,}14$

So use:

$P = 0{,}57\, F_r + 0{,}93\, F_a = (0{,}57 \times 1{,}45) + (0{,}93 \times 11{,}5) = 11{,}52$ kN

Therefore, the load ratio $C/P = 168{,}5/11{,}52 = 14{,}6$

$$L_{10h} = \left(\frac{10^6}{60 \times 3\,000}\right) \left(\frac{168{,}5}{11{,}52}\right)^3 = 17\,400 \text{ h}$$

SKF rating life

$L_{10mh} = a_{SKF}\, L_{10h}$

1. Lubrication condition – the viscosity ratio, κ

$\kappa = \nu/\nu_1$

Given:
oil viscosity grade = ISO VG 68
operating temperature = 85 °C *(185 °F)*

then, using **diagram 13**, **page 100**, $\nu = 13$ mm²/s

Given:
$n = 3\,000$ r/min
$d_m = 0{,}5\,(60 + 130) = 95$ mm

then, using **diagram 14**, **page 101**, $\nu_1 = 7$ mm²/s

Therefore, $\kappa = 13/7 = 1{,}8$

The next higher viscosity grade, ISO VG 100, would give $\kappa = 2,5$. But this would result in the NU 311 ECP bearing having $\kappa > 4$, which, during cold starts in particular, would give unwanted high κ values.

2. Contamination factor, η_c

Given:

- contamination conditions are typical (i.e. open bearings, no filtering, wear particles and ingress from surrounding environment)
- $d_m = 95$ mm

then, using **table 6**, **page 105**, $\eta_c = 0,2$

Given:

$P_u = 2 \times 3,2 = 6,4$ kN
P = 11,52 kN (*Basic rating life*)

then $\eta_c P_u/P = 0,2 \times 6,4/11,52 = 0,11$

3. SKF life modification factor a_{SKF}

Given:

$\kappa = 1,8$
$\eta_c\ P_u/P = 0,11$
7312 BECBP are SKF Explorer bearings

then, using **diagram 9**, **page 96**, $a_{SKF} = 5$

Given:

$L_{10h} = 17\ 400$ h

then $L_{10mh} = 5 \times 17\ 400 = 87\ 000$ h

Conclusion

The pair of 7312 BECBP SKF Explorer bearings are of a suitable size.

Lubrication

The pump has an oil bath. This is typical of process pumps, because of their requirement for long service intervals. In this pump, for simplicity, the locating and the non-locating support bearings are lubricated by the same oil bath.

As determined in previous steps, κ is 1,8 for the pair of angular contact ball bearings and 2,8 for the cylindrical roller bearing, and so the viscosity grade of the selected oil is adequate.

Operating temperature and speed

Determine whether a detailed thermal analysis is required (*Thermal equilibrium*, **page 131**) by checking that:

- the rotational speed is less than 50% of the bearing limiting speed:
  - This is true for the non-locating support.
  - For the locating support, it is 56%, which is just slightly above the limit. That is, for a pair of single row angular contact ball bearings, the limiting speed is reduced by 20% (*Permissible speed*, **page 402**), and so 3 000/(0,8 × 6 700) = 0,56.
- the load ratio C/P > 10:
  - This is true for the locating and non-locating supports.
- there is no pronounced external heat input:
  - The pump is located in an environment where the ambient temperature is 20 to 30 °C *(70 to 85 °F)*.
  - The pump medium is at ambient temperature, so no additional heat flows to the bearings.

Therefore, no further thermal analysis is needed.

Bearing interfaces

Because the loads on the bearings will be greater, as a result of the modification to the pump, you should check the bearing seat tolerances to make sure the bearings are mounted with adequate fits.

Given the standard steel shaft and cast iron housing, the bearing loads, speeds and temperatures, which are all within standard conditions, you can apply *Seat tolerances for standard conditions*, **page 148**.

Shaft tolerances

You can find shaft tolerances for seats for radial ball bearings in **table 5**, **page 148**, and for radial roller bearings in **table 6**, **page 149**.

Given:

	NU 311 ECP	7312 BECBP
Condition of rotation	rotating inner ring load	rotating inner ring load
P/C ratio	0,02	0,07
Bore diameter	55 mm	60 mm

Results:

	Bearing seat			
Bearing	**Dimensional tolerance**	**Total radial run-out tolerance**	**Total axial run-out tolerance**	**Ra**
NU 311 ECP	k6Ⓔ	IT5/2	IT5	0,8 µm
7312 BECBP	k5Ⓔ	IT4/2	IT4	0,8 µm

Housing tolerances

Any wear developing during service may lead to imbalance of the impeller, leading to an indeterminate direction of load on the outer rings of both bearings.

You can find tolerances for seats for cast iron and steel housings, for radial ball bearings, in table 8, page 151.

Given:

	NU 311 ECP	7312 BECBP
Condition of rotation	indeterminate direction of load	indeterminate direction of load
P/C ratio	0,02	0,07
Outer diameter	120 mm	130 mm

Results:

Bearing	**Dimensional tolerance**	**Total radial run-out tolerance**	**Total axial run-out tolerance**	**Ra**
NU 311 ECP	K7Ⓔ	IT6/2	IT6	3,2 µm
7312 BECBP	K7Ⓔ	IT6/2	IT6	3,2 µm

Axial location

The current design has adequate axial location. Make sure that the lock nut that locates the inner rings of the angular contact ball bearings is sufficiently tightened. Apply the clamp force uniformly around the circumference and respect the abutment dimensions (product data for 7312 BECBP is on page 414). To avoid distortion of the inner rings and to achieve the desired axial clearance in the bearing pair, limit the clamping force. For centrifugal pumps, a clamping force of $C_0/4$ (19 kN) is recommended.

Bearing execution

Checking the initial internal clearance

The current design uses bearings with Normal initial clearance. The fits for the inner and outer rings, and a temperature difference between the inner and outer rings of 10 °C (*20 °F*), reduce the internal clearance. Other influences on the internal clearance are negligible.

1. Initial internal clearance

	NU 311 ECP	Pair of 7312 BECBP
min./avg./max.	**40 / 55 / 70 µm**	**22 / 32 / 27 µm**
	Refer to *Bearing data*, page 504. Values obtained from table 3, page 506.	Refer to *Bearing data*, page 392. Axial values obtained from table 4, page 394, converted to radial (axial × tan 40°).

2. Clearance reduction caused by interference fits

Use:

$\Delta r_{fit} = \Delta_1 f_1 + \Delta_2 f_2$ (*Clearance reduction caused by interference fits*, page 184)

Obtain values for:

- factors f_1 and f_2 (diagram 2, page 184)
- probable fits for shafts, Δ_1 (table 14, page 160)
- probable fits for housings, Δ_2 (table 20, page 172)

Results:

		NU 311 ECP	Pair of 7312 BECBP
d/D		0,46	0,46
f_1		0,78	0,78
f_2		0,86	0,86
Δ_1	min./avg./max.	–32 / –19 / –6 µm	–26 / –16 / –6 µm
Δ_2	min./avg./max.	–20 / 0 / 20 µm	–21 / 1 / 23 µm
Δr_{fit}	min./avg./max.	**–42 / –15 / –5 µm**	**–38 / –12 / –5** µm

3. Clearance reduction caused by temperature difference

Use:

$\Delta r_{temp} = 0{,}012\ \Delta T\ d_m$ (*Clearance reduction caused by temperature difference between shaft, bearing rings and housing*, page 184)

Results:

	NU 311 ECP	Pair of 7312 BECBP
d_m	87,5 mm	95 mm
Δr_{temp}	**–11 µm**	**–11 µm**

4. Operating clearance

	NU 311 ECP	Pair of 7312 BECBP
min./avg./max.	**–13 / 30 / 55 µm**	**–27 / 17 / 4 µm**

For a cylindrical roller bearing, negative clearance (i.e. preload) is generally not recommended.

Pairs of angular contact ball bearings should have an average operating clearance close to zero (ranging between small clearance and light preload), particularly when the pairs are loaded predominantly axially. A small range is required to:

- limit preload – to limit friction (increased friction results in higher temperatures, and therefore reduced viscosity and reduced bearing life)
- limit clearance – to avoid ball skidding

This manual calculation does not consider smoothing of the mating surfaces, nor elastic deflection under load, nor the probability of extreme values occurring at the same time.

Analysis using more advanced SKF software gives operating clearance results:

	NU 311 ECP	Pair of 7312 BECBP
min./avg./max.	**3 / 34 / 59 µm**	**–10 / 11 / 24 µm**

These results indicate that Normal internal clearance is suitable.

Cage selection

Given the estimated operating temperature of 85 °C *(185°)* (i.e. the higher temperature of the two bearing supports), a speed of well below the limiting speed, and considering availability and price, the standard rolling element guided polyamide cages are confirmed as adequate.

For historical reasons, in some geographical areas, brass cages are preferred for angular contact ball bearings. These are available as standard from SKF. This also applies to the cylindrical roller bearings.

Conclusion

Non-locating support

The NU 311 ECP bearing, currently used in the centrifugal pump, is adequate. As an alternative, the NU 311 ECM bearing could be used. Downsizing of the bearing is possible.

Bearing execution is described by suffixes in the bearing designation (*Designation system*, page 514).

Designation suffixes:

	Suffix	Description
Internal design	EC	optimized internal design incorporating more and/or larger rollers and with a modified roller end / flange contact designed to minimize friction
Cage design	P	glass fibre reinforced PA66 cage, roller centred
	M	machined brass cage, riveted, roller centred
Clearance class	–	Normal

Locating support

The pair of universally matchable 7312 BECBP bearings, currently used in the centrifugal pump, are adequate. As an alternative, the 7312 BECBM bearing could be used.

Bearing execution is described by suffixes in the bearing designation (*Designation system*, page 404).

Designation suffixes:

	Suffix	Description
Internal design	B	40° contact angle
	E	optimized internal design – reinforced rolling element set
External design / clearance class	CB	bearing for universal matching; two bearings arranged back-to-back or face-to-face; have Normal axial internal clearance
Cage design	P	glass fibre reinforced PA66 cage, ball centred
	M	machined brass cage, ball centred

Fig. 2

Radial shaft seal, design HMS5

Fig. 3

Radial shaft seal, design HMSA10

Sealing, mounting and dismounting

Sealing

The current pump design uses radial shaft seals to keep the oil bath lubricant in the pump and to protect the bearings from contamination (**fig. 1**, **page 228**). You can use seals SKF HMS5 (**fig. 2**) or HMSA10 (**fig. 3**). These are suitable for both oil and grease lubricated applications. The temperature range and speed capability of the nitrile rubber compound used for these seals suits the operating conditions of the pump.

When the seal counterface becomes worn, you can repair the shaft with a wear sleeve, such as SKF Speedi-Sleeve.

Hot mounting of the bearings

The bearings are mounted with an interference fit on the shaft and a transition fit in their housings. You can mount the bearings easily by heating their inner rings to 100 °C *(210 °F)* and the housing seats to 50 °C *(160 °F)*. For heating the inner rings, use an SKF induction heater or electric hot plate.

Shaft alignment

To maximize pump life, the pump and its electric motor need to be well aligned. SKF alignment tools can help.

Overall conclusions

The existing bearings can be used in combination with the new impeller design.

Downsizing of the cylindrical roller bearing is recommended.